PROBLEM 1:
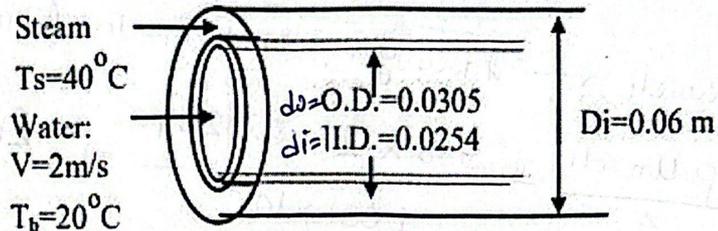
A double pipe heat exchanger is used to condense steam at 40°C saturation temperature. Water at average bulk

Steam →
$T_s=40°C$
Water:
$V=2m/s$
$T_b=20°C$
$d_o=O.D.=0.0305$
$d_i=I.D.=0.0254$
$D_i=0.06$ m

temperature of 20°C flows at 2 m/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/m²K, and the thermal resistance of a surface scale on the outer surface of the copper pipe is 0.000176 m²K/W.

a. Determine the overall heat transfer coefficient between the steam and the water based on the outer area of the copper pipe. (Thermal Conductivity of copper at 40°C: $K_c=383$ W/mK)

b. Evaluate the temperature at the inner surface of the pipe.

~~c. Estimate the length required to condense 0.5 kg/s of steam.~~

⚘ Properties of water at 20°C:

$\rho = 998 \ kg/m^3$ $k = 0.599 \ W/m \cdot K$

$\mu = 1.007 \times 10^{-3} \ kg/m \cdot s$ $Pr = 7.05$

$\nu = 1.009 \times 10^{-6} \ m^2/s$

⚘ Properties of steam 40°C:

$h_{fg} = 2573.45 - 167.5 = 2406 \ kJ/kg$

$h_o = 6000 \ w/m^2K \ ; \quad R_{fo} = 1.76 \times 10^{-4} \ m^2k/w$

Assumption ⟹ $R_{fi} = 0$

a-) Overall Heat Transfer Coefficient ៖ (based on outer surface)

$$U_o \cdot A_o = \cfrac{1}{\cfrac{1}{h_i A_i} + \cfrac{\ln(r_o/r_i)}{2\pi \cdot K_c \cdot L} + \cfrac{Rf_o}{A_o} + \cfrac{1}{h_o A_o}}$$

$$\boxed{h_i = \frac{Nu \cdot k}{d_i}}$$

?

⇒ Nusselt ← Flow Type ← Reynolds Number.

$$Re = \frac{\rho \cdot U_m \cdot d_i}{\mu} = \frac{998 \times 2 \times 0.0254}{1.007 \times 10^{-3}} \Rightarrow Re = 50346 > 2300$$

↓
Turbulent Flow

from Table 3.3:

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

$$f = (1.58 \cdot \ln(Re_b) - 3.28)^{-2} \Rightarrow f = [1.58 \cdot \ln(50346) - 3.28]^{-2} \Rightarrow \boxed{f = 0.00523}$$

$$Nu_b = \frac{\left(\frac{0.00523}{2}\right) \cdot 50346 \cdot (7.05)}{1 + 8.7 \left(\frac{0.00523}{2}\right)^{1/2} \cdot (7.05 - 1)} \Rightarrow \boxed{Nu_b = 251.5}$$
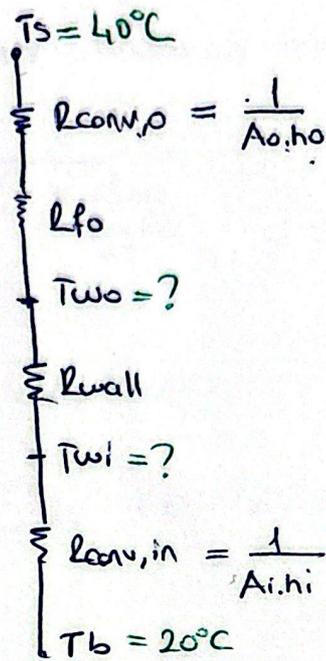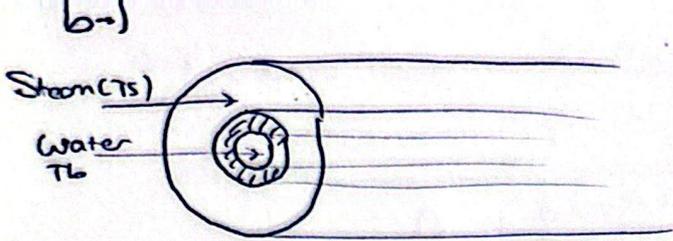
$$h_i = \frac{Nu \cdot k}{d_i} \Rightarrow \frac{251.5 \times 0.599}{0.0254} \Rightarrow \boxed{h_i = 5931 \frac{W}{m^2 K}}$$

→ Find Overall Heat Transfer Coof. U_o ៖ (Dividing by "A_o")

$$U_o = \cfrac{1}{\cfrac{d_o}{h_i \cdot d_i} + \cfrac{d_o \cdot \ln(d_o/d_i)}{2 K_c} + Rf_o + \cfrac{1}{h_o}}$$

$$U_o = \cfrac{1}{\cfrac{0.0305}{5931 \times 0.0254} + \cfrac{0.0305 \cdot \ln[0.0305/0.0254]}{2 \times 383} + 0.000176 + \cfrac{1}{6000}}$$

$$\boxed{U_o = 1827 \; W/m^2 K}$$

b-)

$T_s = 40°C$

$R_{conv,o} = \dfrac{1}{A_o.h_o}$

R_{fo}

$T_{wo} = ?$

R_{wall}

$T_{wi} = ?$

$R_{conv,in} = \dfrac{1}{A_i.h_i}$

$T_b = 20°C$

?

$$\dfrac{T_s - T_b}{R_{conv,o} + R_{fo} + R_{wall} + R_{conv,in}} \neq \dfrac{\boxed{T_{wi} - T_b}}{R_{conv,in}}$$

$$T_{wi} = \dfrac{R_{conv,in}(T_s - T_b)}{\dfrac{R_{conv,o} + R_{fo} + R_{wall} + R_{conv,in}}{A_o}} + T_b$$

$$T_{wi} = \dfrac{\dfrac{1\,A_i}{A_i.h_i}(T_s - T_b)}{A_i \times \left(\dfrac{1}{A_o.h_o} + \dfrac{R_{fo}}{A_o} + \dfrac{\ln(d_o/d_i)}{2\pi k L} + \dfrac{1}{A_i.h_i}\right)} + T_b$$

$\left\{ \dfrac{A_i}{A_o} = \dfrac{d_i}{d_o}\right)$

$A_i = \pi . d_i . L$

$$T_{wi} = \dfrac{\dfrac{(T_s - T_b)}{h_i}}{\dfrac{d_i}{d_o}.\dfrac{1}{h_o} + \dfrac{d_i R_{fo}}{d_o} + \dfrac{d_i.\ln(d_o/d_i)}{2 \times k_c} + \dfrac{1}{h_i}} + T_b$$

$$T_{wi} = \dfrac{\left[\dfrac{40-20}{5931}\right]}{\dfrac{0.0254}{0.0305}.\dfrac{1}{6000} + \dfrac{0.0254}{0.0305}.0.000176 + \dfrac{0.0254.\ln[0.0305/0.0254]}{2 \times 383} + \dfrac{1}{5931}} + 20$$

$$\boxed{T_{wi} = 21.4°C}$$

PROBLEM 2:

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 22 m 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 mild steel (ε=0.0445 mm). The sea water flow rate is 120 m³/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 = H_m);

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.).
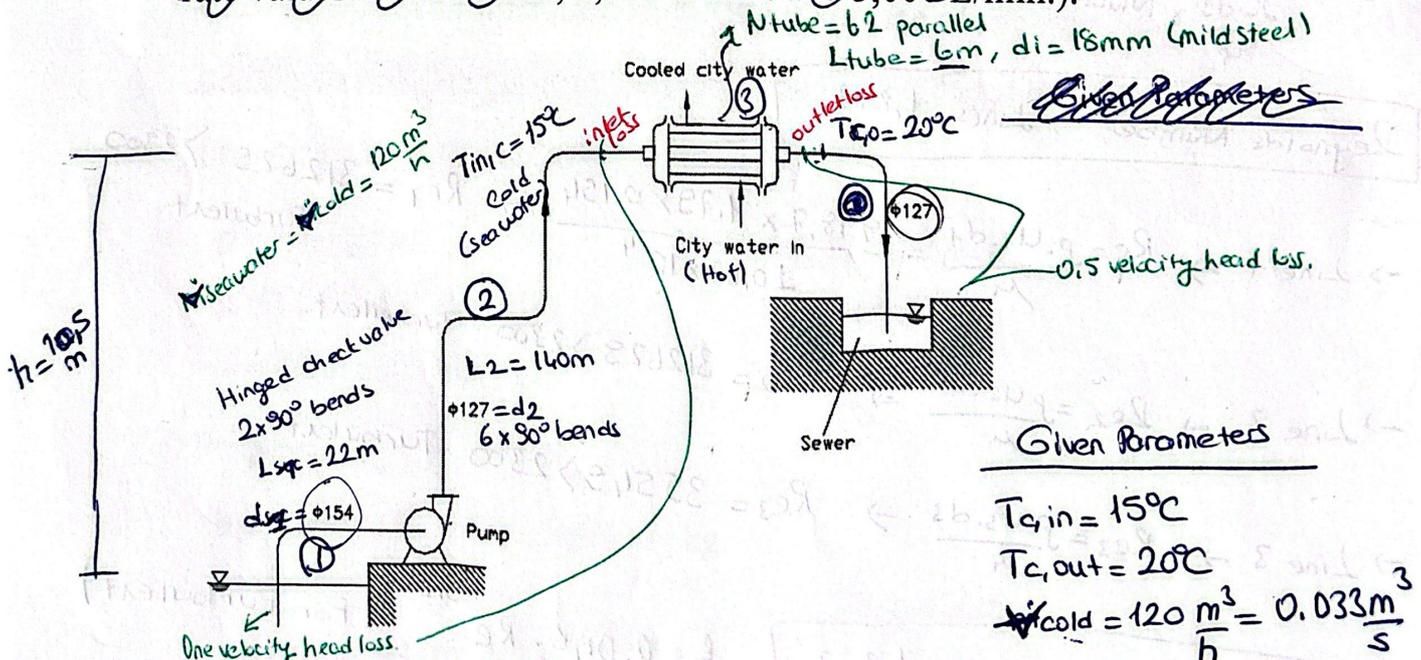
Properties of water at $T_m = \dfrac{15+20}{2} = 17.5°C$:

$\rho = 998.7$ kg/m³ $\mu = 10.67 \times 10^{-4}$ Pa.s

90° bend → Steel R/d =1.0

Inlet loss = $(\Delta P_{inlet}) = \dfrac{1}{2}\rho . u^2$

Outlet . loss = $(\Delta P_{outlet}) = 0.5 \times \left(\dfrac{1}{2}\rho . u^2\right)$

Handwritten annotations on figure:

- $N_{tube} = 62$ parallel
- $L_{tube} = 6m$, $d_i = 18mm$ (mild steel)
- Cooled city water
- inlet loss
- outlet loss
- $T_{c,o} = 20°C$
- Given Parameters
- $T_{in,c} = 15°C$ cold (sea water)
- $\dot{V}_{seawater} = \dot{V}_{cold} = 120 \frac{m^3}{h}$
- Hinged check valve
- $2 \times 90°$ bends
- $L_{sp} = 22m$
- $d_{sp} = \phi 154$
- Pump
- $L_2 = 140m$
- $\phi 127 = d_2$
- $6 \times 90°$ bends
- City water In (Hot)
- $\phi 127$
- Sewer
- 0.5 velocity head loss.
- One velocity head loss.
- $h = 10.5 m$

Given Parameters

$T_{c,in} = 15°C$

$T_{c,out} = 20°C$

$\dot{V}_{cold} = 120 \dfrac{m^3}{h} = 0.033 \dfrac{m^3}{s}$

Pipe and lines

$d_1 = 154$ mm ; $L_1 = 22m$

$d_2 = 127$ mm ; $L_2 = 140m$

$d_3 = \dfrac{18}{}$ mm ; $L_3 = 6m$

Heat exchanger

$N_{tube} = 62$

Volumetric flow rate of seawater $\Rightarrow \dot{V}_{cold} = 120 \frac{m^3}{h} = 0.033 \frac{m^3}{s}$

$\llcorner$ Mass flow rate of seawater $= \dot{m}_{cold} = \dot{V}_{cold} \times \rho$

$$\dot{m}_{cold} = 0.033 \times 998.7 \rightarrow \boxed{\dot{m}_{cold} = 33,29 \frac{kg}{s}}$$

$\boxed{\text{Velocities} \Rightarrow \text{Lines } 1, 2, 3}$ $\left\{ \dot{m} = \rho \cdot V \cdot A = \rho \cdot \dot{V} \right\} \rightarrow \left(V = \frac{\dot{V}}{A} \right)$

$\Rightarrow$ In suction Line (Line 1) $d_1 = 154 mm = 0.154 m$
$L_1 = 22 m$

$U_1 = \dfrac{\dot{V}_{cold}}{\frac{\pi \cdot d_1^2}{4}} \Rightarrow U_1 = \dfrac{0.033}{\frac{\pi \cdot 0.154^2}{4}} \rightarrow \boxed{U_1 = 1.79 \, m/s}$

$\rightarrow$ In the pipe from pump to heat exchanger (Line 2) $d_2 = 127 mm = 0.127 m$
$L_2 = 140 m$

$U_2 = \dfrac{\dot{V}_{cold}}{\frac{\pi \cdot d_2^2}{4}} \rightarrow \boxed{U_2 = 2.63 \, m/s}$

$\Rightarrow$ In the heat exchanger tubes (Line 3)

$U_3 = \dfrac{\dot{V}_{cold}}{\frac{\pi \cdot d_3^2}{4} \times N_{Tubes}} \Rightarrow \boxed{U_3 = 2.11 \, m/s}$

$\boxed{\text{Reynolds Number} \rightarrow \text{Lines } 1, 2, 3}$

$\Rightarrow$ Line 1 $\rightarrow$ $R_e = \dfrac{\rho \cdot U_1 \cdot d_1}{\mu} = \dfrac{998.7 \times 1.79 \times 0.154}{10.67 \times 10^{-4}} \rightarrow R_{e_1} = 312629 > 2300$
Turbulent

$\rightarrow$ Line 2 $\rightarrow$ $R_{e_2} = \dfrac{\rho \cdot U_2 \cdot d_2}{\mu} \Rightarrow$ $R_{e_2} = 312629 > 2300$ Turbulent

$\rightarrow$ Line 3 $\rightarrow$ $R_{e_3} = \dfrac{\rho \cdot U_3 \cdot d_3}{\mu} \rightarrow$ $R_{e_3} = 35549 > 2300$. Turbulent

$\boxed{\text{Friction coefficients} \rightarrow \text{Lines } 1, 2, 3}$ $f = 0.046 \cdot R_e^{-0.2}$ (For Turbulent)

$f_1 = 0.046 \cdot (R_{e_1})^{-0.2} \Rightarrow \boxed{f_1 = 0.0038}$

$\boxed{f_2 = 0.0037}$

$\boxed{f_3 = 0.0056}$

PROBLEM 3: (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°C

Water outlet temperature, 45°C

Condensation temperature, 105°C

After 4 months of operation, under the same conditions, the exit temperature of water drop to 40°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.

Specific heat for water at mean temperature T_m ~~$\frac{T_{c1}+T_{c2}}{2} = \frac{35+95}{2} = 65°C$~~

$c_{p,c} = 4178$ J/kg.K
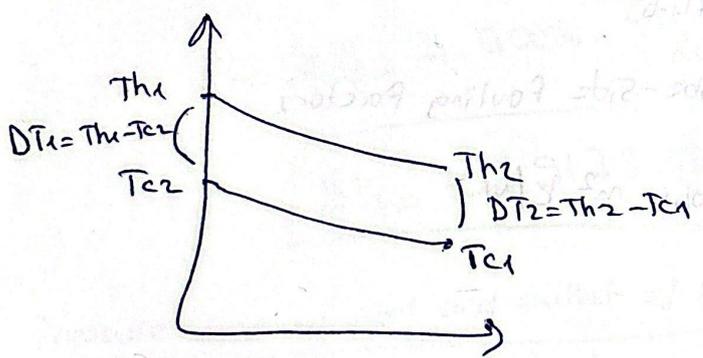
Given's

- water inlet temp. $(T_{c1}) = 20°C$
- " outlet " $(T_{c2}) = 45°C$
- Steam temp. $(T_h) = 105°C$
- Mass flow rate of Cold water $(\dot{m}_c) = 70$ kg/s

- outer diameter of tube $(d_o) = 1.9$ cm
- Inner " " " $(d_i) = 1.6$ cm
- tot. length of tube $(L) = 3$ m
- After four months $(T_{c2}) = 40°C$

Find $\Rightarrow$ Tot. fouling resistance (R_{ft})

$$Q = \dot{m}_c \times C_{pc}(T_{c2} - T_{c1}) = 70 \times 4178 \times (45-20) = 7311.9 \text{ kw}$$

$A_o \Rightarrow$ Tot. heat transfer area $= \pi d_o L \cdot N = \pi \times 0.019 \times 3 \times 840 = 150.3 \text{ m}^2$

HEX log mean temperature;

$$\Delta T_m = \frac{\Delta T_L - \Delta T_2}{\ln(\Delta T_L / \Delta T_2)} = \frac{(105)-20) - (105-45)}{\ln\left(\frac{105-20}{105-45}\right)} = 71.8\,°C$$

$\Big\downarrow$

$\Delta T_L = (105-20)$

$\Delta T_2 = (105-45)$

$$Q = U_c A_o \Delta T_m \Rightarrow U_c = \frac{Q}{A_o \cdot \Delta T_m} = \frac{7311.9 \times 10^3}{150.3 \times 71.8} = 677.1 \text{ w/m}^2\text{.k}$$
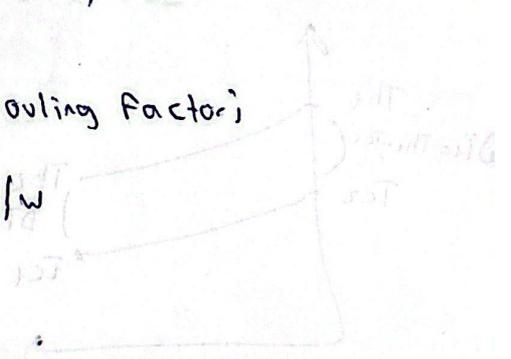
After 4 months;

~~test~~

$$Q_f = \dot{m}_c C_{pc}(t_{c2_f} - T_{c1}) = 70 \times 4179 \times (40-20) = 5850.6 \text{ kw}$$

$$\Delta T_m = \frac{\Delta T_L - \Delta T_2}{\ln(\Delta T_L / \Delta T_2)} = \frac{(105-20) - (105-40)}{\ln\left(\frac{105-20}{105-40}\right)} = 74.6\,°C$$

$$Q_f = U_f A_o \Delta T_m \Rightarrow U_f = \frac{Q_f}{A_o \Delta T_m} = \frac{5850.6 \times 10^3}{150.4 \times 74.6} = 521.5 \text{ w/m}^2\text{.k}$$

Shell-side fouling is negligible, so the tube-side fouling factor;

$$R_{ft} = \frac{1}{U_f} - \frac{1}{U_c} = \frac{1}{521.5} - \frac{1}{677.1} = 0.00044 \text{ m}^2\text{.k/w}$$

Pressure Drop in Lines

Line 1 $\Rightarrow$ Hinged check value
  2 × 90° Bends
  $f_1 = 0.0038$
  $d_1 = 0.154 \, m$
  $L_1 = 22 \, m$
  $u_1 = 1.79 \, m/s$

Line 2 $\Rightarrow$ 6 × 90° Bends
  $d_2 = 0.127 \, m$
  $L_2 = 140 \, m$
  $u_2 = 2.63 \, m/s$

From Table 4.3 $\Rightarrow$ 90° Bend $\rightarrow$ R/D = 1 ise $L/di = 16.5$ oluyor

$\Rightarrow$ Hinged check value $\rightarrow L/di = 110$

$$\Delta P = 4 \cdot f \cdot \rho \cdot \frac{u^2}{2} \sum \left(\frac{L}{di} \right) \sim \text{General form.}$$

$\rightarrow$ Line 1 $\Rightarrow$ $\Delta P_1 = 4 \times 0.0038 \cdot (998.7) \times \frac{1.79^2}{2} \cdot \left(\underset{\text{Hinged check value}}{110} + \underset{90° \text{ Bend}}{2 \times 16.5} + \underset{\text{Line}}{\frac{22}{0.154}} \right)$

$$\boxed{\Delta P_1 = 6951.9 \, N/m^2}$$

$\rightarrow$ Line 2 $\Rightarrow$ $\Delta P_2 = 4 \times 0.0037 \cdot 998.7 \cdot \frac{2.63^2}{2} \cdot \left[6 \times 16.5 + \frac{140}{0.127} \right]$

$$\boxed{\Delta P_2 = 56351 \, N/m^2}$$

$\rightarrow$ Line 3 $\Rightarrow$ $\Delta P_3 = 4 \cdot 0.0056 \cdot 998.7 \cdot \frac{2.11^2}{2} \cdot \left(\frac{\overset{N_{tube} \quad L_{tube}}{62 \times 6}}{\underset{d_{tube}}{0.018}} \right)$

$$\boxed{\Delta P_3 = 1023173.1 \, N/m^2}$$

Pressure Loss at inlet and outlet of HEX:

$\Delta P_{inlet} + \Delta P_{outlet} = (1 + 0.5) \cdot \frac{1}{2} \cdot \rho \cdot u_2^2 \Rightarrow 1.5 \times \frac{1}{2} \cdot 998.7 \cdot 2.63^2 \Rightarrow \boxed{\Delta P_{loss} = 5181 \, N/m^2}$

Pressure Loss due to elevation difference:

$\Delta P_{elevation} = \rho \cdot g \cdot h = 998.7 \times 9.81 \times 10.5 \Rightarrow \boxed{\Delta P_{elevation} = 102976 \, N/m^2}$

$\Delta P_{total} = \Delta P_1 + \Delta P_2 + \Delta P_3 + \Delta P_{loss} + \Delta P_{elevation}$

$= 6951.9 + 56351 + 1023173.1 + 5181 + 102976 \rightarrow \boxed{\Delta P_{total} = 1205693.9 \, Pa \\ = 1205.69 \, kPa}$