Mean Temperature Difference in Multipass Exchangers

In an exchanger with one shell pass and several tube-side passes, the fluids in the tubes and shell will flow cocurrently in some of the passes and countercurrently in the others.

For given inlet and outlet temperatures, the mean temperature difference for countercurrent flows greater than that for cocurrent or parallel flow.

Underwood and Bowman et al. have presented graphical methods for calculating the true mean temperature difference in terms of the value of θ_m which would be obtained for countercurrent flow, and a correction factor F_t.

Following points are considered in the derivation of the temperature correction factor F_t,

(a)The shell fluid temperature is uniform over the cross-section considered as constituting a pass.

(b) There is equal heat transfer surface in each pass.

(c) The overall heat transfer coefficient U is constant throughout the exchanger.

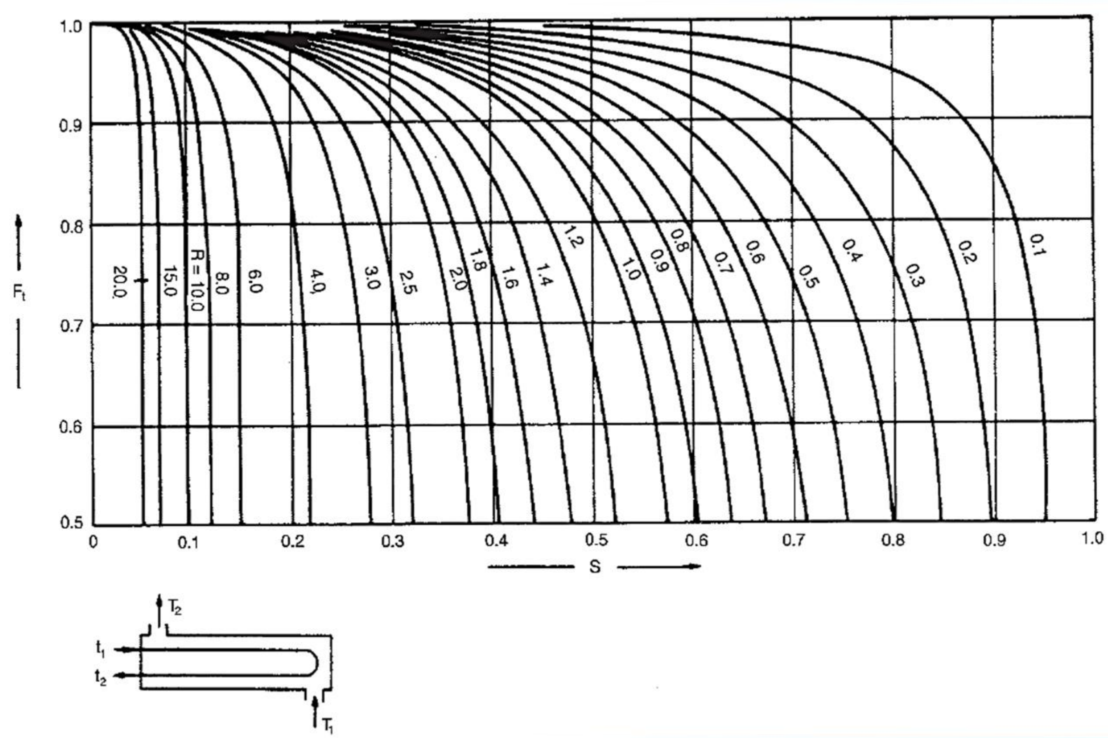
(d) The heat capacities of the two fluids are constant over the temperature range.

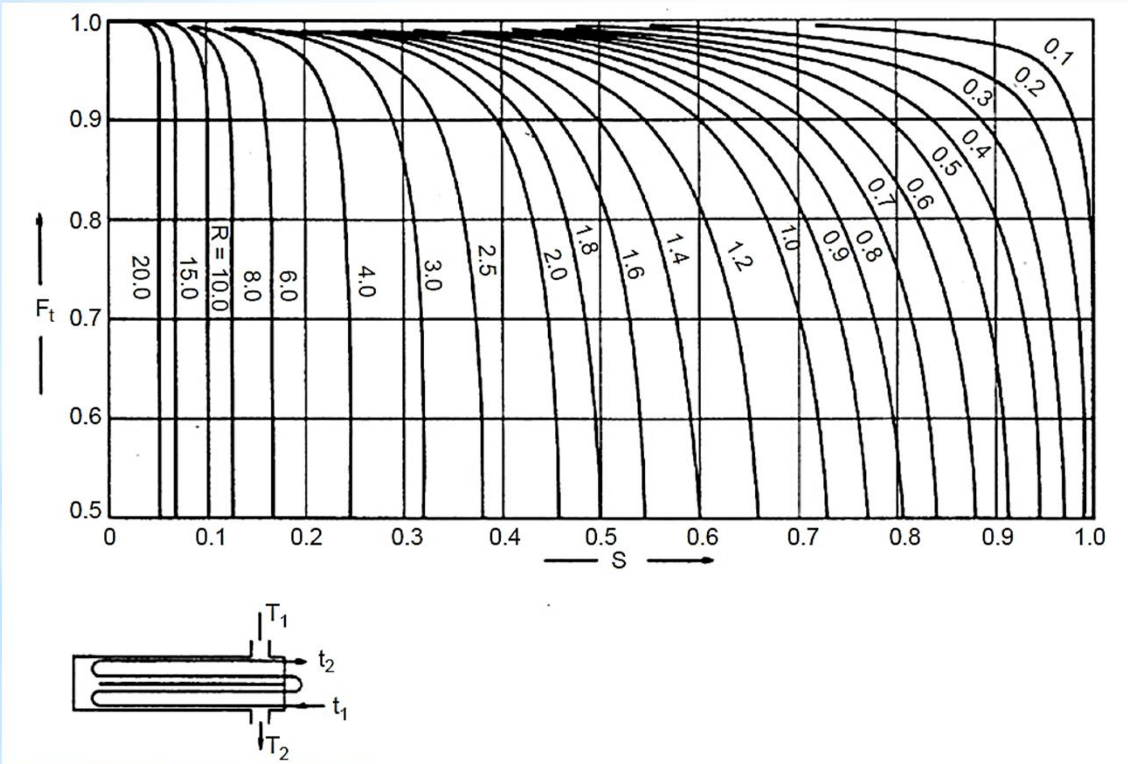
(e) There is no change in phase of either fluid.

(f) Heat losses from the unit are negligible.

F_t can be found from the curves shown in Figs

Temperature correction factor: one shell pass; two or more even tube passes





Temperature correction factor: two shell passes; four or multiples of 4 tube passes

Shell and Tube Exchangers: General Design Considerations

Fluid Allocation: Shell or Tubes

Where no phase change occurs, the following factors will determine the allocation of the fluid streams to the shell or tubes.

Corrosion: The more corrosive fluid should be allocated to the tube side. This will reduce the cost of expensive alloy or clad components.

Fouling: The fluid that has the greatest tendency to foul the heat transfer surfaces should be placed in the tubes.

This will give better control over the design fluid velocity, and the higher allowable velocity in the tubes will reduce fouling. Also, the tubes will be easier to clean.

Fluid temperatures: If the temperatures are high enough to require the use of special alloys, placing the higher temperature fluid in the tubes will reduce the overall cost.

At moderate temperatures, placing the hotter fluid in the tubes will reduce the shell surface temperatures and hence the need for lagging to reduce heat loss or for safety reasons.

Operating pressures: The higher pressure stream should be allocated to the tube side. High-pressure tubes will be cheaper than a high-pressure shell.

Pressure drop: For the same pressure drop, higher heat transfer coefficients will be obtained on the tube side than the shell side, and fluid with the lowest allowable pressure drop should be allocated to the tube side.

Viscosity: Generally, a higher heat transfer coefficient will be obtained by allocating the more viscous material to the shell side, providing the flow is turbulent.

The critical Reynolds number for turbulent flow in the shell is in the region of 200.

If turbulent flow cannot be achieved in the shell, it is better to place the fluid in the tubes, as the tube-side heat transfer coefficient can be predicted with more certainty.

Stream flow rates: Allocating the fluids with the lowest flow rate to the shell side will normally give the most economical design.

Shell and Tube Fluid Velocities

High velocities will give high heat transfer coefficients but also a high-pressure drop.

The velocity must be high enough to prevent any suspended solids settling, but not so high as to cause erosion.

High velocities will reduce fouling. Plastic inserts are sometimes used to reduce erosion at the tube inlet. Typical design velocities are given next:

Liquids

Tube-side: process fluids: 1 to 2 m/s, maximum 4 m/s if required to reduce fouling;

water: 1.5 to 2.5 m/s.

Shell-side: 0.3 to 1 m/s.

Vapors

For vapors, the velocity used will depend on the operating pressure and fluid density; the lower values in the following ranges will apply to high molecular weight materials:

Vacuum	50 to 70 m/s
Atmospheric pressure	10 to 30 m/s
High pressure	5 to 10 m/s

Stream Temperatures

The closer the temperature approach used, the larger will be the heat transfer area required for a given duty.

[the difference between the outlet temperature of one stream and the inlet temperature of the other stream]

As a general guide,

- the greater temperature difference should be at least 20°C;
- the least temperature difference, 5 to 7°C for coolers using cooling water; 3 to 5°C using refrigerated brines.

The maximum temperature rise in recirculated cooling water is limited to around 30°C.

When the heat exchange is between process fluids for heat recovery, the optimum approach temperatures will normally not be lower than 20°C.

Pressure Drop

When the designer is free to select the pressure drop, an economic analysis can be made to determine the exchanger design that gives the lowest operating costs, taking into consideration both capital and pumping costs.

General guide for designs that are near the optimum.

Liquids

Viscosity	<1 mN s/m^2	35 kN/m^2
	1 to 10mN s/m^2	50–70 kN/m^2

Gas and Vapors

High vacuum	0.4–0.8 kN/m^2
Medium vacuum	0.1 absolute pressure
1 to 2 bar	0.5 system gauge pressure
Above 10 bar	0.1 system gauge pressure

When a high pressure drop is used, care must be taken to ensure that the resulting high fluid velocity does not cause erosion or flow-induced tube vibration.

Tube-side Heat Transfer Coefficient and Pressure Drop (Single Phase)

Heat Transfer

Turbulent Flow

Heat transfer data for turbulent flow inside conduits of uniform cross-section are usually correlated by an equation of the form

$$Nu = CRe^{a}Pr^{b}\left(\frac{\mu}{\mu_w}\right)^{c}$$

A general equation that can be used for exchanger design is

$$Nu = CRe^{0.8}Pr^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

where

C = 0.021 for gases;

= 0.023 for nonviscous liquids;

= 0.027 for viscous liquids.

Butterworth (1977) gives the following equation, which is based on the ESDU work:

$$St = ERe^{-0.205}Pr^{-0.505}$$

where

St = Stanton number = (Nu/RePr) = $(h_i/\rho u_t C_p)$;

and E = 0.0225 exp(-0.0225(ln Pr)^2).

Equation is applicable at Reynolds numbers greater than 10,000.

Laminar Flow

Reynolds number less than 2000, the flow in pipes will be laminar.

The following equation can be used to estimate the film heat transfer coefficient:

$$Nu = 1.86(RePr)^{0.33}\left(\frac{d_e}{L}\right)^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

In laminar flow, the length of the tube can have a marked effect on the heat transfer rate for length to diameter ratios less than 500.

Transition Region

In the flow region between laminar and fully developed turbulent flow, heat transfer coefficients cannot be predicted with certainty, as the flow in this region is unstable.

The transition region should be avoided in exchanger design.

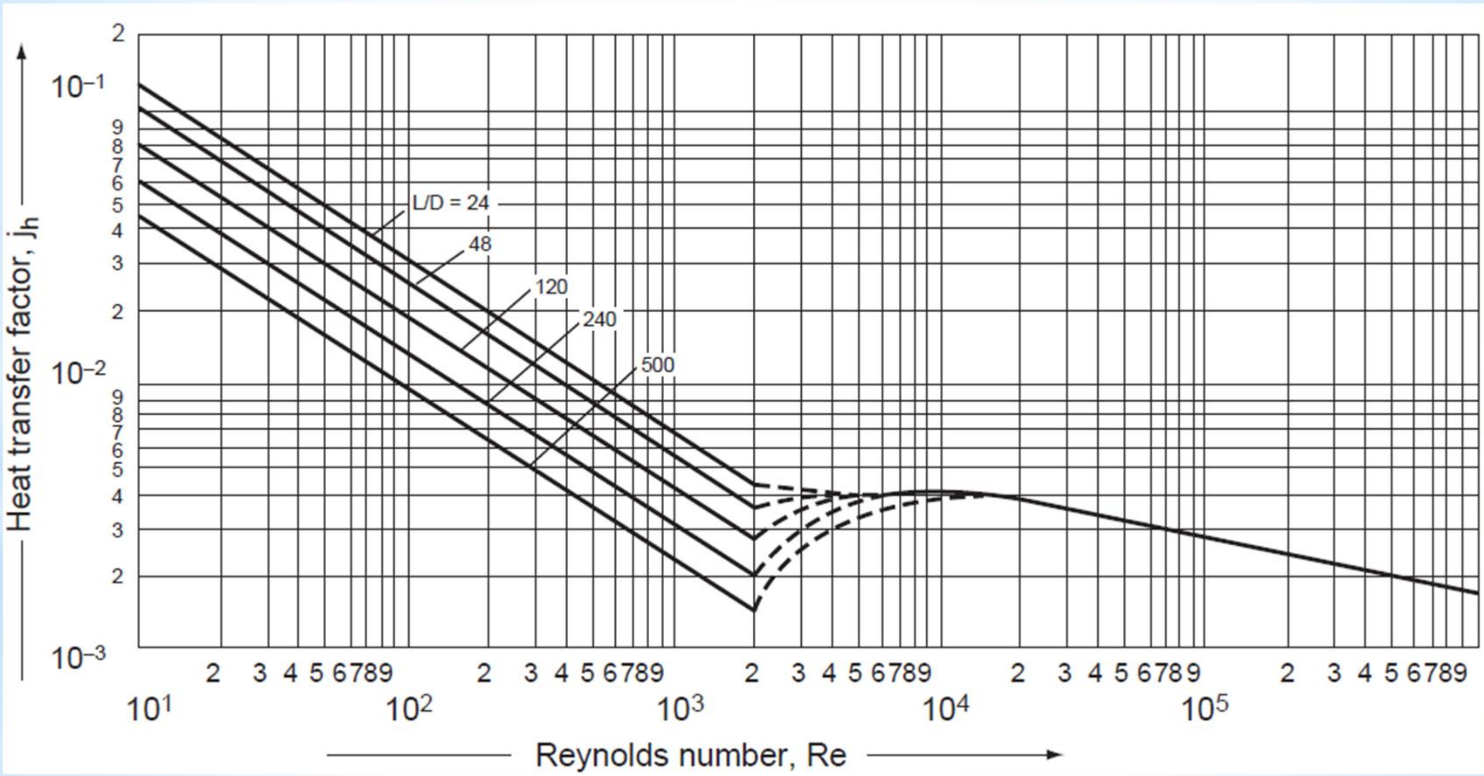
Heat Transfer Factor, j_h

It is often convenient to correlate heat transfer data in terms of a heat transfer j factor, which is similar to the friction factor used for pressure drop.

The heat transfer factor is defined by

$$j_h = St Pr^{0.67} \left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

The use of the j_h factor enables data for laminar and turbulent flow to be represented on the same graph



The j_h values obtained from Figure can be used with equation to estimate the heat transfer coefficients for heat exchanger tubes and commercial pipes.

The coefficient estimated for pipes will normally be higher, as pipes are rougher than the tubes used for heat exchangers.

Equation can be rearranged to a more convenient form:

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

Kern (1950) and other workers define the heat transfer factor as

$$j_H = Nu Pr^{-1/3} \left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

The relationship between j_h and j_H is given by

$$j_H = j_h Re$$

Viscosity Correction Factor

The viscosity correction factor will normally be significant only for viscous liquids.

To apply the correction, an estimate of the wall temperature is needed.

This can be made by first calculating the coefficient without the correction and using the following relationship to estimate the wall temperature:

$$h_i(t_w - t) = U(T - t)$$

Usually, an approximate estimate of the wall temperature is sufficient, but trial-and-error calculations can be made to obtain a better estimate if the correction is large.

Coefficients for Water

More accurate estimate can be made by using equations developed specifically for water.

The following equation has been adapted from data given by Eagle and Ferguson (1930):

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

Tube-Side Pressure Drop

There are two major sources of pressure loss on the tube side:

- The friction loss in the tubes
- The losses due to the sudden contraction and expansion and flow reversals that the fluid experiences in flow through the tube arrangement.

The tube friction loss can be calculated using the familiar equations for pressure-drop loss in pipes.

The basic equation for isothermal flow in pipes (constant temperature) is

$$\Delta P = 8 j_f \left(\frac{L'}{d_i}\right) \frac{\rho u_t^2}{2}$$

where

j_f is the dimensionless friction factor and L' is the effective pipe length.

The flow in a heat exchanger will clearly not be isothermal, and this is allowed for by including an empirical correction factor to account for the change in physical properties with temperature.

Normally, only the change in viscosity is considered:

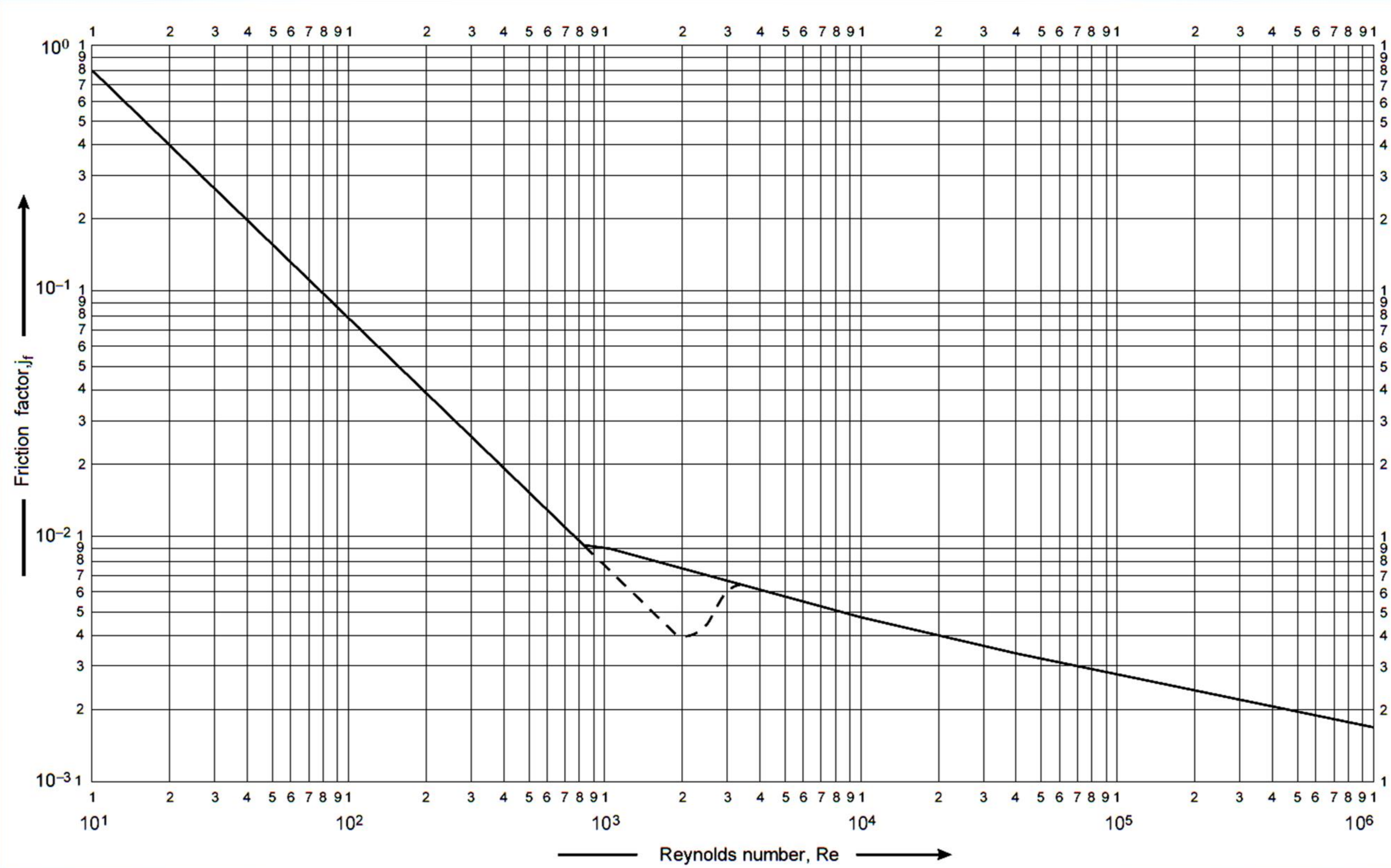
$$\Delta P = 8j_f(L'/d_i)\rho\frac{u_t^2}{2}\left(\frac{\mu}{\mu_w}\right)^{-m}$$

m = 0.25 for laminar flow, Re $<$ 2100;

= 0.14 for turbulent flow, Re $>$ 2100.

Values of j_f for heat exchanger tubes can be obtained from Figure given in next slide; for commercial pipes.

The pressure losses due to contraction at the tube inlets, expansion at the exits, and flow reversal in the headers can be a significant part of the total tube-side pressure drop. There is no entirely satisfactory method for estimating these losses.


Friction factor, j_f
Reynolds number, Re
10^0
10^{-1}
10^{-2}
10^{-3}
10^1
10^2
10^3
10^4
10^5
10^6

The loss in terms of velocity heads can be estimated by counting the number of flow contractions, expansions, and reversals, and using the factors for pipe fittings to estimate the number of velocity heads lost.

For two tube passes, there will be two contractions, two expansions, and one flow reversal.

The head loss for each of these effects is contraction 0.5, expansion 1.0, 180° bend 1.5; so for two passes the maximum loss will be

$$2 \times 0.5 + 2 \times 1.0 + 1.5 = 4.5 \text{ velocity heads}$$

From this, Frank's recommended value of 2.5 velocity heads per pass is the most realistic value to use.

Combining this factor with equation gives

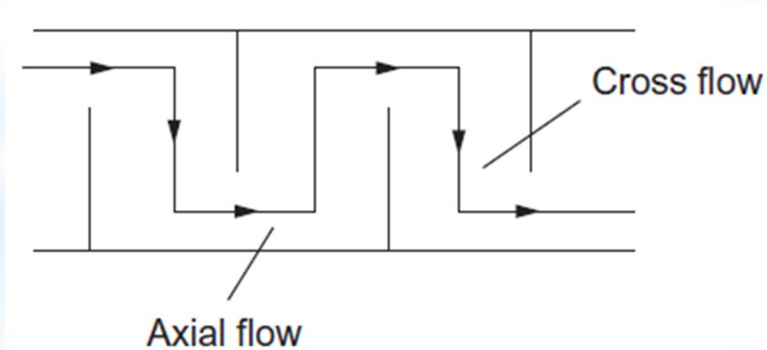
$$\Delta P_t = N_p \left[8 j_f \left(\frac{L}{d_i} \right) \left(\frac{\mu}{\mu_w} \right)^{-m} + 2.5 \right] \frac{\rho u_t^2}{2}$$

where
ΔP_t = tube-side pressure drop, N/m2 (Pa);
N_p = number of tube-side passes;
u_t = tube-side velocity, m/s;
L = length of one tube.

Shell-side Heat Transfer and Pressure Drop (Single Phase)

The prediction of pressure drop, and indeed heat transfer coefficients in the shell is very difficult due to the complex nature of the flow pattern in the segmentally baffled unit.

Whereas, the baffles are intended to direct fluid across the tubes, the actual flow is a combination of cross-flow between the baffles and axial or parallel flow in the baffle windows as shown in Figure.



Although even this does not represent the actual flow pattern because of leakage through the clearances necessary for the fabrication and assembly of the unit.

This more realistic flow pattern is shown in Figure which is based on the work of Tinker who identifies the various streams in the shell as follows:

A—fluid flowing through the clearance between the tube and the hole in the baffle.

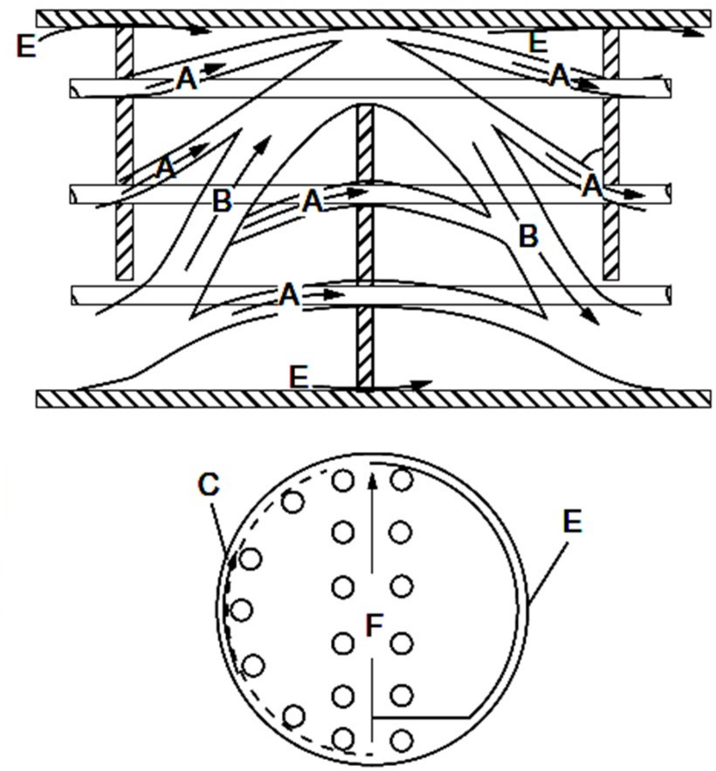
B—the actual cross-flow stream.

C—fluid flowing through the clearance between the outer tubes and the shell.

E—fluid flowing through the clearance between the baffle and the shell.

F—fluid flowing through the gap between the tubes because of any pass-partition plates.

This is especially significant with a vertical gap.



Design Methods

Kern's Method

Kern's method was based on experimental work on commercial exchangers with standard tolerances and will give a reasonably satisfactory prediction of the heat transfer coefficient for standard designs.

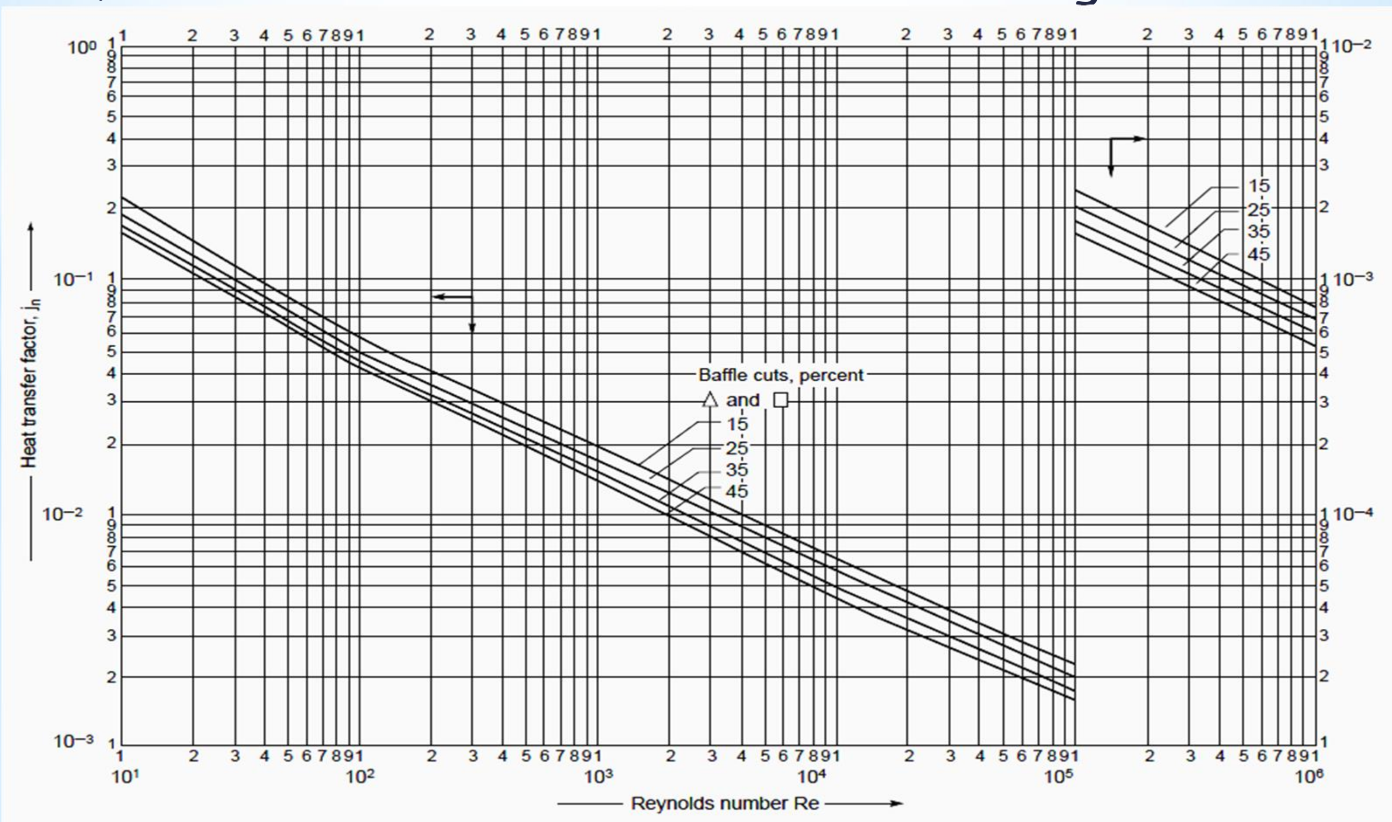
The prediction of pressure drop is less satisfactory, as pressure drop is more affected by leakage and bypassing than heat transfer.

The shell-side heat transfer and friction factors are correlated in a similar manner to those for tube-side flow by using a hypothetical shell velocity and shell diameter.

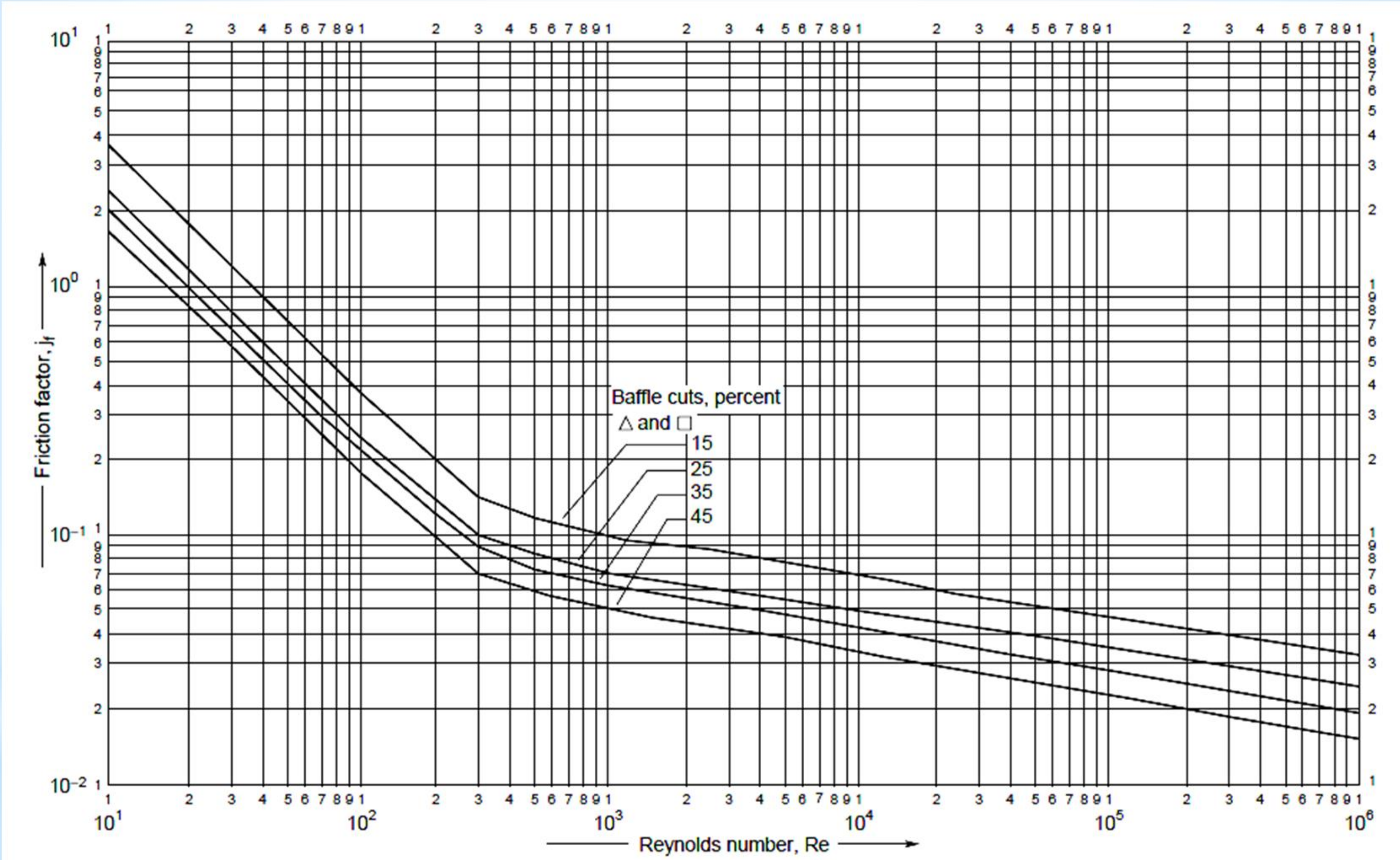
As the cross-sectional area for flow will vary across the shell diameter, the linear and mass velocities are based on the maximum area for cross-flow: that at the shell equator.

The shell equivalent diameter is calculated using the flow area between the tubes taken in the axial direction (parallel to the tubes) and the wetted perimeter of the tubes.

Shell-side j_h and j_f factors for use in this method are given in Figures, for various baffle cuts and tube arrangements.



Abhishek Kumar Chandra



The procedure for calculating the shell-side heat transfer coefficient and pressure drop for a single shell pass exchanger is given from next slide.