CHAPTER 7
Heat Exchanger

Heat Transfer

- How is the heat transfer?
- Mechanism of Convection
- Applications .
- Mean fluid Velocity and Boundary and their effect on the rate of heat transfer.
- Fundamental equation of heat transfer
- Logarithmic-mean temperature difference.
- Heat transfer Coefficients.
- Heat flux and Nusselt correlation
- Simulation program for Heat Exchanger

How is the heat transfer?

- Heat can transfer between the surface of a solid conductor and the surrounding medium whenever temperature gradient exists.

  Conduction

  Convection

    Natural convection

    Forced Convection

Mechanisms of heat transfer

There exist 3 basic mechanisms of heat transfer between different bodies (or inside a continuous body)

- Conduction in solids or stagnant fluids
- Convection inside moving fluids, but first of all we shall discuss heat transfer from flowing fluid to a solid wall
- Radiation (electromagnetic waves) the only mechanism of energy transfer in an empty space

Aim of analysis is to find out relationships between heat flows (heat fluxes) and driving forces (temperature differences)

- Natural and forced Convection
  - Natural convection occurs whenever heat flows between a solid and fluid, or between fluid layers.
  - As a result of heat exchange
  - Change in density of effective fluid layers taken place, which causes upward flow of heated fluid.
- If this motion is associated with heat transfer mechanism only, then it is called Natural Convection

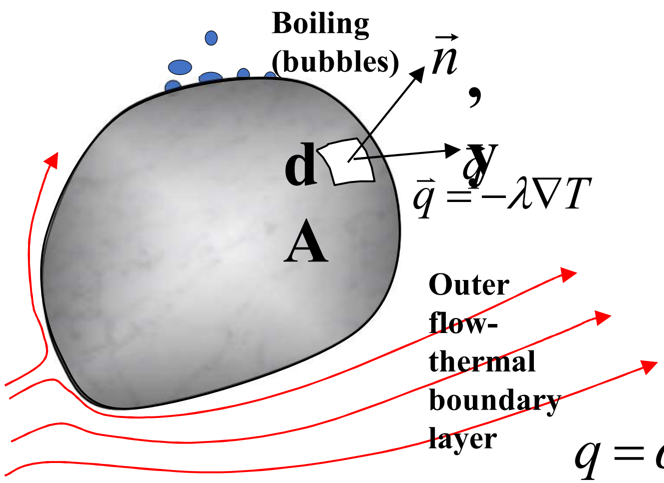
Forced Convection

- If this motion is associated by mechanical means such as pumps, gravity or fans, the movement of the fluid is enforced.
- And in this case, we then speak of Forced convection.

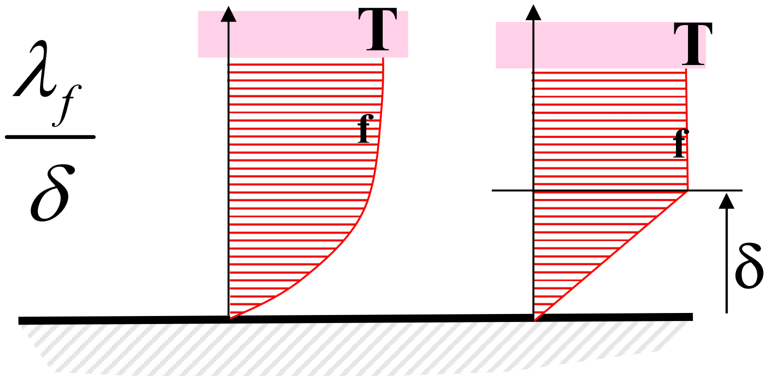
Convection

Calculation of heat flux q from flowing fluid to a solid surface requires calculation of temperature profile in the vicinity of surface (for example temperature gradients in attached bubbles during boiling, all details of thermal boundary layer,…).

Engineering approach simplifies the problem by introducing the idea of stagnant homogeneous layer of fluid, having an equivalent thermal resistance (characterized by the heat transfer coefficient α [W/(m^{2}K)])



$$\alpha = \frac{\lambda_f}{\delta}$$



$$q = \alpha (T_f - T_w) = \lambda_f \frac{\partial T}{\partial y} = \frac{\lambda_f}{\delta}(T_f - T_w)$$

Convection – Nu,Re,Pr

Heat transfer coefficient α depends upon the flow velocity (u), thermodynamic parameters of fluid (λ) and geometry (for example diameter of sphere or pipe D). Value α is calculated from engineering correlation using dimensionless criteria

$$Nu = \frac{hD}{k}$$ Nusselt number (dimensionless α, reciprocal thickness of boundary layer)

$$\mathrm{Re} = \frac{uD\rho}{\mu}$$ Reynolds number (dimensionless velocity, ratio of intertial and viscous forces)

$$\mathrm{Pr} = \frac{\nu}{a}$$ Prandl number (property of fluid, ratio of viscosity and temperature diffusivity)

Rem: μ is dynamic viscosity [Pa.s], ν kinematic viscosity [m^2/s],
ν=μ/ρ

And others

Pe=Re.Pr	**Péclet number**
Gz=Pe.D/L	**Graetz number (D-diameter, L-length of pipe)**
	Rayleigh
De=Re√D/D_c	**Dean number (coiled tube, D_c diameter of curvature)**

Convection Turbulent flow

Turbulent flow is characterised by the energy transport by turbulent eddies which is more intensive than the molecular transport in laminar flows. Heat transfer coefficient and the Nusselt number is greater in turbulent flows. Basic differences between laminar and turbulent flows are:

- Nu is proportional to $\sqrt[3]{u}$ in laminar flow, and $u^{0.8}$ in turbulent flow.
- Nu doesn't depend upon the length of pipe in turbulent flows significantly (unlike the case of laminar flows characterized by rapid decrease of Nu with the length L)
- Nu doesn't depend upon the shape of cross section in the turbulent flow regime (it is possible to use the same correlations for eliptical, rectangular…cross sections using the concept of equivalent diameter – this cannot be done in laminar flows)

The simplest correlation for hydraulically smooth pipe designed by Dittus Boelter is frequently used (and should be memorized)

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{m}$$

m=0.4 for heating

m=0.3 for cooling

important!

Similar result follows from the Colburn analogy $Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{1/3}$

Convection Turbulent flow

- Dittus Boelter correlation assumes that Nu is independent of L. For very short pipes (L/D<60) Hausen's correlation can be applied

$$Nu = 0{,}037\left[1+\left(\frac{D}{L}\right)^{2/3}\right]\left(Re^{3/4}-180\right)Pr^{0{,}42}\left(\frac{\mu}{\mu_W}\right)^{0{,}14},$$

$2300 < Re < 10^5$ **$0{,}6 < Pr < 500$**

- Effect of wall roughness can be estimated from correlations based upon analogies between momentum and heat transfer (Reynolds analogy, Colburn analogy, Prandtl Taylor analogy). Results from hydraulics (pressure drop, friction factor λ_f) are used for heat transfer prediction. Example is correlation Pětuchov (1970) recommended in VDI Warmeatlas

Friction factor

$$Nu = \frac{\lambda_f \, Re\, Pr}{8.56 + 35.92\left(Pr^{2/3}-1\right)\sqrt{\lambda_f}}\left(\frac{\mu}{\mu_W}\right)^n$$

$n=0{,}11$ $T_W > T$ (*heating*)

$n=0{,}25$ $T_W < T$ (*cooling*)

$10^4 < Re < 5\cdot 10^6$ $0{,}5 < Pr < 2000$ $0{,}08 < \mu_W/\mu < 40$.

Heat Exchangers Defination

- A device whose primary purpose is the transfer of energy between two fluids is named a Heat Exchanger.

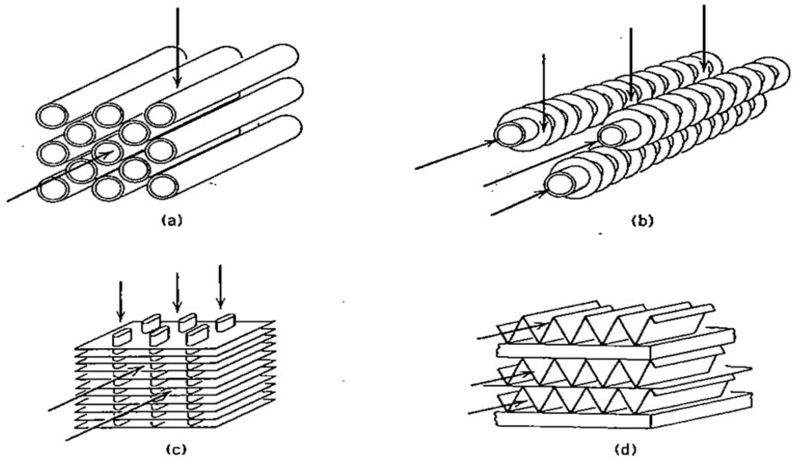
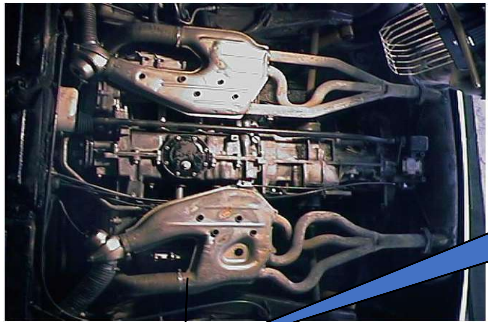


Figure 22.4 Compact heat-exchanger configurations.

Applications of Heat Exchangers



Heat Exchangers prevent car engine overheating and increase efficiency

Heat exchangers are used in AC and furnaces

Heat exchangers are used in Industry for heat transfer

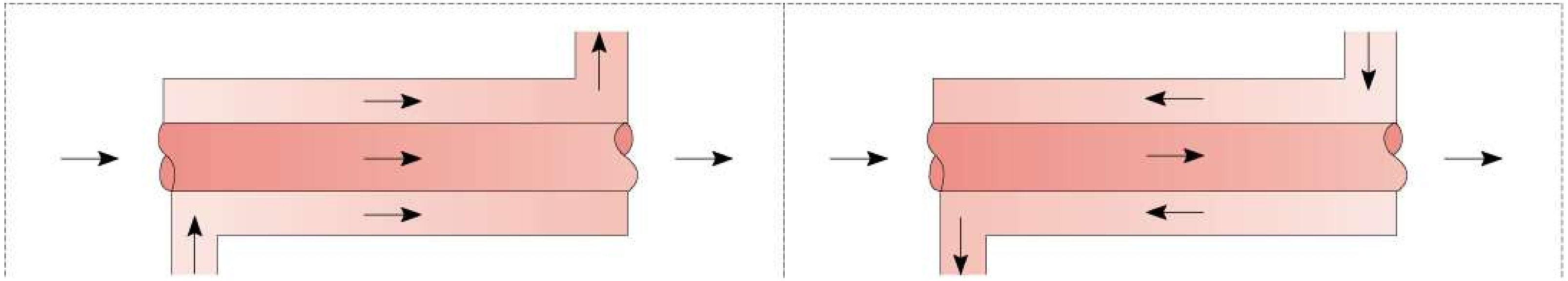
What are heat exchangers for?

- To get fluid streams to the right temperature for the next process
  - reactions often require feeds at high temp.
- To condense vapours
- To evaporate liquids
- To recover heat to use elsewhere
- To reject low-grade heat
- To drive a power cycle

Heat Exchanger Types

Heat exchangers are uired for energy conversion and utilization. They involve heat exchange between two fluids separated by a solid and encompass a wide range of flow configurations.

- **Concentric-Tube Heat Exchangers**

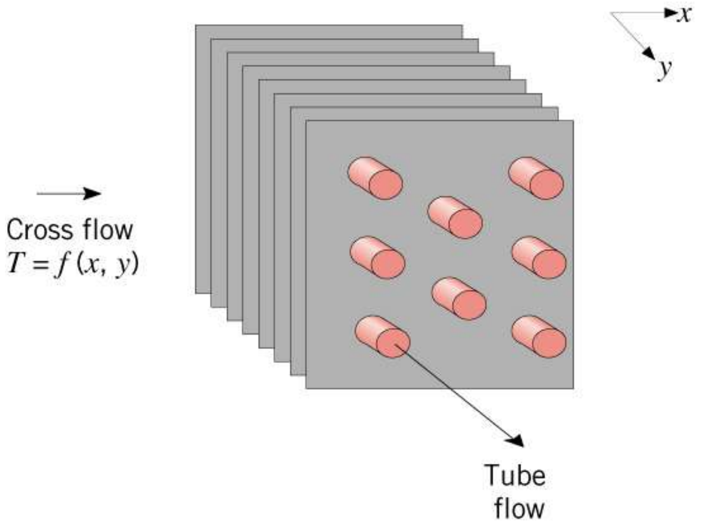


Parallel Flow **Counterflow**

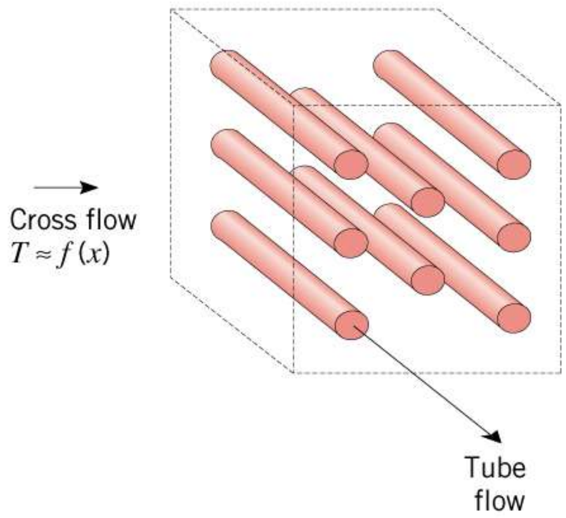
- **Simplest configuration.**

- **Superior performance associated with counter flow.**

- **Cross-flow Heat Exchangers**



Finned-Both Fluids Unmixed

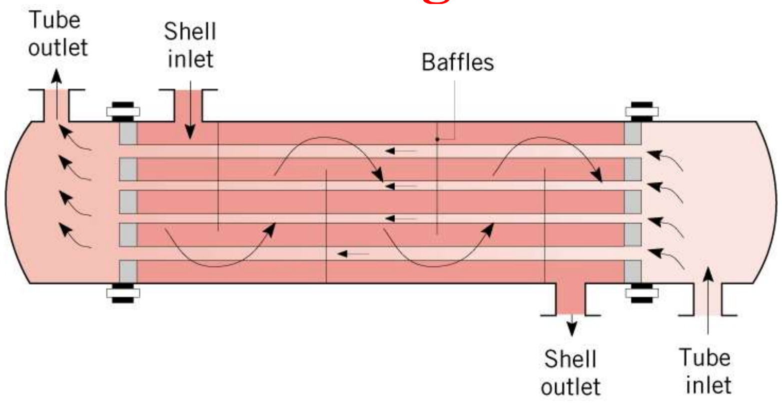


Unfinned-One Fluid Mixed the Other Unmixed

- **For cross-flow over the tubes, fluid motion, and hence mixin direction (*y*) is prevented for the finned tubes, but occurs for**

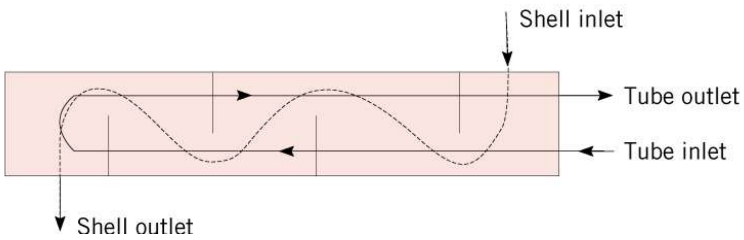
- **Heat exchanger performance is influenced by mixing.**

- **Shell-and-Tube Heat Exchangers**

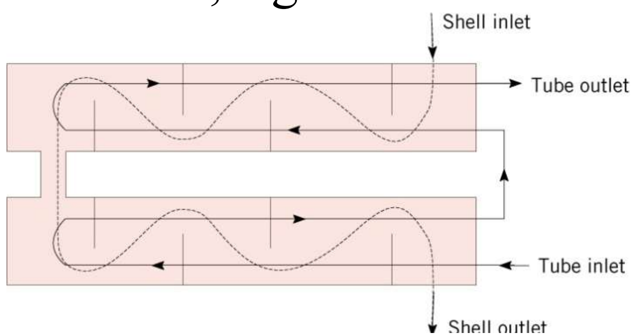


One Shell Pass and One Tube Pass

- **Baffles are used to establish a cross-flow and to induce turbulent mixing of the shell-side fluid, both of which enhance convection.**
  - The number of tube and shell passes may be varied, e.g.:

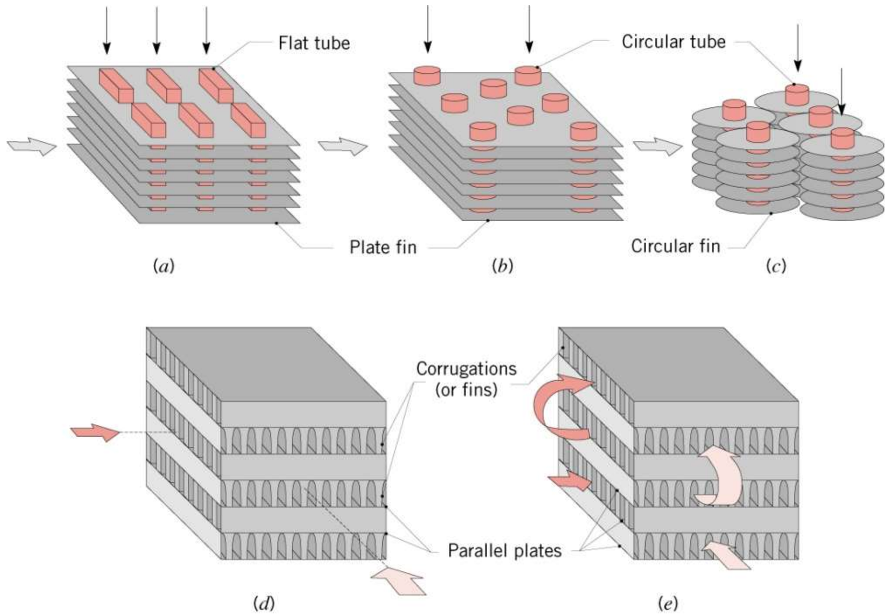


One Shell Pass,
Two Tube Passes



Two Shell Passes,
Four Tube Passes

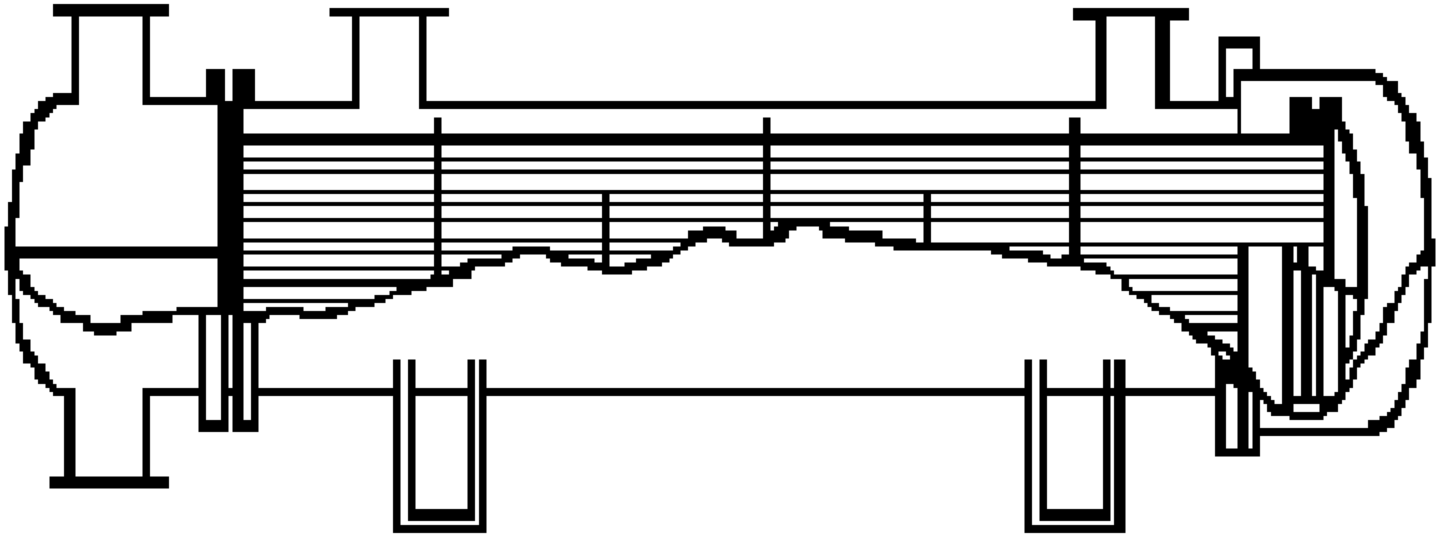
- **Compact Heat Exchangers**
  - **Widely used to achieve large heat rates per unit volume, particularly when one or both fluids is a gas.**



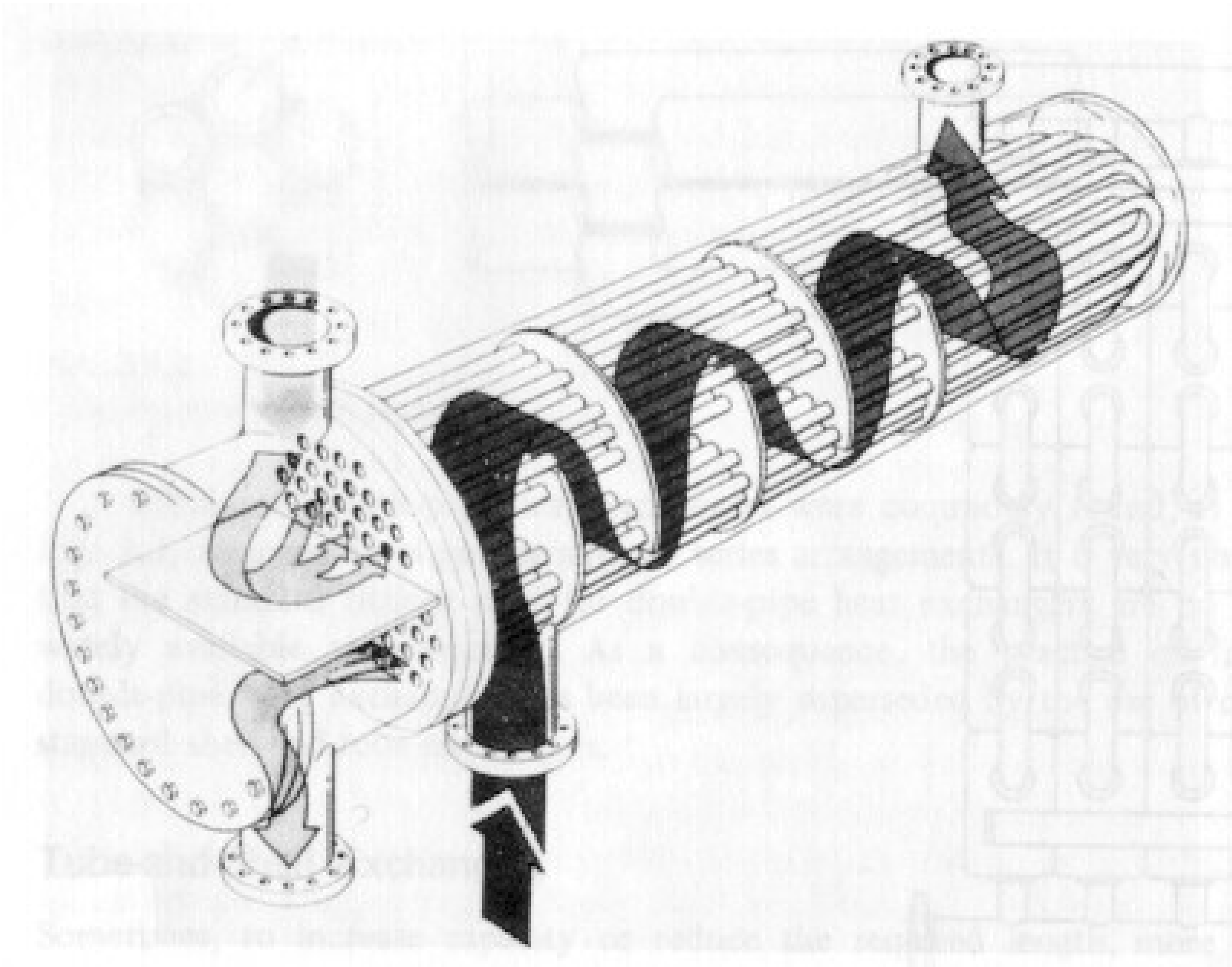
(a) Fin-tube (flat tubes, continuous plate fins)
(b) Fin-tube (circular tubes, continuous plate fins)
(c) Fin-tube (circular tubes, circular fins)
(d) Plate-fin (single pass)
(e) Plate-fin (multipass)

Shell and Tube

Typical shell and tube exchanger as used in the process industry

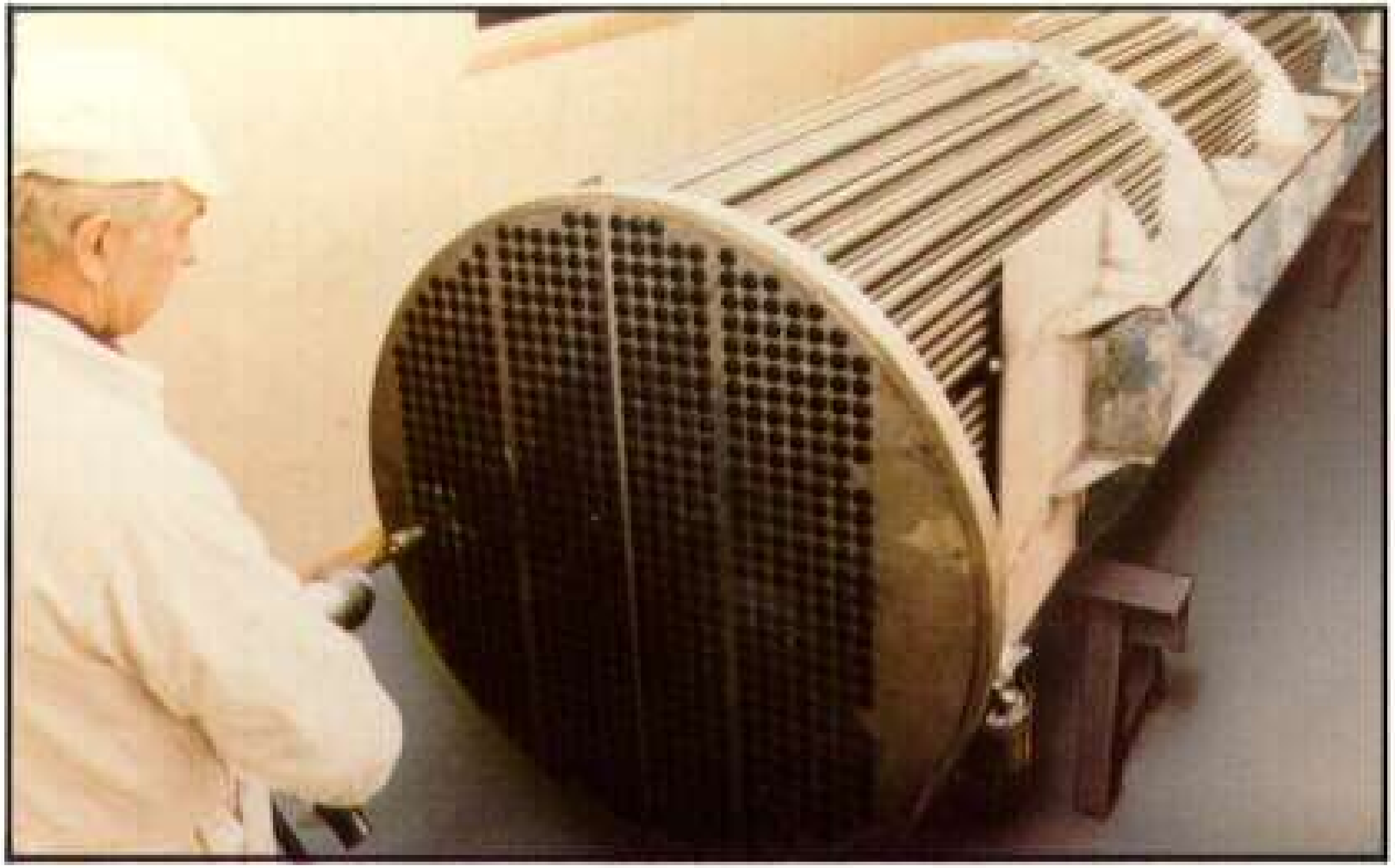


Shell-side flow



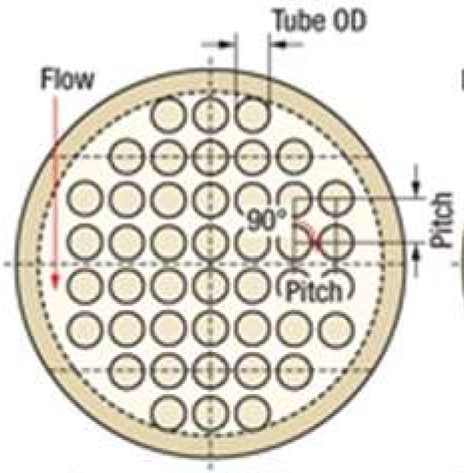
Complete shell-and-tube

Example of an exchanger

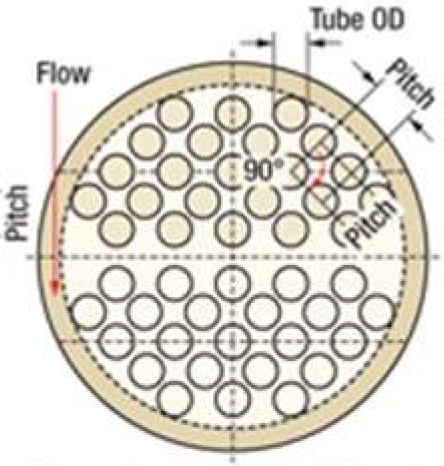


Bundle for shell-and-tube exchanger

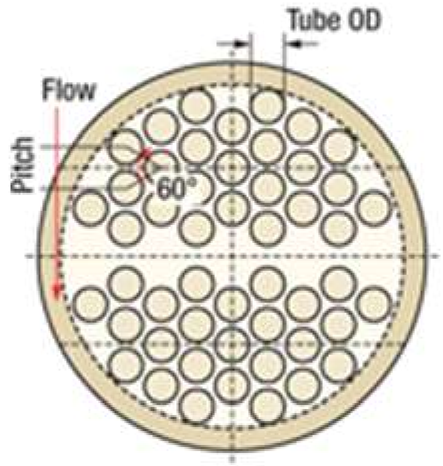
Coverage of Shell Area



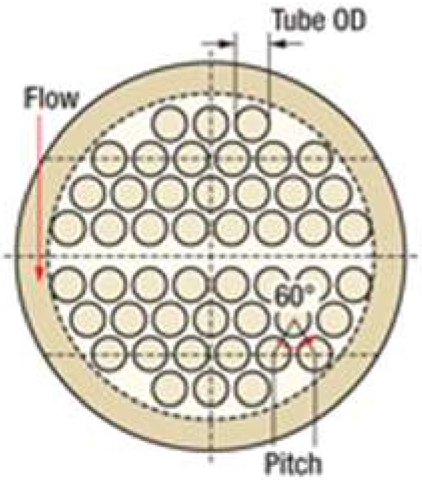
Square pitch, 90° layout



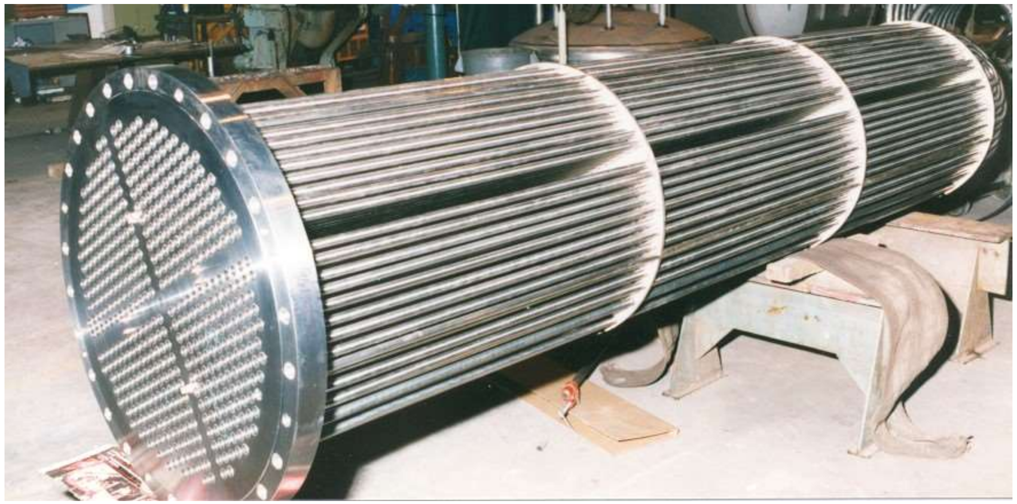
Diamond square pitch, 45° layout



Inline triangular pitch, 60° layout



Triangular pitch, 30° layout



Overall Heat Transfer Coefficient for the Heat Exchanger

The overall heat transfer coefficient for clean surface (*U_c*) is given by

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

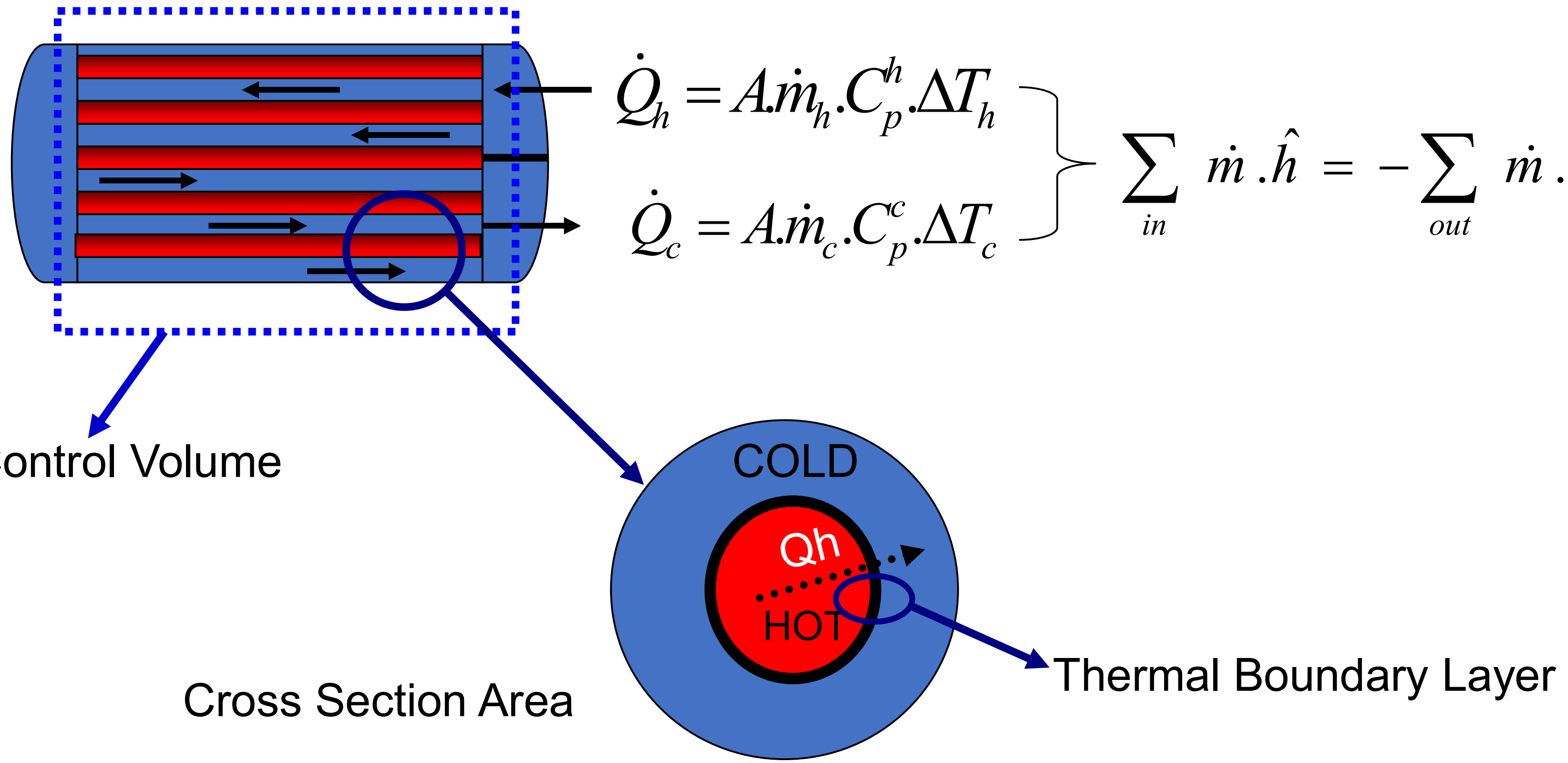
Considering the total fouling resistance, the heat transfer coefficient for fouled surface (*U_f*) can be calculated from the following expression:

$$\frac{1}{U_f} = \frac{1}{U_c} + R_{ft}$$

Principle of Heat Exchanger

- First Law of Thermodynamic: *“Energy is conserved.”*

$$\cancelto{0}{\frac{dE}{dt}} = \left(\sum_{in} \dot{m}\,.\hat{h}_{in} - \sum_{out} \dot{m}\,.\hat{h}_{out} \right) + \cancelto{0}{\dot{Q}} + \cancelto{0}{\dot{w}_s} + \cancelto{0}{\dot{e}_{generated}}$$



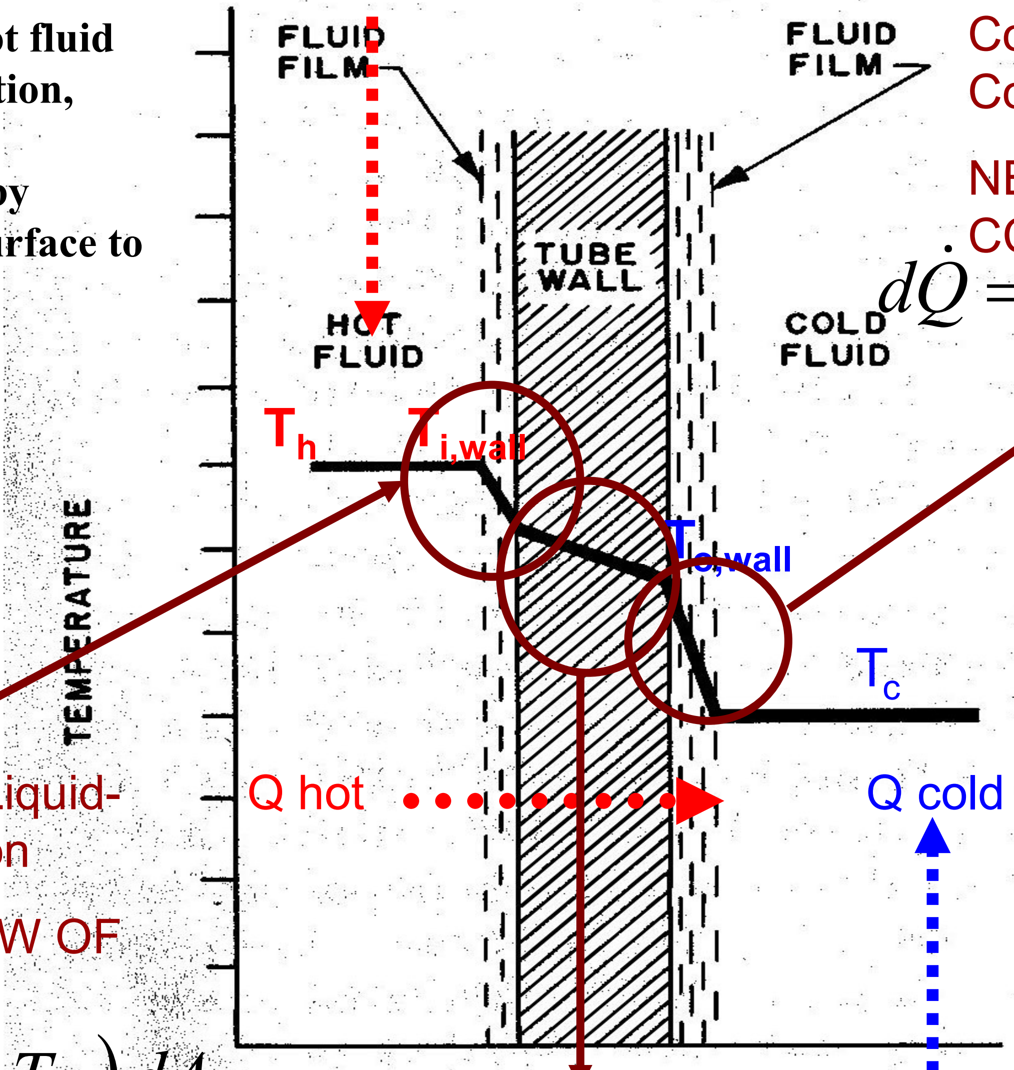
$$\dot{Q}_h = A.\dot{m}_h.C_p^h.\Delta T_h$$

$$\dot{Q}_c = A.\dot{m}_c.C_p^c.\Delta T_c$$

$$\sum_{in} \dot{m}\,.\hat{h} = -\sum_{out} \dot{m}\,.\hat{h}$$

Thermal boundary layer

Energy moves from hot fluid to a surface by convection, through the wall by conduction, and then by convection from the surface to the cold fluid.



Region I : Hot Liquid-Solid Convection

NEWTON'S LAW OF CCOLING

$$d\dot{Q} = h_h.(T_h - T_{iw}).dA$$

Region II : Conduction Across Copper Wall

FOURIER'S LAW

$$d\dot{q} = -k.\frac{dT}{dr}$$

Region III: Solid – Cold Liquid Convection

NEWTON'S LAW OF CCOLING

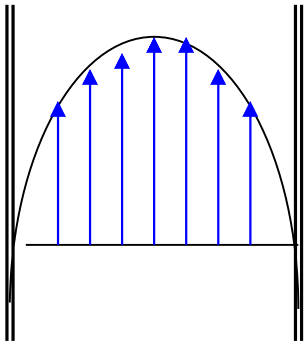
$$d\dot{Q} = h_c.(T_{ow} - T_c).dA$$

- **Velocity distribution and boundary layer**

When fluid flow through a circular tube of uniform cross-suction and fully developed,

The velocity distribution depend on the type of the flow.

In laminar flow the volumetric flowrate is a function of the radius.



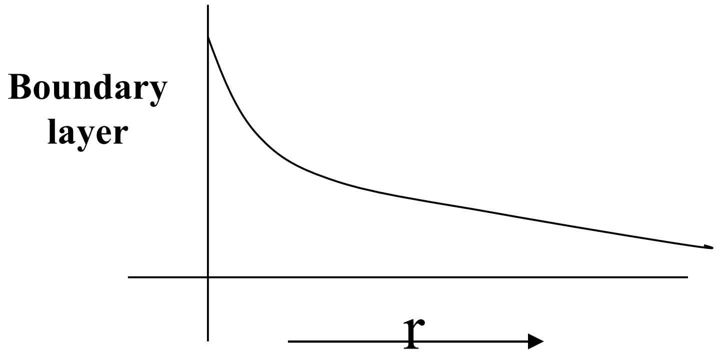
$$V = \int_{r=0}^{r=D/2} u 2\pi r dr$$

V = volumetric flowrate

u = average mean velocity

- In turbulent flow, there is no such distribution.

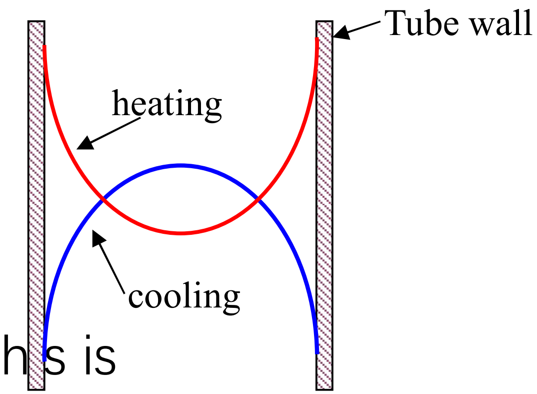
- The molecule of the flowing fluid which adjacent to the surface have zero velocity because of mass-attractive forces. Other fluid particles in the vicinity of this layer, when attempting to slid over it, are slow down by viscous forces.

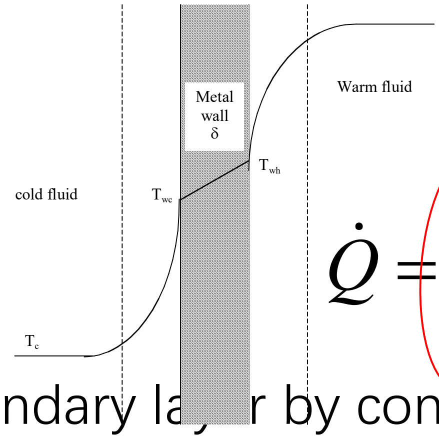


- Accordingly the temperature gradient is larger at the wall and through the viscous sub-layer, and small in the turbulent core.

$$\dot{Q} = hA\Delta T$$

$$\dot{Q} = hA(T_w - T)$$





$$\dot{Q} = \frac{k}{\delta} A(T_w - T)$$

h

- The reason for this is

  1) Heat must transfer through the boundary layer by conduction.

  2) Most of the fluid have a low thermal conductivity (k)

  3) While in the turbulent core there are a rapid moving eddies, which they are equalizing the temperature.

U = The Overall Heat Transfer Coefficient [W/m.K]

Region I : Hot Liquid – Solid Convection

$$\dot{Q} = h_{hot}\left(T_h - T_{iw}\right).A \;\longrightarrow\; T_h - T_{iw} = \frac{\dot{Q}}{h_h . A_i}$$

Region II : Conduction Across Copper Wall

$$\dot{Q} = \frac{k_{copper}.2\pi L}{\ln\frac{r_o}{r_i}}\left(T_{w,i} - T_{w,o}\right) \;\longrightarrow\; T_{i,w} - T_{o,w} = \frac{\dot{Q}.\ln\left(\frac{r_o}{r_i}\right)}{k_{copper}.2\pi L}$$

Region III : Solid – Cold Liquid Convection

$$\dot{Q} = h_c\left(T_c - T_{w,i}\right)A_o \;\longrightarrow\; T_{o,wall} - T_c = \frac{\dot{Q}}{h_c . A_o}$$

+

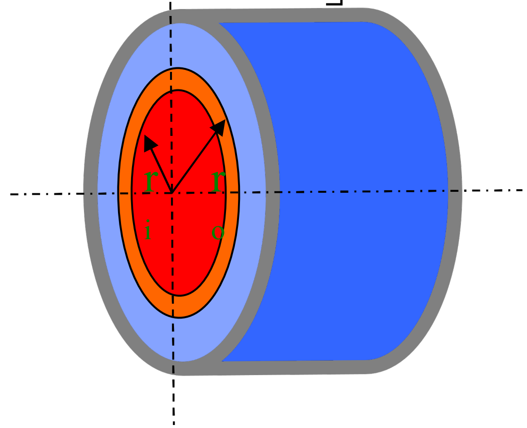
$$T_h - T_c = \dot{Q}\left[\frac{1}{h_h.A_i} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{k_{copper}.2\pi L} + \frac{1}{h_c.A_o}\right]$$

$$T_h - T_c = \frac{\dot{Q}}{R_1 + R_2 + R_3}$$

$$\dot{Q} = U.A.\left(T_h - T_c\right)$$

$$U = \frac{1}{A.\Sigma R}$$

$$U = \left[\frac{r_o}{h_{hot}.r_i} + \frac{r_o.\ln\left(\frac{r_o}{r_i}\right)}{k_{copper}.r_i} + \frac{1}{h_{cold}}\right]^{-1}$$



Calculating Using Log Mean Temperature

Hot Stream : $d\dot{Q}_h = \dot{m}_h . C_p^h . dT_h$

Cold Stream: $d\dot{Q}_c = \dot{m}_c . C_p^c . dT_c$

$$d(\Delta T) = dT_h - dT_c$$

$$\Delta T = T_h - T_c$$

$$\rightarrow \quad d(\Delta T) = \left(\frac{d\dot{Q}_h}{m_h . C_p^h} - \frac{d\dot{Q}_c}{m_c . C_p^c} \right)$$

$$d\dot{Q} = -d\dot{Q}_{hot} = d\dot{Q}_{cold}$$

$$-d\dot{Q} = -U.\Delta T.dA$$

$$d(\Delta T) = -U.\Delta T.dA \left(\frac{1}{m_h . C_p^h} + \frac{1}{m_c . C_p^c} \right)$$

$$\int_{\Delta T_1}^{\Delta T_2} \frac{d(\Delta T)}{\Delta T} = -U . \left(\frac{\Delta T_h}{\dot{Q}_h} + \frac{\Delta T_c}{\dot{Q}_c} \right) . \int_{A_1}^{A_2} dA$$

$$\int_{\Delta T_1}^{\Delta T_2} \frac{d(\Delta T)}{\Delta T} = -U . \left(\frac{1}{m_h . C_p^h} + \frac{1}{m_c . C_p^c} \right) . \int_{A_1}^{A_2} dA$$

$$\ln\left(\frac{\Delta T_2}{\Delta T_1} \right) = -\frac{U.A.}{\dot{Q}} (\Delta T_h + \Delta T_c) = -\frac{U.A}{\dot{Q}} \left[\left(T_h^{in} - T_h^{out} \right) - \left(T_c^{in} - T_c^{out} \right) \right]$$

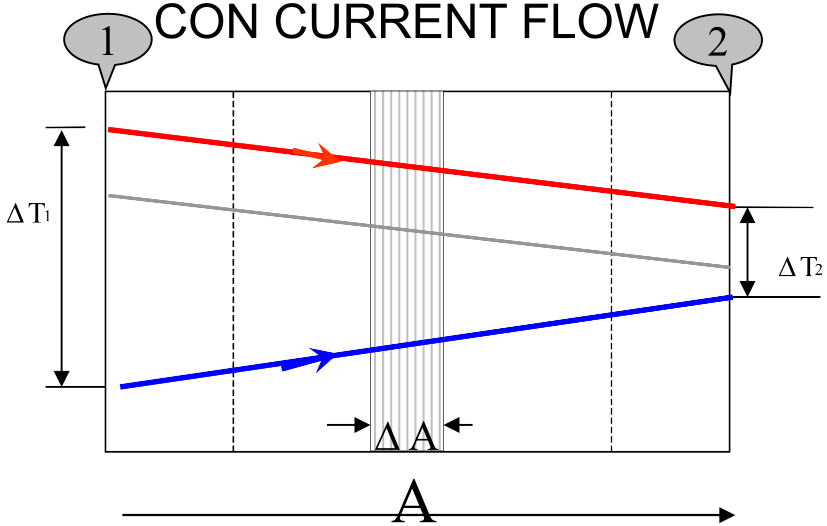
$$\dot{Q} = U . A \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1} \right)}$$

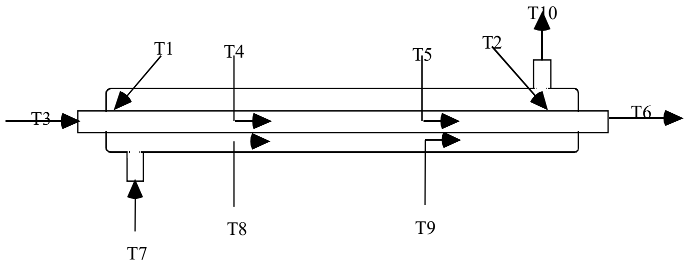
Log Mean Temperature

Log Mean Temperature evaluation

$$\Delta T_{Ln} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} \rightarrow U = \frac{\dot{m}_h . C_p^h .(T_3 - T_6)}{A.\Delta T_{Ln}} = \frac{\dot{m}_c . C_p^c .(T_7 - T_{10})}{A.\Delta T_{Ln}}$$

CON CURRENT FLOW



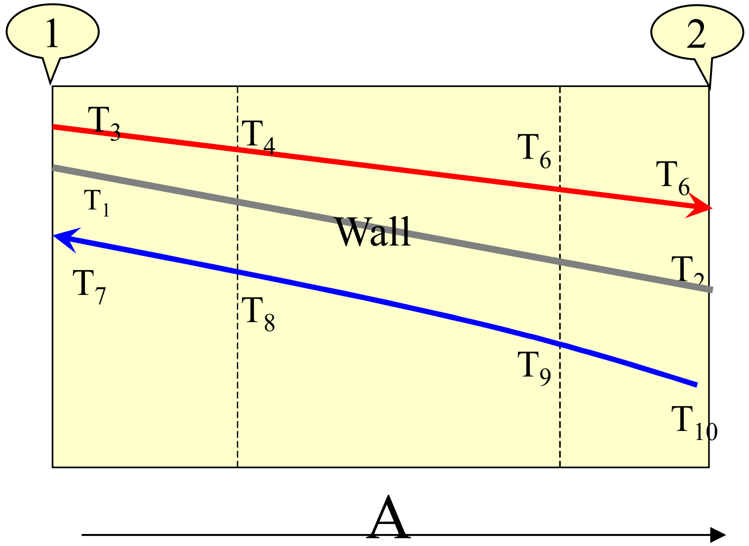


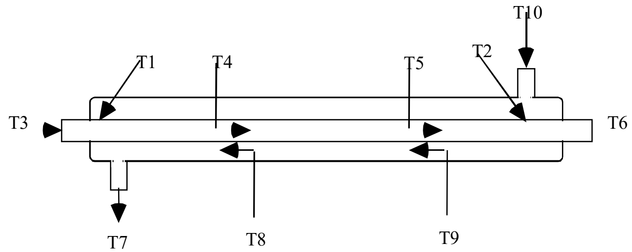
Parallel Flow

$$\Delta T_1 = T_h^{in} - T_c^{in} = T_3 - T_7$$

$$\Delta T_2 = T_h^{out} - T_c^{out} = T_6 - T_{10}$$

COUNTER CURRENT FLOW





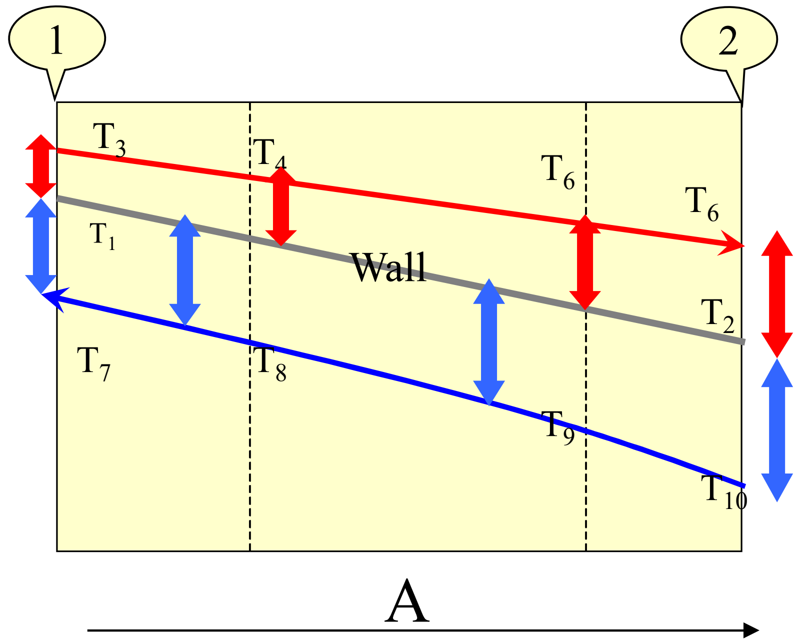
Counter - Current Flow

$$\Delta T_1 = T_h^{in} - T_c^{out} = T_3 - T_7$$

$$\Delta T_2 = T_h^{out} - T_c^{in} = T_6 - T_{10}$$

$$\dot{Q} = h_h \ A_i \ \Delta T_{lm}$$

$$\Delta T_{lm} = \frac{(T_3 - T_1) - (T_6 - T_2)}{\ln \dfrac{(T_3 - T_1)}{(T_6 - T_2)}}$$



$$\dot{Q} = h_c \ A_o \ \Delta T_{lm}$$

$$\Delta T_{lm} = \frac{(T_1 - T_7) - (T_2 - T_{10})}{\ln \dfrac{(T_1 - T_7)}{(T_2 - T_{10})}}$$

DIMENSIONLESS ANALYSIS TO CHARACTERIZE A HEAT EXCHANGER

$$Nu = f\,(\mathrm{Re}, \mathrm{Pr}, L/D, \mu_b/\mu_o)$$

$$Nu = \frac{h.D}{k} \qquad \mathrm{Re} = \frac{v.D.\rho}{\mu} \qquad \mathrm{Pr} = \frac{C_p.\mu}{k}$$

•**Further Simplification:**

$$Nu = a.\mathrm{Re}^b.\mathrm{Pr}^c$$

Can Be Obtained from 2 set of experiments

One set, run for constant Pr

And second set, run for constant Re

$$\dot{Q} = \frac{k}{\delta} A(T_w - T)$$

$\frac{k}{\delta}$ → h

•Empirical Correlation

- For laminar flow

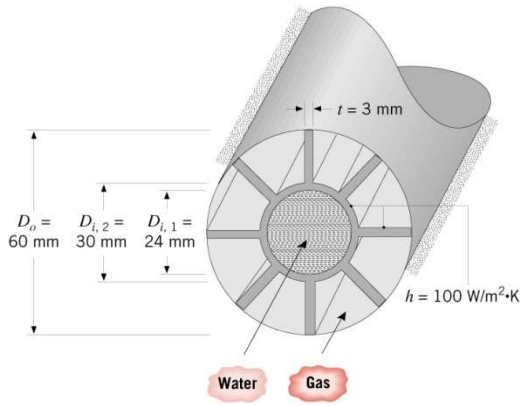
Nu = 1.62 (Re*Pr*L/D)

- For turbulent flow

$$Nu_{Ln} = 0.026.\mathrm{Re}^{0.8}.\mathrm{Pr}^{1/3}.\left(\frac{\mu_b}{\mu_o}\right)^{0.14}$$

- Good To Predict within 20%
- Conditions: L/D > 10
  0.6 < Pr < 16,700
  Re > 20,000

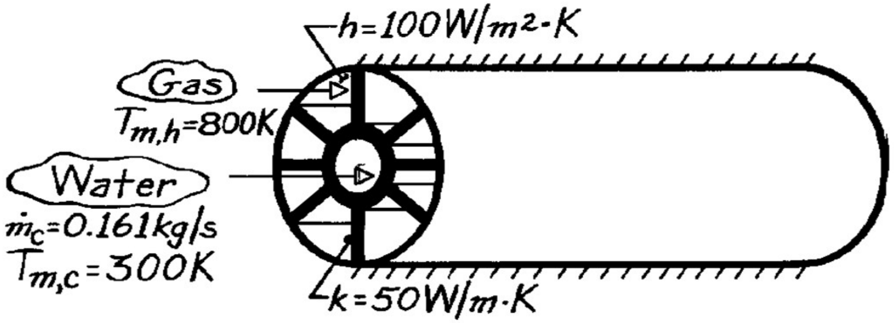
Determination of heat transfer per unit length for heat recovery device involving hot flue gases and water.



KNOWN: Geometry of finned, annular heat exchanger. Gas-side temperature and convection coefficient. Water-side flowrate and temperature.

FIND: Heat rate per unit length.

SCHEMATIC:



$D_o = 60$ mm
$D_{i,1} = 24$ mm
$D_{i,2} = 30$ mm
$t = 3$ mm $= 0.003$m
$L = (60\text{-}30)/2$ mm $= 0.015$m

ASSUMPTIONS: (1) Steady-state conditions, (2) Constant properties, (3) One-dimensional conduction in strut, (4) Adiabatic outer surface conditions, (5) Negligible gas-side radiation, (6) Fully-developed internal flow, (7) Negligible fouling.

PROPERTIES: *Table A-6*, Water (300 K): k = 0.613 W/m·K, Pr = 5.83, $\mu = 855 \times 10^{-6}$ N·s/m^2.

ANALYSIS: The heat rate is

$$q = (UA)_c \left(T_{m,h} - T_{m,c}\right)$$

where

$$1/(UA)_c = 1/(hA)_c + R_w + 1/(\eta_o hA)_h$$

$$R_w = \frac{\ln\left(D_{i,2}/D_{i,1}\right)}{2\pi kL} = \frac{\ln(30/24)}{2\pi(50\ W/m\cdot K)1m} = 7.10\times10^{-4}\,K/W.$$

With

$$\mathrm{Re_D} = \frac{4\dot{m}}{\pi D_{i,1}\mu} = \frac{4\times 0.161\ \mathrm{kg/s}}{\pi(0.024\mathrm{m})855\times10^{-6}\ \mathrm{N\cdot s/m^2}} = 9990$$

the internal flow is turbulent and the Dittus-Boelter correlation gives

$$h_c = \left(k/D_{i,1}\right)0.023\,\mathrm{Re}_D^{4/5}\,\mathrm{Pr}^{0.4} = \left(\frac{0.613\ \mathrm{W/m\cdot K}}{0.024\mathrm{m}}\right)0.023(9990)^{4/5}(5.83)^{0.4} = 1883\ \mathrm{W/m^2\cdot K}$$

$$(hA)_c^{-1} = \left(1883\ \mathrm{W/m^2\cdot K}\times\pi\times0.024\mathrm{m}\ \times 1\mathrm{m}\ \right)^{-1} = 7.043\times10^{-3}\ \mathrm{K/W}.$$

The overall fin efficiency is

$$\eta_o = 1-\left(A_f/A\right)\left(1-\eta_f\right)$$

$$A_f = 8\times2(L\cdot w) = 8\times2(0.015\mathrm{m}\times1\mathrm{m}) = 0.24\mathrm{m}^2$$

$$A = A_f + \left(\pi D_{i,2} - 8t\right)w = 0.24\mathrm{m}^2 + (\pi\times0.03\mathrm{m} - 8\times0.003\mathrm{m}) = 0.31\mathrm{m}^2.$$

From Eq. 11.4,

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

where

$$m = [2h/kt]^{1/2} = \left[2 \times 100\ W/m^2 \cdot K / 50\ W/m \cdot K \left(0.003m\right)^{1/2}\right] = 36.5\,m^{-1}$$

$$mL = \left(2h/kt\right)^{1/2} L = 36.5\,m^{-1} \times 0.015m = 0.55$$

$$\tanh\left[\left(2h/kt\right)^{1/2} L\right] = 0.499.$$

Hence

$$\eta_f = 0.800/1.10 = 0.907$$

$$\eta_o = 1 - \left(A_f / A\right)\left(1 - \eta_f\right) = 1 - \left(0.24/0.31\right)\left(1 - 0.907\right) = 0.928$$

$$\left(\eta_o hA\right)_h^{-1} = \left(0.928 \times 100\ W/m^2 \cdot K \times 0.31m^2\right)^{-1} = 0.0347\ K/W.$$

It follows that

$$\left(UA\right)_c^{-1} = \left(7.043 \times 10^{-3} + 7.1 \times 10^{-4} + 0.0347\right) K/W$$

$$\left(UA\right)_c = 23.6\ W/K$$

and

$$q = 23.6\ W/K\left(800 - 300\right)K = 11{,}800\ W$$ <

for a 1m long section.

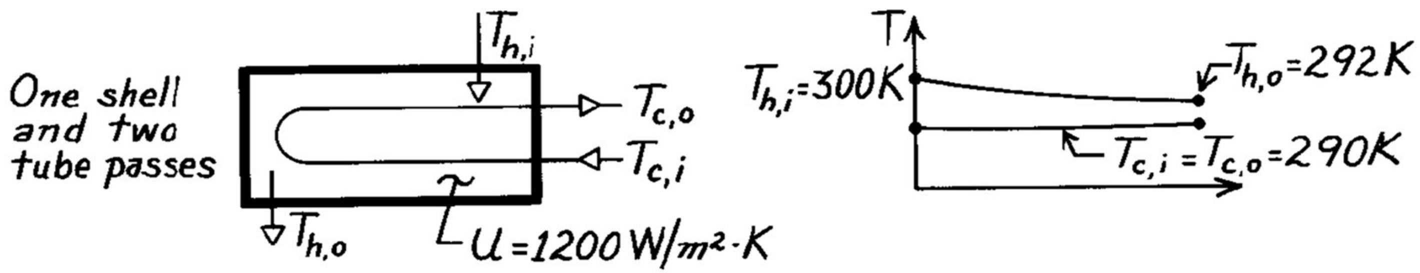
COMMENTS: (1) The gas-side resistance is substantially decreased by using the fins $\left(A'_f >> \pi D_{i,2}\right)$ and q is increased.

(2) Heat transfer enhancement by the fins could be increased further by using a material of larger k, but material selection would be limited by the large value of $T_{m,h}$.

Problem : Design of a two-pass, shell-and-tube heat exchanger to supply vapor for the turbine of an ocean thermal energy conversion system based on a standard (Rankine) power cycle. The power cycle is to generate 2 MW_e at an efficiency of 3%. Ocean water enters the tubes of the exchanger at 300K, and its desired outlet temperature is 292K. The working fluid of the power cycle is evaporated in the tubes of the exchanger at its phase change temperature of 290K, and the overall heat transfer coefficient is known.

FIND: (a) Evaporator area, (b) Water flow rate.

SCHEMATIC:



Overall Heat Transfer Coefficient for the Heat Exchanger

The overall heat transfer coefficient for clean surface (U_c) is given by

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

Considering the total fouling resistance, the heat transfer coefficient for fouled surface (U_f) can be calculated from the following expression:

$$\frac{1}{U_f} = \frac{1}{U_c} + R_{ft}$$

Outlet Temperature Calculation and Length of the Heat Exchanger

The outlet temperature for the fluid flowing through the tube is

$$T_{c2} = \frac{(\dot{m}c_p)_h (T_{h1} - T_{h2})}{(\dot{m}c_p)_c} + T_{c1}$$

The surface area of the heat exchanger for the fouled condition is :

$$A_f = \frac{\dot{Q}}{U_f (F)(LMTD)}$$

and for the clean condition

$$A_c = \frac{Q}{U_c(F)(LMTD)}$$

where the *LMTD* is always for the counter flow.

The over surface design (OS) can be calculated from :

$$OS = \frac{A_f}{A_c} = \frac{U_c}{U_f}$$

The length of the heat exchanger is calculated by

$$L = \frac{A_f}{N_t \pi d_o}$$

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

PROPERTIES: *Table A-6*, Water ($\overline{T}_m$ = 296 K): c_p = 4181 J/kg·K.

ANALYSIS: (a) The efficiency is

$$\eta = \frac{\dot{W}}{q} = \frac{2\,\text{MW}}{q} = 0.03.$$

Hence the required heat transfer rate is

$$q = \frac{2\,\text{MW}}{0.03} = 66.7\,\text{MW}.$$

Also

$$\Delta T_{\ell m,CF} = \frac{(300-290)-(292-290)\,°C}{\ell n \dfrac{300-290}{292-290}} = 5°C$$

t F = 1. Hence

$$A = \frac{q}{U\,F\,\Delta T_{\ell m,CF}} = \frac{6.67\times 10^{7}\ \text{W}}{1200\ \text{W/m}^2\cdot\text{K}\times 1\times 5°C}$$

$$A = 11{,}100\,\text{m}^2.$$

b) The water flow rate through the evaporator is

$$\dot{m}_h = \frac{q}{c_{p,h}\left(T_{h,i} - T_{h,o}\right)} = \frac{6.67 \times 10^7 \text{ W}}{4181 \text{ J/kg} \cdot \text{K}\left(300 - 292\right)}$$

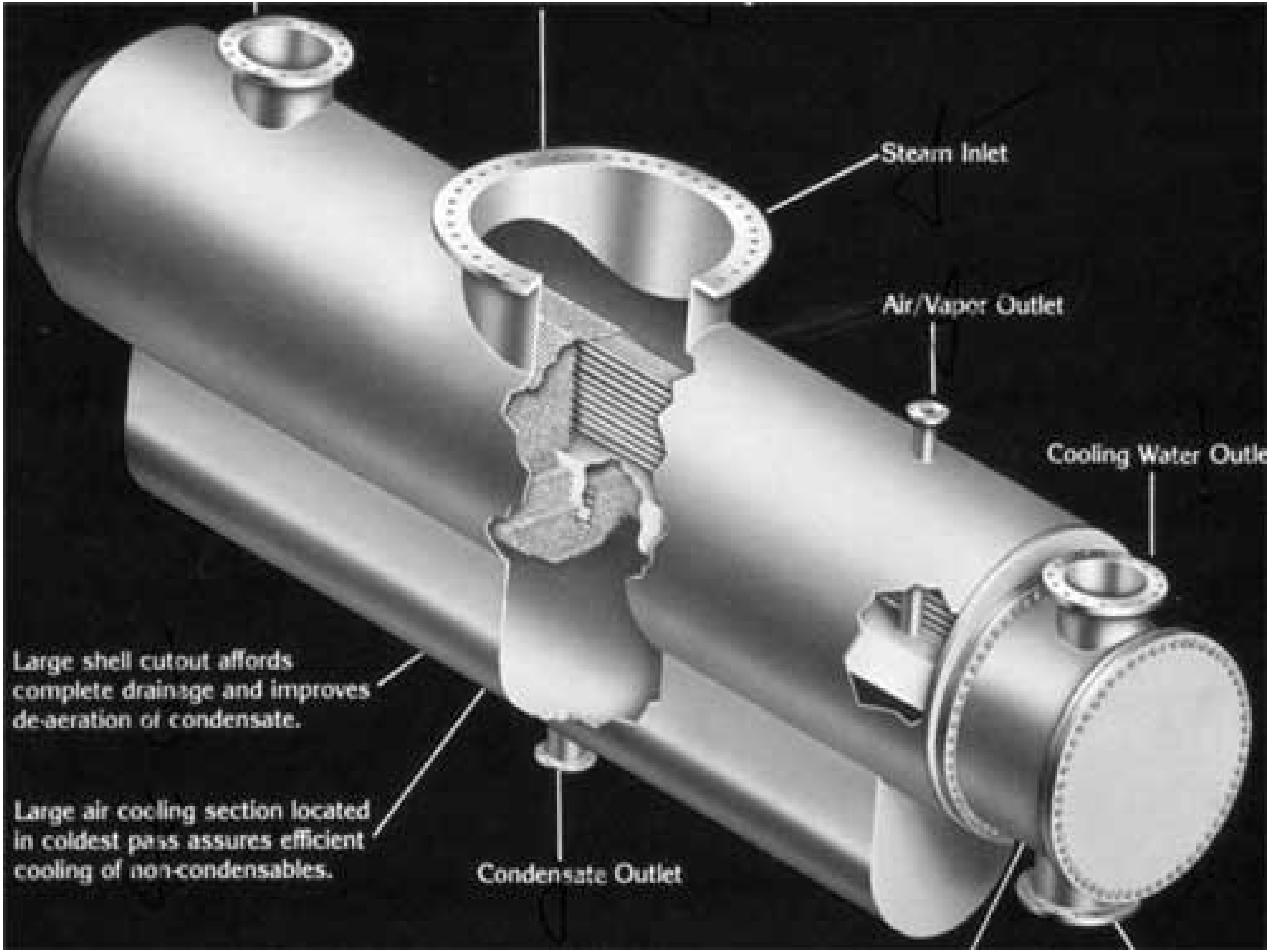
$$\dot{m}_h = 1994 \text{ kg/s.}$$

COMMENTS: (1) The required heat exchanger size is enormous due to the small temperature differences involved,

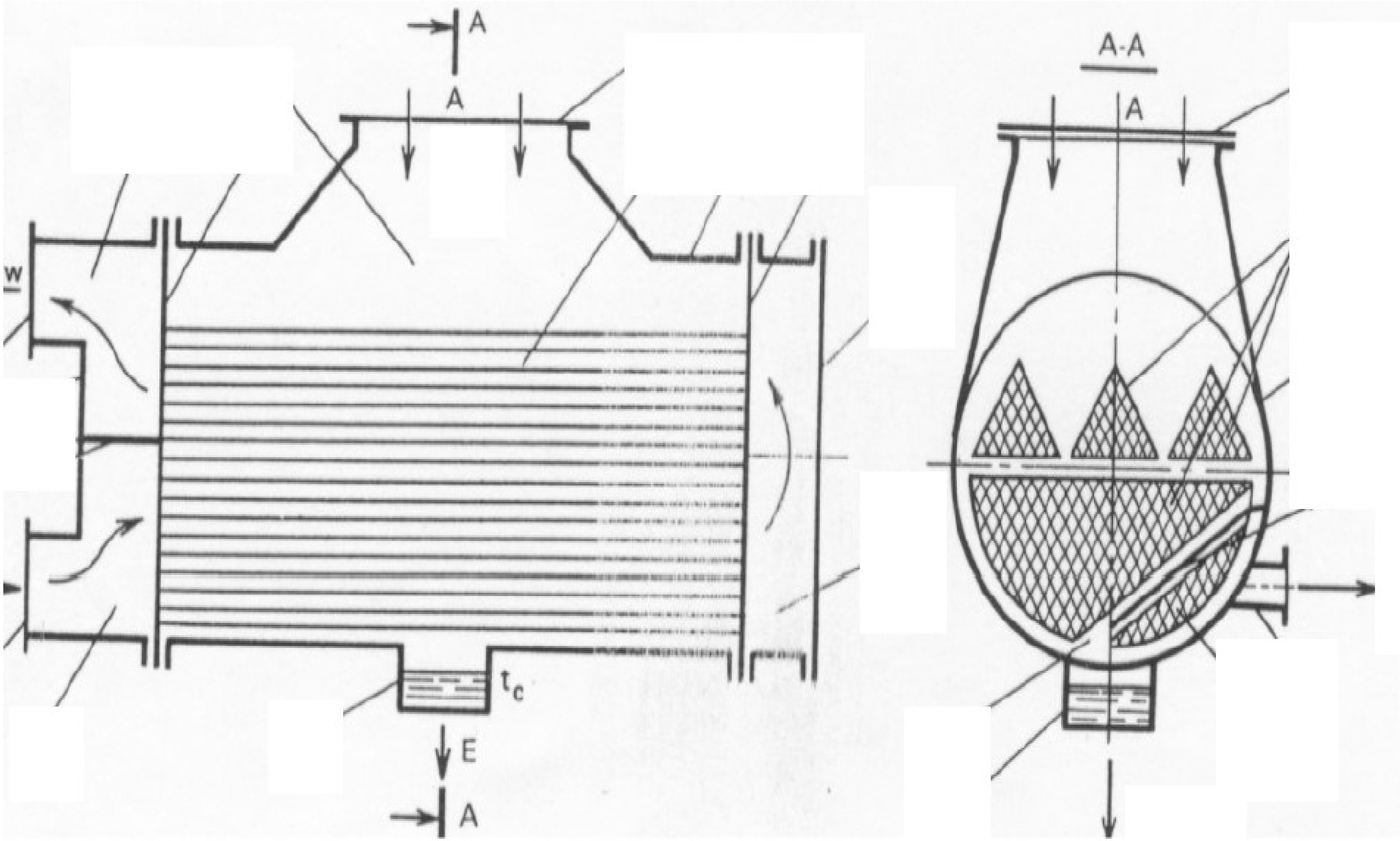
Performance Analysis of Condensers

Steam Condenser

- Steam condenser is a closed space into which steam exits the turbine and is forced to give up its latent heat of vaporization.
- It is a necessary component of a steam power plant because of two reasons.
- It converts dead steam into live feed water.
- It lowers the cost of supply of cleaning and treating of working fluid.
- It is far easier to pump a liquid than a steam.
- It increases the efficiency of the cycle by allowing the plant to operate on largest possible temperature difference between source and sink.
- The steam's latent heat of condensation is passed to the water flowing through the tubes of condenser.
- After steam condenses, the saturated water continues to transfer heat to cooling water as it falls to the bottom of the condenser called, hotwell.
- This is called subcooling and certain amount is desirable.
- The difference between saturation temperature corresponding to condenser vaccum and temperature of condensate in *hotwell* is called condensate depression.


Steam Inlet
Air/Vapor Outlet
Cooling Water Outle
Large shell cutout affords
complete drainage and improves
de-aeration of condensate.
Large air cooling section located
in coldest pass assures efficient
cooling of non-condensables.
Condensate Outlet

Two-Pass Surface Condenser



Energy Balance of A Condenser

- Energy balance:

$$\dot{m}_c\left(h_c - h_c^e\right) = \dot{m}_{CW}\, C_W\left(T_{We} - T_{Wi}\right)$$

- The temperature rise of cooling water:
- 6 to 7 degree C for single pass.
- 7 to 9 degree C for single pass.
- 10 to 12 degree C for four pass.

Overall Heat Transfer Coefficient for the Condenser

The overall heat transfer coefficient for clean surface (U_c) is given by

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

Considering the total fouling resistance, the heat transfer coefficient for fouled surface (U_f) can be calculated from the following expression:

$$\frac{1}{U_f} = \frac{1}{U_c} + R_{ft}$$

Cooling Water Outlet Temperature Calculation

The outlet temperature for the fluid flowing through the tube is

$$T_{cw,out} = \frac{\dot{m}_{steam}\left(h_{fg}\right)}{\dot{m}_{cw} c_{p,cw}} + T_{cw,in}$$

The surface area of the heat exchanger for the fouled condition is :

$$\dot{Q}_{transfer} = A_{surface} U_f F\left[LMTD\right] = \dot{m}\Delta h$$

Correlations for Condensing Heat Transfer

- Choice of a correlation depend on whether you are looking at horizontal or vertical tubes, and whether condensation is on the inside or outside.
- **Preliminaries**
- The *condensate loading* on a tube is the mass flow of condensate per unit length that must be traversed by the draining fluid.
- The length dimension is perpendicular to the direction the condensate flows;
- the perimeter for vertical tubes,
- the length for horizontal tubes.

Condensate Loading

$$\Gamma = \frac{\text{Mass flor of condensate}}{\text{Perimeter}}$$

$$\Gamma = \frac{\dot{m}_{condensate}}{\pi d_0} \quad \text{for vertical tubes.}$$

$$\Gamma = \frac{\dot{m}_{condensate}}{L_{tube}} \quad \text{for horizontal tubes.}$$

This can be used to calculate a Reynolds number

$$\text{Re}_{condensation} = \frac{4\Gamma}{\mu_{film}}$$

•Flow is considered laminar if this Reynolds number is less than 1800.
•The driving force for condensation is the temperature difference between the cold wall surface and the bulk temperature of the saturated vapor

$$\Delta T_{driving} = T_{sat} - T_{wall} \approx T_{vapour} - T_{surface}$$

The viscosity and most other properties used in the condensing correlations are evaluated at the *film temperature*, a weighted mean of the cold surface (wall) temperature and the (hot) vapor saturation temperature

$$T_{film} = T_{sat} - \frac{3}{4}\left(T_{saturation} - T_{wall}\right) = T_{sat} - \frac{3\Delta T_{driving}}{4}$$

Wall Temperatures

- It is often necessary to calculate the wall temperature by an iterative approach.
- The summarized procedure is:

1. Assume a film temperature, T_f
2. Evaluate the fluid properties (viscosity, density, etc.) at this temperature
3. Use the properties to calculate a condensing heat transfer coefficient (using the correlations to be presented)
4. Calculate the wall temperature. The relationship will typically be something like

$$T_{wall} = T_{sat} - \left\{ \frac{1/UA}{1/h_o A_o} \right\} \left(T_{sat} - T_{coolant} \right)$$

5. Use the wall temperature to calculate a film temperature
6. Compare the calculated film temperature to that from the initial step. If not equal, reevaluate the properties and repeat.

Laminar Flow Outside Vertical Tubes

If condensation is occurring on the outside surface of vertical tubes, with a condensate loading such that the condensate Reynolds Number is less than 1800, the recommended correlation is:

$$h_{cond} = \frac{1.47}{\sqrt[3]{\mathrm{Re}_{condensation}}} \left\{ \frac{k_f^3 \rho_f \left(\rho_f - \rho_v\right) g}{\mu_f^{\ 2}} \right\}^{1/3}$$

- Since the vapor density is usually much smaller than that of the condensate film, some authors neglect it and use the film density squared in the denominator.
- The presence of ripples (slight turbulence) improves heat transfer, so some authors advocate increasing the value of the coefficient by about 20%.

Another form of writing *h* is :

$$h_{cond} = 0.925\left\{\frac{k_f^3 \rho_f \left(\rho_f - \rho_v\right) g}{\mu_f \Gamma}\right\}^{1/3}$$

this may also be compensated for rippling (0.925*1.2=1.11).

Turbulent Flow Outside Vertical Tubes

When the condensate Reynolds Number is greater than 1800, the recommended correlation is :

$$h_{cond} = 0.0076\,\mathrm{Re}^{0.4}\left\{\frac{k_f^3 \rho_f \left(\rho_f - \rho_v\right) g}{\mu_f^{\,2}}\right\}^{1/3}$$

Laminar Flow Outside Horizontal Tubes

When vapor condenses on the surface of horizontal tubes, the flow is almost always laminar.
The flow path is too short for turbulence to develop. Again, there are two forms of the same relationship:

$$h_{cond} = \frac{1.51}{\sqrt[3]{\mathrm{Re}_{condensation}}} \left\{ \frac{k_f^3 \rho_f (\rho_f - \rho_v) g}{\mu_f^{\;2}} \right\}^{1/3}$$

$$h_{cond} = 0.725 \left\{ \frac{k_f^3 \rho_f (\rho_f - \rho_v) g h_{fg}}{\mu_f \Delta T_{driving} d_0} \right\}^{1/4}$$

The constant in the second form varies from 0.725 to 0.729.
The rippling condition (add 20%) is suggested for condensate Reynolds Numbers greater than 40.

Condenser tubes are typically arranged in banks, so that the condensate which falls off one tube will typically fall onto a tube below.

The bottom tubes in a stack thus have thicker liquid films and consequently poorer heat transfer.

The correlation is adjusted by a factor for the number of tubes, becoming for the Nth tube in the stack

$$h_{cond} = 0.725\left\{\frac{k_f^3 \rho_f \left(\rho_f - \rho_v\right) g h_{fg}}{N \mu_f \Delta T_{driving} d_0}\right\}^{1/4} = \frac{h_{top}}{\sqrt[4]{N}}$$

Splashing of the falling fluid further reduces heat transfer, so some authors recommend a different adjustment

$$h_{cond} = 0.725\left\{\frac{k_f^3 \rho_f \left(\rho_f - \rho_v\right) g h_{fg}}{N \mu_f \Delta T_{driving} d_0}\right\}^{1/4} = \frac{h_{top}}{\sqrt[6]{N}}$$

Overall Heat Transfer Coefficient for the Condenser

The overall heat transfer coefficient for clean surface (U_c) is given by

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

Considering the total fouling resistance, the heat transfer coefficient for fouled surface (U_f) can be calculated from the following expression:

$$\frac{1}{U_f} = \frac{1}{U_c} + R_{ft}$$