
بسم الله الرحمن الرحیم

Designing Shell-and-Tube Heat Exchangers Using Softwares

Lecture 1: Kern's Method

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2014

Schedule – Kern’s Method

Kern’s method

- Introduction to Kern’s method
- Algorithm of design procedure for shell-and-tube heat exchangers
- Design procedure steps along with an example

Objectives

- This lecture on designing shell-and-tube HEs serves as an introduction lecture to the subject, and covers:

  - Introduction to “Kern’s method” definition along with its advantages and disadvantages

  - Developing an algorithm for the design of shell-and-tube exchangers

  - Finally, following up the procedure set out in the algorithm in an example

Introduction to Kern's method

- **Kern's was based on experimental work on commercial exchanger**
- **Advantages:**
  - **Giving reasonably satisfactory prediction of the heat-transfer coefficient for standard design**
  - **Simple to apply**
  - **Accurate enough for preliminary design calculations**
  - **Accurate enough for designs when uncertainty in other design parameter is such that the use of more elaborate method is not justified**
- **Disadvantage:**
  - **The prediction of <u>*pressure drop*</u> is less satisfactory, as pressure drop is more affected by leakage and bypassing than heat transfer**
  - **The method does not take account of the <u>*bypass*</u> and <u>*leakage*</u> streams**

Design procedure for shell-and-tube heat exchangers (Kern's method)

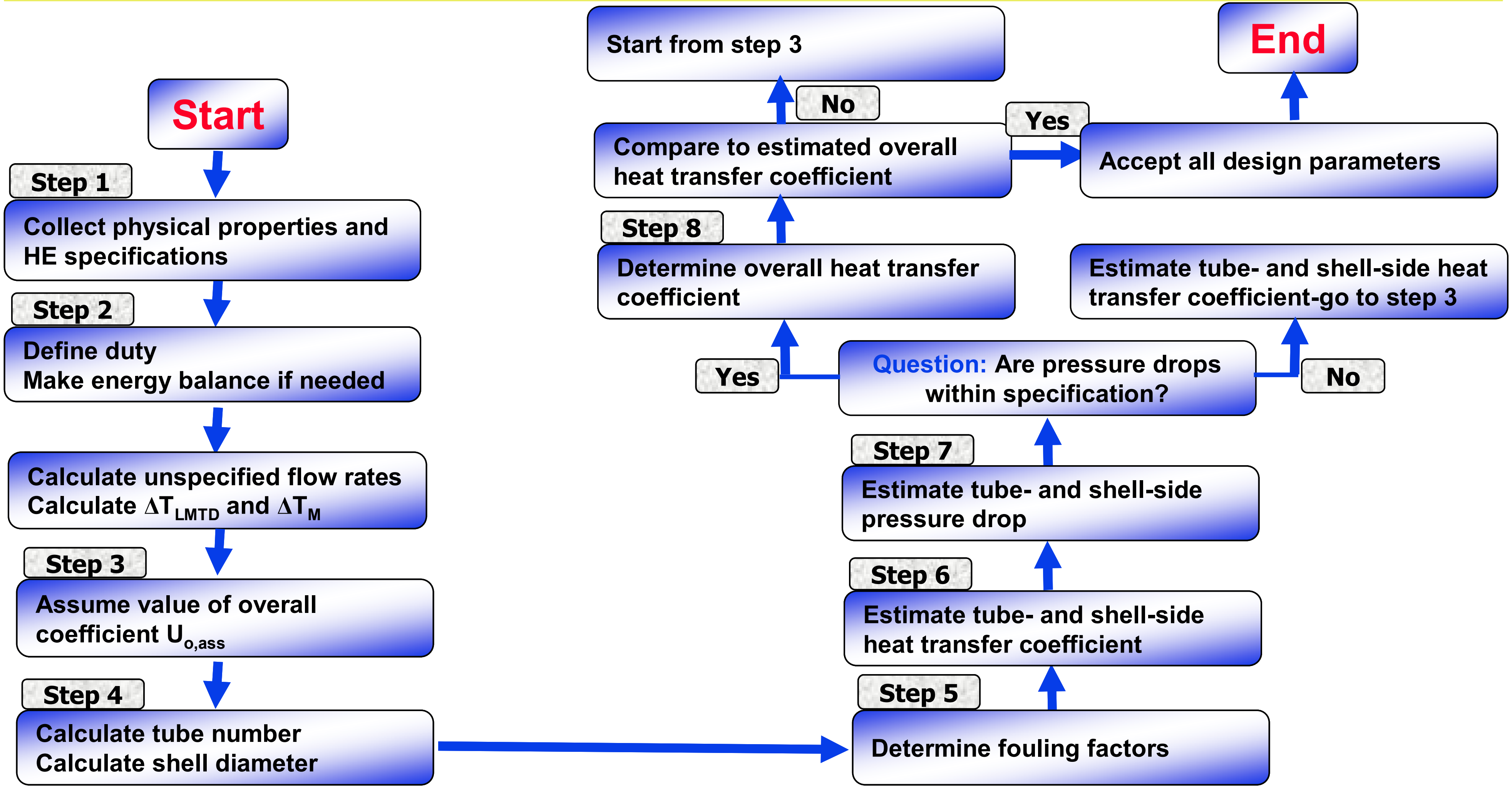

Fig. 1: Algorithm of design procedure

Kern's Method Design Example

- Design an exchanger to sub-cool condensate from a methanol condenser from 95 °C to 40 °C

  - Flow-rate of methanol 100,000 kg/h

  - Brackish water (seawater) will be used as the coolant, with a temperature rise from 25° to 40 °C

Solution: Step 1

Collect physical properties and HE specifications:

- Physical properties

Table 1

Physical properties at <u>fluid mean temperature</u>	Methanol	Water
C_p (Kj/Kg °C)	2.84	4.2
μ (mNs/m^2)	0.34	0.8
k_f (W/m °C)	0.19	0.59
ρ (Kg/m^3)	750	995

- HE specifications:
  - Coolant (brackish water) is corrosive, so assign to tube-side.
  - Use one shell pass and two tube passes.
  - At shell side, fluid (methanol) is relatively clean. So, use 1.25 triangular pitch (pitch: distance between tube centers).

Tube Arrangements

- The tubes in an exchanger are usually arranged in an equilateral triangular, square, or rotated square pattern (Fig. 2)

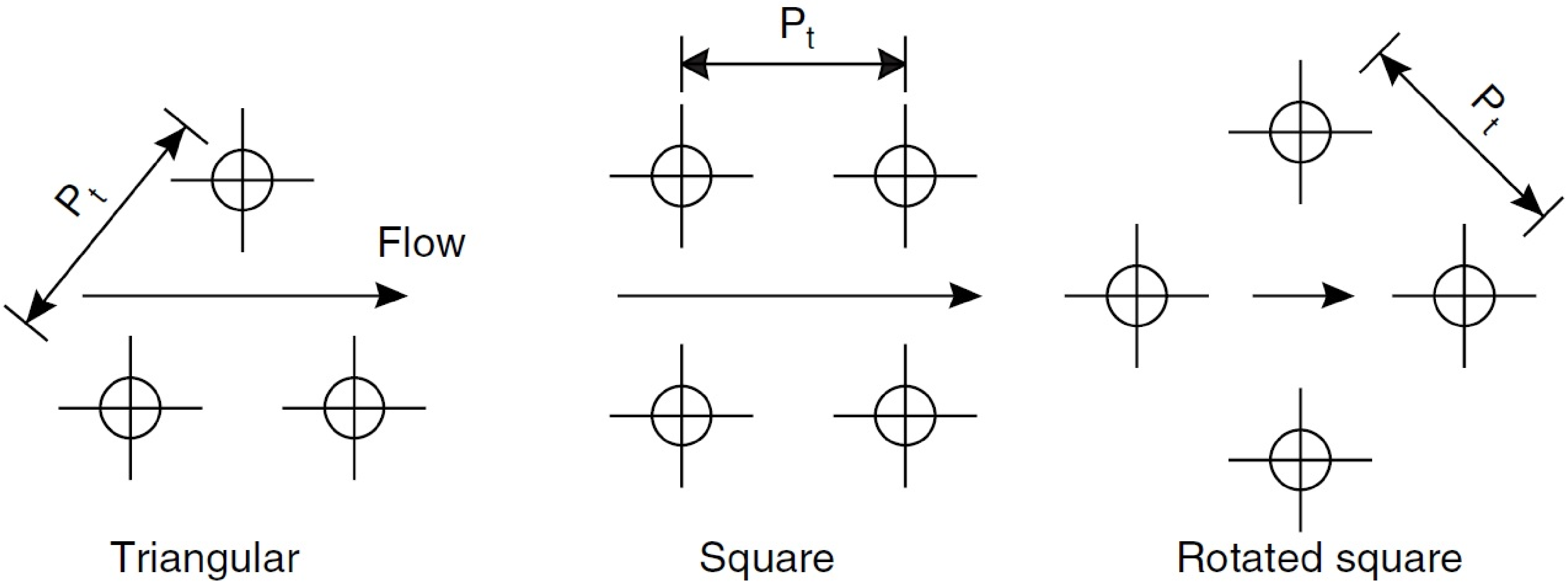

Fig. 2: Tube patterns

Tube Pattern Applications

- The triangular and rotated square patterns give **higher heat-transfer** rates, but at the expense of a **higher pressure drop** than the square pattern.
- A square, or rotated square arrangement, is used for **heavily fouling fluids**, where it is necessary to mechanically clean the outside of the tubes.
- The recommended tube ***pitch is 1.25 times the tube outside diameter***; and this will normally be used unless process requirements dictate otherwise.

Step 2

➤ Define duty, Make energy balance if needed

- To start step 2, the duty (heat transfer rate) of methanol (the hot stream or water, the cold stream) needed to be calculated.

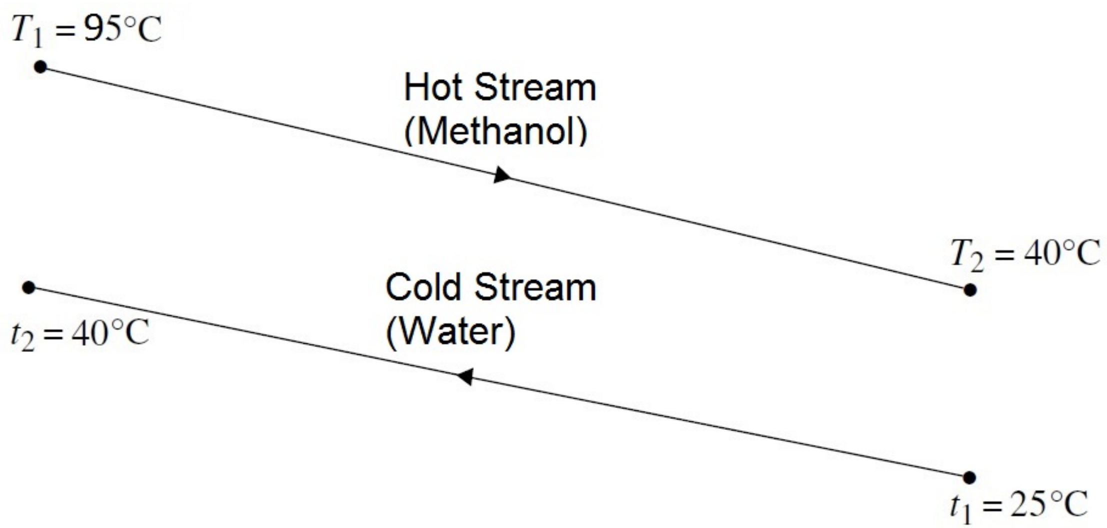

Fig. 3: Streams definitions.

$$\text{Heat load} = Q = \dot{m}_h \; C_{ph}(T_1 - T_2) = \frac{100000}{3600} \times 2.84\,(95 - 40) = 4340\ \text{kW}$$

Step 2 (Cont'd)

- The cold and the hot stream heat loads are equal. So, cooling water flow rate is calculated as follow:

$$\text{Cooling water flow} = \dot{m}_c = \frac{Q}{C_{P_c}\,(t_2 - t_1)} = \frac{4340}{4.2\,(40-25)} = 68.9\ \text{kg/s}$$

- The well-known "logarithmic mean" temperature difference (LMTD or lm) is calculated by:

$$\Delta T_{LMTD} = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln \dfrac{T_1 - t_2}{T_2 - t_1}} = \frac{(95-40)-(40-25)}{\ln \dfrac{(95-40)}{(40-25)}} = 31^\circ C$$

Mean Temperature Difference

- The usual practice in the design of shell and tube exchangers is to estimate the "***true temperature difference***" from the logarithmic mean temperature by applying a correction factor to allow for the ***departure from true counter-current flow***:

$$\Delta T_m = F_t \, \Delta T_{LMTD}$$

- **Where:**

  - **ΔT_m = true temperature difference,**
  - **F_t = the temperature correction factor.**

Temperature Correction Factor

- The correction factor (F_t) is a function of the ***shell and tube fluid temperatures***, and the **number of tube and shell passes**.
- It is normally correlated as a function of two dimensionless temperature ratios:

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$

$$S = \frac{t_2 - t_1}{T_1 - t_1}$$

Step 2

➤ For a ***1 shell : 2 tube pass exchanger***, the correction factor is plotted in Fig. 4.

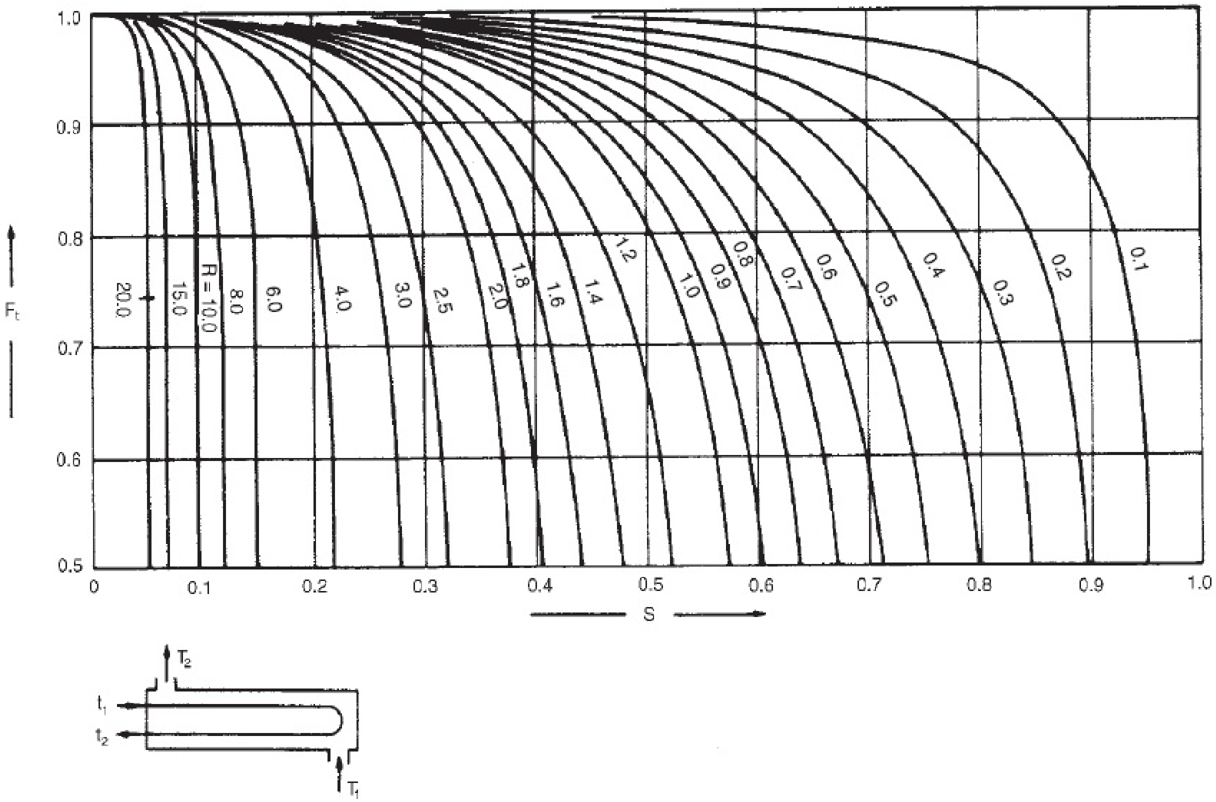

Fig. 4: Temperature correction factor: one shell pass; two or more even tube passes (available in TEMA)

Step 2 (Cont'd)

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{95 - 40}{40 - 25} = 3.67$$

$$S = \frac{(t_2 - t_1)}{(T_1 - t_1)} = \frac{40 - 25}{95 - 25} = 0.21$$

- **From Fig. 4, the correction factor (F_t) is 0.85.**

$$\Delta T_m = F_t \, \Delta T_{LMTD} = 0.85 \times 31 = 26 \ ^\circ C$$

Step 3

- **Assume value of overall coefficient $U_{o,ass}$**
  - **Typical values of the overall heat-transfer coefficient for various types of heat exchanger are given in Table 1.**
  - **Fig. 5 can be used to estimate the overall coefficient for tubular exchangers (shell and tube).**
  - **The film coefficients given in Fig. 5 include an allowance for fouling.**
  - **The values given in Table 1 and Fig. 5 can be used for the preliminary sizing of equipment for process evaluation, and as trial values for starting a detailed thermal design.**
- **From Table 2 or Fig. 5:** $U = 600\ W/m^2°C$

Step 3 (Cont'd)

Table 2: Typical overall coefficients

Shell and tube exchangers		
Hot fluid	Cold fluid	U (W/m^2 °C)
Heat exchangers		
Water	Water	800–1500
Organic solvents	Organic solvents	100–300
Light oils	Light oils	100–400
Heavy oils	Heavy oils	50–300
Gases	Gases	10–50
Coolers		
Organic solvents	Water	250–750
Light oils	Water	350–900
Heavy oils	Water	60–300
Gases	Water	20–300
Organic solvents	Brine	150–500
Water	Brine	600–1200
Gases	Brine	15–250
Heaters		
Steam	Water	1500–4000
Steam	Organic solvents	500–1000
Steam	Light oils	300–900
Steam	Heavy oils	60–450
Steam	Gases	30–300
Dowtherm	Heavy oils	50–300
Dowtherm	Gases	20–200
Flue gases	Steam	30–100
Flue	Hydrocarbon vapours	30–100
Condensers		
Aqueous vapours	Water	1000–1500
Organic vapours	Water	700–1000
Organics (some non-condensables)	Water	500–700
Vacuum condensers	Water	200–500
Vaporisers		
Steam	Aqueous solutions	1000–1500
Steam	Light organics	900–1200
Steam	Heavy organics	600–900
Air-cooled exchangers		
Process fluid		
Water		300–450
Light organics		300–700
Heavy organics		50–150
Gases, 5–10 bar		50–100
10–30 bar		100–300
Condensing hydrocarbons		300–600
Immersed coils		
Coil	Pool	
Natural circulation		
Steam	Dilute aqueous solutions	500–1000
Steam	Light oils	200–300
Steam	Heavy oils	70–150
Water	Aqueous solutions	200–500
Water	Light oils	100–150

(continued)

Table 1. (*continued*)

Immersed coils		
Coil	Pool	U (W/m^2 °C)
Agitated		
Steam	Dilute aqueous solutions	800–1500
Steam	Light oils	300–500
Steam	Heavy oils	200–400
Water	Aqueous solutions	400–700
Water	Light oils	200–300
Jacketed vessels		
Jacket	Vessel	
Steam	Dilute aqueous solutions	500–700
Steam	Light organics	250–500
Water	Dilute aqueous solutions	200–500
Water	Light organics	200–300
Gasketed-plate exchangers		
Hot fluid	Cold fluid	
Light organic	Light organic	2500–5000
Light organic	Viscous organic	250–500
Viscous organic	Viscous organic	100–200
Light organic	Process water	2500–3500
Viscous organic	Process water	250–500
Light organic	Cooling water	2000–4500
Viscous organic	Cooling water	250–450
Condensing steam	Light organic	2500–3500
Condensing steam	Viscous organic	250–500
Process water	Process water	5000–7500
Process water	Cooling water	5000–7000
Dilute aqueous solutions	Cooling water	5000–7000
Condensing steam	Process water	3500–4500

Step 3 (Cont'd)

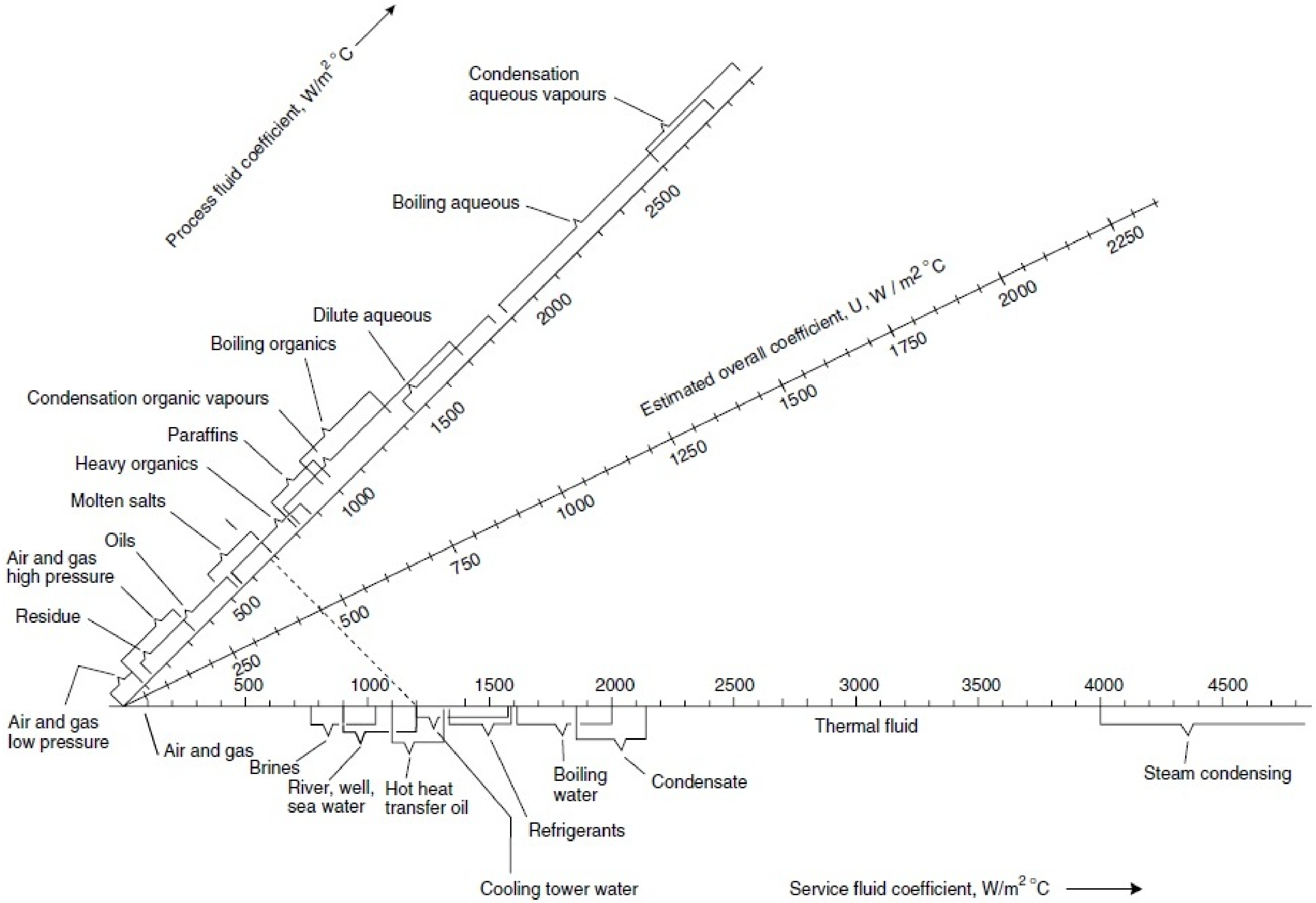

Fig. 5: Overall coefficients (join process side duty to service side and read U from centre scale)

Step 4

Calculate tube number, Calculate shell diameter

- **Provisional area:**

$$A = \frac{Q}{U\,\Delta T_M} = \frac{4340 \times 10^3}{600 \times 26} = 278\ m^2$$

- **So, the total outside surface area of tubes is 278 m^2**

- **Choose *20 mm o.d.* (outside diameter), *16 mm i.d.* (inside diameter), *4.88-m-long tubes* ($\frac{3}{4}$ in. $\times$ 16 ft), cupro-nickel.**
- **Allowing for tube-sheet thickness, take tube length: L= 4.83 m**

- **Surface area of one tube: A = πDL = 4.83 x 20 x $10^{-3}\pi$ = 0.303 m^2**

- $$\text{Numbers of tubes} = \frac{\text{Total outside surface area of tubes (Provisional area)}}{\text{Outside surface area of one tube}} = \frac{278}{0.303} = \underline{\underline{918}}$$

Step 4 (Cont'd)

- An estimate of the bundle diameter D_b can be obtained from equation below which is an empirical equation based on standard tube layouts. The constants for use in this equation, for triangular and square patterns, are given in Table 3.

$$D_b = d_o \left(\frac{N_t}{K_1}\right)^{1/n_1}$$

- where D_b = bundle diameter in mm, do = tube outside diameter in mm., N_t = number of tubes.
- As the shell-side fluid is relatively clean use 1.25 triangular pitch.
- So, for this example:

$$\text{Bundle diameter} \quad D_b = 20 \left(\frac{918}{0.249}\right)^{1/2.207} = 826 \text{ mm}$$

Step 4 (Cont'd)

Table 3: Constants K_1 and n_1

Triangular pitch, $p_t = 1.25d_o$					
No. passes	1	2	4	6	8
K_1	0.319	0.249	0.175	0.0743	0.0365
n_1	2.142	2.207	2.285	2.499	2.675
Square pitch, $p_t = 1.25d_o$					
No. passes	1	2	4	6	8
K_1	0.215	0.156	0.158	0.0402	0.0331
n_1	2.207	2.291	2.263	2.617	2.643

Step 4 (Cont'd)

- Use a split-ring floating head type for Fig. 6.
- From Fig. 6, bundle diametrical clearance is 68 mm.
- Shell diameter (D_s):

  D_s= Bundle diameter + Clearance = 826 + 68 = 894 mm.
- Note 1: nearest standard pipe size are 863.6 or 914.4 mm.
- Note 2: Shell size could be read from standard tube count tables [Kern (1950), Ludwig (2001), Perry et al. (1997), and Saunders (1988)].

Step 4 (Cont'd)

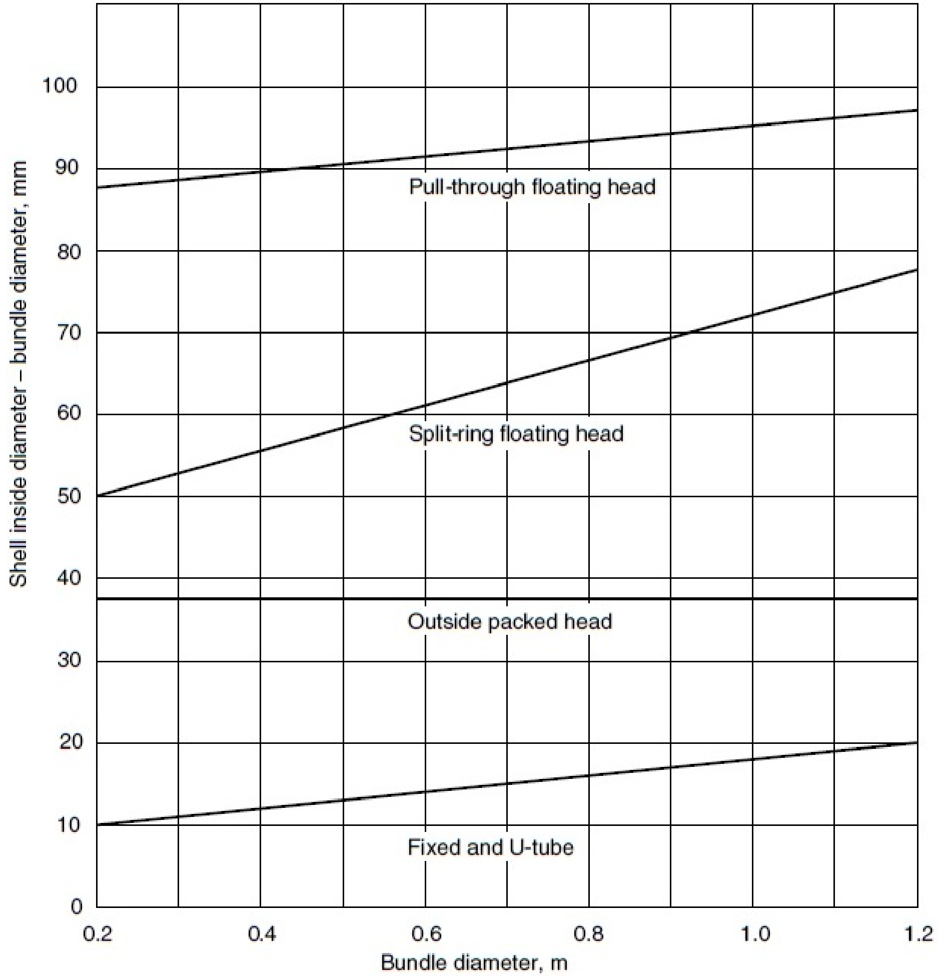

Fig. 6: Shell-bundle clearance

Step 6

➤ Estimate tube- and shell-side heat transfer coefficient

- **Tube-side heat transfer coefficient:**

$$\text{Mean water temperature } (T_{avg}) = \frac{40+25}{2} = 33\ ^{\circ}\text{C} \Rightarrow \rho = 995\ \text{kg}/\text{m}^3$$

$$\text{Tube cross - sectional area (a)} = \frac{\pi}{4} D^2 = \frac{\pi}{4} \times 16^2 = 201\,\text{mm}^2$$

- Since we have two tubes pass, we divide the total numbers of tubes by two to find the numbers of tubes per pass, that is:

$$\text{Tubes per pass} = \frac{918}{2} = 459$$

- Total flow area is equal to numbers of tubes per pass multiply by tube cross sectional area:

$$\text{Total flow area} = 459 \times (201 \times 10^{6}) = 0.092\ \text{m}^2$$

Step 6 (Cont'd)

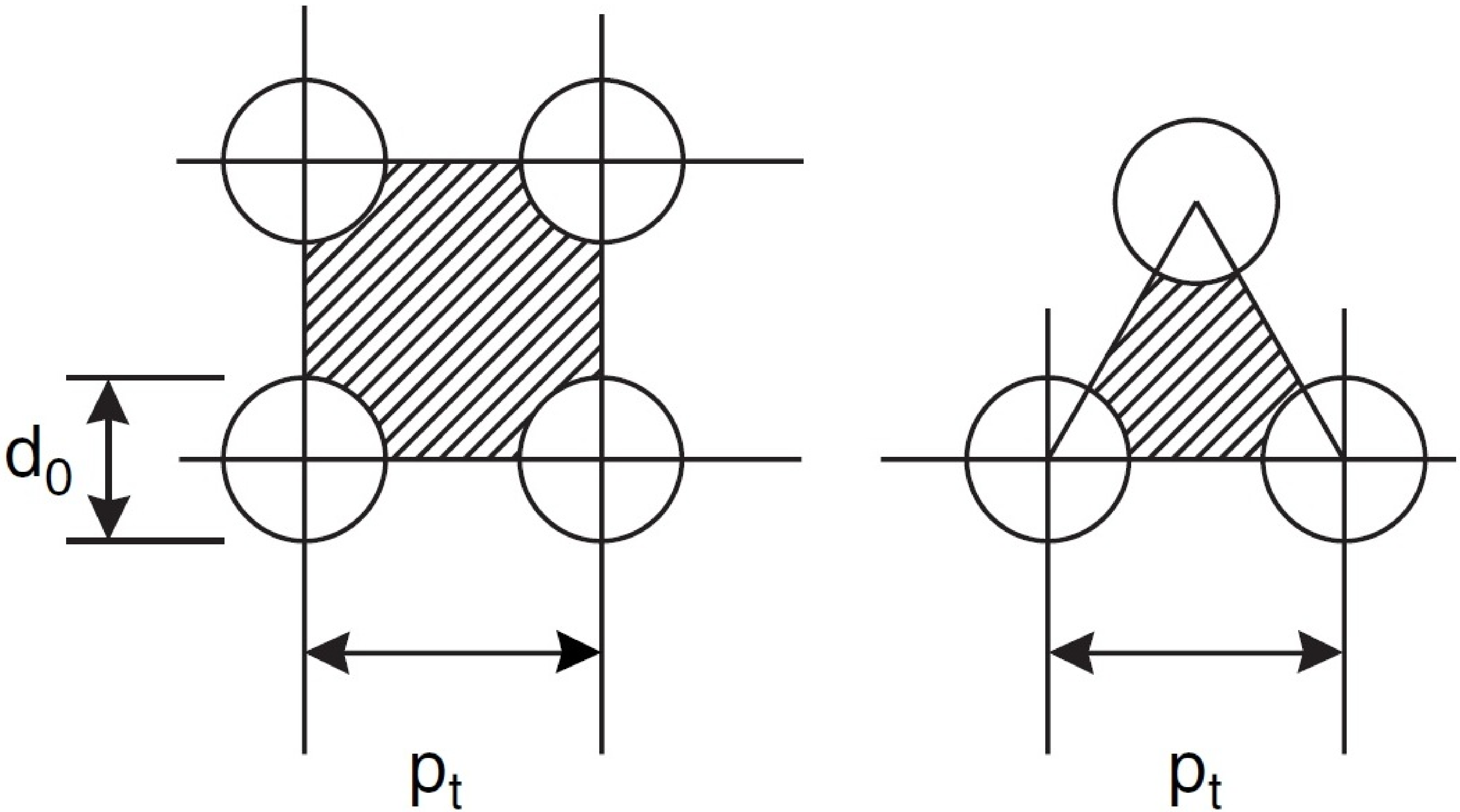

Fig. 7: Equivalent diameter, cross-sectional areas and wetted perimeters.

Step 6 (Cont'd)

- Water mass velocity $= \dfrac{\text{Cooling water flow}}{\text{Total flow area}} = \dfrac{68.9}{0.092} = 749\ \mathrm{kg/s\,m^2}$
- Water linear velocity $(u_t) = \dfrac{\text{Water mass velocity }(G_t)}{\text{Water density }(\rho)} = \dfrac{749}{995} = 0.75\ \mathrm{m/s}$

- ***Coefficients for water***: a more accurate estimate can be made by using equations developed specifically for water.
- The physical properties are conveniently incorporated into the correlation. The equation below has been adapted from data given by Eagle and Ferguson (1930):

$$h_i = \frac{4200\,(1.35 + 0.02t)\,u^{0.8}}{d_i^{0.2}}$$

  - where h_i = inside coefficient, for water, W/m² °C,
  - $\boxed{t = \text{water temperature, °C,}}$
  - u_t = water linear velocity, m/s,
  - d_i = tube inside diameter, mm.

Step 6 (Cont'd)

- $$h_i = \frac{4200\,(1.35 + 0.02t)\,u^{0.8}}{d_i^{0.2}} = \frac{4200\,(1.35 + 0.02 \times 33)\,0.75^{0.8}}{16^{0.2}} = 3852\ \mathrm{W/m^2\,^\circ C}$$

- The equation can also be calculated using equation below; this is done to illustrate use of this method.

$$\frac{h_i\,d_i}{k_f} = j_h\,Re\,Pr^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

  - where h_i = inside coefficient, for water, W/m² °C,
  - d_i = tube inside diameter, mm
  - k_f = fluid thermal conductivity, W/m² °C
  - j_h = heat transfer factor, dimensionless
  - Re = Reynolds number, dimensionless
  - Pr = Prandtl number, dimensionless
  - μ = viscosity of water, N s/m²
  - μ_w = viscosity of water at wall temperature, N s/m²

Step 6 (Cont'd)

- Viscosity of water (μ) from Table 1 $= 0.8\,\mathrm{mNs/m^2}$

  Fluid thermal conductivity from Table 1 $= 0.59\ \mathrm{W/m^\circ C}$

$$Re = \frac{\rho u d_i}{\mu} = \frac{995 \times 0.75 \times 16 \times 10^{\;3}}{8 \times 10^{-3}} = 14925$$

$$Pr = \frac{C_p \mu}{k_f} = \frac{4.2 \times 10^{3} \times 0.8 \times 10^{-3}}{0.59} = 5.7$$

- Neglect $\left(\frac{\mu}{\mu_w}\right)$

- $\frac{L}{d_i} = \frac{4.83 \times 10^3}{16} = 302 \Rightarrow$ From Fig. 8, $j_h = 3.9 \times 10^{-3}$

- $h_i = \frac{k_f}{d_i} j_h \, Re \, Pr^{0.33} \left(\frac{\mu}{\mu_w}\right)^{0.14} = \frac{0.59}{16 \times 10^{-3}} \times 3.9 \times 10^{-3} \times 14925 \times 5.7^{0.33} \times 1^{\,0.14} = 3812\ \mathrm{W/m^{2\circ} C}$

- Check reasonably the previously calculated value 3812 W/m^2°C with value calculated, 3852 W/m^2°C.

Step 6 (Cont'd)

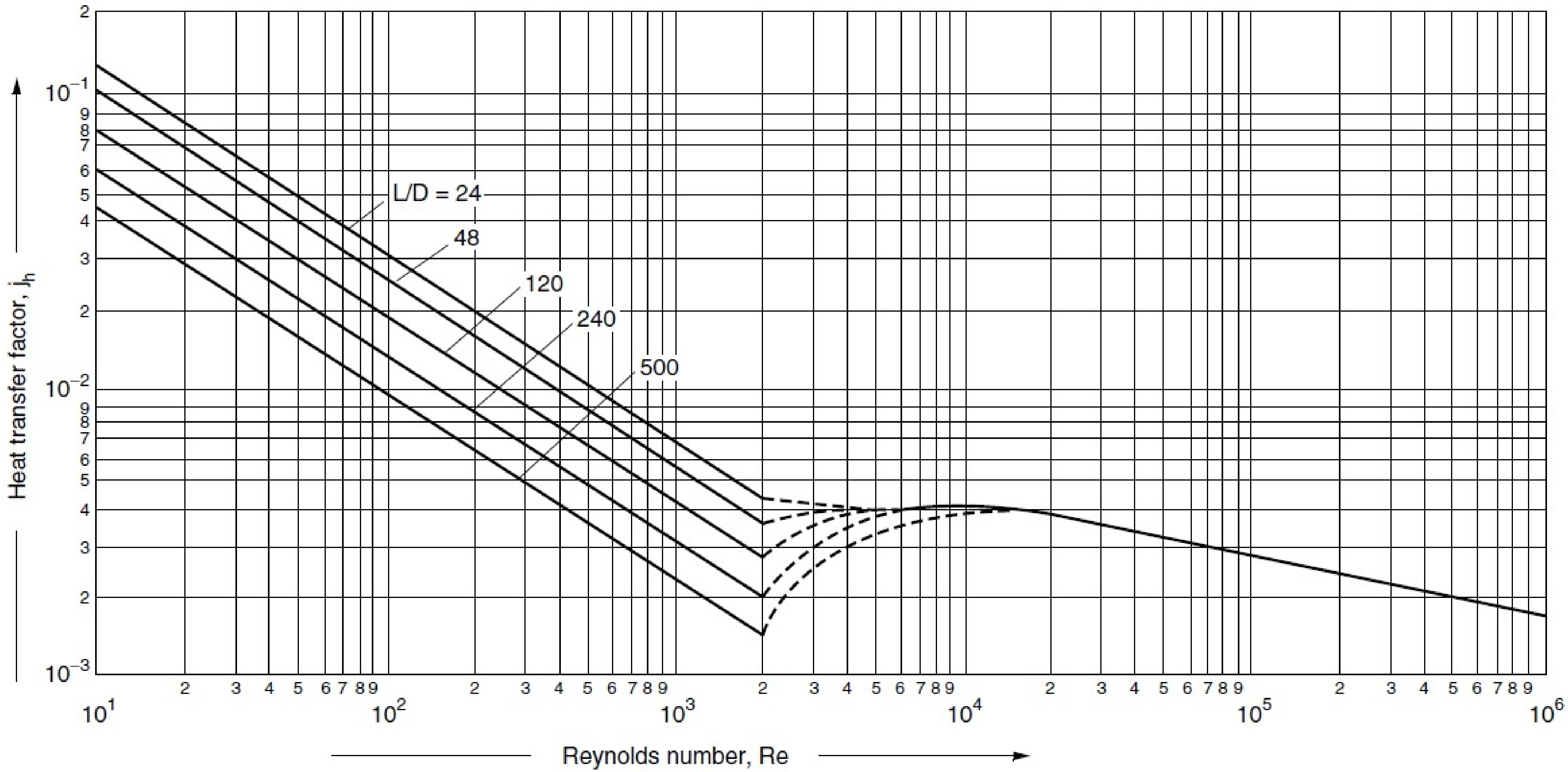

Fig. 8: Tube-side heat-transfer factor

Step 6 (Cont'd)

- **Shell-side heat transfer coefficient:**
- ***Baffle spacing:*** The baffle spacings used range from 0.2 to 1.0 shell diameters.
- A close baffle spacing will give higher heat transfer coefficients but at the expense of higher pressure drop.
- ***Area for cross-flow:*** calculate the area for cross-flow A_s for the hypothetical row at the shell equator, given by:

$$\boxed{A_s = \frac{(p_t - d_o)D_s l_b}{p_t}}$$

- Where p_t = tube pitch (distance between the centers of two tubes, Fig. 7).
- d_o = tube outside diameter, m,
- D_s = shell inside diameter, m,
- l_b = baffle spacing, m.
- Note: the term $(p_t - d_o)/p_t$ is the ratio of the clearance between tubes and the total distance between tube centers.

Step 6 (Cont'd)

- ***Baffle spacing:***

  Choose baffle spacing = 0.2 D_s=0.2 □ 894 = 178 mm

- ***Tube pitch:***

  P_t = 1.25 d_o= 1.25 □ 20 = 25 mm

- ***Cross-flow area:***

$$A_s = \frac{(p_t - d_o)}{p_t} D_s l_b = \frac{(25-20)}{25} \times 894 \times 178 \times 10^{-6} = 0.032 \ m^2$$

Step 6 (Cont'd)

- ***Shell-side mass velocity G_s and the linear velocity u_t:***

$$G_s = \frac{W_s}{A_s}$$

$$u_s = \frac{G_s}{\rho}$$

- Where W_s = fluid flow-rate on the shell-side, kg/s,
- ρ = shell-side fluid density, kg/m^3.

- ***Shell equivalent diameter (hydraulic diameter):*** calculate the shell-side equivalent diameter, see Fig. 7. For an equilateral triangular pitch arrangement:

$$d_e = \frac{4\left(\frac{p_t}{2} \times 0.87 p_t - \frac{1}{2}\pi\frac{d_o^2}{4}\right)}{\frac{\pi d_o}{2}} = \frac{1.10}{d_o}\left(p_t^2 - 0.917\, d_o^2\right)$$

- Where d_e = equivalent diameter, m.

Step 6 (Cont'd)

- ***Shell-side mass velocity G_s:***

$$\text{Mass velocity, } G_s = \frac{W_s}{A_s} = \frac{100000}{3600} \times \frac{1}{0.032} = 868 \, {}^{kg}\!/_{s\,m^2}$$

- ***Shell equivalent diameter (hydraulic diameter):***

$$d_e = \frac{1.10}{d_o}(p_t^2 - 0.917\, d_o^2) = \frac{1.1}{20}(25^2 - 0.917 \times 20^2) = 14.4 \text{ mm}$$

Step 6 (Cont'd)

- Mean shell side temperature $= \dfrac{95+40}{2} = 68\ {}^\circ\mathrm{C}$

  Methanol density (ρ) from Table 1 $= 750\ \mathrm{kg/m^3}$)

  Viscosity of methanol (μ from Table 1 $= 0.34\ \mathrm{mNs/m^2}$)

  Heat capacity from Table 1 $= 2.84\ \mathrm{kJ/kg^\circ C}$

  Thermal conductivity Table 1 $= 0.19\ \mathrm{W/m^\circ C}$

  $$\mathrm{Re} = \frac{\rho u_s d_e}{\mu} = \frac{G_s d_e}{\mu} = \frac{868 \times 14.4 \times 10^{-3}}{0.34 \times 10^{-3}} = 36762$$

  $$\mathrm{Pr} = \frac{C_p \mu}{k_f} = \frac{2.84 \times 10^{3} \times 0.34 \times 10^{-3}}{0.19} = 5.1$$

- Choose 25 per cent baffle cut, from Fig. 9

  $$j_h = 3.3 \times 10^{-3}$$

Step 6 (Cont'd)

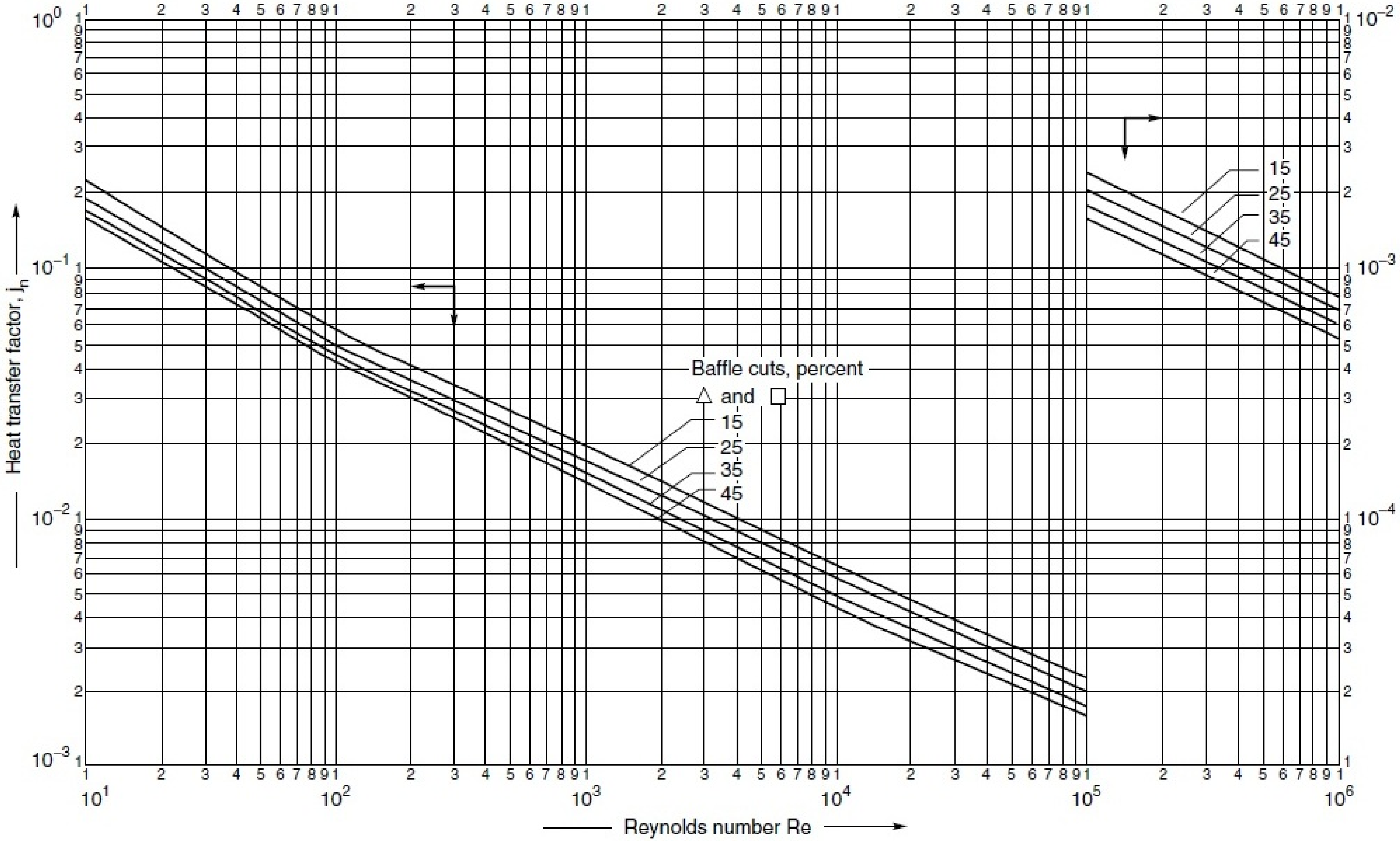

Fig. 9: Shell-side heat-transfer factors, segmental baffles

Step 6 (Cont'd)

- For the calculated Reynolds number, the read value of j_h from Fig. 9 for 25 per cent baffle cut and the tube arrangement, we can now calculate the shell-side heat transfer coefficient h_s from:

$$Nu = \frac{h_s d_e}{k_f} = j_h \, Re \; Pr^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \quad \text{(without viscosity correction term)} \rightarrow h_s = \frac{0.19}{1.44 \times 10^{-3}} \times 3.3 \times 10^{-3} \times 36762 \times 5.1^{1/3}$$
$$= 2740$$

- The tube wall temperature can be estimated using the following method:

  Mean temperature difference across all resistance: 68 -33 =35 °C

$$\text{across methanol film} = \frac{U}{h_o} \times \Delta T = \frac{600}{2740} \times 35 = 8\ ^\circ C$$

- Mean wall temperature = 68 – 8 = 60 °C

$$\mu = 0.37 \text{ mNs/m}^2$$

$$\left(\frac{\mu}{\mu_w}\right)^{0.14} = 0.99$$

Which shows that the correction for low-viscosity fluid is not significant.

Step 7 (Cont'd)

Pressure drop

- Tube side: From Fig. 10, for Re = 14925

$$j_f = 4.3 \square 10^{-3}$$

- Neglecting the viscosity correction term:

$$\Delta P_t = N_p[8j_f(\frac{L}{d_i})(\frac{\mu}{\mu_w})^{-m} + 2.5]\frac{\rho u_t^2}{2}$$

$$= 2\,(8 \times 4.3 \times 10^{-3}\,(\frac{4.83 \times 10^3}{16}) + 2.5)\,\frac{995 \times 0.75^2}{2}$$

$$= 7211\,\mathrm{N/m^2} = 7.2\,\mathrm{kPa}\ (1.1\ \mathrm{psi})$$

- low, could consider increasing the number of tube passes.
- Shell side

$$\text{Linear velocity} = \frac{G_s}{\rho} = \frac{868}{750} = 1.16\ \mathrm{m/s}$$

- From Fig. 11, for Re = 36762

$$j_f = 4 \square 10^{-2}$$

- Neglect viscosity correction

Step 7 (Cont'd)

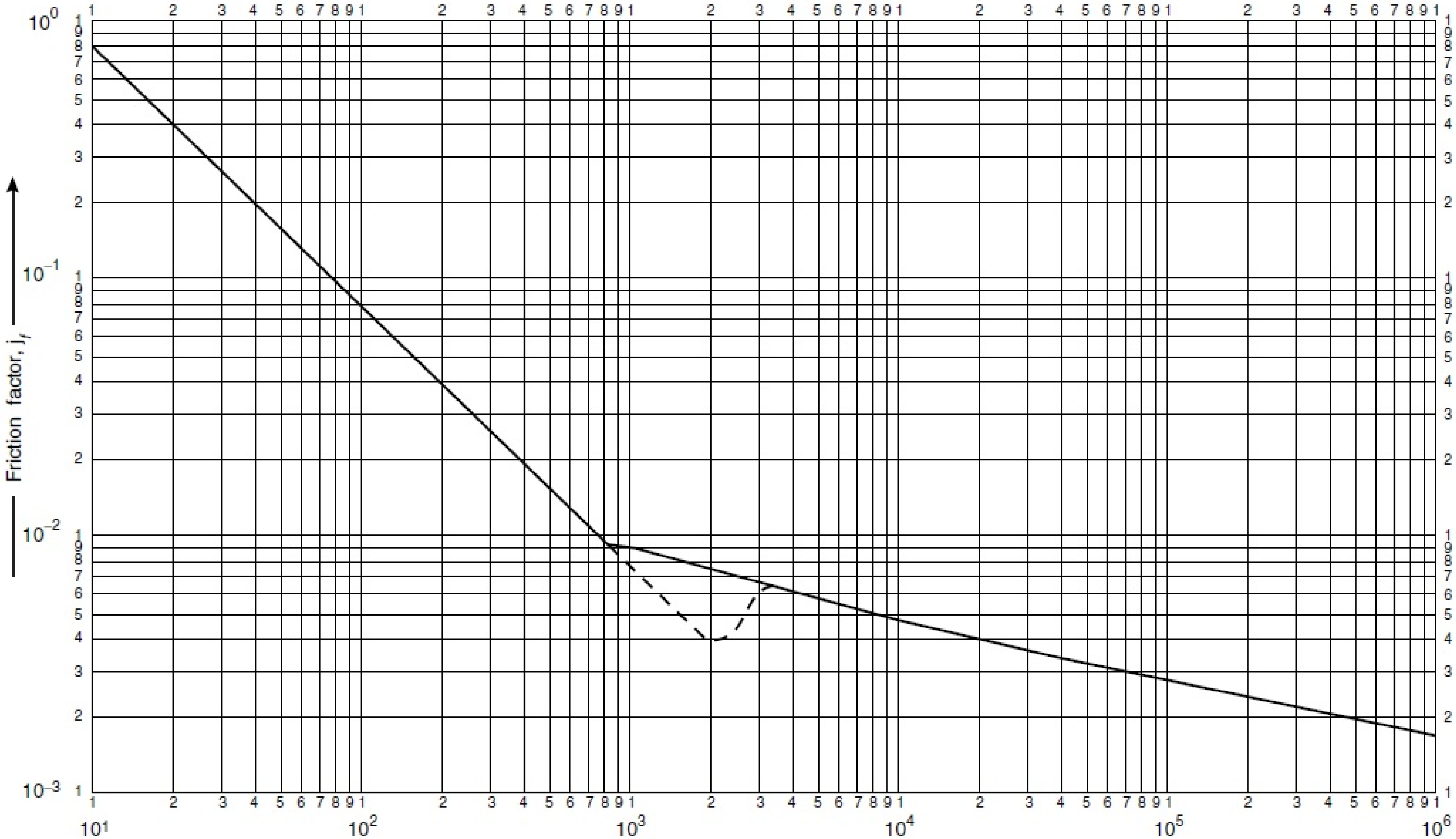

Fig. 10: Tube-side friction factors

Step 7 (Cont'd)

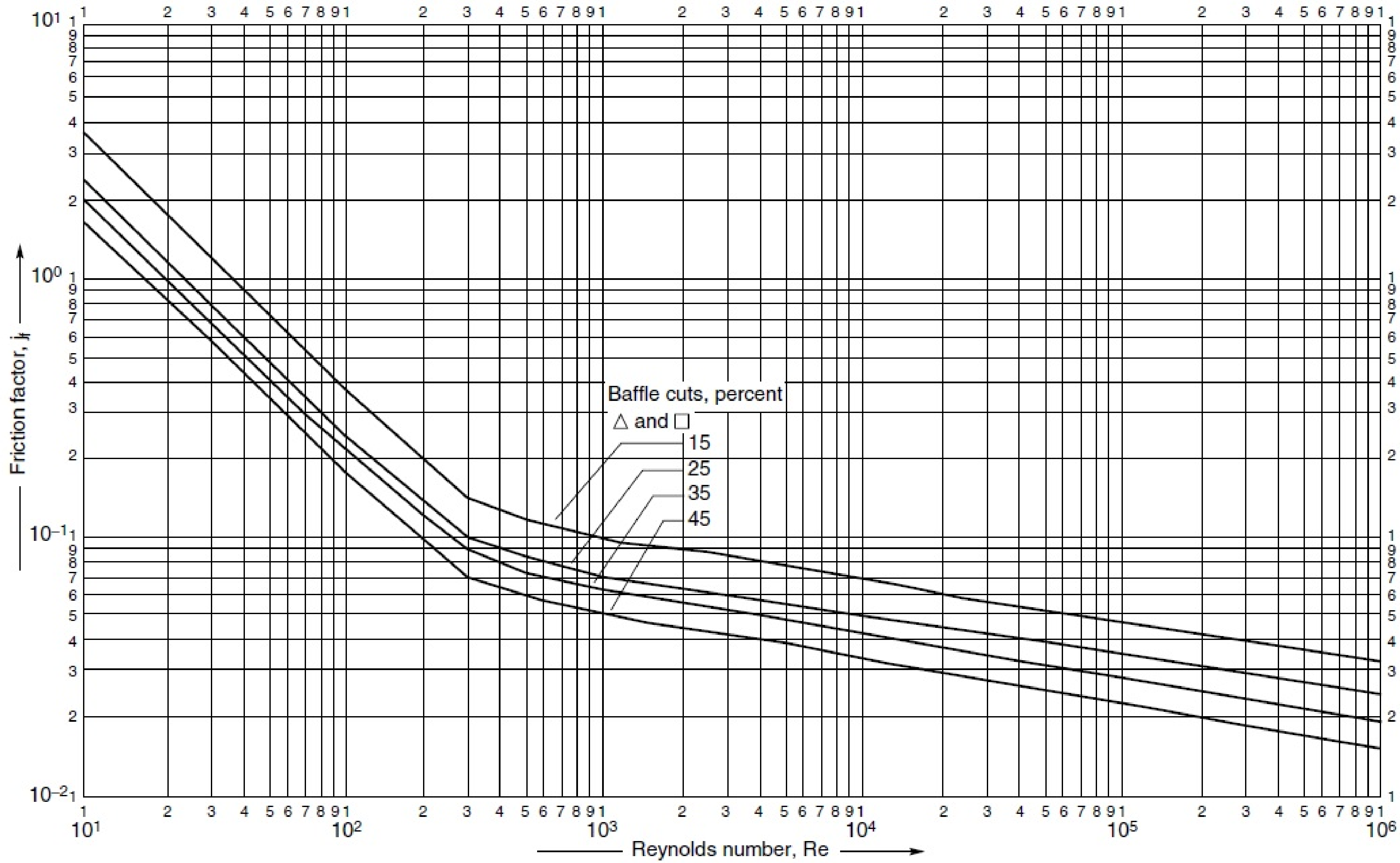

Fig. 11: Shell-side friction factors, segmental baffles

Step 7 (Cont'd)

$$\Delta P_s = 8j_f\left(\frac{D_s}{d_e}\right)\left(\frac{L}{L_s}\right)\frac{\rho u_t^2}{2} = 8 \times 4 \times 10^{-2}\left(\frac{894}{14.4}\right)\left(\frac{4.83 \times 10^3}{178}\right)\frac{750 \times 1.16^2}{2}$$
$$= 272019\ N/m^2$$
$$= 272\ kPa\ (39\ psi)\ \text{too high,}$$

- could be reduced by increasing the baffle pitch. Doubling the pitch halves the shell side velocity, which reduces the pressure drop by a factor of approximately $(1/2)^2$

$$\Delta P_s = \frac{272}{4} = 68\ kPa\ (10psi),\ \text{acceptable}$$

- This will reduce the shell-side heat-transfer coefficient by a factor of $(1/2)^{0.8}(h_o \propto Re^{0.8} \propto u_s^{0.8})$

$$h_o = 2740 \ \square\ (1/2)^{0.8} = 1573\ W/m^2{}^\circ C$$

- This gives an overall coefficient of 615 W/m²°C – still above assumed value of 600 W/m²°C

Step 8 (Cont'd)

- Take the thermal conductivity of cupro-nickel alloys from Table 1, 50 W/m°C, the fouling coefficients from Table 3; methanol (light organic) 5000 $Wm^{-2}°C^{-1}$, brackish water (sea water), take as highest value, 3000 $Wm^{-2}°C^{-1}$

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{1}{h_{od}} + \frac{d_o \ln \frac{d_o}{d_i}}{2K_w} + \frac{d_o}{d_i} \times \frac{1}{h_{id}} + \frac{d_o}{d_i} \times \frac{1}{h_i}$$

$$\frac{1}{U_o} = \frac{1}{2740} + \frac{1}{5000} + \frac{20 \times 10^{-3} \ln \frac{20}{16}}{2 \times 50} + \frac{20}{16} \times \frac{1}{3000} + \frac{20}{16} \times \frac{1}{3812}$$

$$U_o = 738 \ W/m^2 {}^\circ C$$

- Well above assumed value of 600 $Wm^{-2}°C$

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Thank You

Step 5

Table. 3: Fouling factors (coefficients), typical values

Fluid	Coefficient ($W/m^2\,°C$)	Factor (resistance) ($m^2\,°C/W$)
River water	3000–12,000	0.0003–0.0001
Sea water	1000–3000	0.001–0.0003
Cooling water (towers)	3000–6000	0.0003–0.00017
Towns water (soft)	3000–5000	0.0003–0.0002
Towns water (hard)	1000–2000	0.001–0.0005
Steam condensate	1500–5000	0.00067–0.0002
Steam (oil free)	4000–10,000	0.0025–0.0001
Steam (oil traces)	2000–5000	0.0005–0.0002
Refrigerated brine	3000–5000	0.0003–0.0002
Air and industrial gases	5000–10,000	0.0002–0.0001
Flue gases	2000–5000	0.0005–0.0002
Organic vapours	5000	0.0002
Organic liquids	5000	0.0002
Light hydrocarbons	5000	0.0002
Heavy hydrocarbons	2000	0.0005
Boiling organics	2500	0.0004
Condensing organics	5000	0.0002
Heat transfer fluids	5000	0.0002
Aqueous salt solutions	3000–5000	0.0003–0.0002