King Abdulaziz University

Mechanical Engineering Department

MEP 460

Heat Exchanger Design

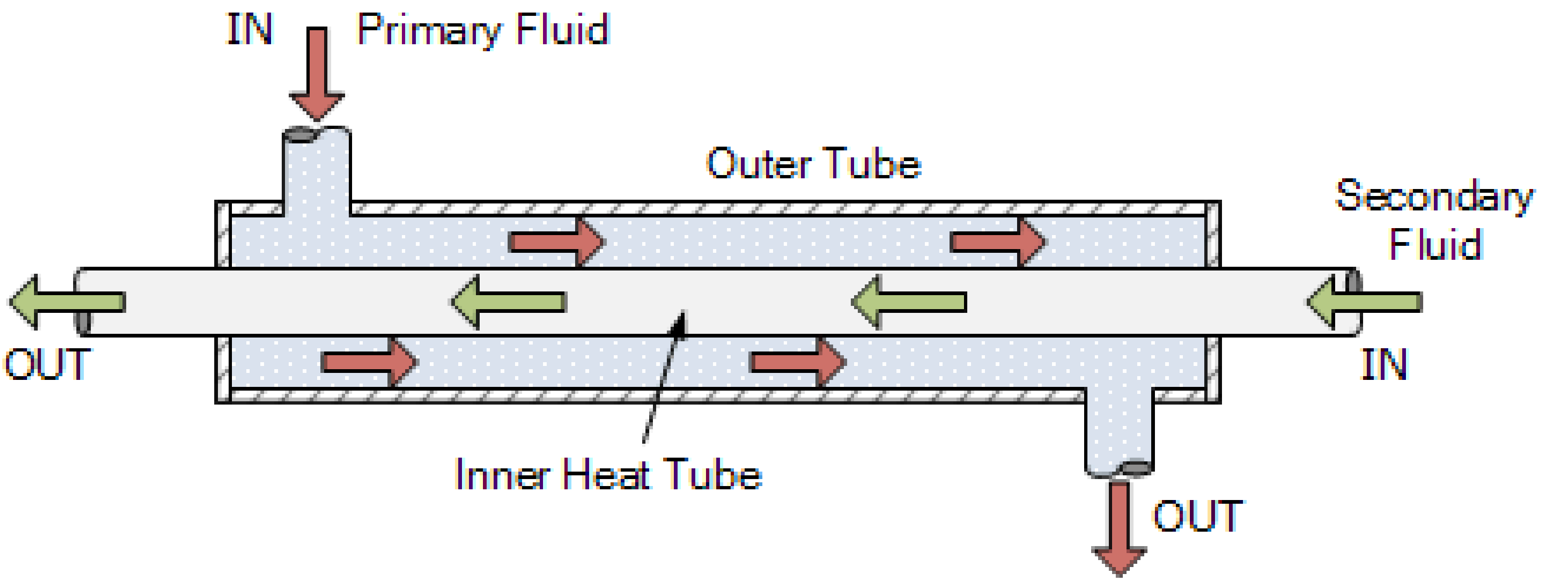
Double pipe heat Exchanger

Feb. 2019

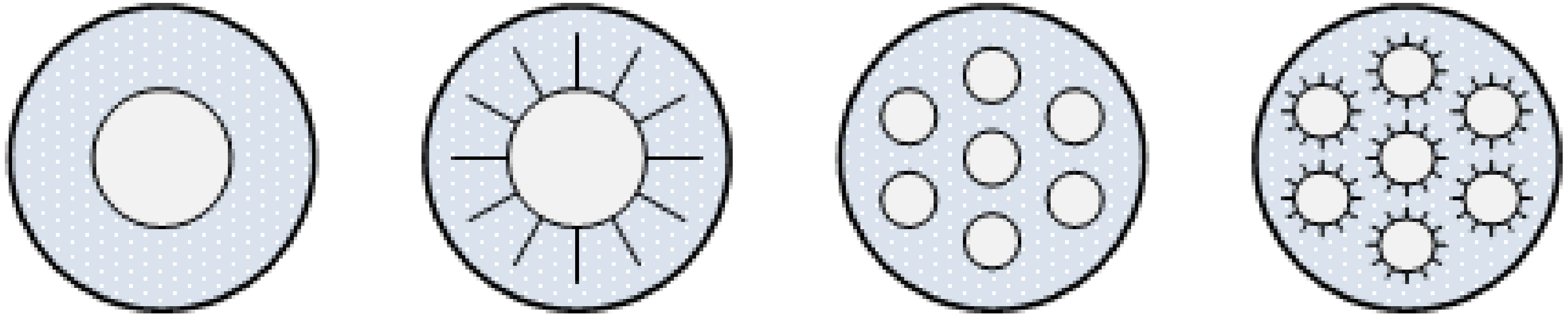
Ch. 7 Double pipe heat exchangers

1-Introduction
2-Advantages and disadvantages of double pipe HX
3-Double pipe HX geometry
4-Unfinned outside surface of inner pipe
5-Example 7.1
6-Finned outer surface of the inner pipe
7-Example 7.2
8-Series & parallel combination of hairpins HX
9-Total pressure drop in double pipe HX

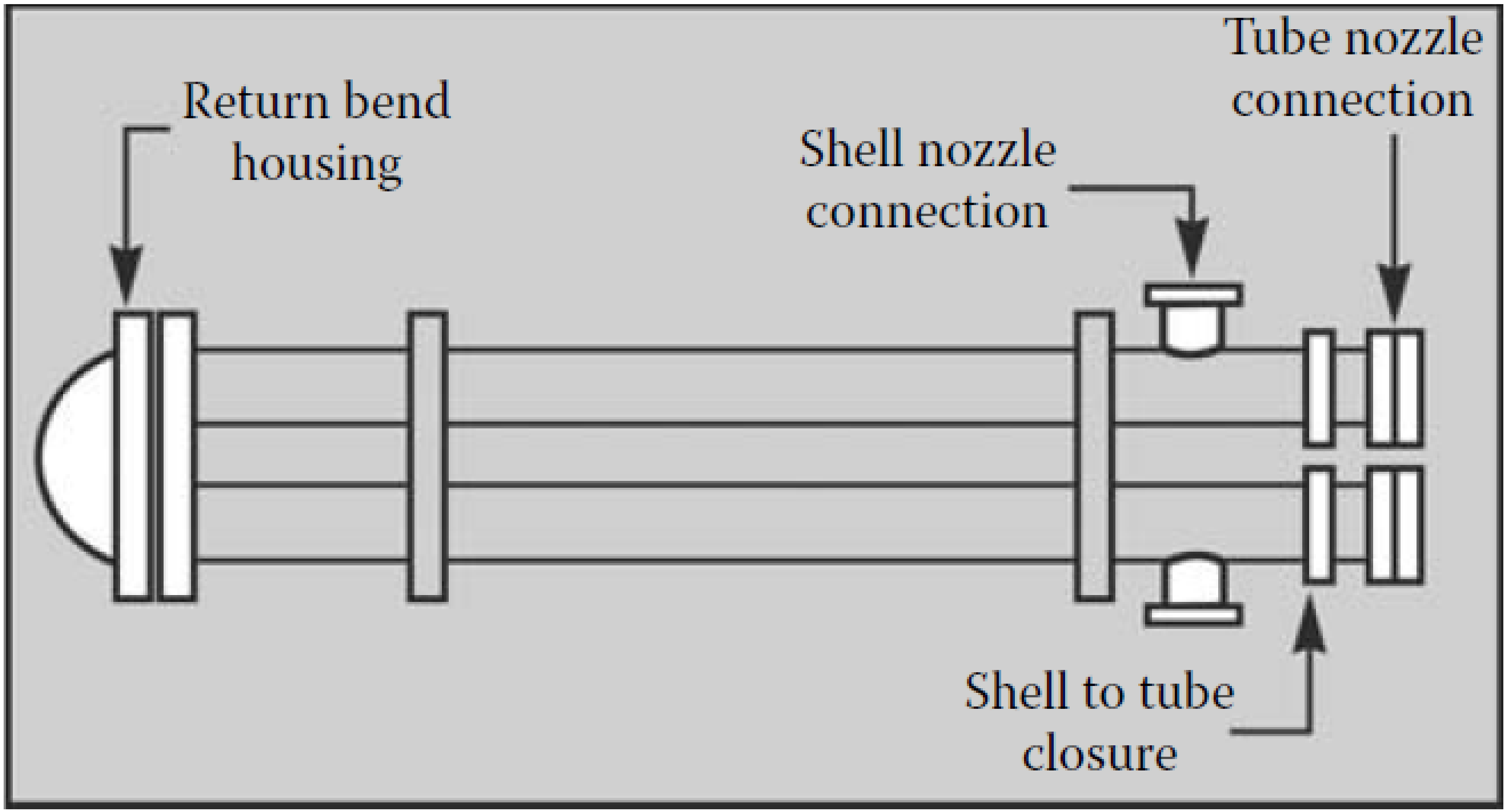
1-Introduction



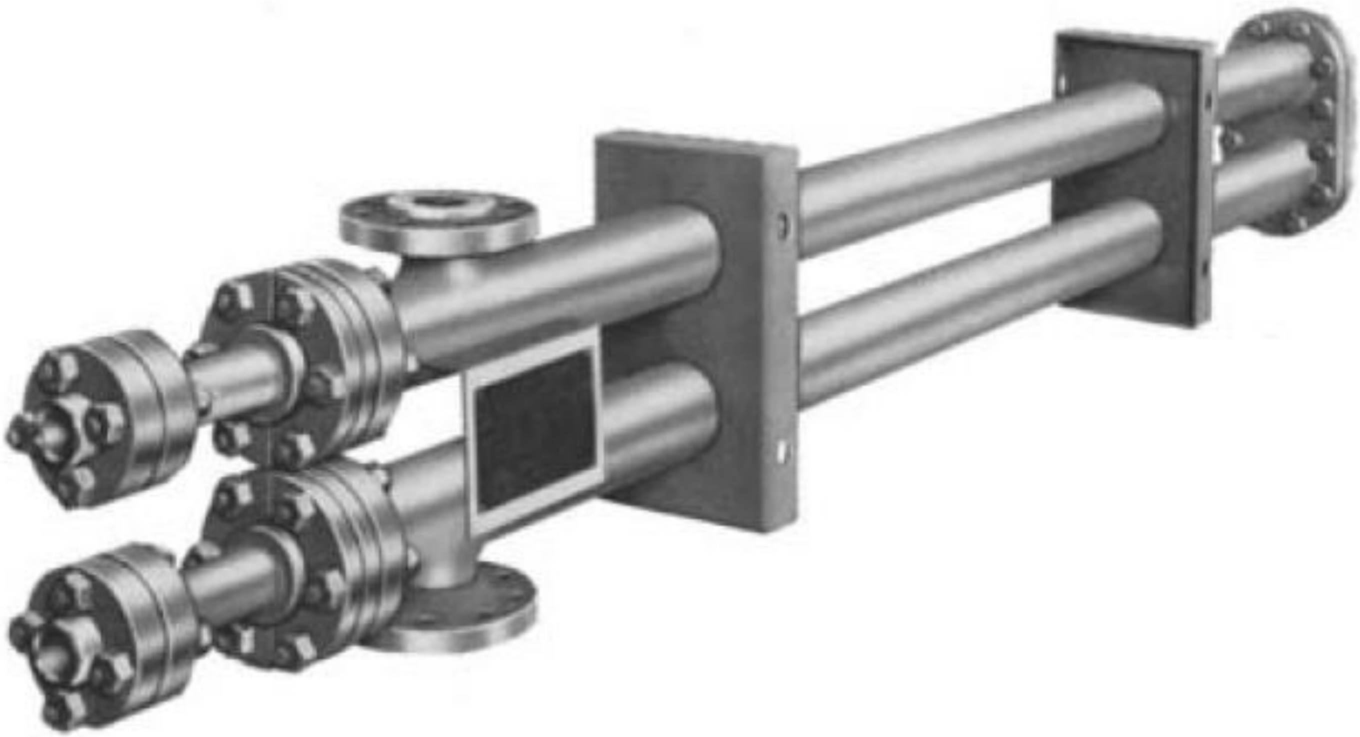
Tubular Heat Pipe Configurations of Inner Tubes



Hairpin double pipe heat exchangers

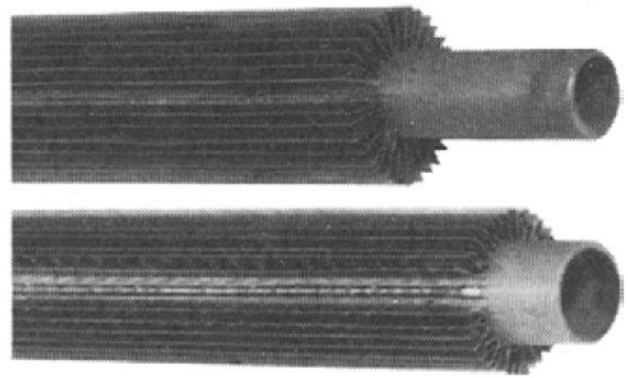


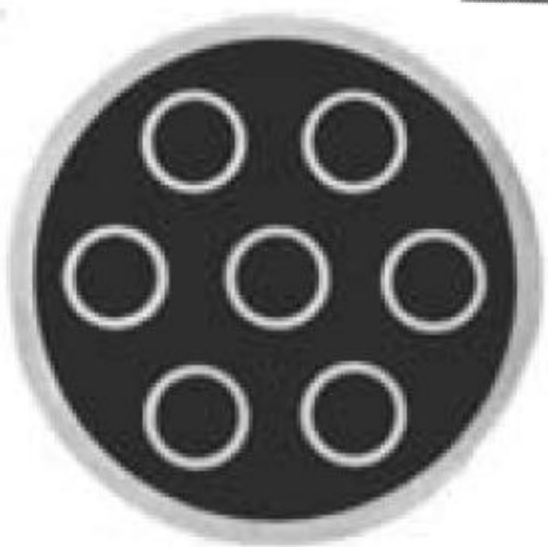
1-Introduction



Double-pipe hairpin sections—Available in both finned and bare tube designs. Shell sizes range from 2" to 6" IPS. Applications: Ideal for all severe operating conditions. A simple, low-cost section is also available for standard petroleum, petrochemical or chemical service.

Finned and un-finned double pipe heat exchangers

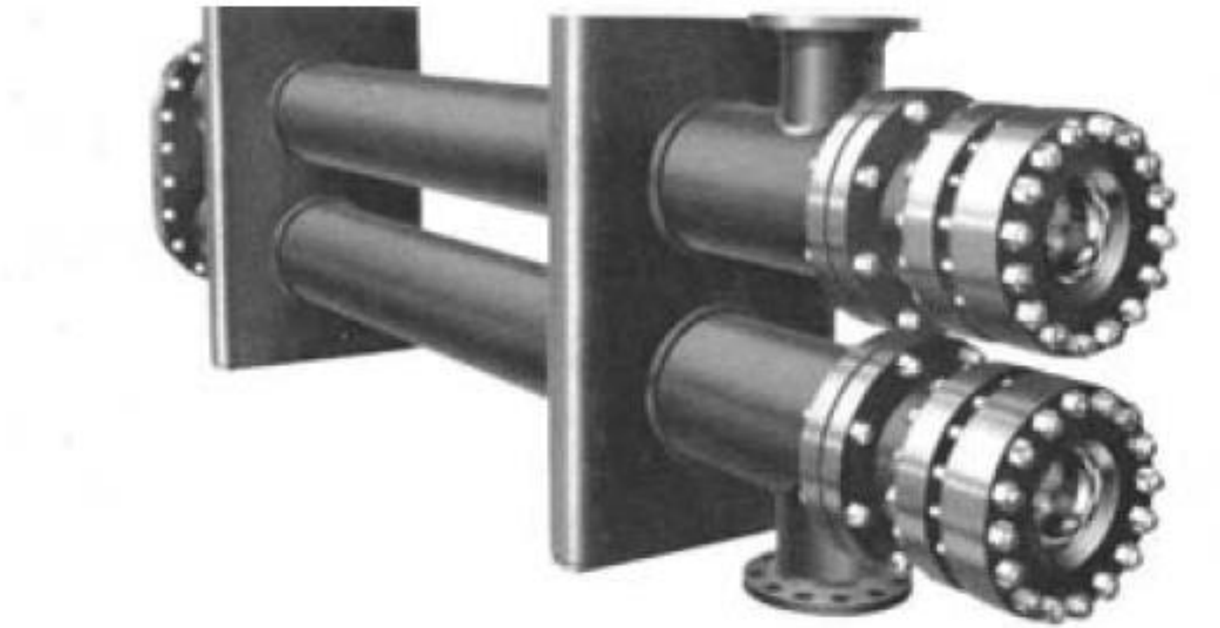




Cross section view of bare tubes inside shell

Cross section view of fintubes inside shell

Multi tube double pipe heat exchangers



Multi-tube hairpin sections—Available in both finned and bare tube designs. Shell sizes range from 3" to 16" IPS. Applications: Designs are available to meet most heat exchanger requirements; from heating heavy oils and asphalts to light chemicals, gasoline, butanes, and other hydrocarbons. Also available are designs for high-pressure-gas heat transfer and for processing lethal or hard-to-contain fluids and gases such as hydrogen, dowhterm and acids.

2-Advantages and disadvantages of double pipe heat exchanges

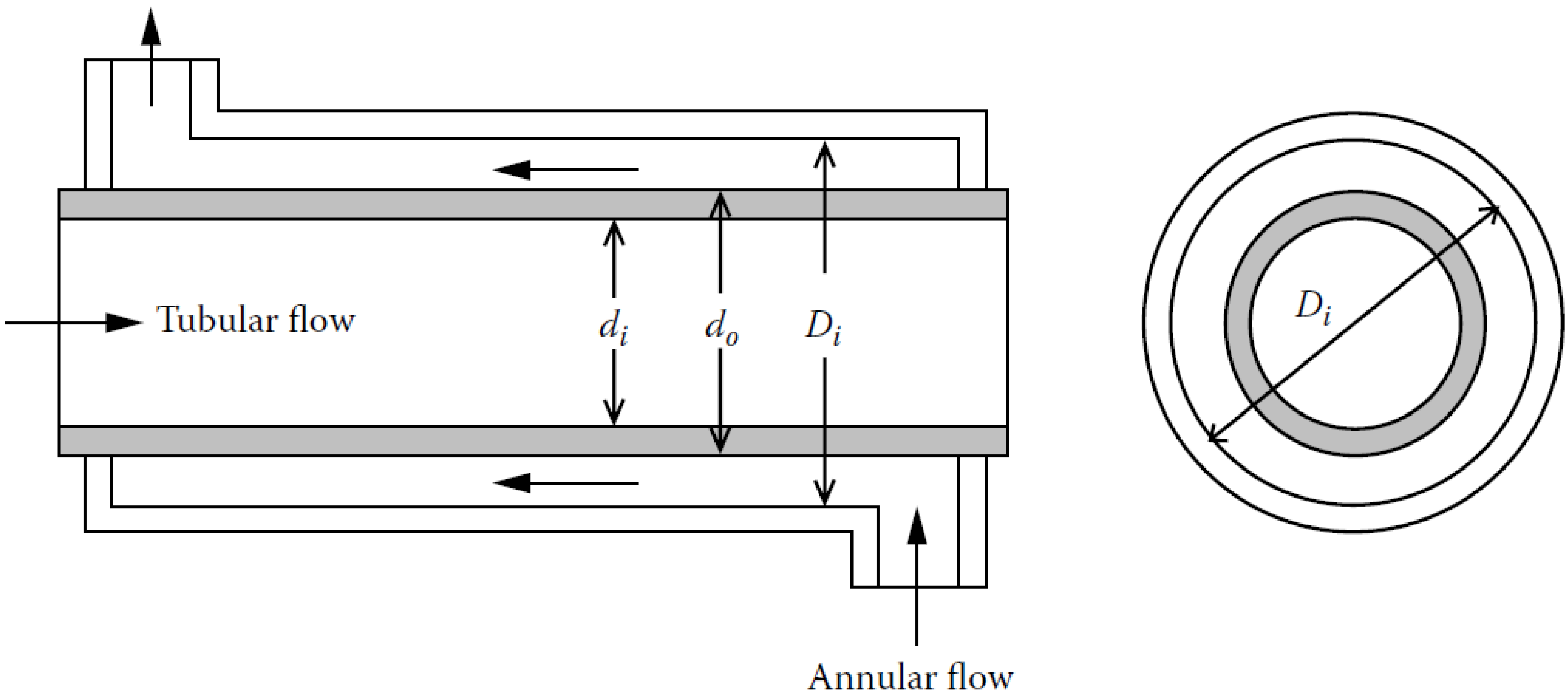
advantages

Good for sensible heating or cooling
Up to 50 ^{2} heat transfer area
Good for one or two fluids at high pressure
Can be used at severe fouling condition since it can easily be cleaned
Can achieve pure parallel or counter flow arrangement
U tube or hairpin heat exchanger can handle thermal expansion
Extra units can be easily added

Disadvantages

Bulky
Expensive

3-Double pipe geometry



Cross section area for annulus flow $$A_c = \left(\frac{\pi}{4}\right)\left[D_i^2 - d_o^2\right]$$

Hydraulic diameter for heat transfer calculations and pressure drop calculations for annulus flow

D_e for heat transfer calculations

By definition the hydraulic diameter

$$D_h = \frac{4\ min.\ flow\ area}{wetted\ premiter}$$

Perimeter for heat transfer calculations

$$P_h = \pi d_o$$

Hydraulic diameter for heat transfer calculations

$$D_e = \frac{4A_c}{P_h} \qquad A_c = \left(\frac{\pi}{4}\right)\left[D_i^2 - d_o^2\right]$$

$$D_e = \frac{4(\pi D_i^2/4\ - \pi d_o^2/4)}{\pi d_o} = \frac{D_i^2 - d_o^2}{d_o} \qquad Nu = \frac{hD_e}{k}$$

Annulus flow velocity

$$V_a = \frac{\dot{m}_a}{\rho_a A_c} \qquad R_{e_a} = \left(\frac{\rho_a V_a D_h}{\mu_a}\right)$$

Hydraulic diameter for heat transfer calculations and pressure drop calculations for annulus flow

D_h for pressure drop calculations

Perimeter for pressure drop calculations

$$P_w = \pi(D_i + d_o)$$

Cross section area

$$A_c = (\pi/4)(D_i^2 - d_o^2)$$

Hydraulic diameter for pressure calculations

$$D_h = \frac{4A_c}{P_w}$$

$$D_h = \frac{4(\pi D_i^2/4 - \pi d_o^2/4)}{\pi(D_i + d_o)} = D_i - d_o$$

$$Re_{Dh} = \frac{\rho_a V_a D_h}{\mu_a}$$

Pressure drop calculations

Inside the tubes

$$\Delta p_t = f\frac{2L}{d_i}\rho\frac{u_m^2}{2}N_{hp} = f\frac{2L}{d_i}\frac{G^2}{2\rho}N_{hp}$$

Pumping power

$$P_t = \frac{\Delta p_t \dot{m}_t}{\eta_p \rho}$$

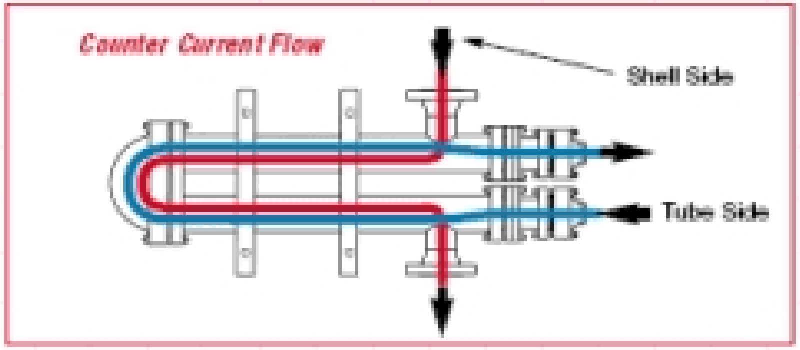
Flow in the annulus

$$\Delta P_a = f\frac{2L}{D_h}\rho\frac{u_a^2}{2}N_{hp}$$

Assuming the length of hairpin HX to be L

Pumping power

$$P_a = \frac{\Delta p_a \dot{m}_a}{\eta_p \rho}$$



N_{hp} is the number of hairpins HX

Example 7.1

Example 7.1

Water at a flow rate of 5,000 kg/h 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 3.5 m hairpins of 3 in. (ID = 0.0779 m) by 2 in. (ID = 0.0525 m, OD = 0.0603 m) counterflow double-pipe heat exchangers with annuli and pipes, each connected in series, will be used. Hot water flows through the inner tube. Fouling factors are: $R_{fi} = 0.000176\ \mathrm{m^2 \cdot K/W}$, $R_{fo} = 0.000352\ \mathrm{m^2 \cdot K/W}$. Assume that the pipe is made of carbon steel ($k = 54\ \mathrm{W/m \cdot K}$). The heat exchanger is insulated against heat losses.

1. Calculate the number of hairpins.
2. Calculate the pressure drops.

Example 7.1 continue

Inner tube side heat transfer coefficient—First calculate the Reynolds number to determine if the flow is laminar or turbulent.

From the appendix, the properties of hot water at $T_b = 132.5°C$ are

$$\rho = 932.53\ \text{kg/m}^3, \qquad c_p = 4.268\ \text{kJ/kg}\cdot\text{K}$$
$$k = 0.687\ \text{W/m}\cdot\text{K}, \qquad \mu = 0.207\times 10^{-3}\ \text{Pa}\cdot\text{s}$$
$$Pr = 1.28$$

The hot water mass flow rate is calculated as follows:

$$Q = (\dot{m}c_p)_c\,\Delta T_c = (\dot{m}c_p)_h\,\Delta T_h$$
$$\dot{m}_h = \frac{(\dot{m}c_p)_c\,\Delta T_c}{c_p\Delta T_h} = \frac{(5{,}000/3{,}600)\times 4.179\times(35-20)}{4.268\times(15)} = 1.36\ \text{kg/s}$$

Example 7.1 continue

The hot water mass flow rate is calculated as follows:

$$Q = (\dot{m}c_p)_c \Delta T_c = (\dot{m}c_p)_h \Delta T_h$$

$$\dot{m}_h = \frac{(\dot{m}c_p)_c \Delta T_c}{c_p \Delta T_h} = \frac{(5{,}000/3{,}600) \times 4.179 \times (35-20)}{4.268 \times (15)} = 1.36 \text{ kg/s}$$

where $c_p = 4.179$ kJ/kg · K for cold water at $T_b = 27.5°C$.
The velocity and Reynolds number are calculated as follows:

$$u_m = \frac{\dot{m}_h}{\rho_h A_c} = \frac{1.36}{(932.53)\left(\frac{\pi}{4}\right)(0.0525)^2} = 0.673 \text{ m/s}$$

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{4\dot{m}_h}{\pi \mu d_i} = \frac{4 \times 1.36}{\pi \times 0.207 \times 10^{-3} \times 0.0525} = 159{,}343$$

Hence, the flow is turbulent; a correlation can be selected from Chapter 3. Prandtl's correlation is used here with constant properties:

Example 7.1 continue

To determine the heat transfer coefficient in the annulus, the following properties at $T_b = 27.5°C$ from the appendix are used:

$$\rho = 996.4 \text{ kg/m}^3, \qquad c_p = 4.179 \text{ kJ/kg}\cdot\text{K}$$
$$k = 0.609 \text{ W/m}\cdot\text{K}, \qquad \mu = 8.41\times10^{-6} \text{ Pa}\cdot\text{s}$$
$$Pr = 5.77$$

The velocity of cold water, u_m, through the annulus is calculated:

$$u_m = \frac{\dot{m}}{A_c\rho} = \frac{5{,}000/3{,}600}{\frac{\pi}{4}(0.0779^2 - 0.0603^2)(996.4)} = 0.729 \text{ m/s}$$

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

$$Re = \frac{\rho u_m D_h}{\mu} = \frac{996.4\times0.729\times0.0176}{0.841\times10^{-3}} = 15{,}201$$

Therefore, the flow is turbulent. Prandtl's correlation is also used for the annulus:

$$f = (3.64 \log_{10} Re_b - 3.28)^{-2} = [3.64 \log_{10}(15{,}201) - 3.28]^{-2} = 7.021\times10^{-3}$$

$$f/2 = 3.51\times10^{-3}$$

$$Nu_b = \frac{(f/2)(Re_b)Pr_b}{1+8.7(f/2)^{1/2}(Pr_b - 1)}$$

$$Nu_b = \frac{(3.51\times10^{-3})(15{,}201)(5.77)}{1+8.7(3.51\times10^{-3})^{1/2}(5.77-1)} = 89.0$$

Example 7.1 continue

$$Nu_b = \frac{(f/2)(Re_b)Pr_b}{1+8.7(f/2)^{1/2}(Pr_b - 1)}$$

where

$$\begin{aligned} f &= (1.58\ln\ Re - 3.28)^{-2} \\ &= [1.58\ln(1,59,343) - 3.28]^{-2} \\ &= 4.085\times10^{-3} \end{aligned}$$

$$Nu_b = \frac{(0.004085/2)(1,59,343)(1.28)}{1+8.7\left(\dfrac{0.004085}{2}\right)^{1/2}(1.28-1)} = 375.3$$

$$h_i = \frac{Nu_b k}{d_i} = \frac{375.3\times0.687}{0.0525} = 4911\ \mathrm{W/m^2\cdot K}$$

To determine the heat transfer coefficient in the annulus, the following properties at $T_b = 27.5°C$ from the appendix are used:

Example 7.1 continue

The equivalent diameter for heat transfer, from Equation 7.7, is

$$D_e = \frac{D_i^2 - d_o^2}{d_o} = \frac{0.0779^2 - 0.0603^2}{0.0603} = 0.0403 \text{ m}$$

and

$$h_o = \frac{Nu_b k}{D_e} = \frac{89.0 \times 0.609}{0.0403} = 1345 \text{ W/m}^2 \cdot \text{K}$$

The overall heat transfer coefficient based on the outside area of the inner tube is

$$\frac{1}{U_f} = \frac{d_o}{d_i h_i} + \frac{d_o R_{fi}}{d_i} + \frac{d_o \ln(d_o/d_i)}{2k} + R_{fo} + \frac{1}{h_o}$$

$$= \frac{0.0603}{0.0525 \times 4,911} + \frac{0.0603 \times 1.76 \times 10^{-4}}{0.0525}$$

$$+ \frac{0.0603 \ln\left(\frac{603}{525}\right)}{2 \times 54} + 3.52 \times 10^{-4} + \frac{1}{1345}$$

$$U_f = 622 \text{ W/m}^2 \cdot \text{K}$$

The heat transfer surface area can be calculated as follows:

Example 7.1 continue

The heat transfer surface area can be calculated as follows:

$$A_o = \frac{Q}{U_o \Delta T_m}$$

$$\Delta T_m = \Delta T_1 = \Delta T_2 = 105°\text{C}$$

The heat duty of the heat exchanger is

$$Q = \left(\dot{m}c_p\right)_c \Delta T_c = 1.389 \times 4.179 \times 15 = 87.1 \text{ kW}$$

So,

$$A_o = \frac{87.1 \times 1{,}000}{622 \times 105} = 1.33 \text{ m}^2$$

The heat transfer area per hairpin is

$$A_{hp} = 2\pi d_o L = 2\pi \times 0.0603 \times 3.5 = 1.325 \text{ m}^2$$

$$\frac{A_o}{A_{hp}} = \frac{1.33}{1.325} = 1$$

Therefore, the number of the hairpins, N_{hp}, equals 1.

Example 7.1 continue

To determine the pressure drop in the tube side, the frictional pressure drop can be calculated from Equation 4.16:

$$\Delta p_t = 4f\frac{2L}{d_i}N_{hp}\frac{\rho u_m^2}{2}$$

$$\Delta p_i = 4\times 4.085\times 10^{-3}\frac{2\times 3.5}{0.0525}\times 1\times 932.53\frac{(0.673)^2}{2} = 460.1\ \text{Pa}$$

To calculate the pumping power for the tube stream, Equation 4.19 for pumping power is used:

$$P_t = \frac{\Delta p_t \cdot \dot{m}_h}{\eta_p \rho_h} = \frac{460.1\times 1.36}{0.80\times 932.33} = 0.84\ \text{W}$$

For pressure drop in the annulus side,

$$\Delta p_a = 4f\frac{2L}{D_h}\rho\frac{u_m^2}{2}N_{hp}$$

$$\Delta p_a = 4\times 7.02\times 10^{-3}\frac{2\times 3.5}{0.0176}\times 996.41\frac{(0.719)^2}{2}\times 1 = 2876.4\ \text{Pa}$$

Example 7.1 continue

The clean heat transfer coefficient based on the outside heat transfer area is

$$\frac{1}{U_c} = \frac{d_o}{d_i h_i} + \frac{d_o \ln(d_o/d_i)}{2k} + \frac{1}{h_o}$$

$$= \frac{0.0603}{0.0525 \times 4{,}911} + \frac{0.0603 \ln(603/525)}{2 \times 54} + \frac{1}{1345}$$

$$U_c = 948 \text{ W/m}^2 \cdot \text{K}$$

The cleanliness factor can also be calculated

$$CF = \frac{U_f}{U_c} = \frac{622}{948} = 0.66$$

as can be the percentage over surface:

$$OS = 100 U_c R_{ft}$$

$$R_{ft} = \frac{1 - CF}{U_c CF} = \frac{1 - 0.66}{948 \times 0.66} = 0.543 \times 10^{-3} \text{ m}^2 \cdot \text{K/W}$$

$$OS = 100 \times 948 \times 0.543 \times 10^{-3} = 51.5\%$$

Kakac used expression for finding the heat transfer coefficient for both inside the pipe and in the annulus

See Ch. 3 & 4

$$Nu_b = \frac{(f/2)Re_b Pr_b}{1+8.7(f/2)^{1/2}(Pr_b - 1)}$$

Based on three-layer turbulent boundary layer model, $Pr > 0.5$

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

Valid for $2{,}300 < Re_b < 5 \times 10^6$ and $0.5 < Pr_b < 2{,}000$

Or in log form

$$f = \left(3.64 \log_{10} Re_b - 3.28\right)^{-2}$$

$$\Delta p_t = 4 f \frac{2L}{d_i} N_{hp} \frac{\rho u_m^2}{2}$$

For hairpin of length L.

Remember $f = f_{Kakac}/4$

Incropera Expression for turbulent flow heat transfer coefficient

$$Nu_D = \frac{(f/8)(Re_D - 1000)\,Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}$$

(8.62)[d] Turbulent, fully developed, $0.5 \lesssim Pr \lesssim 2000$, $3000 \lesssim Re_D \lesssim 5 \times 10^6$, $(L/D) \gtrsim 10$

$$f = (0.790 \ln Re_D - 1.64)^{-2}$$

(8.21)[c] Turbulent, fully developed, smooth walls, $3000 \lesssim Re_D \lesssim 5 \times 10^6$

$$\frac{1}{\sqrt{f}} = -2.0 \log\left[\frac{e/D}{3.7} + \frac{2.51}{Re_D\sqrt{f}}\right]$$

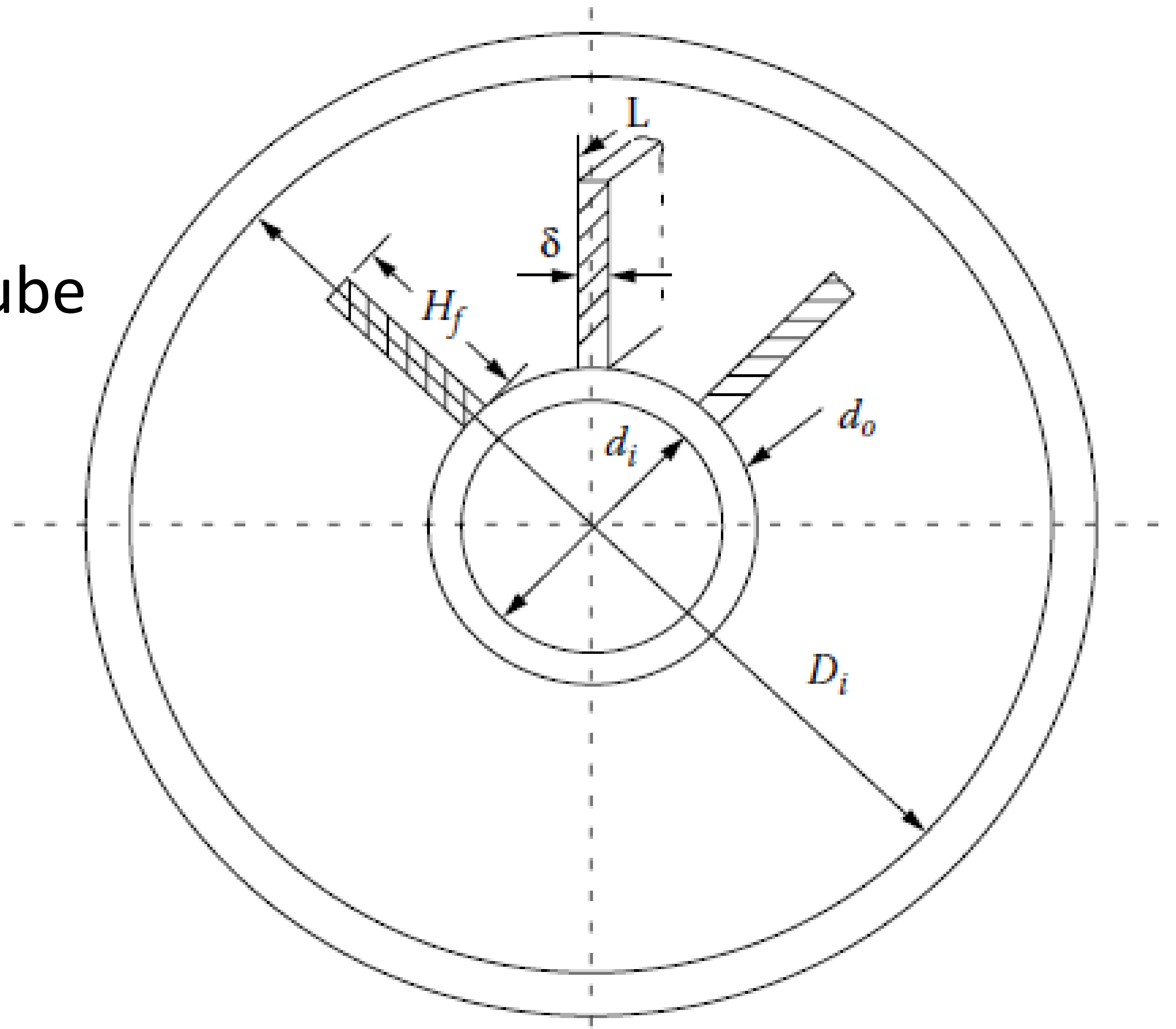
(8.20)[c] Turbulent, fully developed

$$Nu_D = 0.023\, Re_D^{4/5}\, Pr^n$$

(8.60)[d] Turbulent, fully developed, $0.6 \lesssim Pr \lesssim 160$, $Re_D \gtrsim 10{,}000$, $(L/D) \gtrsim 10$, $n = 0.4$ for $T_s > T_m$ and $n = 0.3$ for $T_s < T_m$

6-Finned outer surface of the inner pipe

H_f fin height
δ Fin thickness
N_f no of fins per tube
N_t no of tubes



Hairpin Heat exchanger with multi-tube finned inner pipe

The wetted perimeter for hydraulic diameter

$$P_w = \pi(D_i + d_o N_t) + 2H_f N_f N_t$$

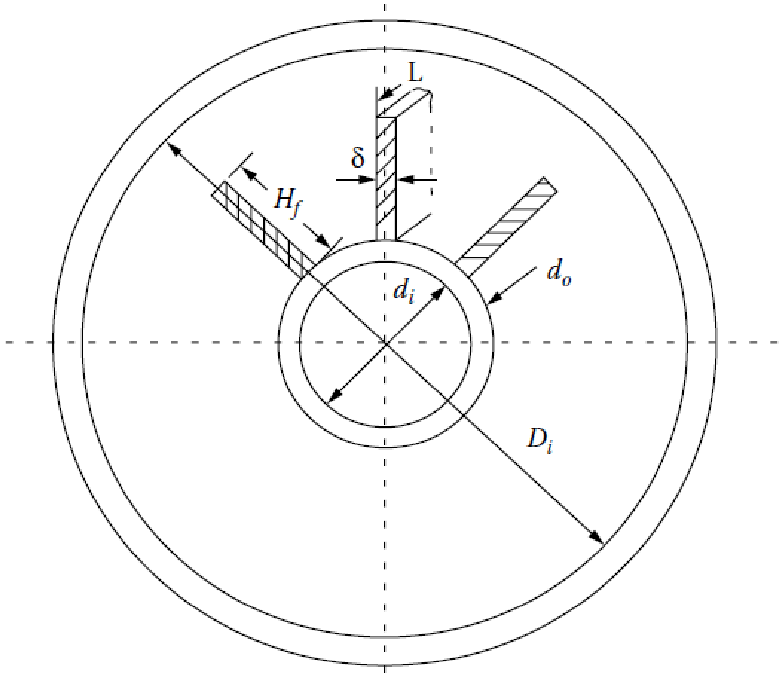
For calculating the heat transfer equivalent diameter

$$P_h = \left(\pi d_o + 2H_f N_f\right) N_t$$

Cross section flow area

$$A_c = \frac{\pi}{4}\left(D_i^2 - d_o^2 N_t\right) - \delta H_f N_t N_f$$

$$D_e = \frac{4A_c}{P_h} \qquad D_h = \frac{4A_c}{P_w}$$



Symbol	meaning
N_t	Number of tubes
N_f	Number of fins/tube
H_f	Fin height
δ	Fin thickness
L	Heat exchanger length

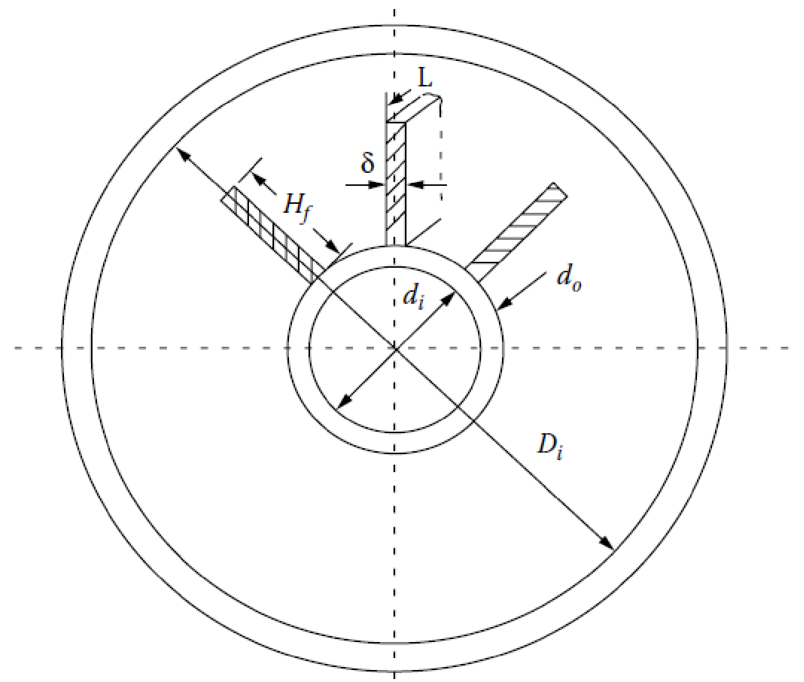
6-Finned outer surface of the inner pipe

Area of the un-finned part $$A_u = 2N_t(\pi d_o L - N_f L\delta)$$

Area of the fins $$A_f = 2N_t N_f L(2H_f + \delta)$$

Total area =Un-finned area + finned are

$$A_t = A_u + A_f = 2N_t L(\pi d_o + 2N_f H_f)$$



Surface overall efficiency

$$\eta_o = 1 - \left(\frac{A_f}{A_t}\right)(1 - \eta_f)$$

Efficiency for straight fin

$$\eta_f = \frac{\tanh(mH_f)}{mH_f} \qquad m = \sqrt{\frac{2h}{\delta k_f}}$$

Clean and fouled overall heat transfer coefficient
Finned double pipe

$$\frac{1}{U_{of}A_t} = \frac{1}{h_i A_i} + \frac{R_{fi}}{A_i} + R_w + \frac{R_{fo}}{A_t \eta_o} + \frac{1}{h_o A_t \eta_o}$$

$$\frac{1}{U_{of}} = \frac{1}{h_i (A_i/A_t)} + \frac{R_{fi}}{(A_i/A_t)} + A_t R_w + \frac{R_{fo}}{\eta_o} + \frac{1}{h_o \eta_o}$$

$$R_w = \frac{\ln(d_o/d_i)}{2\pi k L}$$

$$A_t = A_u + A_f = 2N_t L(\pi d_o + 2N_f H_f)$$

$$A_i = \pi d_i (2L)$$

$$\eta_o = \left[1 - \frac{A_f}{A_t}(1 - \eta_f)\right]$$

You may assume a length for the hairpin HX and then find the number hairpins required.

$$\eta_f = \frac{\tanh(mH_f)}{mH_f}$$

For clean surface

$$\frac{1}{U_c} = \frac{1}{h_i (A_i/A_t)} + A_t R_w + \frac{1}{h_o \eta_o}$$

Cleanliness factor CF

$$CF = \frac{U_{of}}{U_c}$$

$$m = \sqrt{\frac{2h}{\delta k_f}}$$

Percentage Over surface OS

$$OS = 100 U_c R_{ft}$$

where

$$R_{ft} = \frac{1 - CF}{U_c CF}$$

Ch. 3 correlations for convective heat transfer coefficients (**Laminar internal flows**)

TABLE 3.1

Laminar Forced Convection Correlations in Smooth Straight Circular Ducts

Number	Correlation	Limitations and Remarks	Ref.
1	$Nu_T = 1.61(Pe_b d/L)^{1/3}$	$Pe_b d/L > 10^3$, constant wall temperature	1, 2
	$Nu_T = 3.66$	$Pe_b d/L < 10^2$, fully developed flow in a circular duct, constant wall temperature	
2	$Nu_T = [(3.66)^3 + (1.61)^3 Pe_b d/L]^{1/3}$	Superposition of two asymptiocs given in case 1 for the mean Nusselt number, $0.1 < Pe_b d/L < 10^4$	3
3	$Nu_T = 3.66 + \dfrac{0.19(Pe_b d/L)^{0.8}}{1+0.177(Pe_b d/L)^{0.467}}$	Thermal entrance region, constant wall temperature, $0.1 < Pe_b d/L < 10^4$	4
4	$Nu_H = 1.953(Pe_b d/L)^{1/3}$	$Pe_b d/L > 10^2$, constant heat flux	1, 2
	$Nu_H = 4.36$	$Pe_b d/L > 10$, fully developed flow in a circular duct, constant heat flux	1, 2
5	$Nu_T = 0.664\dfrac{1}{(Pr)^{1/6}}\left(Pe_b \dfrac{d}{L}\right)^{1/2}$	$Pe_b d/L > 10^4$, $0.5 < Pr < 500$, simultaneously developing flow	6
6	$Nu_T = Nu + \phi\left(\dfrac{d_o}{D_i}\right)\dfrac{0.19(PeD_h/L)^{0.8}}{1+0.117(PeD_h/L)^{0.467}}$	Circular annular duct, constant wall temperature, thermal entrance region	8
	$\varphi(d_o/D_i) = 1 + 0.14\ (d_o/D_i)^{-1/2}$	Outer wall is insulated, heat transfer through the inner wall	
	$\varphi(d_o/D_i) = 1 + 0.14\ (d_o/D_i)^{0.1}$	Heat transfer through outer and inner wall	
7	$Nu_T = 1.86(Re_b Pr_b d/L)^{1/3}(\mu_b/\mu_w)^{0.14}$	Thermal entrance region, constant wall temperature, $0.48 < Pr_b < 16{,}700$, $4.4 \times 10^{-3} < (\mu_b/\mu_w) < 9.75$, $(Re_b Pr_b d/L)^{1/3}(\mu_b/\mu_w)^{0.14} > 2$	11
8	$Nu_H = 1.86(Re_b Pr_b d/L)^{1/3}(\mu_b/\mu_w)^{0.152}$	Thermal entrance region, constant wall heat flux, for oils $0.8 \times 10^3 < Re_b < 1.8 \times 10^3$, $1 < (T_w/T_b) < 3$	13
9	$Nu_H = 1.23(Re_b Pr_b d/L)^{0.4}(\mu_b/\mu_w)^{1/6}$	Thermal entrance region, constant heat flux, $400 < Re_b < 1{,}900$, $170 < Pr_b < 640$, for oils	14
10	$Nu_b = 1.4(Re_b Pr_b d/L)^{1/3}(\mu_b/\mu_w)^n$	Thermal entrance region, $n = 0.05$ for heating liquids, $n = 1/3$ for cooling liquids	15

Note: Unless otherwise stated, fluid properties are evaluated at the bulk mean fluid temperature, $T_b = (T_i + T_o)/2$.

Ch. 3 correlations for convective heat transfer coefficients (**Turbulent internal flows**)

TABLE 3.3

Correlations for Fully Developed Turbulent Forced Convection Through a Circular Duct with Constant Properties

Number	Correlation[a]	Remarks and Limitations	Ref.
1	$Nu_b = \dfrac{(f/2)Re_b Pr_b}{1+8.7(f/2)^{1/2}(Pr_b - 1)}$	Based on three-layer turbulent boundary layer model, $Pr > 0.5$	23, 24
2	$Nu_b = 0.021\, Re_b^{0.8} Pr_b^{0.4}$	Based on data for common gases; recommended for Prandtl numbers ≈ 0.7	25
3	$Nu_b = \dfrac{(f/2)Re_b Pr_b}{1.07+12.7(f/2)^{1/2}(Pr_b^{2/3} - 1)}$	Based on three-layer model with constants adjusted to match experimental data $0.5 < Pr_b < 2{,}000$, $10^4 < Re_b < 5 \times 10^6$	19
4	$Nu_b = \dfrac{(f/2)Re_b Pr_b}{1.07+9(f/2)^{1/2}(Pr_b - 1)Pr_b^{-1/4}}$	Theoretically based; Webb found case 3 better at high Pr and this one the same at other Pr	20
5	$Nu_b = 5 + 0.015\, Re_b^{m} Pr_b^{n}$ $m = 0.88 - 0.24/(4 + Pr_b)$	Based on numerical results obtained for $0.1 < Pr_b < 10^4$, $10^4 < Re_b < 10^6$	21
	$n = 1/3 + 0.5 \exp(-0.6\, Pr_b)$	Within 10% of case 6 for $Re_b > 10^4$	
	$Nu_b = 5 + 0.012\, Re_b^{0.87}(Pr_b + 0.29)$	Simplified correlation for gases, $0.6 < Pr_b < 0.9$	
6	$Nu_b = \dfrac{(f/2)(Re_b - 1000)Pr_b}{1+12.7(f/2)^{1/2}(Pr_b^{2/3} - 1)}$	Modification of case 3 to fit experimental data at low Re ($2{,}300 < Re_b < 10^4$)	22
	$f = (1.58 \ln Re_b - 3.28)^{-2}$	Valid for $2{,}300 < Re_b < 5 \times 10^6$ and $0.5 < Pr_b < 2{,}000$	
	$Nu_b = 0.0214\,(Re_b^{0.8} - 100)Pr_b^{0.4}$	Simplified correlation for $0.5 < Pr < 1.5$; agrees with case 4 within -6% and $+4\%$	
	$Nu_b = 0.012\,(Re_b^{0.87} - 280)Pr_b^{0.4}$	Simplified correlation for $1.5 < Pr < 500$; agrees with case 4 within -10% and $+0\%$ for $3 \times 10^3 < Re_b < 10^6$	
7	$Nu_b = 0.022\, Re_b^{0.8} Pr_b^{0.5}$	Modified Dittus–Boelter correlation for gases ($Pr \approx 0.5 - 1.0$); agrees with case 6 within 0 to 4% for $Re_b \geq 5{,}000$	23

[a] Properties are evaluated at bulk temperatures.

Incopera 7th ed. Correlations for convective heat transfer coefficient for internal flows

TABLE 8.4 Summary of convection correlations for flow in a circular tube[a,b,e]

Correlation		Conditions
$f = 64/Re_D$	(8.19)	Laminar, fully developed
$Nu_D = 4.36$	(8.53)	Laminar, fully developed, uniform q_s''
$Nu_D = 3.66$	(8.55)	Laminar, fully developed, uniform T_s
$\overline{Nu}_D = 3.66 + \dfrac{0.0668\, Gz_D}{1 + 0.04\, Gz_D^{2/3}}$	(8.57)	Laminar, thermal entry (or combined entry with $Pr \gtrsim 5$), uniform T_s, $Gz_D = (D/x)\, Re_D\, Pr$
$\overline{Nu}_D = \dfrac{\dfrac{3.66}{\tanh[2.264\, Gz_D^{-1/3} + 1.7\, Gz_D^{-2/3}]} + 0.0499\, Gz_D \tanh(Gz_D^{-1})}{\tanh(2.432\, Pr^{1/6}\, Gz_D^{-1/6})}$	(8.58)	Laminar, combined entry, $Pr \gtrsim 0.1$, uniform T_s, $Gz_D = (D/x)\, Re_D\, Pr$
$\dfrac{1}{\sqrt{f}} = -2.0 \log\left[\dfrac{e/D}{3.7} + \dfrac{2.51}{Re_D \sqrt{f}}\right]$	(8.20)[c]	Turbulent, fully developed
$f = (0.790 \ln Re_D - 1.64)^{-2}$	(8.21)[c]	Turbulent, fully developed, smooth walls, $3000 \lesssim Re_D \lesssim 5 \times 10^6$
$Nu_D = 0.023\, Re_D^{4/5}\, Pr^n$	(8.60)[d]	Turbulent, fully developed, $0.6 \lesssim Pr \lesssim 160$, $Re_D \gtrsim 10{,}000$, $(L/D) \gtrsim 10$, $n = 0.4$ for $T_s > T_m$ and $n = 0.3$ for $T_s < T_m$
$Nu_D = 0.027\, Re_D^{4/5}\, Pr^{1/3} \left(\dfrac{\mu}{\mu_s}\right)^{0.14}$	(8.61)[d]	Turbulent, fully developed, $0.7 \lesssim Pr \lesssim 16{,}700$, $Re_D \gtrsim 10{,}000$, $L/D \gtrsim 10$
$Nu_D = \dfrac{(f/8)(Re_D - 1000)\, Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}$	(8.62)[d]	Turbulent, fully developed, $0.5 \lesssim Pr \lesssim 2000$, $3000 \lesssim Re_D \lesssim 5 \times 10^6$, $(L/D) \gtrsim 10$
$Nu_D = 4.82 + 0.0185(Re_D\, Pr)^{0.827}$	(8.64)	Liquid metals, turbulent, fully developed, uniform q_s'', $3.6 \times 10^3 \lesssim Re_D \lesssim 9.05 \times 10^5$, $3 \times 10^{-3} \lesssim Pr \lesssim 5 \times 10^{-2}$, $10^2 \lesssim Re_D\, Pr \lesssim 10^4$
$Nu_D = 5.0 + 0.025(Re_D\, Pr)^{0.8}$	(8.65)	Liquid metals, turbulent, fully developed, uniform T_s, $Re_D\, Pr \gtrsim 100$

Incropera, internal turbulent flow

$\frac{1}{\sqrt{f}} = -2.0 \log\left[\frac{e/D}{3.7} + \frac{2.51}{Re_D\sqrt{f}}\right]$	(8.20)[c]	Turbulent, fully developed
$f = (0.790 \ln Re_D - 1.64)^{-2}$	(8.21)[c]	Turbulent, fully developed, smooth walls, $3000 \lesssim Re_D \lesssim 5 \times 10^6$
$Nu_D = 0.023\, Re_D^{4/5}\, Pr^n$	(8.60)[d]	Turbulent, fully developed, $0.6 \lesssim Pr \lesssim 160$, $Re_D \gtrsim 10{,}000$, $(L/D) \gtrsim 10$, $n = 0.4$ for $T_s > T_m$ and $n = 0.3$ for $T_s < T_m$
$Nu_D = 0.027\, Re_D^{4/5}\, Pr^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14}$	(8.61)[d]	Turbulent, fully developed, $0.7 \lesssim Pr \lesssim 16{,}700$, $Re_D \gtrsim 10{,}000$, $L/D \gtrsim 10$
$Nu_D = \frac{(f/8)(Re_D - 1000)\, Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}$	(8.62)[d]	Turbulent, fully developed, $0.5 \lesssim Pr \lesssim 2000$, $3000 \lesssim Re_D \lesssim 5 \times 10^6$, $(L/D) \gtrsim 10$
$Nu_D = 4.82 + 0.0185(Re_D\, Pr)^{0.827}$	(8.64)	Liquid metals, turbulent, fully developed, uniform q_s'', $3.6 \times 10^3 \lesssim Re_D \lesssim 9.05 \times 10^5$, $3 \times 10^{-3} \lesssim Pr \lesssim 5 \times 10^{-2}$, $10^2 \lesssim Re_D\, Pr \lesssim 10^4$

Incropera-Laminar flow

TABLE 8.4 Summary of convection correlations for flow in a circular tube[a,b,e]

Correlation		Conditions
$f = 64/Re_D$	(8.19)	Laminar, fully developed
$Nu_D = 4.36$	(8.53)	Laminar, fully developed, uniform q_s''
$Nu_D = 3.66$	(8.55)	Laminar, fully developed, uniform T_s
$\overline{Nu}_D = 3.66 + \dfrac{0.0668\, Gz_D}{1 + 0.04\, Gz_D^{2/3}}$	(8.57)	Laminar, thermal entry (or combined entry with $Pr \gtrsim 5$), uniform T_s, $Gz_D = (D/x)\, Re_D\, Pr$
$\overline{Nu}_D = \dfrac{\dfrac{3.66}{\tanh[2.264\, Gz_D^{-1/3} + 1.7\, Gz_D^{-2/3}]} + 0.0499\, Gz_D \tanh(Gz_D^{-1})}{\tanh(2.432\, Pr^{1/6}\, Gz_D^{-1/6})}$	(8.58)	Laminar, combined entry, $Pr \gtrsim 0.1$, uniform T_s, $Gz_D = (D/x)\, Re_D\, Pr$

$$Gz = Graetz\ number = \left(\frac{D}{x}\right) Re\ Pr$$

Internal laminar flow-Kakac used correlation

A simple empirical correlation has been proposed by Seider and Tate[11] to predict the mean Nusselt number for laminar flow in a circular duct for the combined entry length with constant wall temperature as

$$Nu_T = 1.86\left(\frac{Pe_b d_i}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \tag{3.24}$$

which is valid for smooth tubes for $0.48 < Pr_b < 16{,}700$ and $0.0044 < (\mu_b/\mu_w) < 9.75$. This correlation has been recommended by Whitaker[12] for values of

$Peclet\ Number\ Pe = Re\ Pr$

$$\left(\frac{Pe_b d_i}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \geq 2 \tag{3.25}$$

Below this limit, fully developed conditions will be established and Equation 3.7 may be used for a good approximation. All physical properties are evaluated at the fluid bulk mean temperature except μ_w, which is evaluated at the wall temperature.

Example on Finned double pipe heat exchangers

Example 7.2

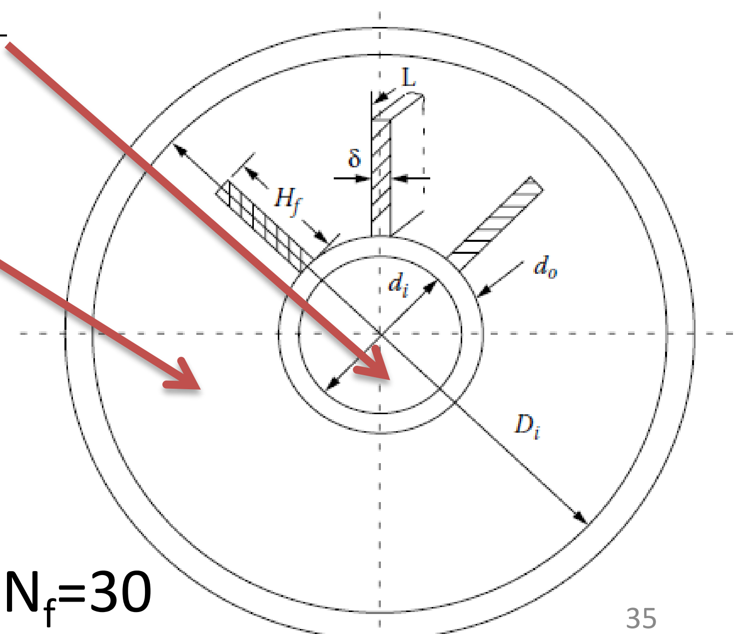
The objective of this example is to design an oil cooler with sea water using a finned tube double-pipe heat exchanger.

Engine oil at a rate of 3 kg/s will be cooled from 65°C to 55°C by sea water at 20°C. The sea water outlet temperature is 30°C, and it flows through the inner tube. The properties are given as

Fluid	Annulus Fluid, Oil	Tube-Side Fluid, Sea Water
Density, kg/m²	885.27	1013.4
Specific heat, kJ/kg · K	1.902	4.004
Viscosity, kg/m · s	0.075	9.64×10^{-4}
Thermal conductivity, W/m · K	0.1442	0.639
Prandtl number, Pr	1,050	6.29

T_{ci}=20 ° C
T_{co}=30 ° C

m_dot=3 kg/s
T_{hi}=65 °C
T_{ho}=55 ° C



Number of tubes N_t=1

Number of fins per tube =N_f=30

Example 7.2 continue

The following design data are selected:

Length of hairpin = 4.5 m
Annulus nominal diameter = 2 in.
Nominal diameter of the inner tube = 3/4 in.
Fin height: $H_f = 0.0127$ m
Fin thickness: $\delta = 0.9$ mm
Number of fins per tube = 30
Material throughout = carbon steel ($k = 52$ W/m · K)
Number of tubes inside the annulus, $N_t = 1$.

See table 9.2 for detailed information about the tubes

Proper fouling factors are to be selected, and the surface area of the heat exchanger as well as the number of hairpins including pressure drops and pumping powers for both streams are to be calculated.

Example 7.2 continue

Procedure:

1-Since all 4 temperatures are known. One can get the main fluid properties at the mean value. The main properties are:

C_p, ρ, k, μ, and Pr

2-Since m_h is known, calculate the heat load q

3-From 2 you can calculate the mass flow rate m_c (in the inner tube)

4-Calculate the cross sectional area for the flow inside the pipe and in the annulus, and the two velocities

5-Calculate the hydraulic diameter for pressure and heat transfer calculation (i.e. D_h, and D_e)

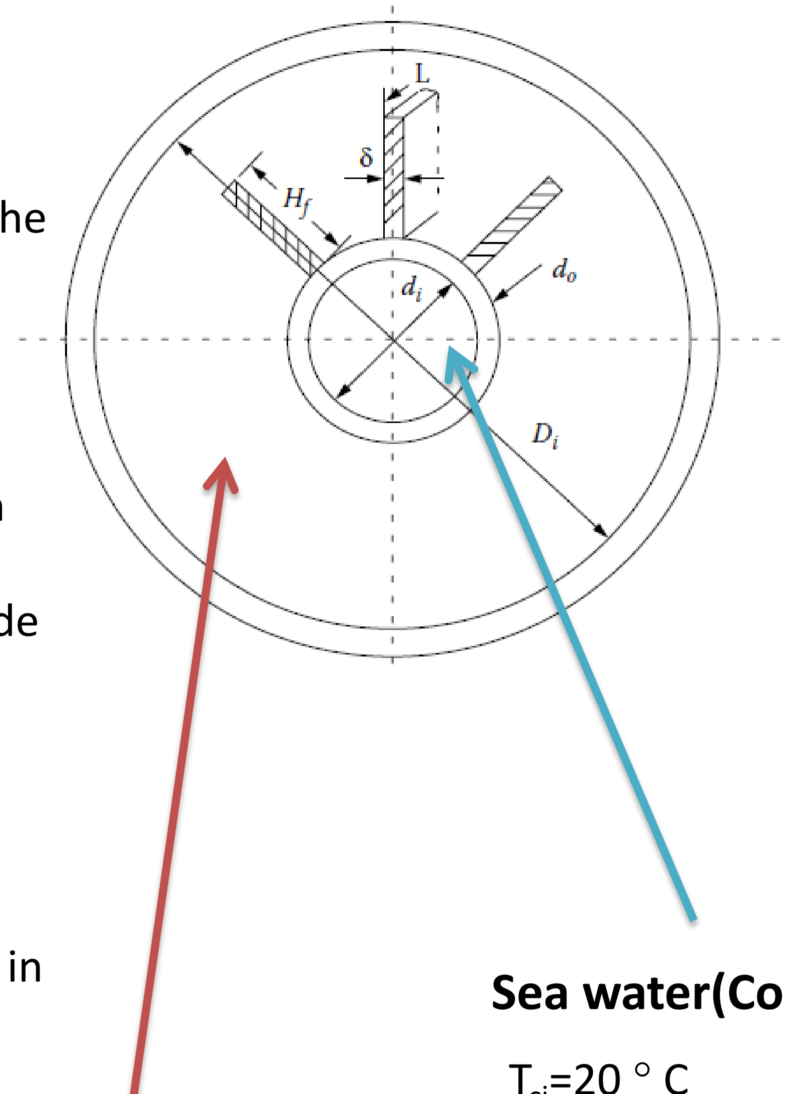
6-Caculate f & Re for both flows and Nu

7-Caculate h_i and h_o

8-Estimate R_{fi} and R_{fo} based on fluid type and tables in Ch. 6 (Fouling)

9-Calculate U_{of} and U_c

10-Find CF and OS



Sea water(Cold)

T_{ci}=20 ° C

T_{co}=30 ° C

Oil (hot)

m_dot=3 kg/s

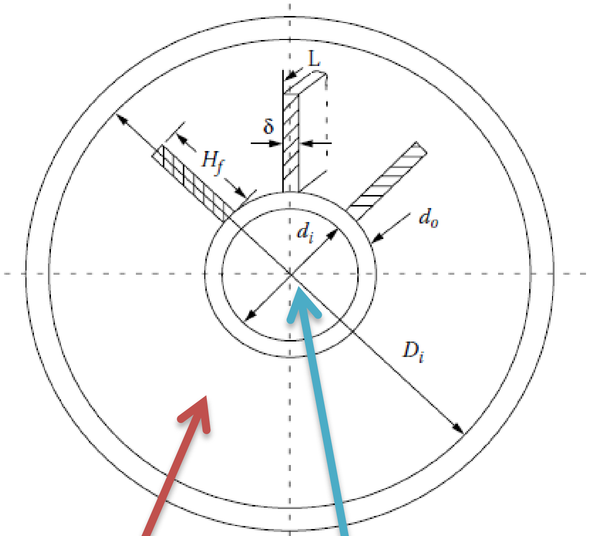
T_{hi}=65 ° C

T_{ho}=55 ° C

Example 7.2 continue

$$\dot{m}_c = \frac{(\dot{m}c_p)_h \Delta T_h}{c_{pc}\Delta T_c}$$

$$= \frac{3\times 1.902\times(10)}{4.004\times(10)} = 1.425\ \text{kg/}$$



Annulus—The net cross-sectional area in the annulus with longitudinal finned tubes is given by Equation 7.10:

$$A_c = \frac{\pi}{4}(D_i^2 - d_o^2 N_t) - (\delta H_f N_f N_t)$$

$$= \frac{\pi}{4}\left[(0.0525)^2 - (0.0266)^2\right] - (0.9\times 10^{-3})(0.0127)(30)(1)$$

$$= 1.263\times 10^{-3}\ \text{m}^2$$

Inner Tube, Sea Water

$$u_m = \frac{\dot{m}_c}{\rho\pi\frac{d_i^2}{4}} = \frac{4\times 1.425}{1013.4\times\pi(0.0209)^2} = 4.1\ \text{m/s}$$

Sea water(Cold)

T_{ci}=20 ° C
T_{co}=30 ° C

Oil (hot)

m_dot=3 kg/s
T_{hi}=65 ° C
T_{ho}=55 ° C

Example 7.2 continue

$$P_w = \pi\left(D_i + d_o N_t\right) + 2H_f N_f N_t$$

$$= \pi\left(0.0525 + 0.0266\right) + 2\left(0.0127\right)\left(30\right)$$

$$= 1.011 \text{ m}$$

$$D_h = \frac{4A_c}{P_w} = \frac{4\left(1.263 \times 10^{-3}\right)}{1.011} = 5.0 \times 10^{-3} \text{ m}$$

The wetted perimeter for heat transfer can be calculated using Equation 7.9:

$$P_h = \pi d_o N_t + 2N_f H_f N_t$$
$$= \pi\left(0.0266\right)\left(1\right) + 2\left(30\right)\left(0.0127\right)\left(1\right) = 0.845 \text{ m}$$

The wetted perimeter for heat transfer can be calculated using Equation 7.9:

$$P_h = \pi d_o N_t + 2N_f H_f N_t$$
$$= \pi\left(0.0266\right)\left(1\right) + 2\left(30\right)\left(0.0127\right)\left(1\right) = 0.845 \text{ m}$$

and the equivalent diameter for heat transfer can be determined as follows:

$$D_e = \frac{4A_c}{P_h} = \frac{4\left(1.263 \times 10^{-3}\right)}{0.845} = 5.98 \times 10^{-3} \text{ m}$$

Example 7.2 continue

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{1013.4 \times 4.1 \times 0.02093}{9.64 \times 10^{-4}} = 90,082$$

Hence, the flow is turbulent. One of the turbulent forced convection correlations given in Chapter 3 can be selected.
The Petukhov–Kirillov correlation is used here:

$$f = (1.58 \ln Re - 3.28)^{-2} = (1.58 \ln 90,082 - 3.28)^{-2} = 0.0046$$

$$Nu_b = \frac{(f/2)(Re_b)\mathrm{Pr}_b}{1.07 + 12.7(f/2)^{1/2}(Pr_b^{2/3} - 1)}$$

$$= \frac{(0.0023)(90,082)(6.29)}{1.07 + 12.7(0.0023)^{1/2}(6.29^{2/3} - 1)} = 513.8$$

$$h_i = \frac{Nu \cdot k}{d_i} = \frac{(513.8 \times 0.639)}{0.02093} = 15,685.9\ \mathrm{W/m^2 \cdot K}$$

Example 7.2 continue

$$u_m = \frac{\dot{m}_h}{\rho A_c} = \frac{3}{885.27 \times 1.263 \times 10^{-3}} = 2.68 \text{ m/s}$$

$$\text{Re} = \frac{\rho u_m D_h}{\mu} = \frac{885.27 \times 2.68 \times 5 \times 10^{-3}}{0.075} = 158.17$$

Laminar

It is a laminar flow.

The Sieder–Tate correlation (Equation 3.24) can be used to calculate the heat transfer coefficient in the annulus:

$$Nu_b = 1.86\left(Re_b\, Pr_b \frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

$$Re_b\, Pr_b \frac{D_h}{L} = (158.17)(1050)\frac{5 \times 10^{-3}}{4.5} = 184.5$$

$$T_w \approx \frac{1}{2}\left(\frac{65+55}{2} + \frac{20+30}{2}\right) = 42.5^\circ \text{ C}$$

$$\mu_w = 0.197 \text{ Pa}\cdot\text{s}$$

$$\left(Re_b Pr_b \frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} = (184.5)^{1/3}\left(\frac{0.075}{0.197}\right)^{0.14} = 4.97 > 2$$

Therefore, the Sieder–Tate correlation is applicable.

$$Nu_b = 1.86(184.5)^{1/3}\left(\frac{0.075}{0.197}\right)^{0.14} = 9.25$$

$$h_o = \frac{Nu \cdot k}{D_e} = \frac{9.15 \times 0.1442}{5.98 \times 10^{-3}} = 223 \text{ W/m}^2 \cdot \text{K}$$

Example 7.2 continue

Note that Equation 3.25 is satisfied using Equation 3.24. Finned and unfinned heat transfer areas are given by Equations 7.11 and 7.12.

$$A_f = 2N_t N_f L(2H_f + \delta) = (2)(1)(30)(4.5)(2\times 0.0127 + 0.9\times 10^{-3})$$
$$A_f = 7.101 \text{ m}^2$$
$$A_u = 2N_t(\pi d_o L - N_f L\delta) = (2)(1)[\pi(0.0266)(4.5) - (30)(4.5)(0.9\times 10^{-3})]$$
$$A_u = 0.509 \text{ m}^2$$

The total area of a hairpin is

$$A_t = A_u + A_f = 7.101 + 0.509 = 7.61 \text{ m}^2$$

$$\eta_f = \frac{\tanh(mH_f)}{mH_f}$$

$$m = \sqrt{\frac{2h_o}{\delta k_f}} = \sqrt{\frac{2(223.05)}{(0.9\times 10^{-3})(52)}} = 97.63$$

$$\eta_f = \frac{\tanh(97.63\times 0.0127)}{97.63\times 0.0127} = 0.682$$

The overall surface efficiency, from Equation 2.14, is

$$\eta_o = \left[1-(1-\eta_f)\frac{A_f}{A_t}\right] = \left[1-(1-0.682)\frac{7.101}{7.610}\right] = 0.703$$

Example 7.2 continue

Fouling factors and overall heat transfer coefficients

Should be Table 6.5 & 6.11

From Tables 5.5 and 5.11, the fouling resistances are

$$R_{fo} = 0.176\times10^{-3}\ \mathrm{m^2\cdot K/W}\quad(\text{engine oil})$$
$$R_{ft} = 0.088\times10^{-3}\ \mathrm{m^2\cdot K/W}\quad(\text{sea water})$$

$$U_{of} = \left[\frac{7.610}{(0.592)(15685.9)} + \frac{7.610}{0.592}(0.088\times10^{-3}) + \frac{7.610\ln\left(\frac{0.0266}{0.0209}\right)}{2(52)\times\pi\times(2\times4.5)}\right.$$
$$\left. + \frac{0.176\times10^{-3}}{0.703} + \frac{1}{(0.703)(223)}\right]^{-1}$$
$$= 108.6\ \mathrm{W/m^2\cdot K}$$

Fouling factors (Kakac Ch. 6)

TABLE 6.5

TEMA Design Fouling Resistances for Industrial Fluids

Industrial Fluids	R_f ($m^2 \cdot K/W$)
Oils	
Fuel oil no. 2	0.000352
Fuel oil no. 6	0.000881
Transformer oil	0.000176
Engine lube oil	0.000176
Quench oil	0.000705
Gases and Vapors	
Manufactured gas	0.001761
Engine exhaust gas	0.001761
Steam (nonoil bearing)	0.000088
Exhaust steam (oil bearing)	0.000264–0.000352
Refrigerant vapors (oil bearing)	0.000352
Compressed air	0.000176
Ammonia vapor	0.000176
CO_2 vapor	0.000176
Chlorine vapor	0.000352
Coal flue gas	0.001761
Natural gas flue gas	0.000881
Liquids	
Molten heat transfer salts	0.000088
Refrigerant liquids	0.000176
Hydraulic fluid	0.000176
Industrial organic heat transfer media	0.000352
Ammonia liquid	0.000176
Ammonia liquid (oil bearing)	0.000528
Calcium chloride solutions	0.000528
Sodium chloride solutions	0.000528
CO_2 liquid	0.000176
Chlorine liquid	0.000352
Methanol solutions	0.000352
Ethanol solutions	0.000352
Ethylene glycol solutions	0.000352

Source: From, *Standards of the Tubular Exchanger Manufacturers Association*, 1988. With permission. ©1988 Tubular Exchanger Manufacturers Association.

TABLE 6.5

TEMA Design Fouling Resistances for Industrial Fluids

Industrial Fluids	R_f ($m^2 \cdot K/W$)
Oils	
Fuel oil no. 2	0.000352
Fuel oil no. 6	0.000881
Transformer oil	0.000176
Engine lube oil	0.000176
Quench oil	0.000705
Gases and Vapors	
Manufactured gas	0.001761
Engine exhaust gas	0.001761
Steam (nonoil bearing)	0.000088
Exhaust steam (oil bearing)	0.000264–0.000352
Refrigerant vapors (oil bearing)	0.000352
Compressed air	0.000176

Fouling factors (Kakac Ch. 6)

TABLE 6.11

Fouling Resistances for Water

Temperature of Heating Medium **Temperature of Water Water Velocity (m/s)**	**Up to 115°C 50°C**		**R_f (m² · K/W) 115°C to 205°C Over 50°C**	
350.000176	**0.9 and Less**	**Over 0.9**	**0.9 and Less**	**Over 0.9**
	0.000528	0.000352	Cooling tower and artificial spray pond	Treated make up 0.0.52
Cooling tower and artificial spray pond				
Treated make up	0.000176	0.000176	0.000352	0.000352
Untreated	0.000528	0.000528	0.000881	0.000705
City or well water	0.000176	0.000176	0.000352	0.000352
River water				
Minimum	0.000352	0.000176	0.000528	0.000352
Average	0.000528	0.000352	0.000705	0.000528
Muddy or silty	0.000528	0.000352	0.000705	0.000528
Hard (over 15 grains/gal)	0.000528	0.000528	0.000881	0.000881
Engine jacket	0.000176	0.000176	0.000176	0.000176
Distilled or closed cycle				
Condensate	0.000088	0.000088	0.000088	0.000088
Treated boiler feedwater	0.000176	0.000088	0.000176	0.000176
Boiler blowdown	0.000352	0.000352	0.000352	0.000352

Source: From *Standards of the Tubular Exchanger Manufacturers Association*, 1988. With permission.

Sea water and brackish water are **missing** from this table

Corrections for Table 6.11

TABLE 6.11

Fouling Resistances for Water

Temperature of Heating Medium Temperature of Water Water Velocity (m/s)	Up to 115°C 50°C		R_f (m² · K/W) 115°C to 205°C Over 50°C	
350.000176	0.9 and Less	Over 0.9	0.9 and Less	Over 0.9
Sea water	0.000088	0.000088	0.00018	0.00018
Brackish water	0.00035	0.00018	0.00053	0.00035
Cooling tower and artificial spray pond				
Treated make up	0.000176	0.000176	0.000352	0.000352
Untreated	0.000528	0.000528	0.000881	0.000705
City or well water	0.000176	0.000176	0.000352	0.000352
River water				
Minimum	0.000352	0.000176	0.000528	0.000352
Average	0.000528	0.000352	0.000705	0.000528
Muddy or silty	0.000528	0.000352	0.000705	0.000528
Hard (over 15 grains/gal)	0.000528	0.000528	0.000881	0.000881
Engine jacket	0.000176	0.000176	0.000176	0.000176
Distilled or closed cycle				
Condensate	0.000088	0.000088	0.000088	0.000088
Treated boiler feedwater	0.000176	0.000088	0.000176	0.000176
Boiler blowdown	0.000352	0.000352	0.000352	0.000352

Source: From *Standards of the Tubular Exchanger Manufacturers Association*, 1988. With permis-

From Fraas book
Heat exchanger design

To convert the fouling factor form $ft^2.hr.\ F/Btu$ to m^2K/W multiply by 5.678

TABLE H5.4 Normal Fouling Factors[a] for Heat Transfer Equipment[b]

Temperature of Heating Medium:	Up to 240°F		240 to 400°F[c]	
Temperature of Water:	125°F or Less		Over 125°F	
	Water Velocity, ft/s		Water Velocity, ft/s	
Types of Water	3 ft/s and Less	Over 3 ft/s	3 ft/s and Less	Over 3 ft/s
Seawater	0.0005	0.0005	0.001	0.001
Distilled	0.0005	0.0005	0.0005	0.0005
Treated boiler feedwater	0.001	0.0005	0.001	0.001
Engine jacket	0.001	0.001	0.001	0.001
City or well water (such as Great Lakes)	0.001	0.001	0.002	0.002
Great Lakes	0.001	0.001	0.002	0.002
Cooling tower and artificial spray pond:				
Treated make-up	0.001	0.001	0.002	0.002
Untreated	0.003	0.003	0.005	0.004
Boiler blowdown	0.002	0.002	0.002	0.002
Brackish water	0.002	0.001	0.003	0.002
River water:				
Minimum	0.002	0.001	0.003	0.002
Mississippi	0.003	0.002	0.004	0.003
Delaware, Schuylkill	0.003	0.002	0.004	0.003
East River and New York Bay	0.003	0.002	0.004	0.003
Chicago sanitary canal	0.008	0.006	0.010	0.008
Muddy or silty	0.003	0.002	0.004	0.003
Hard (over 15 grains/gal)	0.003	0.003	0.005	0.005
Fouling factors—industrial oils:				
Clean recirculating oil				0.001
Machinery and transformer oils				0.001
Vegetable oils				0.003
Quenching oil				0.004
Fuel oil				0.005
Fouling factors—industrial gases and vapors:				
Organic vapors				0.0005
Steam (non-oil bearing)				0.0005
Alcohol vapors				0.0005
Steam, exhaust (oil-bearing from reciprocating engines)				0.001
Refrigerating vapors (condensing from reciprocating compressors)				0.002
Air				0.002
Coke oven gas and other manufactured gas				0.01
Diesel engine exhaust gas				0.01
Fouling factors—industrial liquids:				
Organic				0.001
Refrigerating liquids (heating, cooling or evaporating)				0.001
Brine (cooling)				0.001

[a]Fouling factor $= \frac{1}{\mu}, \frac{h \cdot ft^2 \cdot °F}{Btu}$.

To convert to $\frac{m^2 \cdot K}{W}$: divide by 5.678

[b]Reprinted by permission of Tubular Exchanger Manufacturers Association.
[c]Ratings in columns 3 and 4 are based on a temperature of the heating medium of 240 to 400°F. If the heating medium temperature is over 400°F and the cooling medium is known to scale, these ratings should be modified accordingly.

Example 7.2 continue

$$U_{oc} = \frac{1}{\dfrac{A_t}{A_i h_i} + \dfrac{A_t \ln(d_o/d_i)}{2\pi k \cdot 2L} + \dfrac{1}{\eta_o h_o}}$$

$$= \left[\frac{7.610}{(0.592)(15685.9)} + \frac{7.610 \ln\left(\dfrac{0.0266}{0.0209}\right)}{2(52) \times \pi(2 \times 4.5)} + \frac{1}{(0.703)(223)} \right]^{-1}$$

$$= 127.6 \text{ W/m}^2 \cdot \text{K}$$

The cleanliness factor, *CF*, is determined as follows:

$$CF = \frac{U_{of}}{U_{oc}} = \frac{108.6}{127.6} = 0.85$$

Example 7.2 continue

The total heat transfer surface area is

$$A_o = \frac{Q}{U_o \Delta T_m}$$

$$Q = (\dot{m}c_p)_h (T_{h_1} - T_{h_2}) = (3)(1.902 \times 10^3)(65 - 55) = 57{,}060 \text{ W}$$

$$\Delta T_m = \Delta T_1 = \Delta T_2 = 35^\circ \text{ C}$$

Without fouling,

$$A_{oc} = \frac{Q}{U_{oc} \Delta T_m} = \frac{57{,}060}{(127.6)(35)} = 12.78 \text{ m}^2$$

With fouling,

$$A_{of} = \frac{Q}{U_{of} \Delta T_m} = \frac{57060}{(108.6)(35)} = 15.01 \text{ m}^2$$

The hairpin heat transfer surface area is

$$A_{hp} = A_t = 7.61 \text{ m}^2$$

and the number of hairpins can be determined:

$$N_{hp} = \frac{A_{of}}{A_{hp}} = \frac{15.01}{7.61} = 1.97$$

Example 7.2 continue

Pressure Drops and Pumping Powers—For the inner tube,

$$\Delta p_t = 4f\frac{2L}{d_i}\rho\frac{u_m^2}{2}N_{hp}$$

$$= 4(0.0046)\frac{4.5\times 2}{0.0209}(1013.4)\frac{(4.1)^2}{2}(2) = 135\text{ kPa}$$

Pumping Power

$$W_a = \frac{\Delta p_a \dot{m}_a}{\eta_a \rho_a} = \frac{(135\times 10^3)(1.425)}{(0.80)(1013.4)} = 237.3\text{ W}$$

In the annulus,

$$f_{cp} = \frac{16}{Re} = \frac{16}{158.17} = 0.1011$$

Using Equation 3.21b with Table 3.2, we have

$$f = f_{cp}\left(\frac{\mu_b}{\mu_w}\right)^{-0.5} = 0.1011\left(\frac{0.075}{0.197}\right)^{-0.5} = 0.164$$

$$\Delta p_a = 4f\frac{2L}{D_h}\rho\frac{u_m^2}{2}N_{hp}$$

$$= 4(0.164)\frac{4.5\times 2}{(5\times 10^{-3})}(885.27)\frac{(2.68)^2}{2}(2) = 7.5\text{ MPa}$$

$$w_a = \frac{\Delta p_a \dot{m}_a}{\eta_a \rho_a} = \frac{(7.5\times 10^6)(3)}{(0.80)(885.27)} = 31.8\text{ kW}$$

TABLE 9.2

Heat Exchanger and Condenser Tube Data

Nominal Pipe Size (in.)	Outside Diameter (in.)	Schedule Number or Weight	Wall Thickness (in.)	Inside Diameter (in.)	Surface Area		Cross-Sectional Area	
					Outside ($ft.^2/ft.$)	Inside ($ft.^2/ft.$)	Metal Area ($in.^2$)	Flow Area ($in.^2$)
		40	0.113	0.824	0.275	0.216	0.333	0.533
3/4	1.05	80	0.154	0.742	0.275	0.194	0.434	0.432
		40	0.133	1.049	0.344	0.275	0.494	0.864
1	1.315	80	0.179	0.957	0.344	0.250	0.639	0.719
		40	0.140	1.38	0.434	0.361	0.668	1.496
1-1/4	1.660	80	0.191	1.278	0.434	0.334	0.881	1.283
		40	0.145	1.61	0.497	0.421	0.799	2.036
1-1/2	1.900	80	0.200	1.50	0.497	0.393	1.068	1.767
		40	0.154	2.067	0.622	0.541	1.074	3.356
2	2.375	80	0.218	1.939	0.622	0.508	1.477	2.953
		40	0.203	2.469	0.753	0.646	1.704	4.79
2-1/2	2.875	80	0.276	2.323	0.753	0.608	2.254	4.24
		40	0.216	3.068	0.916	0.803	2.228	7.30
3	3.5	80	0.300	2.900	0.916	0.759	3.106	6.60
		40	0.226	3.548	1.047	0.929	2.680	9.89
3-1/2	4.0	80	0.318	3.364	1.047	0.881	3.678	8.89

8-Series-parallel arrangement of double pipe heat exchangers

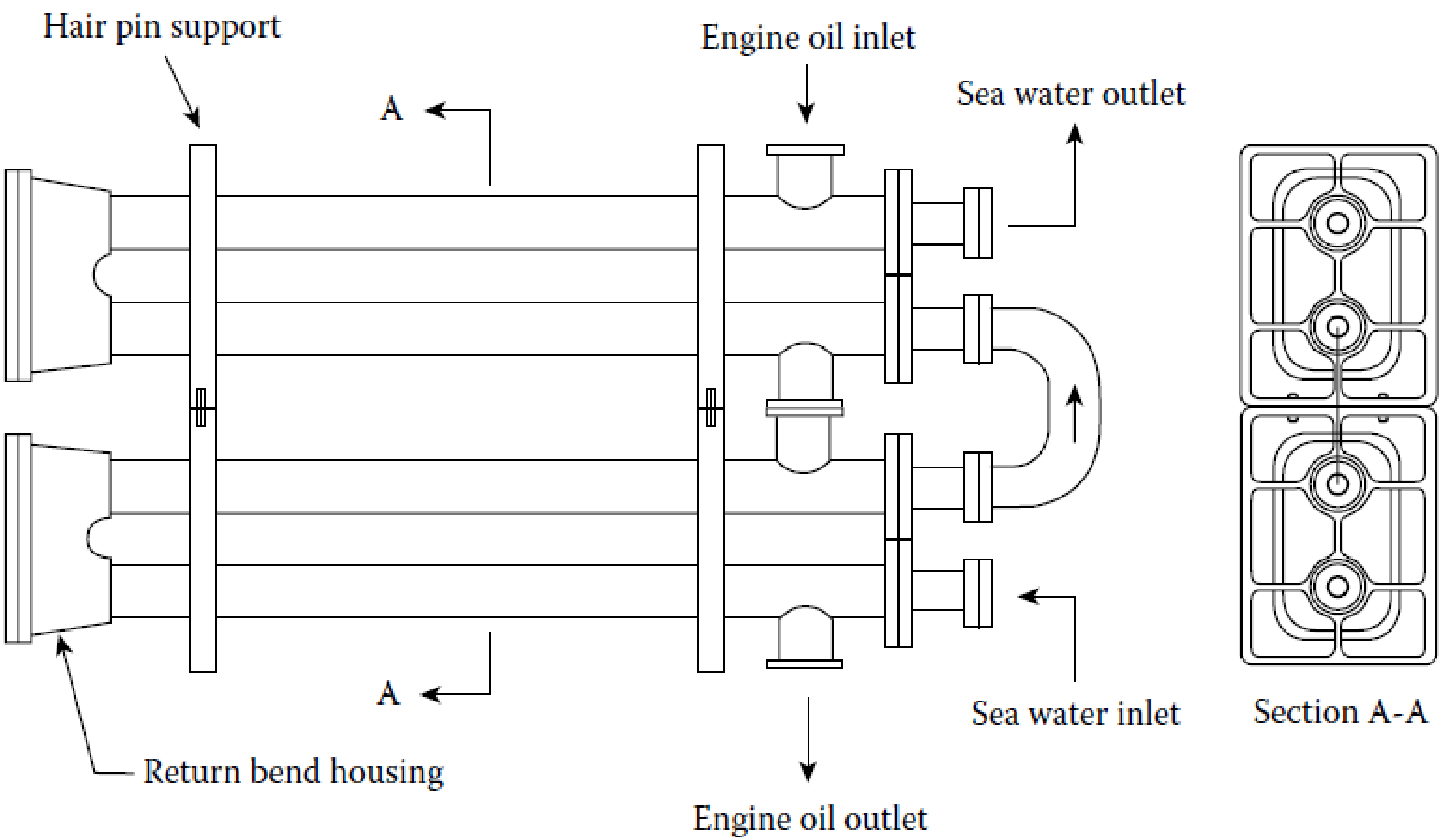


FIGURE 7.4
Two hairpin sections arranged in series.

8-Series-parallel arrangement of double pipe heat exchangers

Hairpin can be connected in series one after the other to get pure counter or pure parallel arrangement

But this may cause high pressure drop and excessive pumping power, therefore hairpin can be connected in parallel /series combinations.

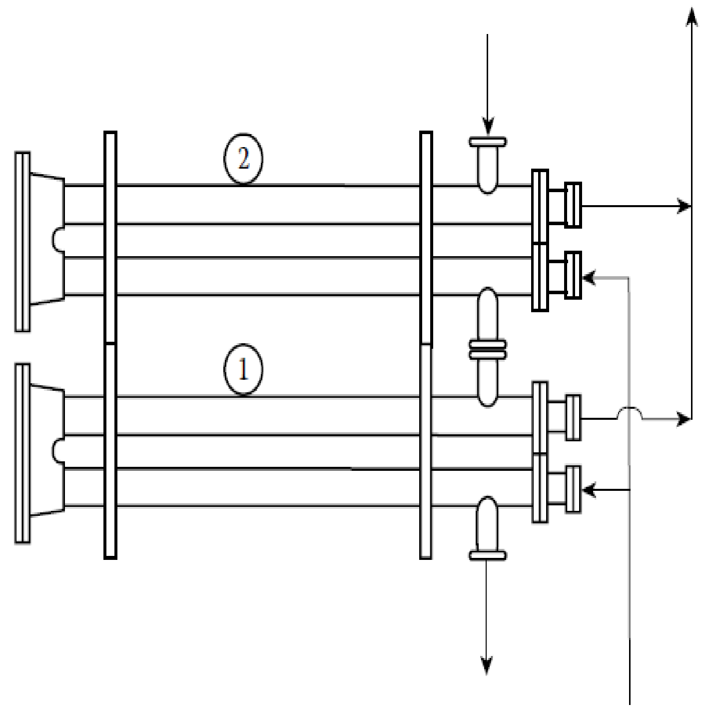
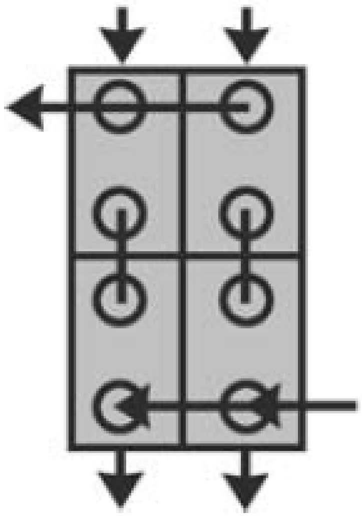


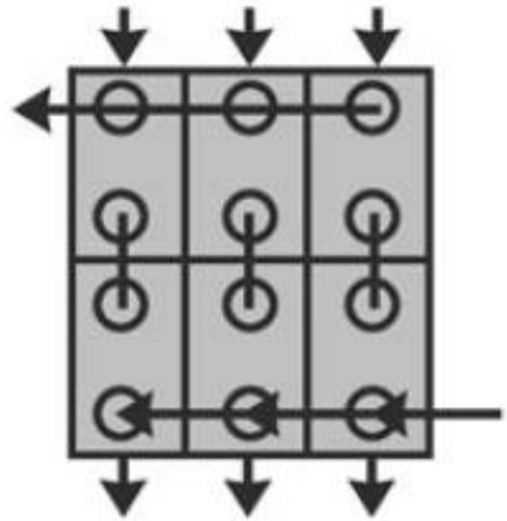
FIGURE 7.6
Two double-pipe units in series on the annulus (shell) side and parallel on the tube side.

A correction factor for the LMTD must then apply

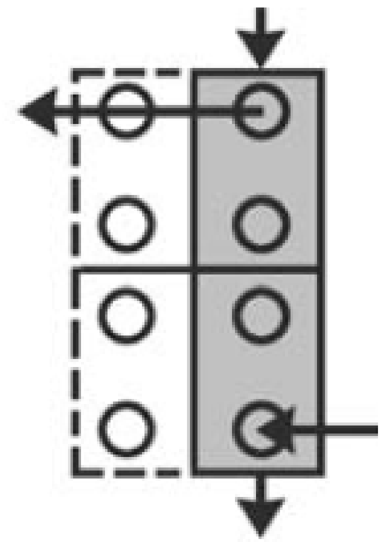
8-Series-parallel arrangement of double pipe heat exchangers



Original installation — four doube pipe sections
2 parallel × 2 series



To increase capacity 50% — merely add one bank
3 parallel × 2 series



To decrease capacity — shut off one bank
1 parallel × 2 series

Parallel series combinations

8-Series-parallel arrangement of double pipe heat exchangers

The LMTD for double pipe HX with combination of series and parallel arrangement must be modified as follows:

$$Q == UA\,\Delta T_m = UAS(T_{hi} - T_{ci})$$

Where S is given by:

For one series hot fluid and n_1 parallel cold stream

$$S = \frac{1 - P_1}{\left(\frac{n_1 R_1}{R_1 - 1}\right) ln\left[\left(\frac{R_1 - 1}{R_1}\right)\left(\frac{1}{P_1}\right)^{1/n_1} + \left(\frac{1}{R_1}\right)\right]}$$

$$P_1 = \frac{T_{h2} - T_{c1}}{T_{h1} - T_{c1}} \qquad R_1 = \frac{T_{h1} - T_{h2}}{n_1(T_{c2} - T_{c1})}$$

For one-series cold stream and n_2 parallel hot streams

$$S = \frac{1 - P_2}{\left(\frac{n_2}{1 - R_2}\right) ln\left[(1 - R_2)\left(\frac{1}{P_2}\right)^{1/n_2} + (R_2)\right]}$$

$$P_2 = \frac{T_{h1} - T_{c2}}{T_{h1} - T_{c1}} \qquad R_2 = \frac{n_2(T_{h1} - T_{h2})}{T_{c2} - T_{c1}}$$

9-Total pressure drop

All pressure losses must be taken into account for calculating the total pressure losses in heat exchanger. These sources of pressure drop must be considered

1-Friction or major losses due to wall friction
2-Minor losses such as pressure drop in inlet and outlet nozzles
3-Pressure drop due elevation difference between inlet and outlet
4-Pressure drop due momentum change

1 $$\Delta P_f = f \frac{2L}{D} \rho \frac{u_m^2}{2} N_{hp}$$

2 $$\Delta P_n = K_c \rho \frac{u_m^2}{2}$$

3 $$\Delta P = \rho g \, \Delta H$$

4 $$\Delta P_m = G^2 \left(\frac{1}{\rho_{out}} - \frac{1}{\rho_{in}}\right)$$