DESIGN OF SHELL AND TUBE HEAT EXCHANGER

A thesis submitted in partial fulfillment of the requirements for the Degree of
Master of Technology
(Refining & Petrochemical Engineering)

By
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Under the guidance of

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College of Engineering
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CERTIFICATE

This is to certify that the work contained in this thesis titled "**DESIGN OF SHELL AND TUBE HEAT EXCHANGER**" has been carried out by **SHILPI SAXENA** under my/our supervision and has not been submitted elsewhere for a degree.

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Mr. Adarsh Kumar Arya

Assistant Professor

Date ...17/05/13..........

ABSTRACT

The objective of this project is to provide thermal, hydraulic and mechanical design of two-fluid single-phase shell and tube heat exchangers before the naphtha vaporizer unit to increase the naphtha's temperature above the dew point of sulphur so that sulphur does not get condensed and corrodes the tubes.

Two important goals were born in mind during the preparation of this project. They are:

- To introduce and apply concepts learned in first courses in heat transfer, fluid mechanics and thermodynamics to develop heat exchanger design theory. Thus, the project will serve as a link between fundamental subjects mentioned and thermal engineering design practice in industry.
- To introduce and apply basic heat exchanger design concepts to the solution of industrial heat exchanger problems. Primary emphasis is placed on fundamental concepts and applications. Also, more emphasis is placed on analysis and less on empiricism.

ACKNOWLEDGEMENT

I owe a deep sense of gratitude to my project guide Mr. **Adarsh Arya** for providing me the opportunity to work on this project, providing the required resources and for his able guidance and constant encouragement which helped me in successful completion of my work. He was very helpful in clearing my doubts regarding my project.

Finally I want to thank all faculty members of U.P.E.S. Dehradun who helped me in completing my M.Tech project on shell and tube heat exchanger design.

SHILPI SAXENA

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CHAPTER 1
INTRODUCTION

A shell-and-tube exchanger consists of a large-diameter pipe (on the order of 12 nominal to 24 nominal and larger), inside a number of tubes is placed (ranging from about 20 to over 1000 tubes!). One fluid is directed through the tubes, and another inside the shell but outside the tubes. Baffles are used to direct the shell fluid past the tubes in such a way that heat transfer is enhanced. Turbulence is caused, and higher heat transfer coefficients, and hence higher rates, result.

Over the past quarter century, the importance of heat exchangers has increased immensely from the viewpoint of energy conservation, conversion, recovery, and successful implementation of new energy sources. Its importance is also increasing from the standpoint of environmental concerns such as thermal pollution, air pollution, water pollution, and waste disposal. Heat exchangers are used in the process, power, transportation, air-conditioning and refrigeration, cryogenic, heat recovery; alternate fuels, and manufacturing industries, as well as being key components of many industrial products available in the marketplace.

The shell and tube exchanger is by far the most commonly used type of heat-transfer equipment used in the chemical and allied industries. The advantages of this type are:

1. The configuration gives a large surface area in a small volume.
2. Good mechanical layout: a good shape for pressure operation.
3. Uses well-established fabrication techniques.
4. Can be constructed from a wide range of materials

1.1 Problem formulation:

Sulfur is one of the foremost corodents which cause problems in the refinery industry. It occurs in crude petroleum at various concentrations, and forms a variety of chemical compounds, including hydrogen sulphide, mercaptans, sulphides, polysulphides, and thiophenes.

In the hydrogen unit, sulfur in presence of hydrogen and catalyst, reacts to form aggressive hydrogen sulphide, which increases the severity of corrosion. In this case the corrosion is detrimental not only because of metal loss but also because of volume of sulphide scale formed that can lead to reactor plugging.

1.2 Innovation:

Hence a shell and tube heat exchanger is used to increase the temperature of naphtha, above the dew point of sulfur, so that sulfur does not get condensed in the naphtha-vaporizer of the hydrogen unit. Hydro cracked diesel is being used to increase the temperature of naphtha.

1.3 Methodology of work:

The optimum thermal design of a shell and tube heat exchanger involves:

Process:

1. Process fluid assignments to shell side or tube side.
2. Selection of stream temperature specifications.
3. Setting shell side and tube side pressure drop design limits.
4. Setting shell side and tube side velocity limits.
5. Selection of fouling coefficients for shell side and tube side.

Mechanical:

1. Selection of heat exchanger TEMA layout and number of passes.
2. Specification of tube parameters - size, layout, pitch and material.
3. Setting upper and lower design limits on tube length.
4. Specification of shell side parameters – materials, baffle cut, baffle spacing.
5. Setting upper and lower design limits on shell diameter, baffle cut and spacing.

CHAPTER 2
THEORETICAL DEVELOPMENT

2.1 Basic design procedure:

The steps in a typical design procedure are given below:

1. Define the duty: heat-transfer rate, fluid flow-rates, and temperatures.
2. Collect together the fluid physical properties required: density, viscosity, thermal conductivity.
3. Decide on the type of exchanger to be used.
4. Select a trial value for the overall coefficient, U.
5. Calculate the mean temperature difference, ΔTm.
6. Calculate the area required from the above equation.
7. Decide the exchanger layout..
8. Calculate the individual coefficients.
9. Calculate the overall coefficient and compare with the trial value. If the calculated value differs significantly from the estimated value, substitute the calculated for the estimated value and return to step 6.
10. Calculate the exchanger pressure drop; if unsatisfactory return to steps 7 or 4 or3, in that order of preference.
11. Optimize the design: repeat steps 4 to 10, as necessary, to determine the cheapest exchanger that will satisfy the duty. Usually this will be the one with the smallest area.

2.2 Fouling factors:

Most process and service fluids will foul the heat-transfer surfaces in an exchanger to a greater or lesser extent. The deposited material will normally have a relatively low thermal conductivity and will reduce the overall coefficient. It is therefore necessary to oversize an exchanger to allow for the reduction in performance during operation. The effect of fouling is allowed for in design by including the inside and outside fouling coefficients. Fouling factors are usually quoted as heat-transfer resistances, rather than coefficients. They are difficult to predict and are usually based on past experience.

2.3 Construction details:

Essentially, a shell and tube exchanger consists of a bundle of tubes enclosed in a cylindrical shell. The ends of the tubes are fitted into tube sheets, which separate the shell-side and tube-side fluids. Baffles are provided in the shell to direct the fluid flow and support the tubes. The assembly of baffles and tubes is held together by support rods and spacers as shown below.

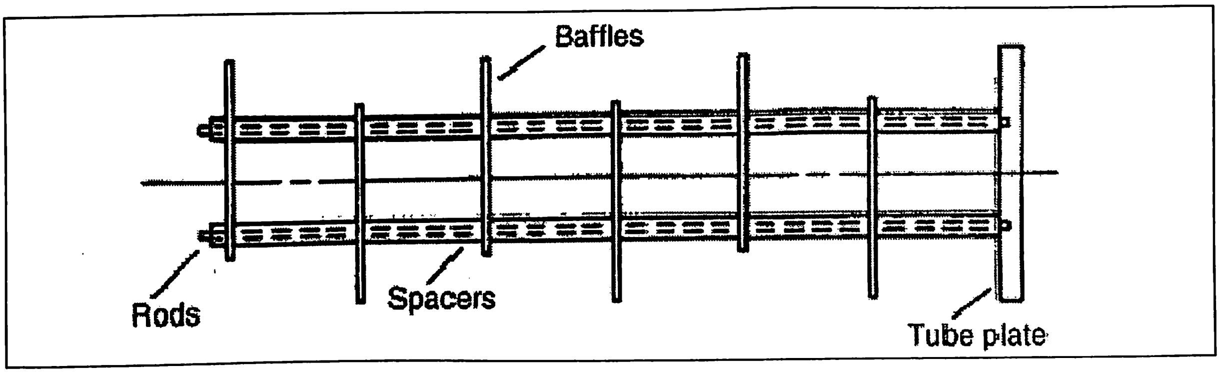

Figure1: Baffle spacers and tie rod

2.4 Parts of a shell and tube heat exchanger:

1. Shell
2. Shell cover
3. Floating-head cover
4. Floating-tube plate
5. Clamp ring
6. Fixed-tube sheet (tube plate)
7. Channel (end-box or header)
8. Channel cover
9. Branch (nozzle) connection
10. Vent connection
11. Drain
12. Test connection
13. Tie rod and spacer
14. Tube bundle
15. Pass partition
16. Floating head
17. Floating-head gland
18. Floating-head gland ring
19. Tube
20. Cross baffle or tube-support plate
21. Lifting Rings
22. Expansion Bellows
23. Support Bracket
24. Impingement Baffles
25. Longitudinal Baffles

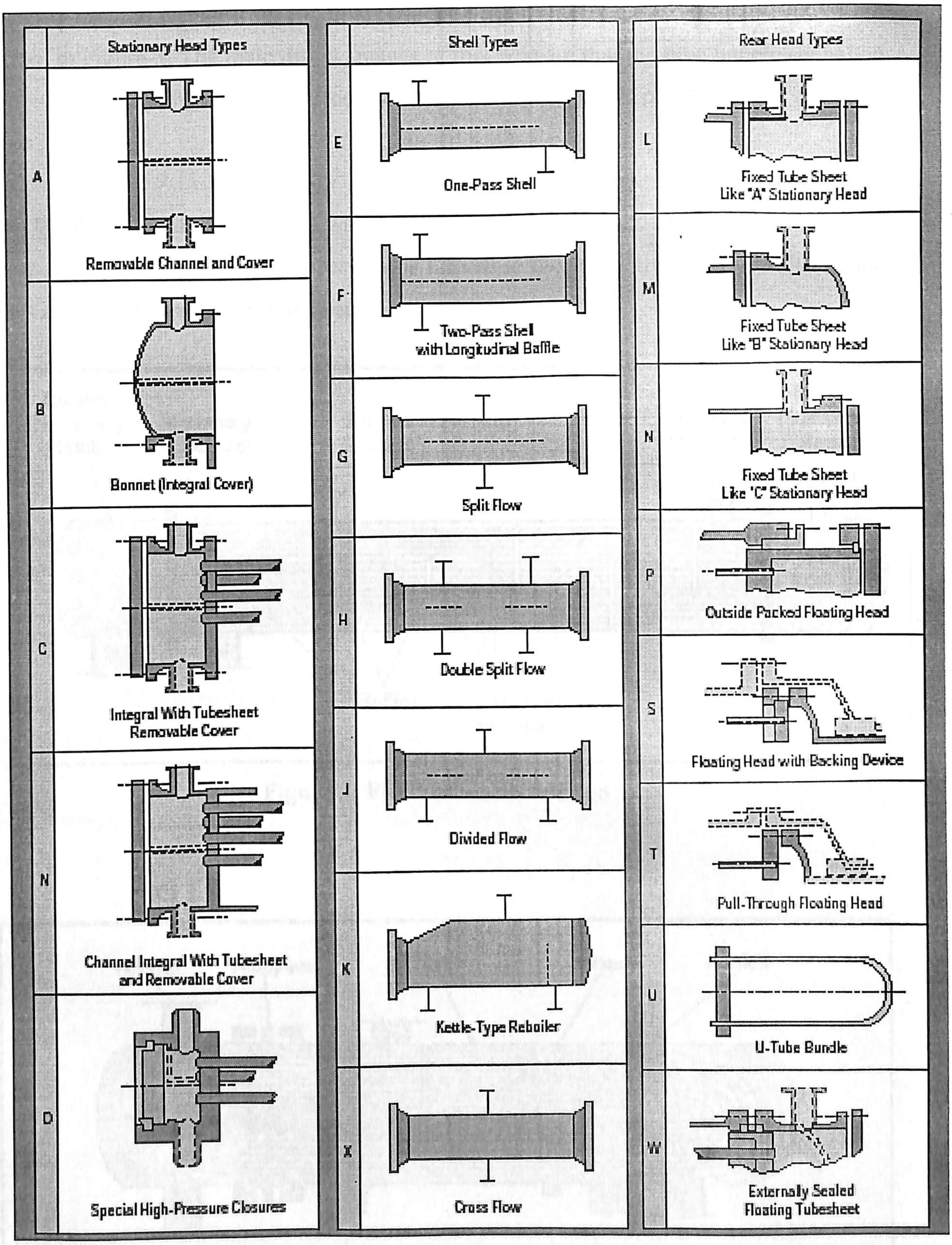

Figure2: Different types of shell and tube heat exchangers

The simplest and cheapest type of shell and tube exchanger is the fixed tube sheet design Shown in Figure 3. The main disadvantages of this type are that the tube bundle cannot be removed for cleaning and there is no provision for differential expansion of the shell and tubes.

The U-tube (U-bundle) type shown in Figure 4, requires only one tube sheet and is cheaper than the floating-head types; but is limited in use to relatively clean fluids as the tubes and bundle are difficult to clean.

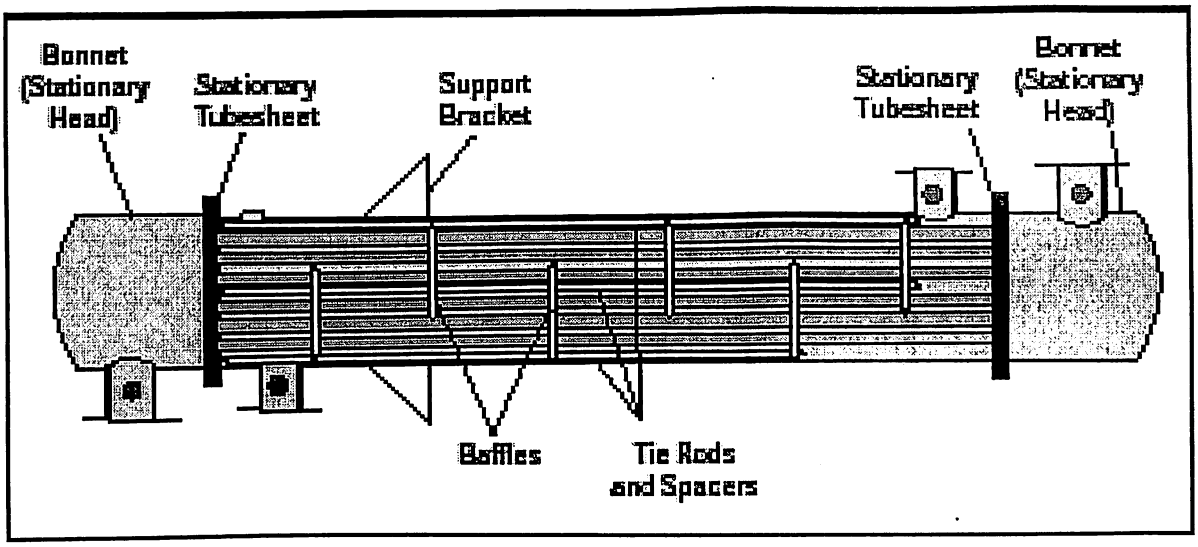

Figure 3: Fixed tube sheet design

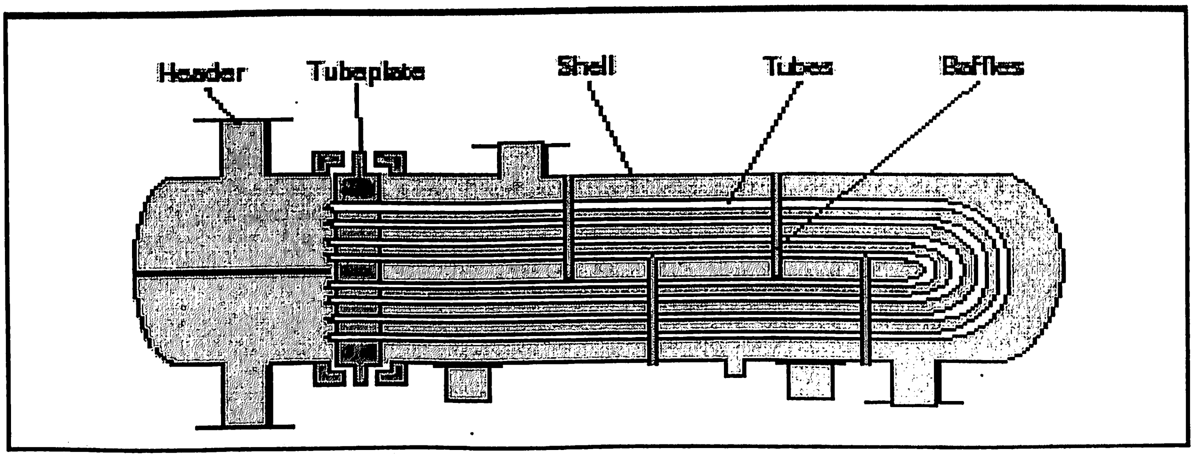

Figure 4: U-Tube design

2.5 Heat Exchanger Standards and Codes:

The mechanical design features, fabrication, materials of construction, and testing of shell and tube exchangers is covered by British Standard, BS 3274. The standards of the American Tubular Heat Exchanger Manufacturers Association, the TEMA standards, are also universally used. The TEMA standards cover three classes of exchanger: class R covers exchangers for the generally severe duties of the petroleum and related industries; class C covers exchangers for moderate duties in commercial and general process applications; and class B covers exchangers for use in the chemical process industries.
The standards give the preferred shell and tube dimensions; the design and manufacturing tolerances; corrosion allowances; and the recommended design stresses for materials of construction.

2.6 Tubes:

2.6.1 Dimensions:

Tube diameters in the range 58 in. (16 mm) to 2 in. (50 mm) are used. The smaller diameters 58 to 1 in. (16 to 25 mm) are preferred for most duties, as they will give more compact, and therefore cheaper, exchangers. Larger tubes are easier to clean by mechanical methods and would be selected for heavily fouling fluids.
The tube thickness (gauge) is selected to withstand the internal pressure and give an adequate corrosion allowance. Steel tubes for heat exchangers are covered by BS 3606 (metric sizes); the standards applicable to other materials are given in BS 3274.
The preferred lengths of tubes for heat exchangers are: 6 ft. (1.83 m), 8 ft (2.44 m), 12 ft (3.66 m), 16 ft (4.88 m) 20 ft (6.10 m), 24 ft (7.32 m). For a given surface area, the use of longer tubes will reduce the shell diameter; which will generally result in a lower cost exchanger, particularly for high shell pressures. The optimum tube length to shell diameter will usually fall within the range of 5 to 10.

2.6.2 Tube arrangements:

The tubes in an exchanger are usually arranged in an equilateral triangular, square, or Rotated square pattern; see Figure 4. 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 (distance between tube centers) is 1.25 times the tube outside diameter; and this will normally be used unless process requirements dictate otherwise. Where a square pattern is used for ease of cleaning, the recommended minimum clearance between the tubes is 0.25 in. (6.4 mm).

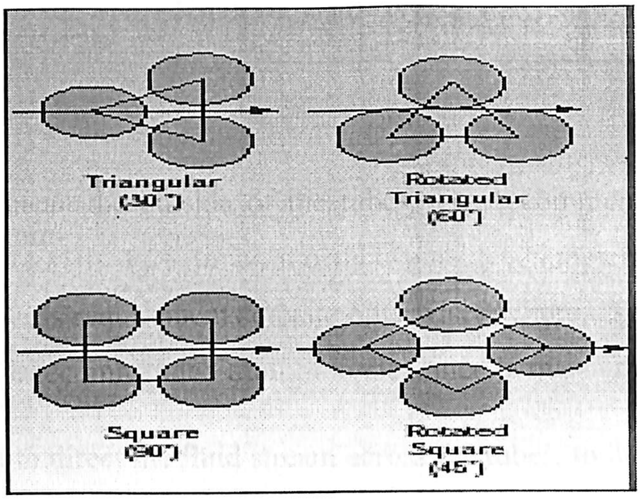

Figure5: Tube pitch

2.6.3 Tube side passes:

The fluid in the tube is usually directed to flow back and forth in a number of "passes" through groups of tubes arranged in parallel, to increase the length of the flow path. The number of passes is selected to give the required tube-side design velocity. Exchangers are built with from one to up to about sixteen tube passes. The tubes are arranged into the number of passes required by dividing up the exchanger headers (channels) with partition plates (pass partitions).

2.7 Shell:

British standard BS 3274 covers exchangers from 6 in. (150 mm) to 42 in. (1067 mm) diameter; and the TEMA standards, exchangers up to 60 in. (1520 mm). Up to about 24 in. (610 mm) shells are normally constructed from standard, close tolerance, pipe; above 24 in. (610 mm) they are rolled from plate.

Tube sheet layout (tube count):

The bundle diameter will depend not only on the number of tubes but also on the number of tube passes, as spaces must be left in the pattern of tubes on the tube sheet to accommodate the pass partition plates. An estimate of the bundle diameter D_b can be obtained from the following equation [2], which is an empirical equation based on standard tube layouts.

$$N_t = K_1(D_b/D_o)^{n_1}$$

Where

N_t =number of the tube.

D_b = bundle diameter, mm

D_o = tube outside diameter, mm

2.8 Baffles:

Baffles are used in the shell to direct the fluid stream across the tubes, to increase the fluid velocity and so improve the rate of transfer. The most commonly used type of baffle is the single segmental baffle shown in Figure 5a, other types are shown in Figures 5b, c and d.

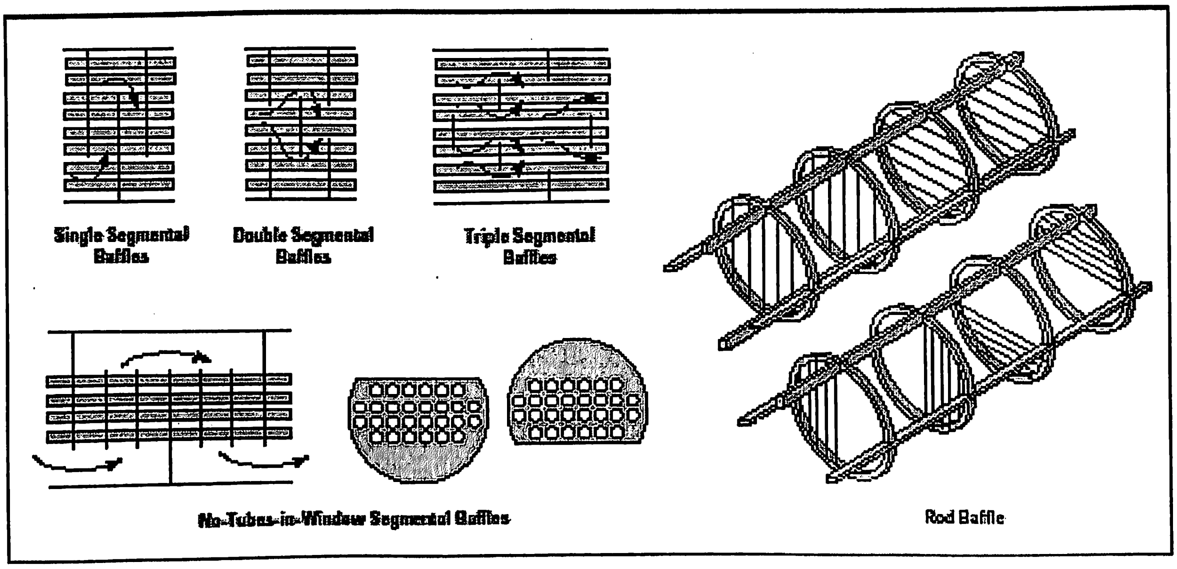

Figure 6: Types of baffle used in the shell and tube exchanger

2.9 Mean Temperature Difference (Temperature Driving Force):

To determine the heat transfer area required for a given duty, an estimate of the mean temperature difference ΔT_m must be made. This will normally be calculated from the terminal temperature differences, the difference in the fluid temperatures at the inlet and outlet of the exchanger. The well-known "logarithmic mean" temperature difference is only applicable to sensible heat transfer in true co-current or counter-current flow (linear temperature enthalpy curves). For counter-current flow, the logarithmic mean temperature is given by [2]:

$$\Delta T_{lm} = [(T_1 - t_2) - (T_2 - t_1)] / \ln[(T_1 - t_2)/(T_2 - t_1)]$$

Where

ΔT_m= log mean temperature difference,

T_1= hot fluid temperature, inlet

T_2= hot fluid temperature, outlet

t_1= cold fluid temperature, inlet

t_2=cold fluid temperature, outlet.

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.

$$\Delta T_{lm} = F_t \Delta T_{lm}$$

Where

ΔT_{lm} =true temperature difference, the mean temperature difference for use in the design equation.

F_T = temperature correction factor

The correction factor 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.

$$R = (T_1 - T_2)/(t_2 - t_1)$$

and

$$S = (t_2 - t_1)/(T_1 - T_2)$$

The temperature correction factor [2] is given by:

$$F_t = \frac{\sqrt{(R^2+1)}\ln\left[\dfrac{(1-S)}{(1-RS)}\right]}{(R-1)\ln\left[\dfrac{2-S[R+1-\sqrt{(R^2+1)}]}{2-S[R+1+\sqrt{(R^2+1)}]}\right]}$$

2.10 General Design Considerations:

2.10.1 Fluid allocation:

(Shell or Tube):

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

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

2. **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.

3. **Fluid temperature**: 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.

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

5. 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.

6. 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.

7. Stream Flow Rates: Allocating the fluids with the lowest flow-rate to the shell-side will normally give the most economical design.

2.10.2 Shell and Tube Fluid Velocity:

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:

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 ranges given below will apply to high molecular weight material

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

2.10.3 Stream Temperature:

The closer the temperature approach used (the difference between the outlet temperature of one stream and the inlet temperature of the other stream) the larger will be the heat-transfer area required for a given duty. The optimum value will depend on the application, and can only be determined by making an economic analysis of alternative designs. As a general guide the greater temperature difference should be at least 20°C, and the least temperature difference 5 to 7°C for coolers using cooling water.

2.10.4 Pressure Drop:

In many applications the pressure drop available to drive the fluids through the exchanger will be set by the process conditions, and the available pressure drop will vary from a few milli bars in vacuum service to several bars in pressure systems. The values suggested below can be used as a general guide, and will normally give designs that are near the optimum.

Liquids:

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

2.10.5 Fluid Physical Property:

The fluid physical properties required for heat-exchanger design are: density, viscosity, thermal conductivity and temperature-enthalpy correlations (specific and latent heats).

2.10.6 Tube Side Heat Transfer Coefficient and Pressure Drop:

1. Turbulent Flow:

Heat-transfer data for turbulent flow inside conduits of uniform cross-section are usually correlated by an equation [2] of the form:

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

Where:

Nu = Nusselt number = $h_i d_e/k_f$

R_e = Reynolds number = $(d_e v\rho/\mu) = (G_t d_e/\mu)$

P_r = Prandtl number = $(C_p\mu/k_f)$

H_i = inside coefficient , W/m^2 ^{0}C

D_e = equivalent diameter, m

D_e = 4 * cross-sectional area of the flow/wetted perimeter

U_t = fluid velocity, m/sec

K_f = fluid thermal conductivity, W/m^2 ^{0}C

G_t = mass velocity, mass flow per unit area, kg/m^2 s,

μ = fluid viscosity at the bulk fluid temperature, Ns/m^2,

μ_w = fluid viscosity at the wall

C_p = fluid specific heat, heat capacity, J/kg ^{0}C.

The index for the Reynolds number is generally taken as 0.8. That for the Prandtl number can range from 0.3 for cooling to 0.4 for heating. The index for the viscosity factor is normally taken as 0.14 for flow in tubes, from the work of Sieder and Tate (1936) [2], but some workers report higher values. 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 non-viscous liquids,

= 0.027 for viscous liquid

2. Laminar Flow:

Below a Reynolds number of about 2000 the flow in pipes will be laminar. Providing the natural convection effects are small, which will normally be so in forced convection, the following equation can be used to estimate the film heat-transfer coefficient [2] :

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

Where L is the length of the tubes.

3. 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, and the transition region should be avoided in exchanger design. If this is not practicable the coefficient should be evaluated using both equations and the least value taken.

2.10.7 Heat Transfer Factor:

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 given equation [2] :

$$\mathbf{J_h = S_t P_r\text{^}.67(\mu/\mu_w)\text{^}(-0.14)}$$

The *jh* values obtained from Figure can be used with the above equation to estimate the heat-transfer coefficients for heat-exchanger tubes and commercial pipes. The above equation can be rearranged in 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) [1], and other workers define the heat transfer factor as:

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

The relationship between j_h and j_H is given by $J_H = J_h R_e$

2.10.8 Viscosity Correction Factor:

The viscosity correction factor will normally only be significant 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 [2] :

$$h_i = (t_w - t) = U(T\text{-}t)$$

Where

t = tube side bulk temperature (mean),

T_W = estimated wall temperature,

T = shell-side bulk temperature (mean).

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.

2.10.9 Tube Side Pressure Drop:

There are two major sources of pressure loss on the tube-side of a shell and tube exchanger: the friction loss in the tubes and 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) [2] is:

$$\Delta P = 8j_f(\, L/d_i)u_t\text{\^{}}2/2$$

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

2.11 Shell Side Heat Transfer and Pressure Drop:

The procedure for calculating the shell-side heat-transfer coefficient and pressure drop for a single shell pass exchanger is given below:

Procedure:

1) Calculate the area for cross-flow As for the hypothetical row of tubes at the shell equator, given by:

$$A_s = (P_t - d_0)D_s l_B / P_t$$

Where

P_t = tube pitch,

d_0 = tube outside diameter

D_s = shell outside diameter, m

l_B = baffle spacing, m

The term $(P_t - d_0)/P_t$ is the ratio of clearance between tubes and the distant between tube centers.

2) Calculate shell side mass velocity G_s and linear velocity u_s:

$$G_s = W_s / A_s \quad \text{and} \quad u_s = G_s / \rho$$

Where

Ws = fluid flow-rate on the shell-side, kg/s,

ρ = shell side fluid density, Kg / m^3

3) Calculate the shell side equivalent diameter. For a square pitch arrangement:

$$d_e = \frac{4\left(\dfrac{p_t^2 - \pi d_o^2}{4}\right)}{\pi d_o} = \frac{1.27}{d_o}(p_t^2 - 0.785 d_o^2)$$

For an equilateral triangular pitch arrangement:

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

Where de = equivalent diameter.

4) Calculate the shell side Reynolds number, given by:

$$R_e = G_s d_e/\mu = u_s d_e \rho/\mu$$

5) For the calculated Reynolds number, read the value of j_h from the figure for the selected baffle cut and tube arrangement.

6) Calculate the shell side heat transfer coefficient h_s from:

$$Nu = h_s d_e/k_f = j_h Re\ Pr^{0.33} (\mu/\mu_W)^{0.14}$$

7) For the calculated shell side Reynolds number, read the friction factor from the figure and calculated the shell side pressure drop from:

$$\Delta P_s = 8 j_f (D_s/D_e)(L/l_B)\rho u_s^2/2(\mu/\mu_W)^{-0.14}$$

Where

L = tube length

l_B = baffle spacing

The term (L/l_B) is the number of times the flow crosses the tube bundle = $(N_B + 1)$, where N_B is the number of baffles.

8) Shell Nozzle Pressure Drop:

The pressure loss in the shell nozzles will normally only be significant with gases. The nozzle pressure drop can be taken as equivalent to 1.5 velocity heads for the inlet and 0.5 for the outlet, based on the nozzle area or the free area between the tubes in the row immediately adjacent to the nozzle, whichever is the least.

CHAPTER 3

CALCULATION BY KERN METHOD

3.1 Given Data:

HOT STREAM: (DIESEL)

Mass flow rate, m_1	= 78400 Kg/ hr
Density of the diesel	= 0.820 gm/cm^3 = 820 Kg/m^3
Inlet temperature	= 165 ^{0}C
Specific heat, C_p	= 0.633 Kcal/Kg ^{0}C

COLD STREAM: (LIGHT NAPHTHA)

Mass flow rate, m_2	= 13862.5 Kg / hr
Inlet temperature	= 40 ^{0}C
Outlet temperature	= 125 ^{0}C
Density of the naphtha	= 0.692 gm/cm^3 = 692 Kg/m^3
Specific heat, C_P	= 0.609 Kcal /Kg ^{0}C

3.2 Sensible Heat Calculation:

Mass flow rate of naphtha	= 13862.5 kg/hr
Mean Specific heat of naphtha	= 0.609 k cal/kg ^{0}C
Initial temperature	= 40 ^{0}C
Final temperature	=125 ^{0}C
Change in temperature, ΔT	= (125–40) = 85 ^{0}C
Therefore sensible heat, $mCp\Delta T$	= 718171.416 kcal/h
	= 3001956.5 kJ/hr
	= 838.307 KW

Calculating outlet temperature of the diesel stream as given below:

From heat balance, we have

Heat lost = Heat gain

$m_1C_P(165\text{-}T_{ho}) = m_2C_P(125\text{-}40)$

$78400 * 0.633(165\text{-}T_{ho}) = 13862.5 * 0.609 * 85$

$T_{ho} = 150.54$

So, outlet temperature of the diesel = 150.54 ^{0}C

3.3 Calculation of Physical Properties:

Various properties of naphtha and diesel are calculated by interpolation and extrapolation.

<u>FOR NAPHTHA:</u>

a) Thermal Conductivity:

$(40\text{-}125) / (0.108\text{-}x) = (125\text{-}136) / (x\text{-}0.096)$

$85(x\text{-}0.096) = 11(0.108\text{-}x)$

$x = 0.0973$

So thermal conductivity of naphtha at 125 ^{0}C is **0.0973 K cal /hr m ^{0}C**

b) Viscosity:

(40-125) / (0.438-x) = (125-136) / (x-0.206)

x - 0.206 = (0.438-x) (0.1294)

x = 0.2325

So, viscosity of naphtha at 125 ^{0}C is **0.2325 cp**

c) Density:

(40-125) / (0.73-x) = (125-136) / (x-0.643)

x = 0.652

So, density of naphtha at 125 ^{0}C is **0.652 Kg/ m^3**

Now, various properties of diesel are calculated at 165^{0}C and at 150.54^{0}C by interpolation and extrapolation.

FOR DIESEL:

a) Viscosity:

(165-151) / (x-0.785) = (151-65) / (0.785-2.699)

86x = 40.714

x = 0.4734 cp

So, at 165 ^{0}C viscosity of diesel is **0.4734 cp.**

b) Thermal Conductivity:

(165-151) / (x-0.095) = (151-65) / (0.095-0.105)

86 x = 8.03

x = 0.093

So, at 165 ^{0}C thermal conductivity of diesel is **0.093 Kcal /hr m ^{0}C**

c) Density:

(165-151) / (x-0.743) = (151-65) / (0.743-0.799)

86x = 63.898-0.784

x =0 .733 Kg /m^3 hence at165^{0}C density is **0.733 Kg/m^3**

Similarly at 150.54 ^{0}C and 157.77 ^{0}C, properties of diesel are calculated.

Property tables for both diesel and naphtha are given below.

Table 1: Physical Property Tables

a) For Diesel:

DIESEL	INLET	MEAN	OUTLET	UNIT
Temperature	165	157.77	150.54	^0C
Specific heat	2.73	2.70	2.66	KJ/Kg ^0C
Thermal conductivity	0.1079	0.1085	0.1091	W/m ^0C
Density	733	737.2	741.4	Kg/ m^3
Viscosity	0.4734	0.60022	0.731	cp

b) For Naphtha:

NAPHTHA	INLET	MEAN	OUTLET	UNIT
Temperature	40	82.5	125	^0C
Specific heat	2.08	2.31	2.54	KJ/Kg ^0C
Thermal conductivity	0.1253	0.1191	0.1129	W/m ^0C
Density	732	692	652	Kg/m^3
Viscosity	0.438	0.335	0.2325	cp

3.4 Calculation of True Temperature Difference:

Now true temperature difference is calculated:

$$R=(T_1-T_2)/(t_2-t_1)$$

and

$$S = (t_2-t_1)/(T_1-t_1)$$

R= (40-125) /(150.54-165) = 0.17

and

S = (150.54-165) /(40-165) = 0.68

For countercurrent flow, log mean temperature difference will be,

$$LMTD = (\Delta t_2-\Delta t_1)/\ln(\Delta t_2/\Delta t_1)$$

LMTD = (110.54-40)/2.303*log(110.54/40)

= **69.39**

F_T is calculated from figure11, S vs. F_T, and the value of F_T is 0.98

$$\Delta t = F_T{*}LMTD$$

Δt = 68.33 °C

Heat Balance:

For diesel,

Q =78400*0.633(165-150.54) = 3001956.5 kJ/hr =**838.307 KW**

For naphtha,

Q = 13862.5*0.609(125-40) = 3001956.5 kJ/hr =**838.307 KW**

3.5 Heat Exchanger Designing:

3.5.1 TRIAL 1:

From table 8, for the overall heat transfer coefficient will be in the range of 300 to500 $W/m^2\,{}^0C$

Assume: $U = 300\ W/m^2\,{}^0C$,

then heat transfer area will be, $A = Q/\ (U\Delta t)$

$A = (838.307*10^3) / (300*68.33) = 40.89\ m^2$

So heat transfer area will be **40.89 m^2**

Layout And Tube Size:

Split size floating head heat exchanger is chosen for better efficiency and cleaning purpose

Taking outside diameter	$d_0 = 25$ mm
Inside diameter	$d_i = 21$ mm
Length of the tube	$L = 7.32$ m (20 ft)

Number Of Tubes:

Area of one tube (neglecting thickness of the tube sheet)	$= \pi d_0 L = 0.574911\ m^2$
Therefore number of tubes , N_t	$= (40.89 / 0.574911) = 72$
So for 2 passes, tube per pass will be	$= 36$
Tube crossectional area	$= \pi\ d_i^2/4 = 4.154*10^{-4}\ m^2$
So area per pass will be	$= 36 * 4.154*10^{-4}\ m^2$
	$= 0.014957 m^2$
Volumetric flow rate of diesel, m / ρ	$= [78400/(737.2*3600)]$
	$= 0.02954\ m^3/s$
Tube side velocity, u_t	$=(0.02954/0.014957)$
	= 1.975 m/s

The velocity is satisfactory between 1 and 2 m/s.

Bundle and Shell Diameter:

From table 5, for 2 tube passes

K_1=0.249 and n_1 = 2.207 and

Nt = number of tube =72

$$D_b = D_0(N_t/K_1)\ \hat{}(1/n_1)$$

So, D_b = 25*(71 / 0.249)^(1 / 2.207) = 323.84 m

For a split ring floating head exchanger, typical shell clearance in figure13, is given as 53 mm.

So, shell inside diameter, Ds= 323.84+53 = 376.844 mm= **377 mm**

Tube Side Heat Transfer Coefficient:

REYNOLDS NUMBER:

$$R_e = dv\rho/\mu$$

R_e = [21*10^(-3)*1.975*737.2] / [0.60022*10^(-3)]

= **5.579*10^4**

PRANDTL NUMBER:

$$P_r = C_P\mu/k$$

P_r = 2.70*{0.60022*10^(-3)}*10^3/0.1085

= **14.93**

LENGTH TO DIAMETER RATIO:

$L/D_i = 318.26$

From figure 7, heat transfer factor, $j_h = 3.1*10^{(-3)}$

$$Nu = hd/k = j_h * Re * Pr^{(1/3)} * (\mu/\mu_W)^{0.14}$$

Nu= 425.86

Therefore, inside heat transfer coefficient = $h_i = (425.86 * 0.1085*10^3/21)$
$= 2200.27\ W/m^2\ {}^0C$

$\mathbf{h_i = 2200.27\ W/m^2{}^\circ C}$

<u>Shell Side Heat Transfer Coefficient:</u>

The shell inside diameter = 377 mm

As a first trial take baffle spacing $l_B = D_s/2.5 = 151$ mm

Calculating area of cross section of the shell equator as given below:

$$A_S = (P_t - d_0)D_s l_B / P_t$$

$A_S = (1.25*25-25)*377*151 / (1.25*25)$
$= 0.0113609\ mm^2$

For equilateral triangle pitch the equivalent diameter is given below:

$$D_e = 1.10(P_t^2 - .917 d_0^2)/d_0$$

$D_e = 1.10*(31.25^2-0.917*25^2) / 25$
$= 17.75$ mm

Volumetric flow rate of the naphtha = 13862.5 / (692*3600)

= 0.005564 m^3/sec

Therefore, shell side velocity = (0.005564 / 0.01136)

= 0.48974 m/sec

REYNOLDS NUMBER:

$$\mathbf{R_e = d_e v \rho / \mu}$$

$R_e = (17.751*10^{-3}*0.4897*692) / (0.60022*10^{-3})$

= 1.79*10^4

PRANDTL NUMBER:

$$\mathbf{P_r = C_P \mu / k}$$

P_r = {2.31*10^3*0.335*10^-3} / 0.1191

= 6.497

Segmental baffles with 25 % cut is used as this gives a assembled heat transfer coefficient without too large pressure drop.

From the figure9, j_H =4.0* 10^(-3)

Neglecting viscosity correction factor, Nusselt number is given as:

$$\mathbf{N_U = hd_e/k}$$

N_U = 4.0*10^-3*1.795*10^4*6.497^(0.33)

= 134.022

So, $h_S = (134.022*0.1191) / (17.751*10^{-3}) = 899.191\ W/m^2\ {}^0C$

So shell side heat transfer coefficient = h_S = **899.19 $W/m^2\ {}^0C$**

Overall Coefficient:

Taking fouling factor from table 4, for naphtha = 5000 $W/m^2\ {}^0C$

and

for diesel = 3000 $W/m^2\ {}^0C$

and k_w = 45 $W/m\ {}^0C$ (from table 7)

$$\mathbf{1/U_0 = 1/h_0 + 1/h_{OD} + d_0 \ln(d_0/d_i)/2k_W + d_0/d_i * h_{iD} + d_0/d_i * h_i}$$

After putting all the given values, U_0 is calculated as,

U_0 = 446.69 $W/m^2\ {}^0C$

Calculating:

$(U_{0CAL} - U_{0ASS})/U_{0ASS}$ = 48.89%

This more than 30%

So this is above the initial estimate of 300 $W/m^2{}^\circ C$. This means the number of tubes could possibly be reduced. But first pressure drops are checked.

Pressure Drop Calculation:

Tube side: 72 tubes, 2 passes, tube inside diameter 21 mm, u_t=1.975,
R_e = 5.57*10^(4), from figure 8, j_f = 3.2*10^(-3)

$$\Delta P_t = N_P[8j_f (L/d_i)(\mu/\mu_W)^{\wedge}(-0.14) + 2.5]\rho u_t^2/2$$

$\Delta P_t = 2(8*3.2*10^{\wedge-3}(7.32*10^3/21)+ 2.5)*737.2*1.975^2/2$

ΔP_t = 0.291 bar

Shell side: j_f= 4.1 *10^-2 from figure 19,

$$\Delta P_S = 8j_f(D_S/d_e)(L/l_B)\rho u^2/2$$

$\Delta P_S = 8* (4.1*10^{\wedge}-2)*(377/17.751)*(7.32*10^{\wedge}3/151)*(692/2)*(0.489)^2$

ΔP_S = 0.280 bar

Viscosity Correction Factor:

The viscosity correction factor $(\mu/\mu_W)^{(0.14)}$ was neglected when calculating the heat transfer coefficients and pressure drops. First an estimate of the temperature at the tube wall, t_W is needed.

The inside area of the tube $= \pi*21*10^{-3}*7.32$

$= 0.5 \text{ m}^2$

Heat flux $= Q/A$

$= (838.30/0.5)$

$= 1584.932 \text{ W/m}^2$

As rough approximation, $(t_W - t)h_i = 1584.932$

Where:

t is the mean bulk fluid temperature. $= 157.77\ ^0C$

$T_w = 158.49\ ^0C$

So, viscosity of diesel at 158.49 ^0C is calculated by interpolation:

At 165 ^0C viscosity of diesel is given as $0.4734*10^{-3}$ Ns/m^2

and at 157.77 $^0C = 0.60022*10^{-3}$ Ns/m^2

By interpolation,

$(157.77-158.49) / (0.60022-x) = (158.49 -165) / (x -0.473)$

$x = 0.5997$ cp

$= 0.5997*10^{(-3)}$ Ns/m^2

So, $(\mu/\mu_W)^{(0.14)} = (0.6002 / 0.5997)^{(0.14)} = 1$

Only a small factor. So the decision to neglect this coefficient is justified.

3.5.2 TRIAL 2:

Taking U = 350 $W/m^{2}\,^{0}C$

A = Q / (UΔt)

= 35.0528 m^2

So heat transfer area will be = **35.0528 m^2**

Layout and Tube Sizes:

Taking, outside diameter, d_0	= 25 mm
Inside diameter, d_i	= 21 mm
Length of the tube, L	= 7.32 m
Taking triangular pitch, P_t	= 31.25 mm (pitch/ diameter= 1.25)

Number Of Tubes:

Area of one tube	= $\pi D_0 L$ = 0.574911 m^2
Therefore number of tubes	= N_t = (35.0528/0.574911) = 60
Taking 2 passes, so tube per pass will be	= 30
Tube cross sectional area	= $\pi\, d_i^2/4 = 4.154*10^{-4}\ m^2$
So area per pass will be	=30* $4.154*10^{-4}$ = 0.00124 m^2

Volumetric flow rate of the diesel, m/ρ	=78400/(737.2*3600)
	= 0.02954 m^3/s

So tube side velocity U_t = (0.02954/0.00124) = 2.37 m/s

Hence tube velocity is not in range of 1-2 m/s

3.5.3 TRIAL 3:

Now taking $U = 350\ W/m^2\ {}^0C$

Heat transfer area will be **$A = 35.0528\ m^2$**

Modified Layout And Tube Size:

Taking outside diameter, d_0	= 30 mm
Inside diameter, d_i	= 26 mm
Length of the tube, L	= 7.32 m
Taking triangular pitch, P_t	= 37.5 mm

Number Of Tubes:

Area of one tube	$= \pi d_0 L = 0.6898\ mm^2$
Number of tubes	$= N_t = (35.0528 / 0.6898) = 52$
Taking 2 passes, so tube per pass will be	= 26
Tube cross sectional area	$= \pi d_i^2/4 = 6.157*10^{-4}\ m^2$
So area per pass will be	$= 0.01600\ m^2$

Volumetric flow rate of diesel, m/ρ	= (78400/[737.2*3600])
	$= 0.02954\ m^3/s$

Tube side velocity = **$U_t = 1.845$ m/s**

Hence within range of velocity 1-2 m/s

Bundle and Shell Diameter:

From table5, for 2 tube passes

K_1=0.249 and n_1 = 2.207

So, D_b = 30*(52 / 0.249)^(1 / 2.207) = 334.512 mm

For a split ring floating head exchanger typical shell clearance from figure13, is 53 mm.

So, shell inside diameter, Ds = 334.512+53

= 387.512 mm

= 388 mm

Tube Side Heat Transfer Coefficient:

REYNOLDS NUMBER:

$$\mathbf{R_e = dv\rho/\mu}$$

$\mathbf{R_e = 6.345 * 10^4}$

PRANDTL NUMBER:

$$\mathbf{P_r = C_P\, \mu/k}$$

$\mathbf{P_r = 14.93}$

LENGTH TO DIAMETER RATIO:

$L/d_i = 281$

From figure7, heat transfer factor, j_h =3.0*10^(-3)

$$\mathbf{Nu = hd/k = j_h * Re * P_r^{(1/3)} * (\mu/\mu_W)^{0.14}}$$

Nu = 468.712

Therefore, inside heat transfer coefficient, h_i = (468.712*0.1085*10^3 / 26)
= 1816.25 W/m^2 ^{0}C

h_i = 1816.25 W/m^2 ^{0}C

__Shell Side Heat Transfer Coefficient:__

The shell inside diameter = 388 mm

As a first trial take baffle spacing, l_B = D_s/2.5
= 155 mm

Calculating area of cross section of the shell equator as given below:

$$\mathbf{A_S = (P_t - d_0) D_s l_B / P_t}$$

A_S =(1.25*30-30)388*155 / (1.25*30)
= 0.01201 m^2

For equilateral triangle pitch the equivalent diameter is given below:

$$\mathbf{D_e = 1.10(P_t\text{^}2 - 0.917 d_0\text{^}2)/d_0}$$

D_e = 1.10(37.5^2-0.917*30^2) / 25
= 21.3015 mm

Volumetric flow rate of the naphtha = 13862.5 / (692*3600*)
= 0.005564 m^3/sec

Therefore, shell side velocity = 0.005564/0.01201
= 0.4632 m/sec

REYNOLDS NUMBER:

R_e = 2.038 *10^4

PRANDTL NUMBER:

P_r = 6.497

Using segmental baffles with 25 % cut
from the figure 9, j_H =4.1* 10^(-3)

Neglecting viscosity correction factor, Nusselt number is calculated as:

N_U = 155.956

So, h_S =871.977 W/m^2 ^{0}C
So shell side heat transfer coefficient, **h_S = 871.977 W/m^2 ^{0}C**

Overall Coefficient:

After putting all the given values U_0 is calculated as,
U_0 = 431.61W/m^2 ^{0}C
So this is above the initial estimate of 350 W/m^2 ^{0}C and (U_{0CAL}- U_{0ASS})/U_{0ASS}= 23.31%
But the number of tubes could possibly be reduced further.

Pressure Drop Calculation:

Tube side: from figure8, j_f = 3.0*10^-3
ΔP_t = 0.252 bar

Shell side: from figure 10, j_f = 4.1 *10^-2
ΔP_S = 0.229 bar

3.5.4 TRIAL 4:

Now taking
$U = 400\ W/m^2\ {}^0C$

Heat transfer area will be **A = 30.671 m^2**

Layout and Tube Size:

Taking outside diameter, d_0	= 30 mm
Inside diameter, d_i	= 26 mm
Length of the tube, L	= 7.32 m
Taking triangular pitch, P_t	= 37.5 mm

Number Of Tubes:

Area of one tube	$= \pi d_0 L = 0.6898 mm^2$
Number of tubes	$= N_t = (30.671/0.6898) = 46$
Taking 2 passes, so tube per pass will be	= 23
Tube cross sectional area	$= \pi d_i^2/4 = 6.157*10^{-4}\ m^2$
So area per pass will be	$= 0.01416\ m^2$
Volumetric flow rate of diesel, m/ρ	= 78400/(737.2*3600)
	$= 0.02954\ m^3/s$

Tube side velocity = **U_t =2.08 m/s**

Hence within range of velocity 1-2 m/s

Bundle and Shell Diameter:

From table5, for 2 tube passes

K_1=0.249 and

n_1 = 2.207

So, D_b = 30*(45 / 0.249)^(1 / 2.207)

= 316 mm

For a split ring floating head exchanger, typical shell clearance from figure13, is 53 mm.

So, shell inside diameter, Ds = 316+53

= 3 mm

= 369 mm

Tube Side Heat Transfer Coefficient:

REYNOLDS NUMBER:

R_e = 7.153*10^4

PRANDTL NUMBER:

P_r = 14.93

LENGTH TO DIAMETER RATIO:

L/d_i = 281

From figure7, heat transfer factor which is j_h =2.9*10^-3

Nu =510.78

Therefore, inside heat transfer coefficient, h_i = 1979.29 $W/m^2\ ^0C$

h_i = 1979.29 $W/m^2\ ^0C$

Shell Side Heat Transfer Coefficient:

The shell inside diameter = 369 mm

As a first trial take baffle spacing, l_B = D_s/2.5

= 148 mm

Calculating area of cross section of the shell equator as:

$$A_S = 0.01092 \text{ m}^2$$

For equilateral triangle pitch, the equivalent diameter is calculated as:

D_e = 21.3015 mm

Volumetric flow rate of the naphtha =13862.5 / (692*3600)

= 0.005564 m^3/sec

Therefore, shell side velocity = 0.005564/0.01092

= 0.5093 m/sec

REYNOLDS NUMBER:

R_e = 2.24 *10 ^4

PRANDTL NUMBER:

P_r = 6.497

Using segmental baffles with 25 % cut

from the figure 9, j_H =4.1* 10^-3

Neglecting viscosity correction factor, Nusselt number is calculated as:

N_U = 171.44

So, $h_S = 958.64\ W/m^2\ {}^0C$

So shell side heat transfer coefficient = $\mathbf{h_S = 958.64\ W/m^2\ {}^0C}$

Overall Coefficient:

After putting all values, U_0 is calculated as,

$U_0 = 473.13 W/m^2\ {}^0C$

Calculating:

$(U_{0CAL} - U_{0ASS})/U_{0ASS} = 18.28\%$

Hence design is accepted.

Pressure Drop Calculation:

Tube side: from figure8, $j_f = 2.90*10^{-3}$

$\Delta P_t = 0.273$ bar

Shell side: from figure 10, $j_f = 4.5*10^{-2}$

$\Delta P_S = 0.271$ bar

Pressure drops in shell side and tube side are within range.

3.5.5 FINAL DESIGN (on the basis of optimization)

So comparing all the designs we got a conclusion that design four will be more economical than others. Since in design four, total number of tubes is less than others and all the design parameters especially pressure drop and velocity are in the proper range. As a process designer our aim is to satisfy the most economical and effective practical procedure and plant requirements to produce a new product or to manufacture an existing product, by new means or more generally to bring about some designated material transformations on a commercial scale to satisfy market need.

3.5.6 SUMMARY:

Table 2: Proposed design

SHELL SIDE(One pass)	TUBE SIDE(Two passes)
Shell side fluid is naphtha	Tube side fluid is diesel
Internal diameter D_S = 369 mm	Tube inside diameter is 26 mm
Baffle spacing is 148, 25 % cut	Tube outside diameter is 30 mm
Number of baffles is 10	Total number of tubes is 45
Shell side velocity is 0.50 m/s	Tube side velocity 2.08 m/s
Shell side coefficient is 958.64W/m^2 ^{0}C	Tube side coefficient is 1979.29 W/m^2 ^{0}C
Reynolds number 7.153*10^4	Reynolds number 2.24*10^4
Prandtl number 6.49	Prandtl number 14.93
$J_H = 4.1*10^{-3}$, $j_f = 4.5*10^{-2}$	$J_H = 2.9*10^{-3}$, $j_f = 2.9*10^{-3}$
Shell side pressure drop 0.273 bar	Tube side pressure drop 0.271 bar

CHAPTER 4

CALCULATION BY BELL'S METHOD

Now using Bell's method, we calculate the shell-side heat transfer coefficient and pressure drop for the exchanger designed.

In Bell's method the heat-transfer coefficient and pressure drop are estimated from correlations for flow over ideal tube-banks, and the effects of leakage, bypassing and flow in the window zone are also allowed by applying correction factors.

This approach will give more satisfactory predictions of the heat-transfer coefficient and pressure drop than Kern's method; and, as it takes into account the effects of leakage and bypassing, it can be used to investigate the effects of constructional tolerances and the use of sealing strips.

Summary of proposed design:

Number of tubes	= 45
Shell I.D.	= 369 mm
Bundle diameter	= 316 mm
Tube O.D.	= 30 mm
Pitch 1.25	= 37.5 mm
Tube length	= 7.32 m

- Values calculated in Kern method:

A_s $= 0.0192\ m^2$

d_0 $= 30*10^{-3}$ m

Pr $= 6.497$

4.1 Calculation for heat transfer coefficient, h_{oc}:

$$Re = G_S\, d_o / \mu = u_s\, d_o\, \rho/\mu$$

$$= 0.005564*692*30*10^{-3}\,(0.01092*0.335*10^{-3})$$

$$= 3.15*10^{4}$$

from figure 14, $j_h = 5.1*10^{-3}$

h_{oc} = heat transfer coefficient calculated for cross-flow over an ideal tube bank, no leakage or bypassing

$$h_{oc} = (k_f / d_o) * j_h * Re * P_r^{(1/3)}$$

where

k_f = 0.1191 W/m ^{0}C

Hence, $\mathbf{h_{oc}}$ **= 1190.07 W/m^2 ^{0}C**

4.2 Calculation for tube row correction factor, Fn:

P't =0.87* Pt for equilateral triangular pitch

Pt'= 0.87*1.25*30= 32.62

Hc= baffle cut height = D_s x B_c, (B_c is the baffle cut as a fraction=0.25)

Hc= 0.25*369=92.25 mm

H_b=height from the baffle chord to the top of the tube bundle

H_b= (D_b/2) - D_s*(0.5-Bc)

= (316/2) - 369*(0.5-0.25)

= 65.75 mm

N_{CV} = number of tube rows crossed (in the cross-flow region)

N_{CV}= (D_b -2H_b)/ Pt'

=(316-2*65.75)/32.62 = 5.656

From figure 15,

Fn = 0.97

Fn is the correction factor to allow for the effect of the number of vertical tube rows.

4.3 Calculation for window correction factor, Fw:

H_b = 65.75 mm

Baffle cut = H_b/D_b = 0.2080 = 20.8%

From figure 24, at cut of 0.208,

Ra'= 0.14

R'a is the ratio of the bundle cross-sectional area in the window zone to the total bundle cross-sectional area

Tubes in one window area = N_W =Nt *Ra'= 45*0.14 = 6.3

Tubes in cross flow area = N_C =Nt –2 Nw = 45-2*6.3 = 32.4

R_W = 2*Nw/Nt= 2*6.3/45 = 0.28

F_W = 1.07

This factor corrects for the effect of flow through the baffle window, and is a function of the heat-transfer area in the window zones and the total heat-transfer area.

4.4 Calculation for bypass correction factor, F_b:

$$F_b=\exp\left[-\alpha * A_b/A_s * \left(1-\{2N_s/N_{cv}\}^{1/3}\right)\right]$$

A_b = clearance area between the bundle and the shell

A_b = $L_b(D_S - D_b)$

= 148(369-316)

= $7.844*10^3$ mm^2

Ab/As = $7.844*10^{-3}/0.01092$

= 0.718

Ns = number of sealing strips encountered by the bypass stream in the cross-flow zone

Ncv = number of constrictions, tube rows, encountered in the cross-flow section

Trying one strip for each five vertical rows:

Hence $N_S / N_{CV} = 1 / 5$

$F_b = \exp \{-1.35*0.718*(1-[2*1/5]^{1/3})\}$

$\mathbf{F_b=0.77}$

This factor corrects for the main bypass stream, the flow between the tube bundle and the shell wall, and is a function of the shell to bundle clearance, and whether sealing strips are used.

4.5 Calculation for leakage correction, F_L:

Using clearances as specified in the Standards,

Tube to baffle = 0.8 mm

Baffle to shell = 4.8 mm

At_b = tube to baffle clearance area, per baffle

$$A\,t_b = C_t\,\pi\,d_o *(N_t-N_w)/2$$
$$= 0.8*\pi*30*(45-6.3)/2$$
$$= 1458.95 \text{ mm}^2$$
$$= 1.458*10^{-3} \text{ m}^2$$

As_b = shell-to-baffle clearance area, per baffle

$As_b = C_S D_S\,(2\pi-\theta b)/2$

From figure 24, at 20 % cut,

$\theta b = 1.95$ rad

$$As_b = 4.8*369(2\pi-1.95)/2$$
$$= 3837.46 \text{ mm}^2$$
$$= 3.837*10^{-3} \text{m}^2$$

AL = total leakage area

$Al = (As_b + At_b) = 5.295*10^{-3} \text{ m}^2$

$Al / As = 5.2*10^{-3} / 0.01092 = 0.48$

Figure 18 ,$\beta = 0.33$

F_l $=1-\beta*[(At_b+2*As_b)/A_l]$

$= 1-0.33\ [(1.458*10^{-3}+2*3.837*10^{-3})/5.29*10^{-3}]$

$\mathbf{F_l}$ $\mathbf{= 0.431}$

This factor corrects for the leakage through the tube-to-baffle clearance and the baffle-to shell clearance

4.6 Calculation heat-transfer coefficient:

The shell-side heat transfer coefficient is given by:

$$\mathbf{hs = h_{oc}*F_n*F_w*F_b*F_l}$$

hs $= 1190.07*0.97*1.07*0.77*0.43$

hs $\mathbf{= 408.96\ W/m^2\ {}^0C}$

Appreciably lower than that predicted by Kern's method.

4.7 Calculation of Pressure drop:

The pressure drops in the cross-flow and window zones are determined separately, and summed to give the total shell-side pressure drop.

<u>Cross flow zone:</u>

The pressure drop in the cross-flow zones between the baffle tips is calculated from correlations for ideal tube banks, and corrected for leakage and bypassing.

$$\mathbf{\Delta P_c = \Delta P_i * F_b' * F_l'}$$

P_c = the pressure drop in a cross-flow zone between the baffle tips, corrected for bypassing and leakage

ΔP_i = the pressure drop calculated for an equivalent ideal tube bank

F'_b = by-pass correction factor

F'_L = leakage correction factor

a) Calculating P_i:

From figure14, at Re =$3.15*10^4$, for 1.25Δ pitch:

$j_f = 5.5*10^{-2}$

$u_s = 0.5093$ m/s

$$\mathbf{P_i = 8* j_f\, Ncv\, \rho\, u_s^2 /2 * (\mu/\mu_w)^{-0.14}}$$

Neglecting viscosity term:

$P_i \quad = 8* 5.5*10^{-2}*5.656*0.5093^2*692/2$

$\quad = 223.34$ N/m^2

b) Calculating F'_b:

$$\mathbf{F_b' = \exp[-\alpha * A_l/A_s* (1-\{2N_s/N_{cv}\}^{1/3})]}$$

$F'_b \quad = \exp[-4*0.48*(1-\{2/5\}^{1/3})]$

$\quad = 0.603$

c) Calculating F'_l:

$$\mathbf{F'_l = 1- \beta'* [(A_{tb}+2*A_{sb})/A_l]}$$

From figure 21,

$\beta' = 0.54$

$F'_l \quad = 1- 0.54* [(1.458*10^{-3}+ 2* 3.837*10^{-3})/5.29*10^{-3}]$

$\quad = 0.0678$

Hence ΔP_c = 223.34 * 0.603 * 0.0678

= 9.13 N/m^2

Window zone

From figure 24,

for baffle cut 20.8 per cent (0.208)

$R_a = 0.15$

$A_w = (\pi/4 * D_s^2 * R_a) - (N_w * d_o^2 * \pi/4)$

$A_w = (\pi*369^2*0.15/4) - (\pi*30^2*6.3/4)$

$= 0.0115\ m^2$

velocity in the window zone, u_w $= 0.005564/0.0115$

$= 0.4838$ m/s

geometric mean velocity ,u_z $= (u_w * u_s)^{1/2}$

$= (0.4838 * 0.5093)^{1/2}$

$= 0.49$ m/s

$N_{wv} = H_b / Pt'$

where N_{wv} = number of restrictions for cross-flow in window zone

H_b = height from baffle chord to the top of tube bundle

$P't$ = vertical tube pitch

$N_{wv} = 65.75/32.62$

$= 2.015$

$$\mathbf{\Delta P_w = F'_L\ (2 + 0.6\ N_{wv}) * \rho u_z^2/2}$$

$\Delta P_w = 0.0678*(2+ 0.6*2.015)*692*0.49^2/2$

$= 18.07\ N/m^2$

End zone

$$\mathbf{\Delta P_e = \Delta P_i\ [(N_{cv}+N_{wv})/N_{cv}]*F'_b}$$

$\Delta P_e = 223.34*[(2.015+5.656)/ 5.656]* 0.603 = 182.65\ N/m^2$

4.8 Total shell-side pressure drop:

Summing the pressure drops over all the zones in series from inlet to outlet gives:

$$\Delta P_s = 2 \text{ end zones} + (N_b - 1) \text{ cross- flow zones} + N_b \text{ window zone}$$

where N_b is the number of baffles $= [(L/l_B) - 1]$

$= 7.32/148 - 1$

$= 48$

$$\Delta P_s = 2\Delta P_e + \Delta P_c (N_b - 1) + N_b \Delta P_w$$

$\Delta P_s = 2*182.65 + 9.13*(48-1) + 18.07*48$

$= 1661$ N/m^2 $= 0.0166$ bar

This is for the exchanger in the clean condition.

Using the factors given in table 6 to estimate the pressure drop in the fouled condition,

$\Delta P_s = 1.40 * 1661 =$ **2325 N/m^2= 0.0232 bar**

Appreciably lower than that predicted by Kern's method. This shows the unsatisfactory nature of Kern method for predicting the shell-side pressure drop.

CHAPTER 5

MECHANICAL DESIGN

Shell and tube heat exchanger data:

a) Shell side data:

Material of construction	: Carbon Steel (corrosion allowance- 3mm)
Number of shell	: 1
Number of passes	: 1
Fluid	: Naphtha
Inlet temperature	: 40 ^{0}C
Outlet temperature	: 125 ^{0}C
Internal diameter D_i	: 369 mm
Internal pressure	: 3 atm = 0.304 N/mm^2
Design pressure	: 0.304 * 1.1 (10% excess pressure) = 0.334 N/mm^2
Permissible stress for carbon steel (f)	: 95 N/mm^2

Segmental baffles (25% cut) with tie rods and spacers

Head	: torispherical
Head: crown radius(Rc)	: 369 mm
Knuckle radius(Rk)	: 6% of Rc = 22 mm

Shell flanges	: male and female tuning
Gasket	: flat metal jacketed asbestos filled
Bolts	: 5% Cr Mo steel
Permissible sress for bolt material	: 140.6 N/mm^2

Nozzle inlet and outlet	: 75 mm
Vent	: 25 mm
Drain	: 25mm
Open for relief valve	: 50mm

b) Tube side data:

Marterail of construction	: stainless steel
Number of tubes	: 45
Outside diameter	: 30mm
Inside diameter	: 26 mm
Length	: 7.32 m
Pitch (triangular)	: 1.25* 30 = 37.5 mm
Fluid	: diesel
Working pressure	: 19 N/mm^2
Design pressure	: 21.5 N/mm^2
Inlet temperature	: 165 ^0C
Outlet temterature	: 153.47 ^0C
Permissible stress	: 100.6 N/mm^2

c) Channel and channel cover:

Material of construction	: carbon steel
Flange joint	: ring type
Gasket	: steel jacketed asbestos
Permissible stress	: 95 N/mm^2
Nozzle (inlet and outlet)	: 75 mm

5.1 Shell Side Calculation

1) Shell thickness:

$$ts= PD_i/ (2fj -P)$$

j = 85%

ts = 0.334* 369 /(2*95*0.85-0.334)

ts = 0.7648 +3 mm(corrosion allowance)

= 3.76 mm

So take ts = 8 mm

2)Thicknesss of head :

$$t_h = PR_C W/ 2fj$$

where

$W = (3+\{Rc / Rk\}^{1/2})*1/4$

$R_c = 369$ mm

$R_k = 22.14064$ mm

$W = 1/4 *(3+(369 /22.14)^{1/2})$

= 1.77

t_h = (0.334* 369*1.77) / (2*95*1)

= 1.148+3

= 4.148 mm

So take 8mm

3)Thickness of nozzle:

$$t_n = Pd / (2fj-P)$$

d = 75mm

j = 1 (seamless pipe)

t_n = 0.334*75 / (2*95*1-0.334)

= 0.132+ 3

= 4mm

4)Tie rods and spacers:

Number of tie rods = 6

Diameter of tie rods = 10mm

5)Shell side flanges(between shell and tube sheet):

a) Gasket size selection:

Gasket –flat metal jacketed

Flange facing - male and female facing

b) Inner dia of gasket, G_i = D_i +2ts

= 369 +2* 3.76

= 376.58 mm

c) Outlet diameter of gasket, G_o = 376.58 +24

= 400.58 mm

d) Mean = 388.58 mm

e)Width of gasket, N = (400.58-376.58)/2

= 12mm

6) Gasket seating width:

$b_o = N/2$

$= 6mm$

$B = b_0 (b_o < 6.3)$

7) Gasket Load:

Gasket seating stress: $Ya = 5.34$ kg/ mm^2

Gasket factor $m = 3.75$

$Wm_1 = \pi\ bGya$

$Wm_2 = \pi 2b\ G\ mP + \pi /(4G^2P)$

$Wm_1 = \pi * 6 * 388.58 * 53.4$

$= 391.131$ KN

$Wm_2 = \pi * 2 * 6 * 388.58 * 3.75 * 0.334 + \pi/(4 * 388.58^2 * 0.334)$

$= 18.348$ KN

8) Thickness of flange:

$K = 1/(0.3 + 1.5\ Wnh_G / HG)$

$Wm_1 = Wn = 391.131 * 10^{3}$ N

$h_G = (B - G)/2$

$B = 525$ mm (from tube side calculation)

$G = 388.58$ mm

$h_G = 136.42$

$H = \pi/4 * (G^2 P)$

$= \pi/4 * (388.58)^2 * 0.334$

$= 158437.225$ N

So substituting these values to get $K = 0.62$

$t_f = G(P/Kf) + C$

$= 388.58(0.334/[95 * 0.62]) + 3$

$= 5.20$ mm

5.2 Tube side calculation:

1)Thickness of tube:

t_t = $Pd_o/(2fj+P)$

P = 21.5 N/mm^2

do = 30 mm

f = 100.6N/mm^2

j = 1

t_t = 21.5* 30/(2*100.6+21.5)

= 2.89

= 3mm

2)Thickness of tube sheet:

ts = FG $(0.25P/f)^{1/2}$

F = 1

G = 388.58mm

F = 100.6 N/mm^2

P = 21.5 N/mm^2

ts = 1*388.58$(0.25*21.5/100.6)^{1/2}$

= 89.819 + 3 mm

= 92.819mm

= 95mm

3)Thickness of channel :

tc = Gc $(KP/f)^{1/2}$

Gc= 388.58mm

K = 0.3(arrow face)

P = 21.5 N/mm^2

F = 95 N/mm^2

Tc = 388.58$(0.3*21.5/95)^{1/2}$

= 100 mm

4) Flange joint (between tube sheet and channel)

a) Gasket size selection:

Gi = di or do	=376.58 mm
Ring gasket width(W)	=22mm
$G_0 = 2(W+G_i/2)$	$= 2(22+376.58/2) = 420.58$ mm
$G = (G_O+G_i)/2$	= 398.58 mm
B_o = W/8 (ring type)	=22/8 = 2.75mm
$B = b_o$	= 2.75 (b_0< 63.75)
ya =126.6 N/mm^2	
m=5.5	

b)Bolt load :

Wm_1 $= \pi\, b\, G_i\, ya$

$= \pi*2.75 *126.6*376.58$ mm

$= 4.118 *10^5$ N

Wm_2 $= \pi*2b * GmP + \pi/4\, G^2P$

$=2*2.75*\pi*5.5*21.5*398.58 + \pi/4 *21.5*398.58^2$

$=3.497*10^6$ N

Am_1 $= 4.118* 10^5/140.6$

$= 2928.876\ mm^2$

Am_2 $= 3.497*10^6/140.6$

$=24871.977\ mm^2$

c)Number of bolts (n) = G in cm / 2.5

= 398.58 / (10*2.5)

= 15.94

Use 16 bolts

d)Diameter of bolt $= \{(Am_2/16) *(4/ \pi)\}^{1/2}= 4.44$ cm

Use M 48 bolt = Pitch diametr = 44.681 mm

Minor diameter =41.795 mm (IS-3139)

Actual bolt area $= (\pi/4)*[(44.681+41.795)/(2*100)]^2$

$=294\ cm^2$

Min pitch circle diameter (B) $= 398.58 + 22 + 2*48$

$= 516.58$

$=525$ mm

5)Flange thickness (tube side):

$t_f = G(P/Kf)+C$

$K = 1/(0.3+ [1.5\ W_n\ h_G/HG])$

$W_n = 3.497*10^6$

$h_G = (B-G)/2$

$= (525-398.58)/2$

$= 63.21$

$H = \pi/4 *(G^2\ P)$

$=\pi/4*(398.58)^2*21.5$

$=2.682*10^6$ N

By above values we get k=1.6285

So $t_f=398.58\ (21.5/\ [1.62*95])^{1/2}$

$= 148.58 + 3$

$=151$

So take $t_f= 155$ mm

CHAPTER 6

COST ESTIMATION

The cost of shell and tube heat exchanger (CE) is calculated from :

$$C_E = C_B * F_D * F_P * F_M$$

Where:

C_B = Base cost (in carbon steel)

F_D = Cost factor for heat exchanger design type

F_P = Cost factor for design pressure

F_M= Cost factor for material of construction

= 1 for carbon steel

BASE COST:

$$C_B = 50 \exp [8.202 + 0.01506 \ln A + 0.06811(\ln A)^2]$$

where

A = heat transfer area m^2 (lower limit: 14 m^2)

(upper limit: 1100 m^2)

Heat transfer area is calculated as = 30.67 m^2

Hence, C_B =Rs 426663.35

EXCHANGER TYPE COST FACTOR:

$$F_D = \exp(-0.9003+0.0906 \ln A)$$

Heat transfer area is calculated as $= 30.67\ m^2$

$F_D = 0.5542$

DESIGN PRESSURE COST FACTOR:

Design pressure cost factor		Pressure range (kpa)
$F_P = 0.8955+ 0.04981 \ln A$	for	200-2100
$F_P = 1.2002+ 0.07140 \ln A$	for	2100-4200
$F_P = 1.4272+ 0.12088 \ln A$	for	4200-6200

Design pressure is calculated as $= 0.334\ N/mm^2$

$F_p = 1.066$

CONSTRUCTION COST FACTOR

$$F_M = Ø_1 + Ø_2 * \ln A$$

Table 3: Material of construction cost factor:

MOC	$Ø_1$	$Ø_2$
Nickel 200	2.4144	0.23456
Monel 400	2.1991	0.15566
Inconel 600	2.1334	0.22177
Incoloy 825	2.3390	0.67888
Titanium	3.2566	0.56666
Hastelloy	3.9879	1.56679

$F_M = 1$ as MOC is CS

Finally C_E is calculated as:

$$C_E = C_B * F_D * F_P * F_M$$

$$C_E = \text{Rs } 426663.35 * 0.5542 * 1.066 * 1$$

$$C_E = \text{Rs } 252066$$

The cost of shell and tube heat exchanger is correlated against heat transfer area only. It excludes such other design, parameter as shell ID, number of tubes, tube length, head type, and other construction details etc. Hence the accuracy of the correlation is limited to preliminary cost estimates only.

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APPENDIX 1

FIGURES:

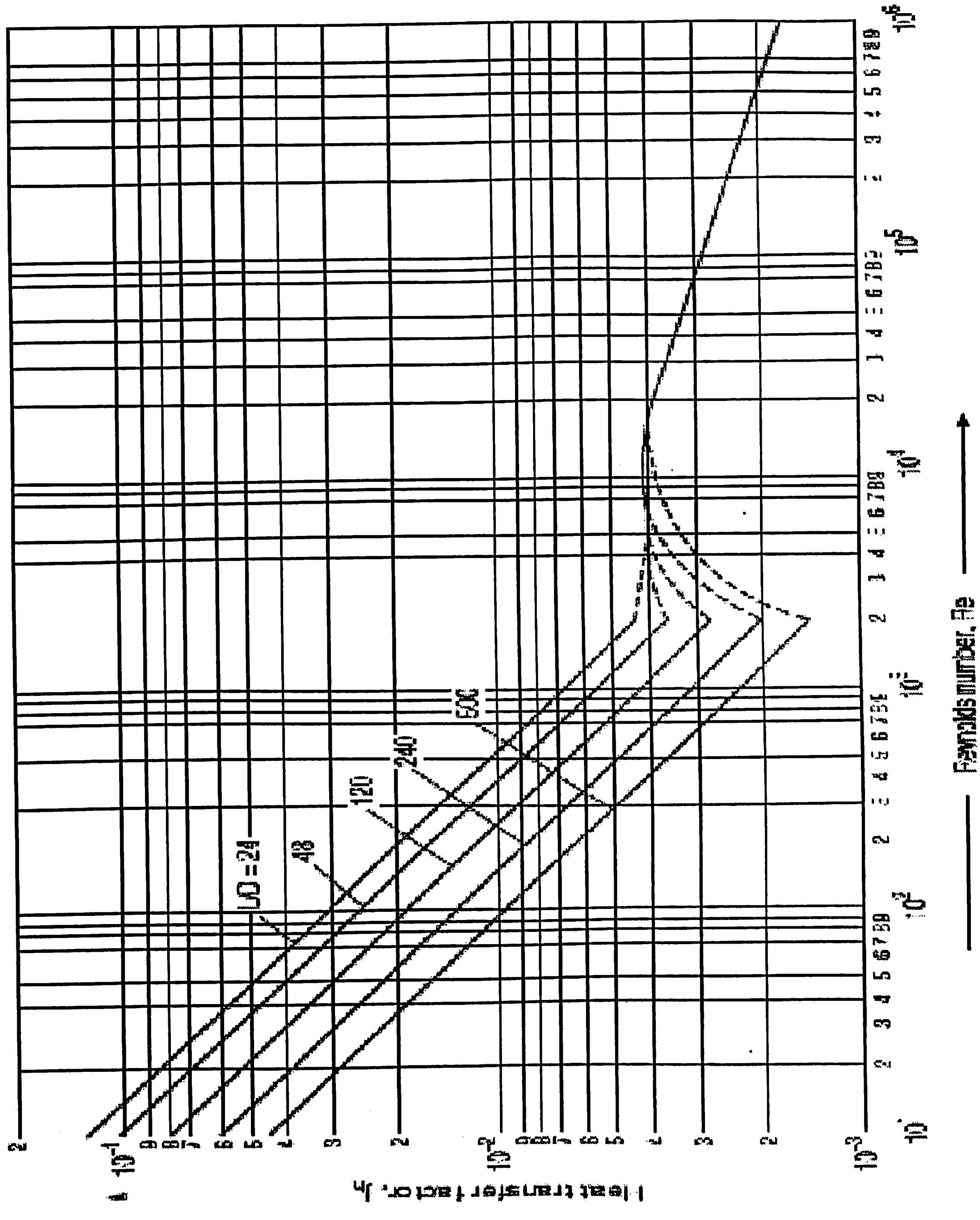


Figure7: Tube side heat transfer factor

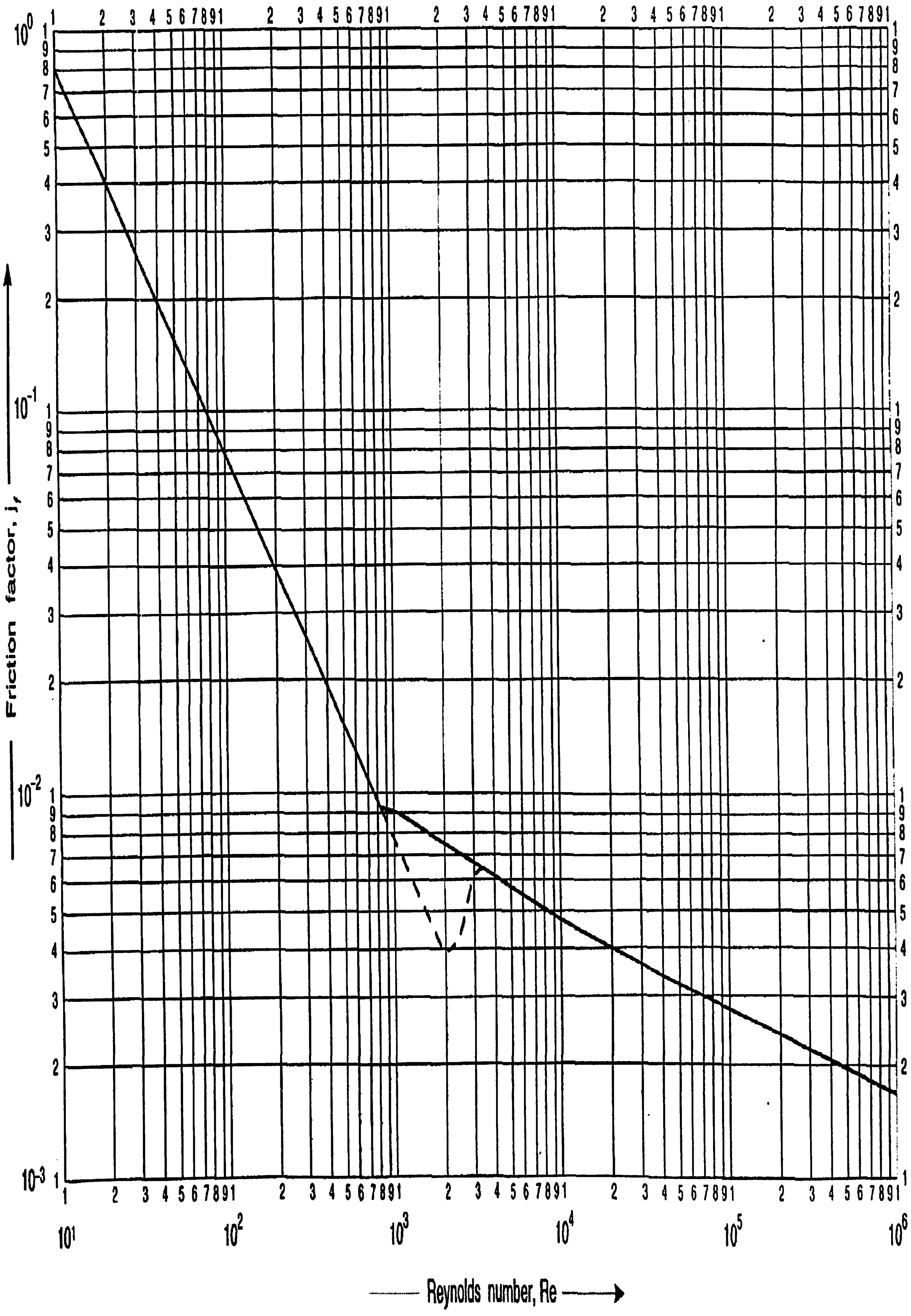


Figure8: Tube side friction factor

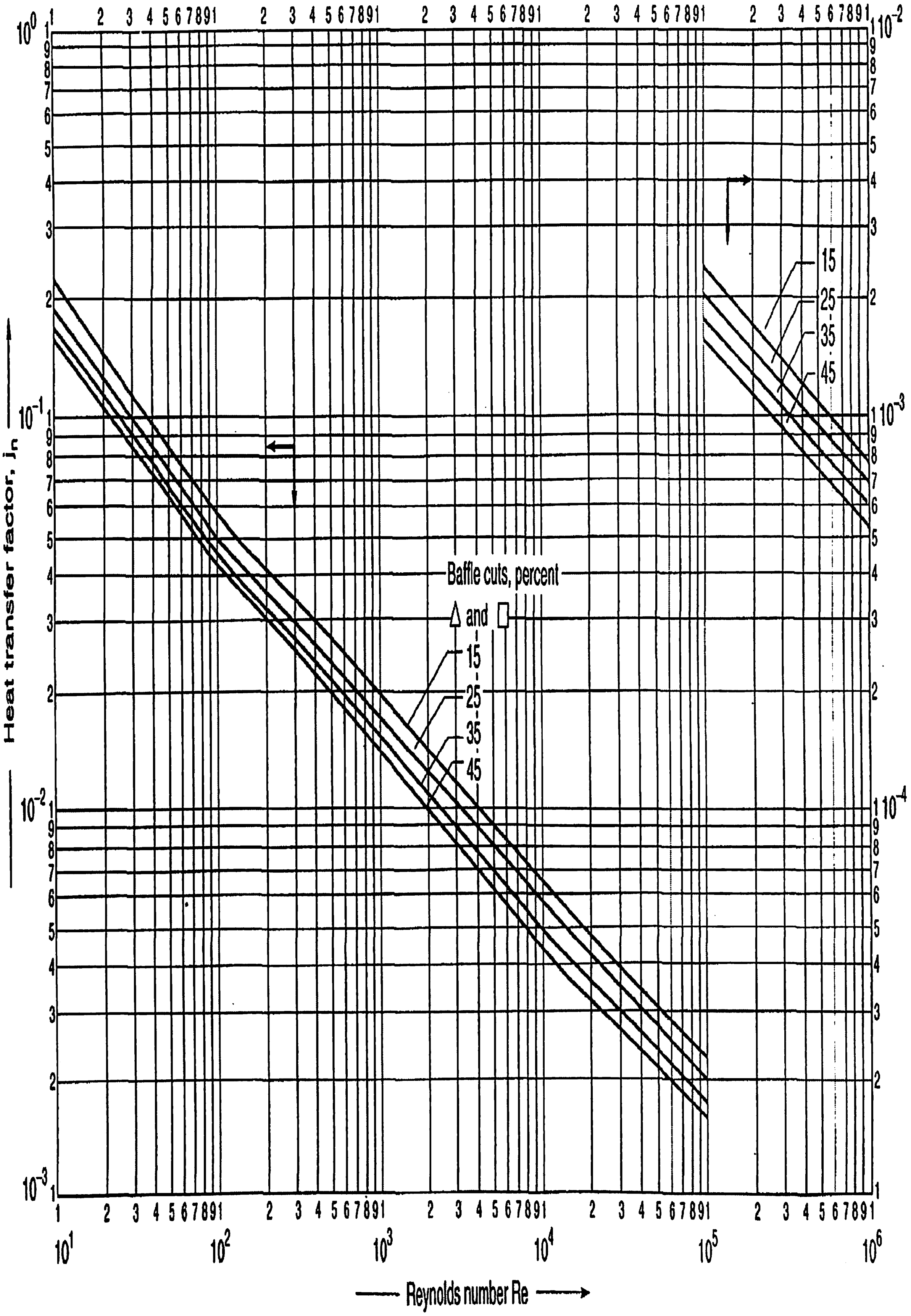


Figure9: Shell side heat transfer factor

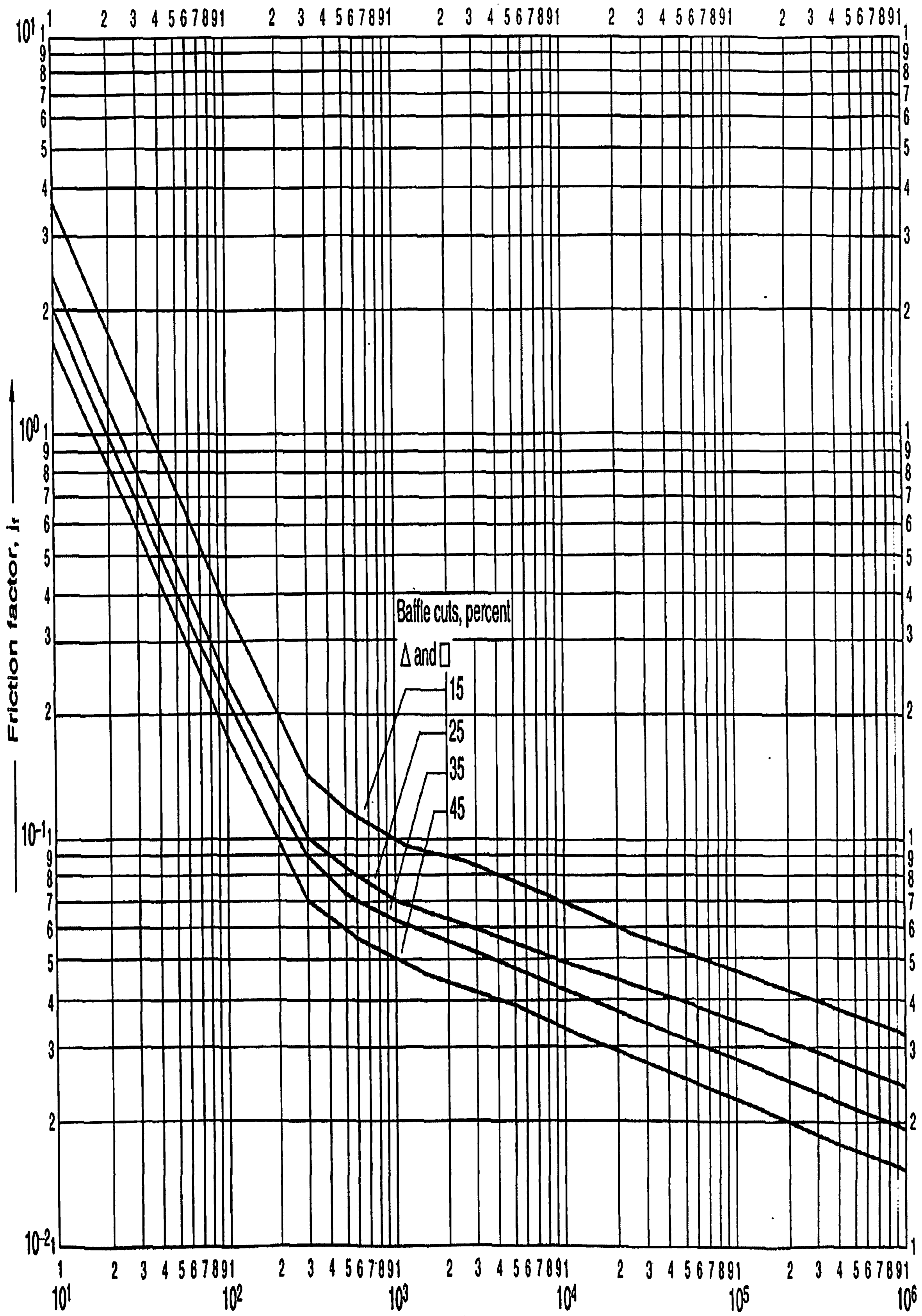


Figure10: Shell side friction factor

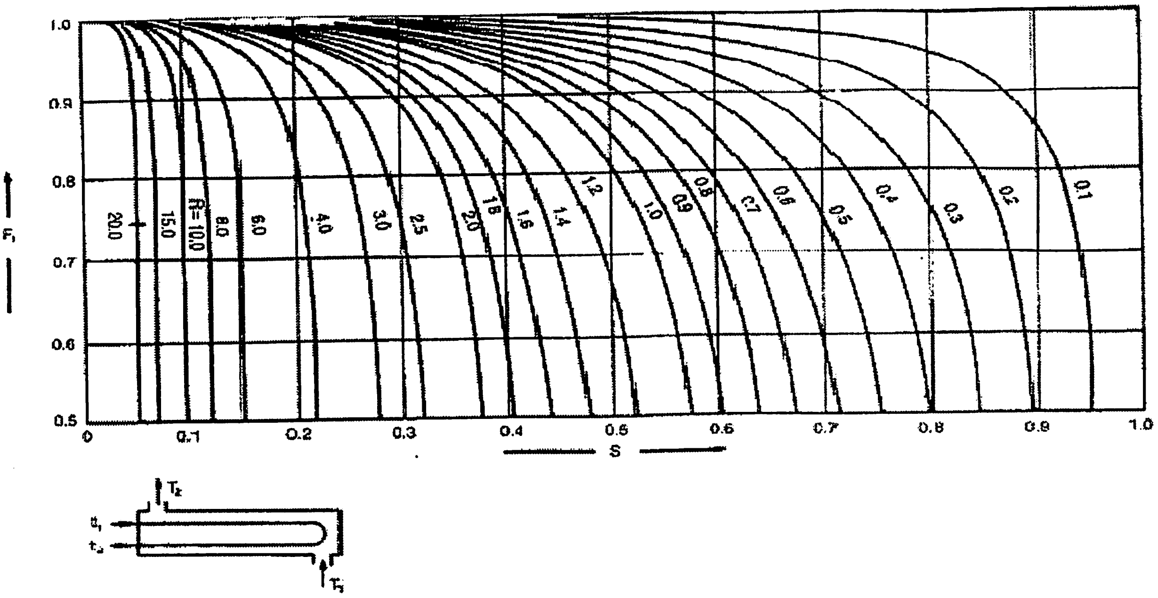


Figure11: Temperature correction factor,1 shell pass, two or more even tube passes

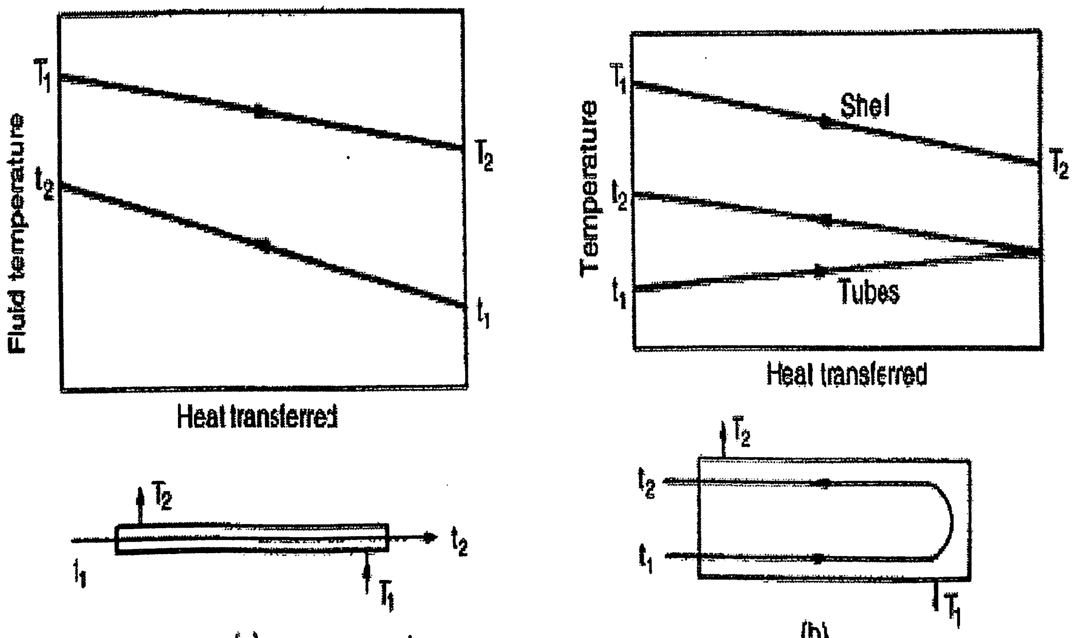


Figure12: Temperature profile (a) Counter current flow (b) 1:2 exchanger

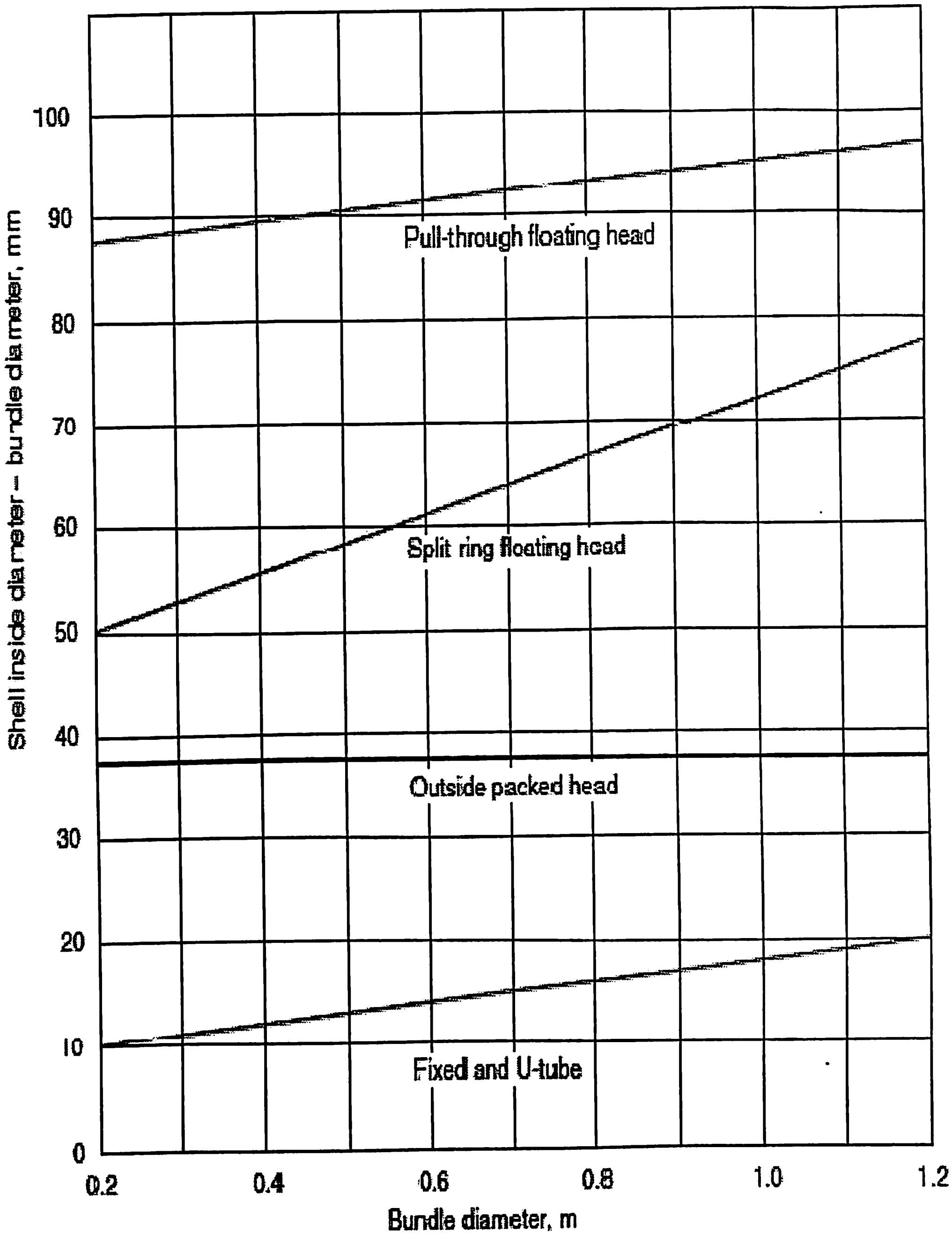


Figure13: Shell bundle clearance

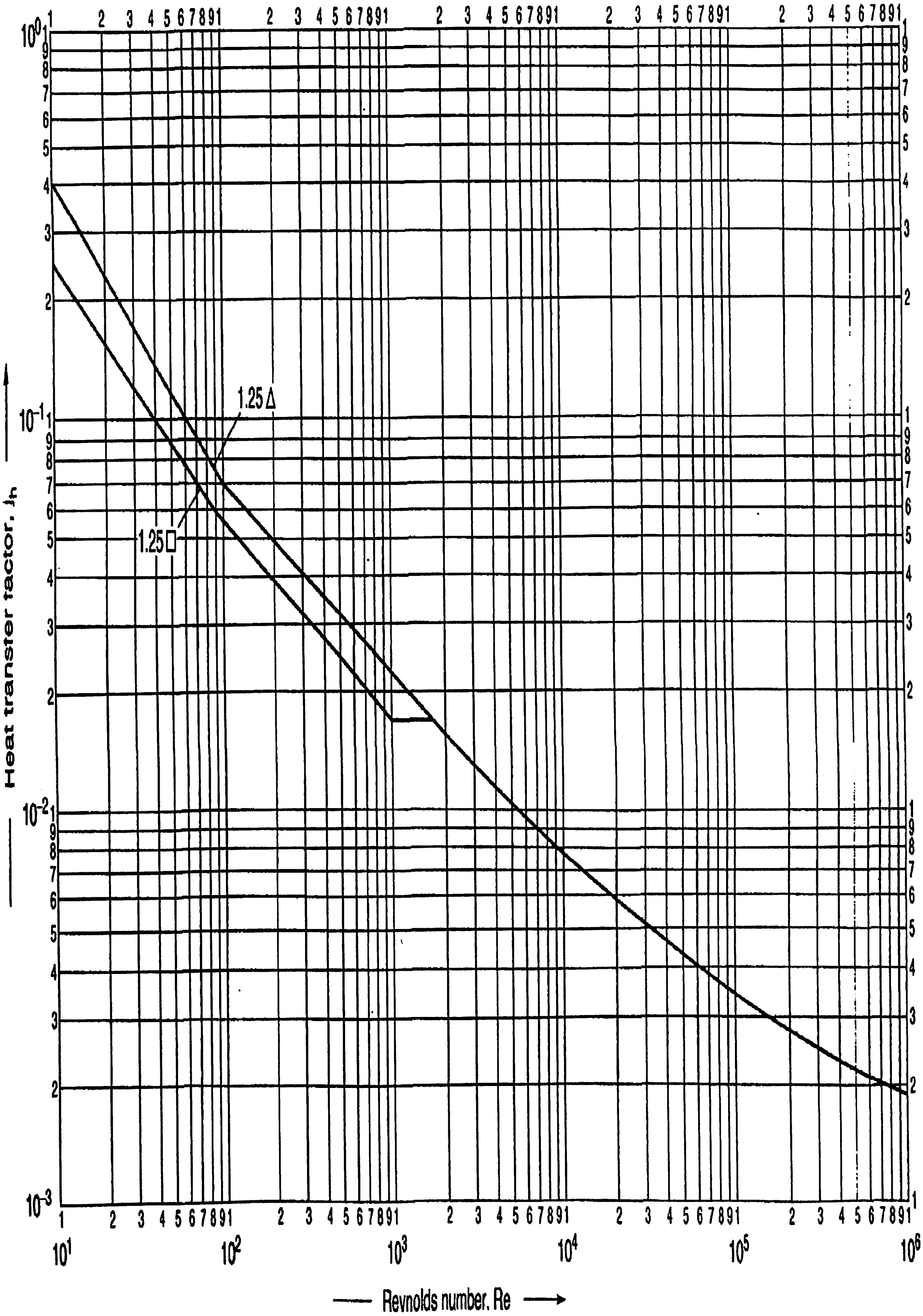


Figure14: Heat-transfer factor for cross-flow tube banks

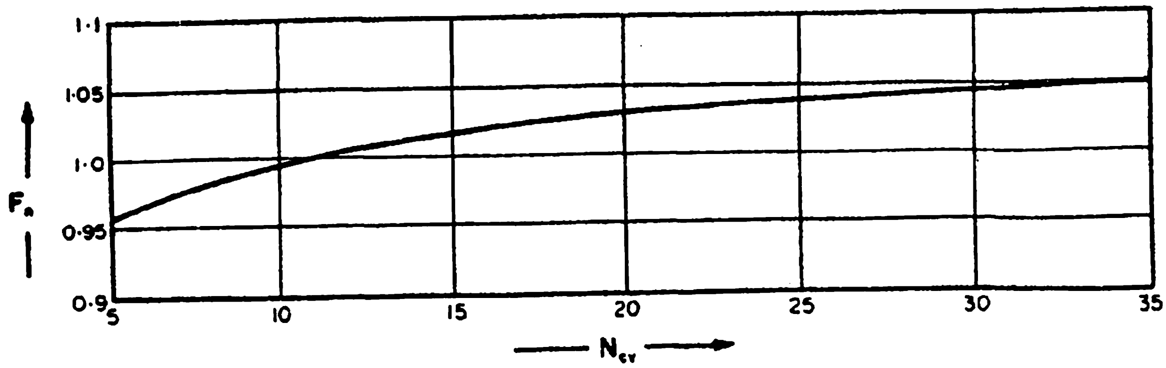


Figure15: Tube row correction factor *Fn*

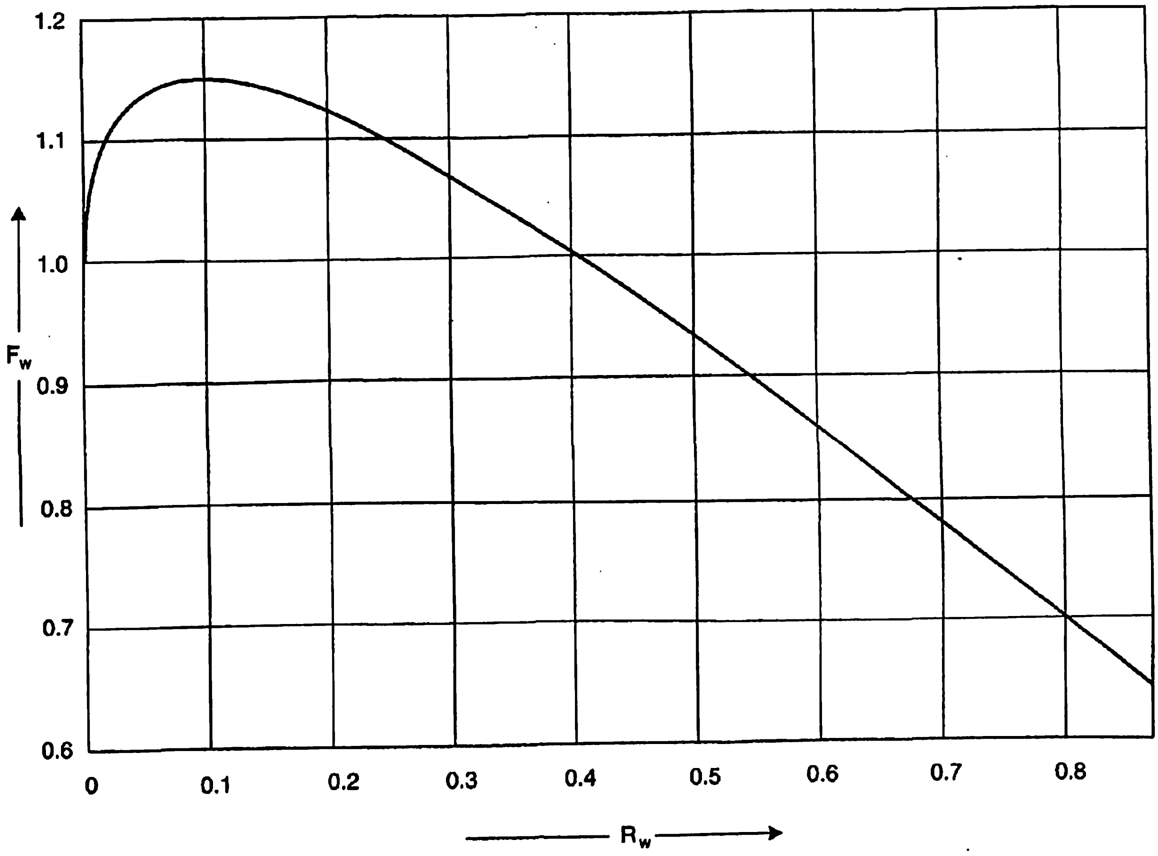


Figure16: Window correction factor

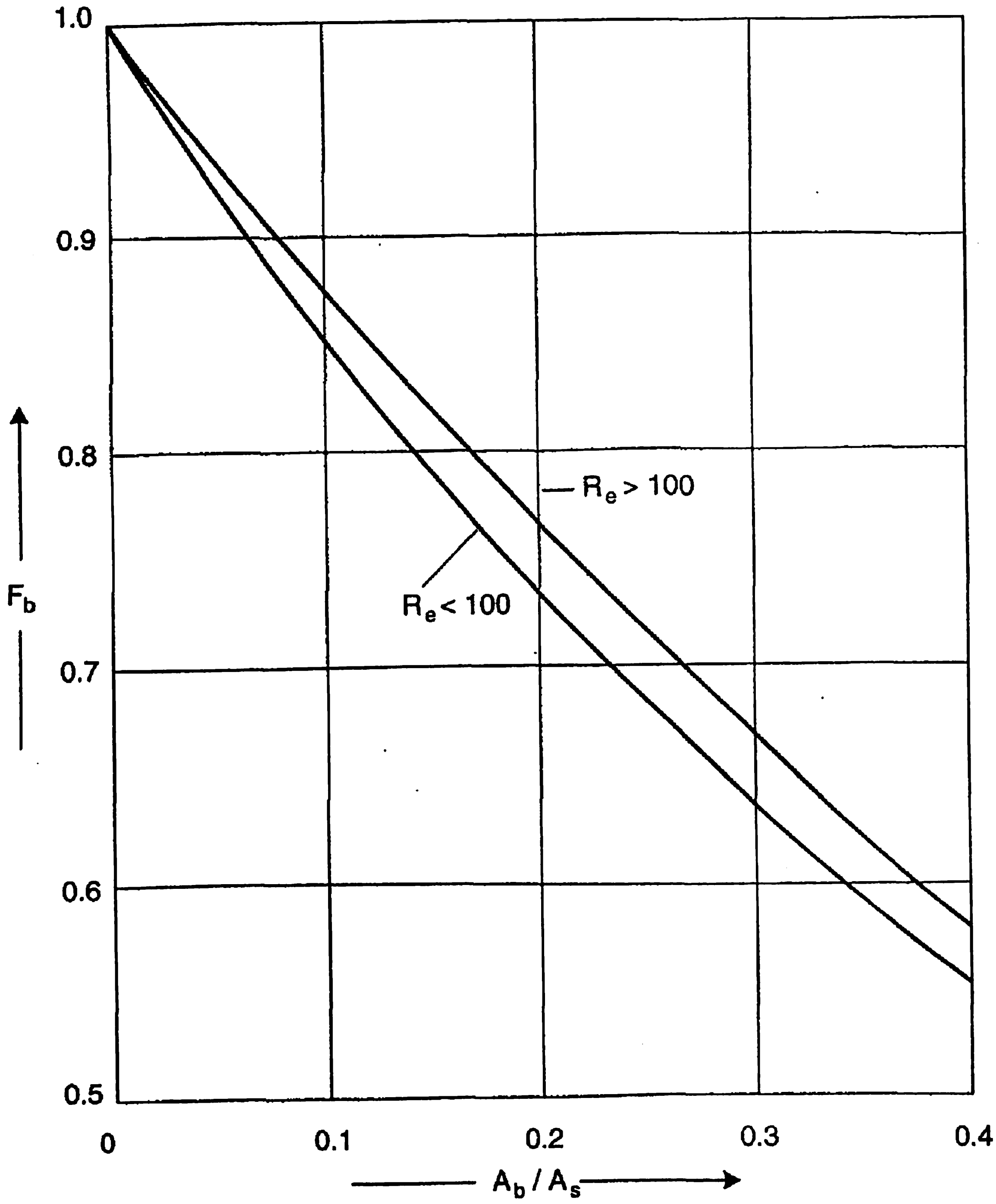


Figure17: Bypass correction factor

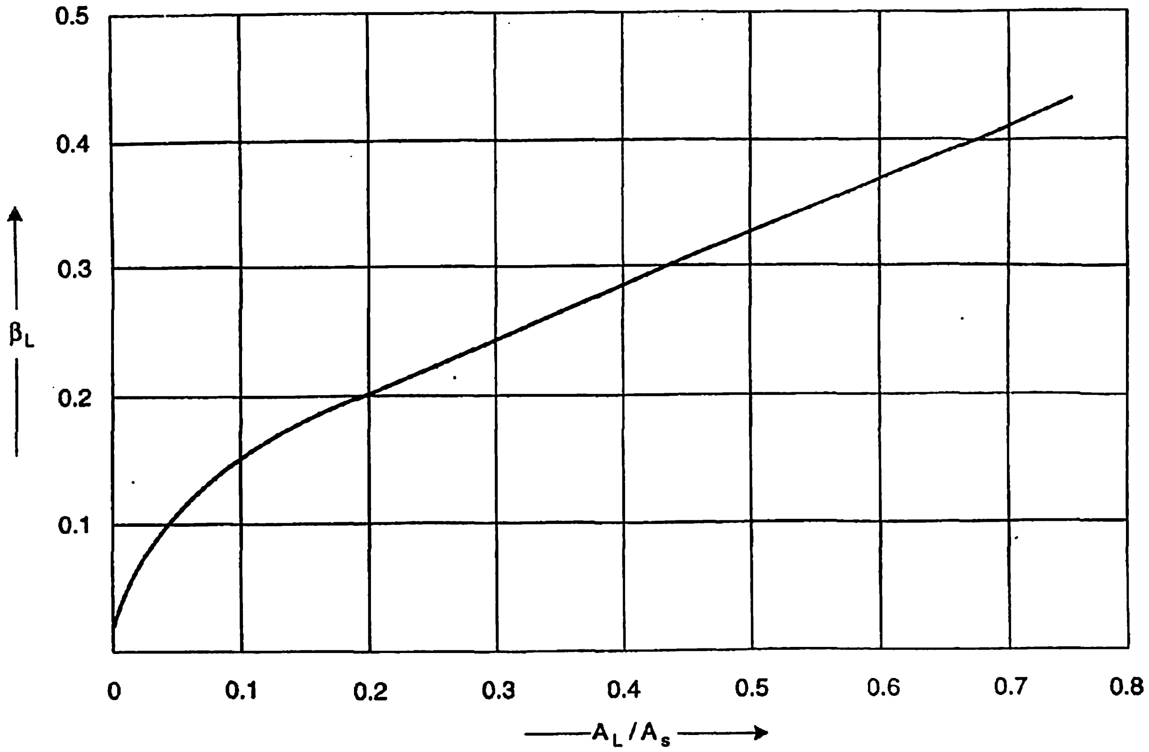


Figure18: Coefficient for F'$_L$, heat transfer

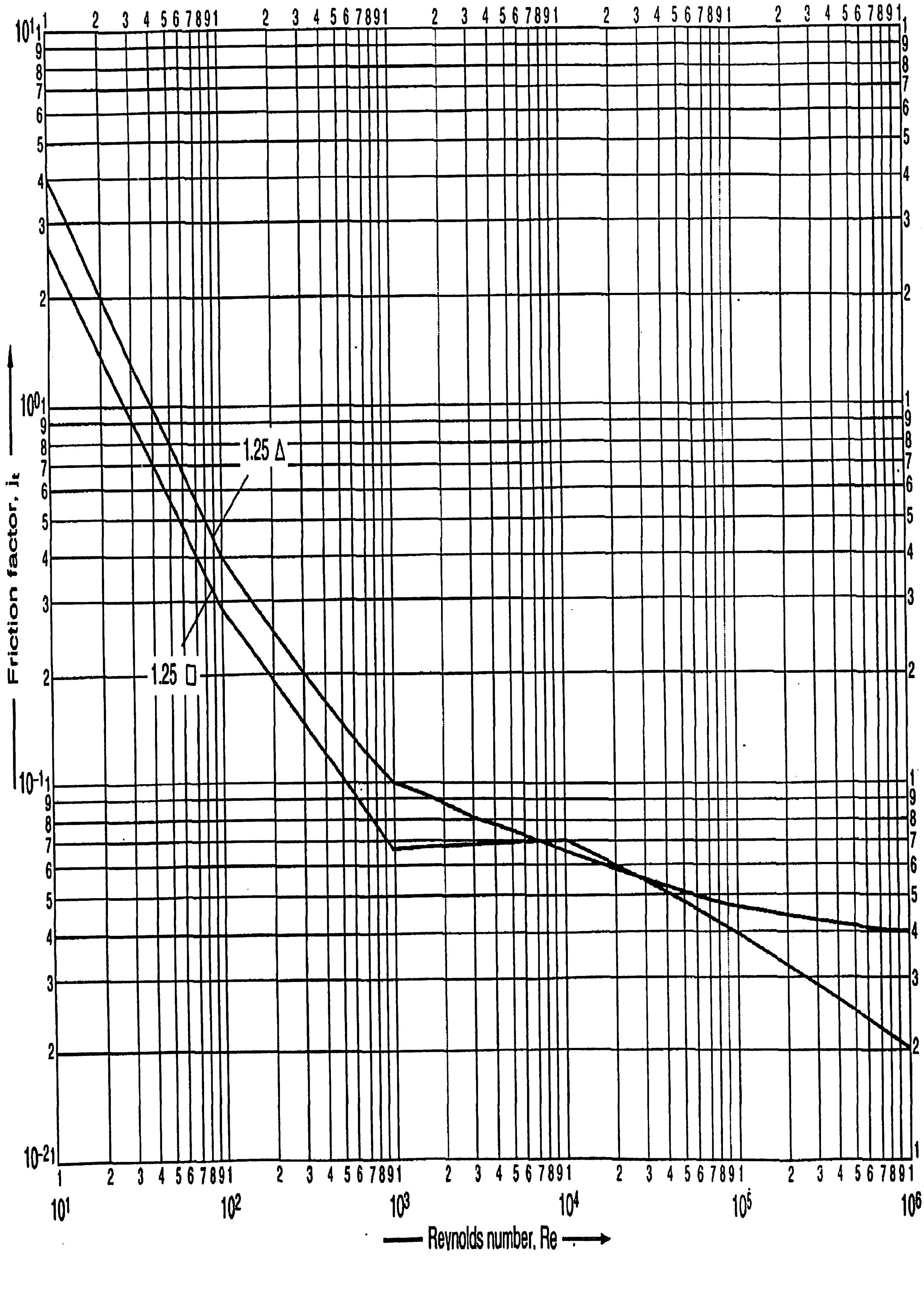


Figure19: Friction factor for cross-flow tube banks

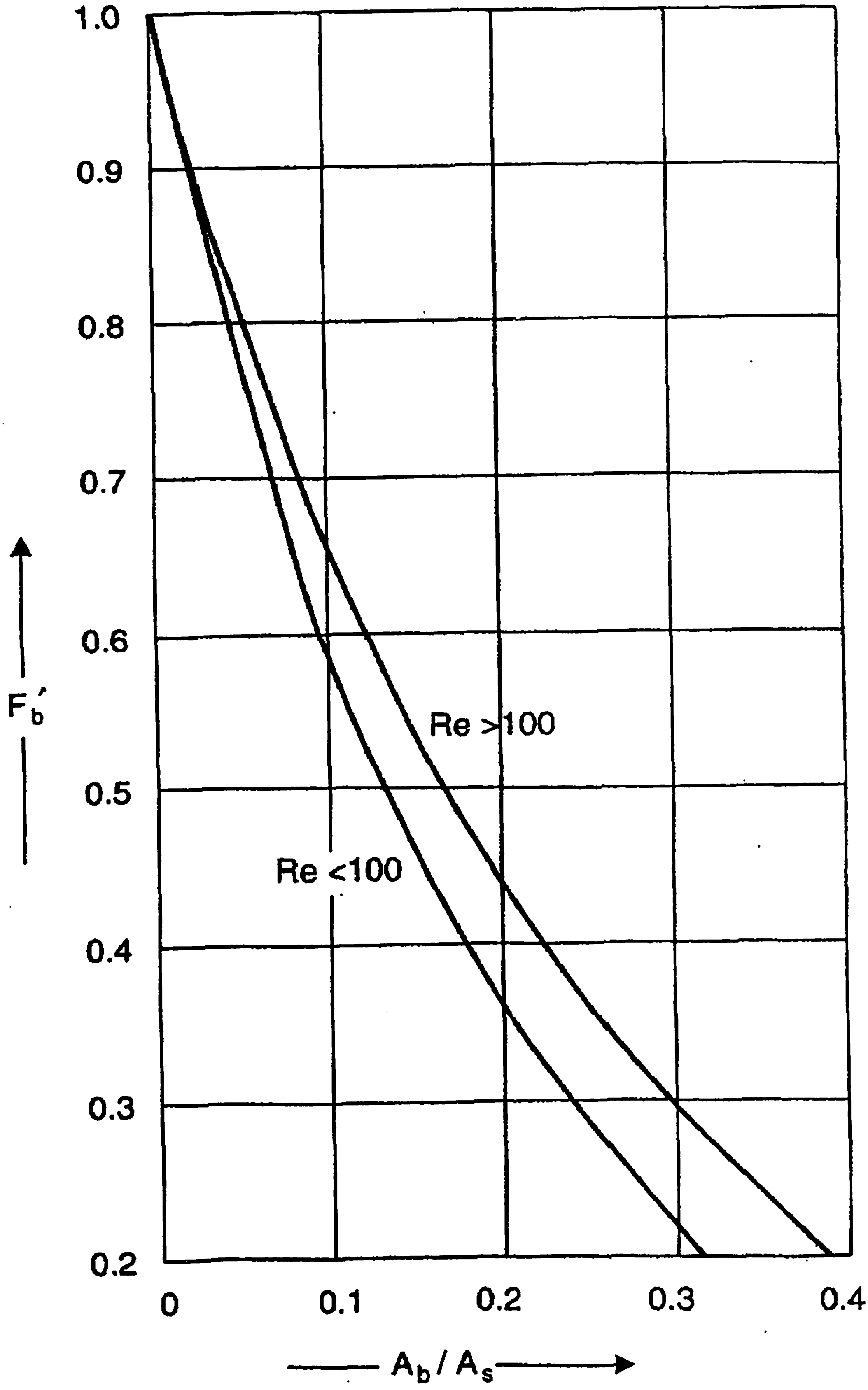


Figure20: Bypass factor for pressure drop *F'b*

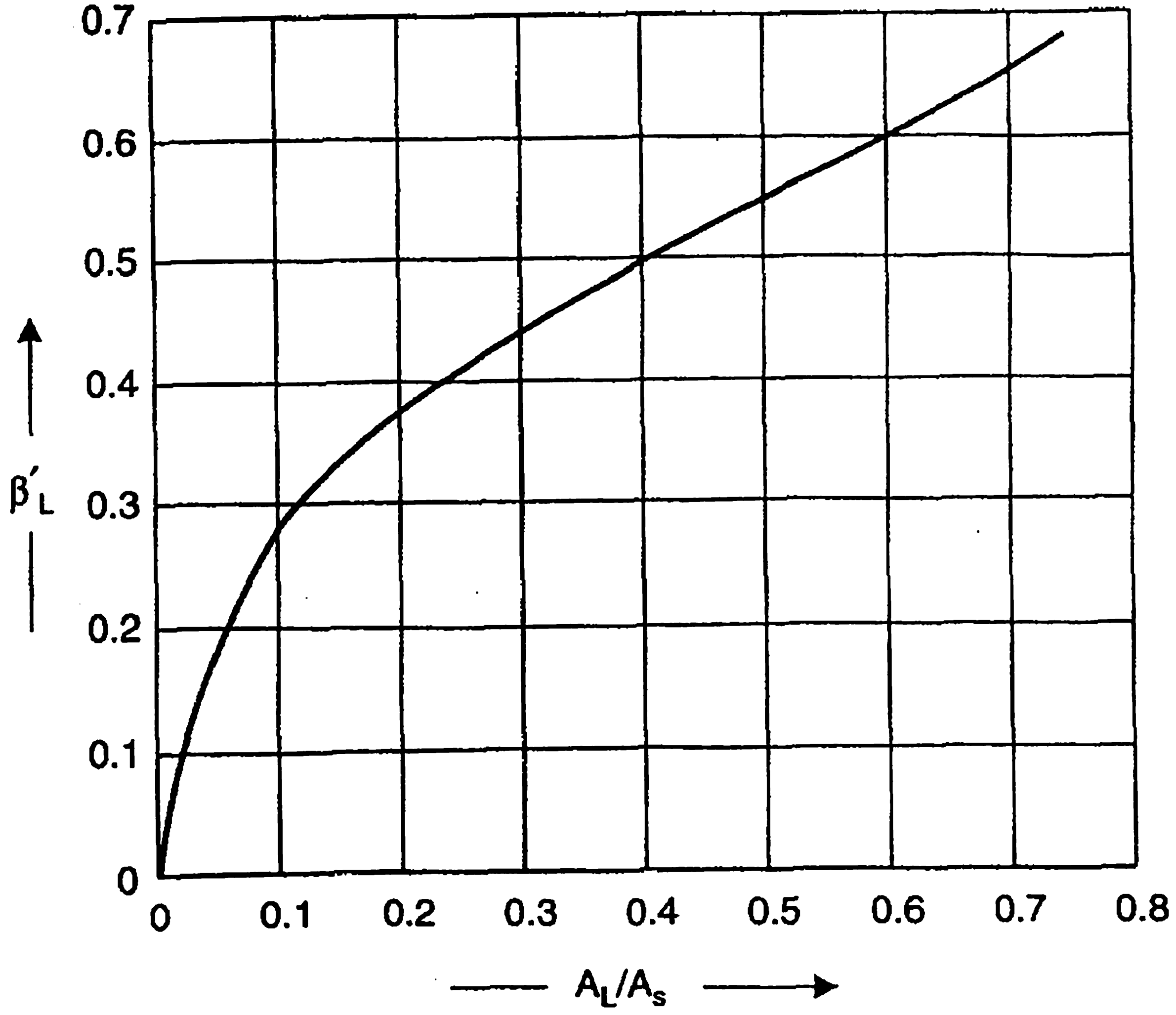
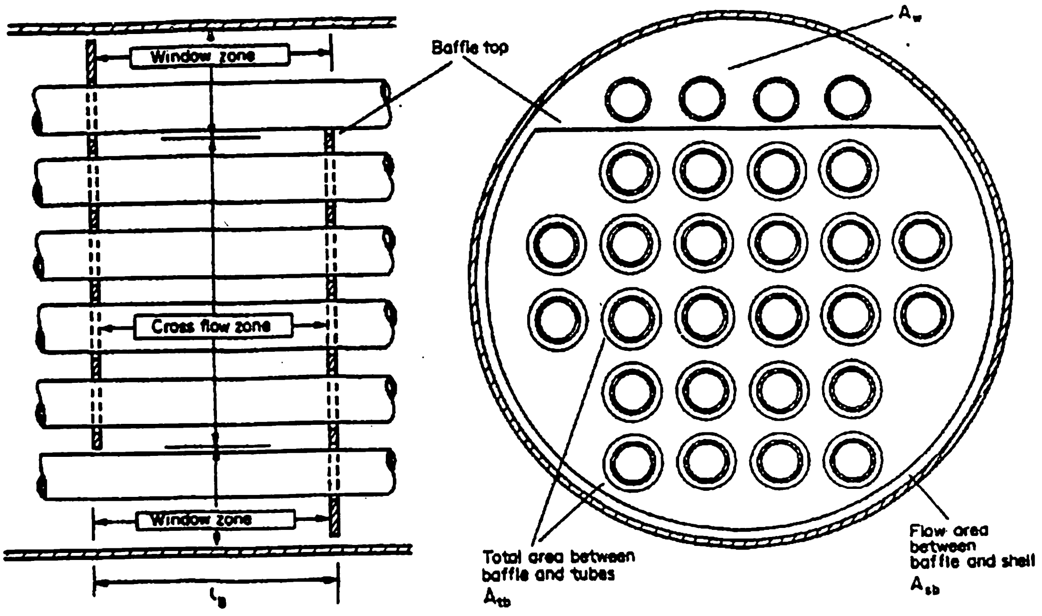


Figure21: Coefficient for *F'L*, pressure drop



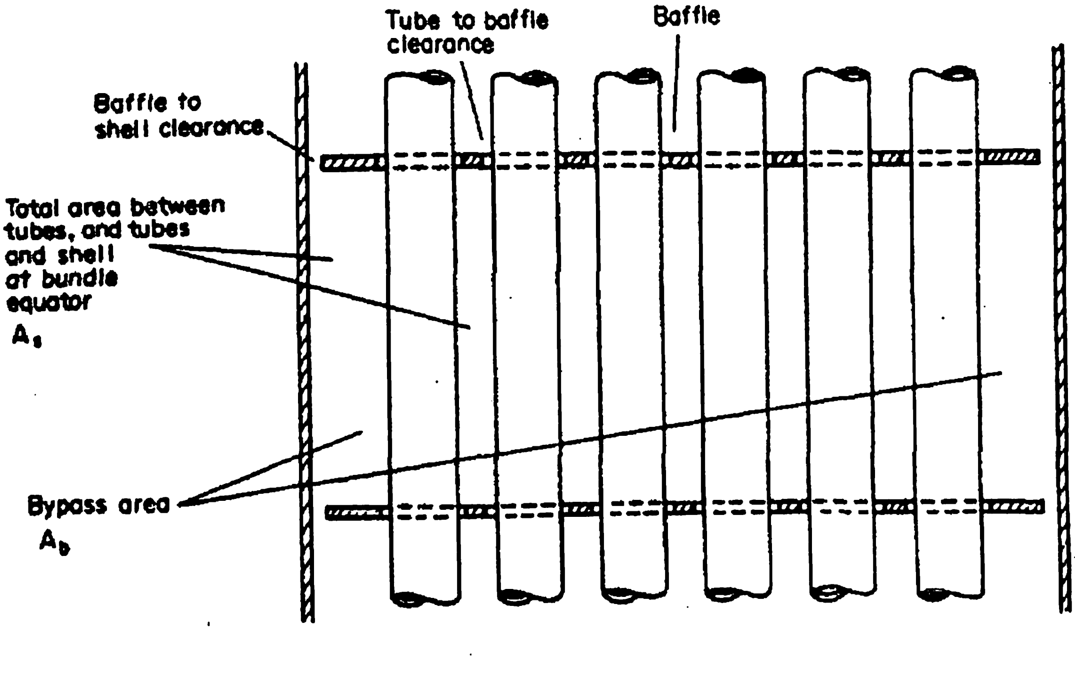


Figure22: Clearance and flow areas in the shell-side of a shell and exchanger

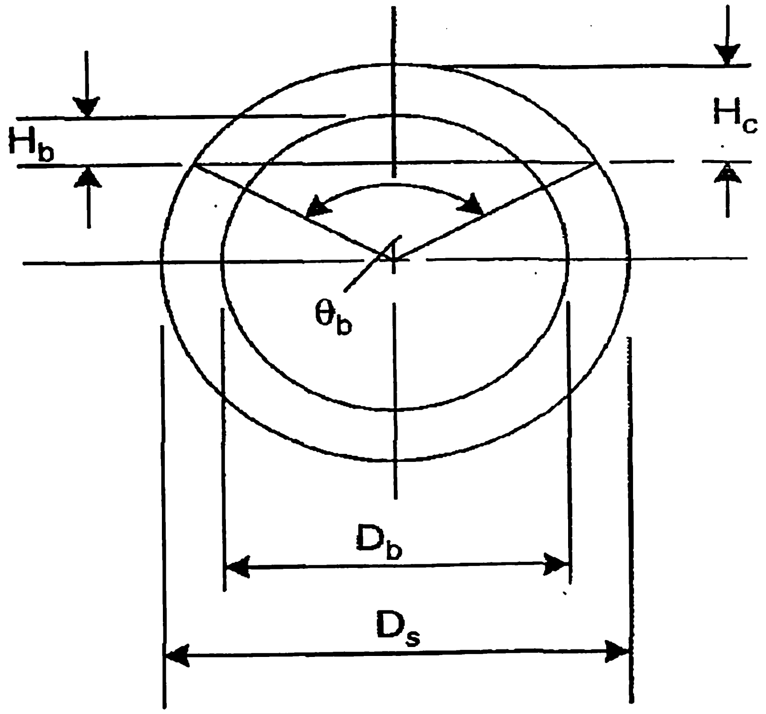


Figure23: Baffle and tube geometry

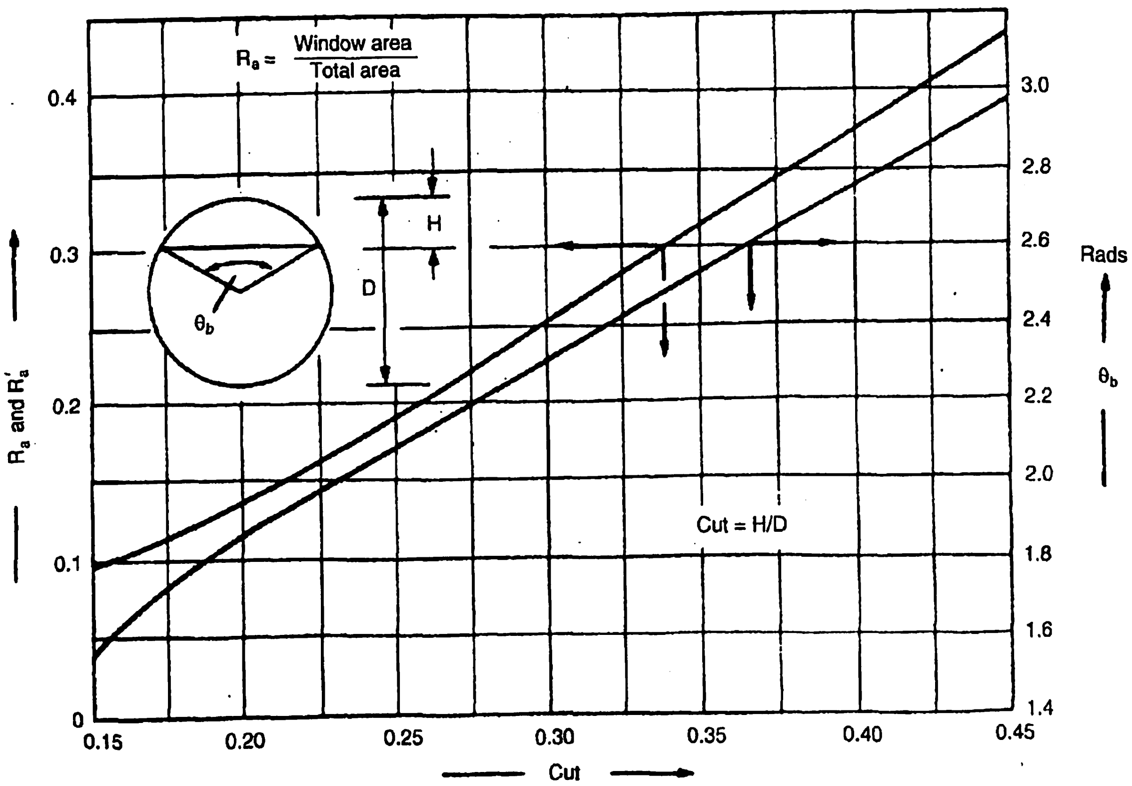


Figure24: Baffle geometrical factors

APPENDIX 2

TABLES:

Table 4: Fouling factors of some typical fluids

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

Table 5: Constants for calculating Shell bundle diameter

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

Table 6: Ratio of fouled to clean pressure drop

Fouling coefficient ($W/m^2 \,°C$)	Shell diameter/baffle spacing 1.0	2.0	5.0
Laminar flow			
6000	1.06	1.20	1.28
2000	1.19	1.44	1.55
<1000	1.32	1.99	2.38
Turbulent flow			
6000	1.12	1.38	1.55
2000	1.37	2.31	2.96
<1000	1.64	3.44	4.77

Table 7: Conductivity of metals

Metal	Temperature (°C)	k_w ($W/m\,°C$)
Aluminium	0	202
	100	206
Brass	0	97
(70 Cu, 30 Zn)	100	104
	400	116
Copper	0	388
	100	378
Nickel	0	62
	212	59
Cupro-nickel (10 per cent Ni)	0–100	45
Monel	0–100	30
Stainless steel (18/8)	0–100	16
Steel	0	45
	100	45
	600	36
Titanium	0–100	16

Table 8: 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