280.371 PROCESS ENGINEERING OPERATIONS

INTEGRATED THERMAL AND HYDRAULIC DESIGN OF CONTINUOUS INDIRECT HEAT EXCHANGERS

STUDY GUIDE

2023

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1 INTRODUCTION

The continuous indirect heat exchanger is a common piece of equipment in the process industries. Its purpose is to enable heat to be transferred from one flowing fluid to another. The thermal design of heat exchangers is important in order to meet the heat transfer duty.

However, power expenditure is required to move fluids through an exchanger because the frictional resistance to flow has to be overcome. The hydraulic design of heat exchangers is therefore equally important.

It is only by integrating thermal and hydraulic design that technically and economically optimal heat exchangers can be built. An optimal design is one that meets the heat transfer duties required and one for which the sum of capital and operating costs is minimized.

The aims of this lecture course are:

- to show how integrated design is carried out,
- to explain the scientific principles upon which it is based, and
- to provide skills for applying the design techniques to common types of process heat exchanger.

1.1 Classification of heat exchangers

Three possible ways (among several) of classifying heat exchangers are by fluid, by flow arrangement and by type.

1.1.1 By fluid

Liquid	-	liquid
Liquid	-	condensing/evaporating fluid
Gas	-	liquid
Gas	-	condensing/evaporating fluid
Gas	-	gas

1.1.2 By flow arrangement

Flow arrangement	Typical fluids
Parallel flow Countercurrent flow Cross flow Mixed flow	liquid - liquid gas - liquid gas - gas
Isothermal	liquid - condensing/evaporating fluid gas - condensing/evaporating fluid

The flow arrangement affects the temperature profile along the heat exchanger (Figure 1).

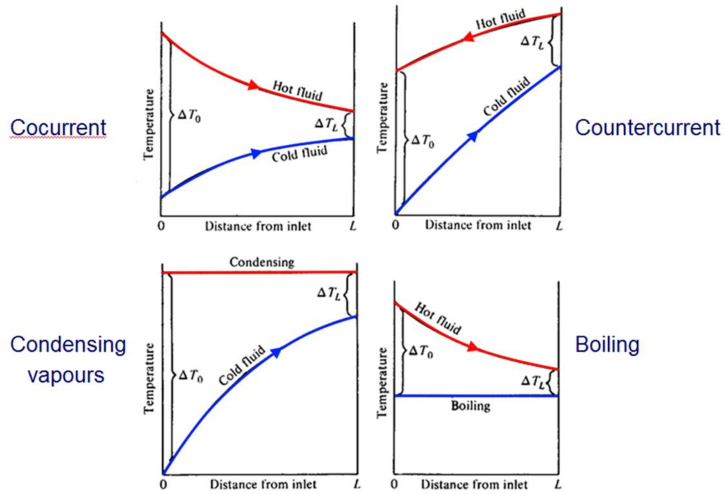


Figure 1 Temperature profiles for pure flow arrangements

1.1.3 By type

The types of greatest importance in the process industries are as follows:

- Double (or triple) concentric pipe (parallel, countercurrent or isothermal)
- Shell and tube (mixed)
- Plate heat exchanger (parallel, countercurrent or isothermal).

The process technologist is most commonly concerned with liquid-liquid and liquid-condensing/evaporating fluid heat exchangers, so gas-liquid, gas-condensing/evaporating fluid and gas-gas exchangers will not be considered. However, it should be noted that the principles to be discussed apply equally to these types.

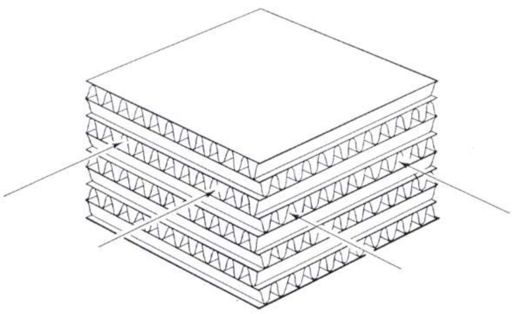


Figure 2 Cross flow heat exchanger (Bayley, 1972).

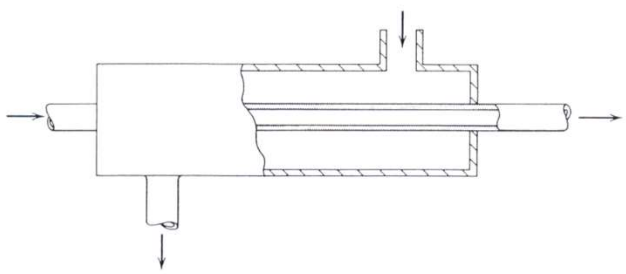
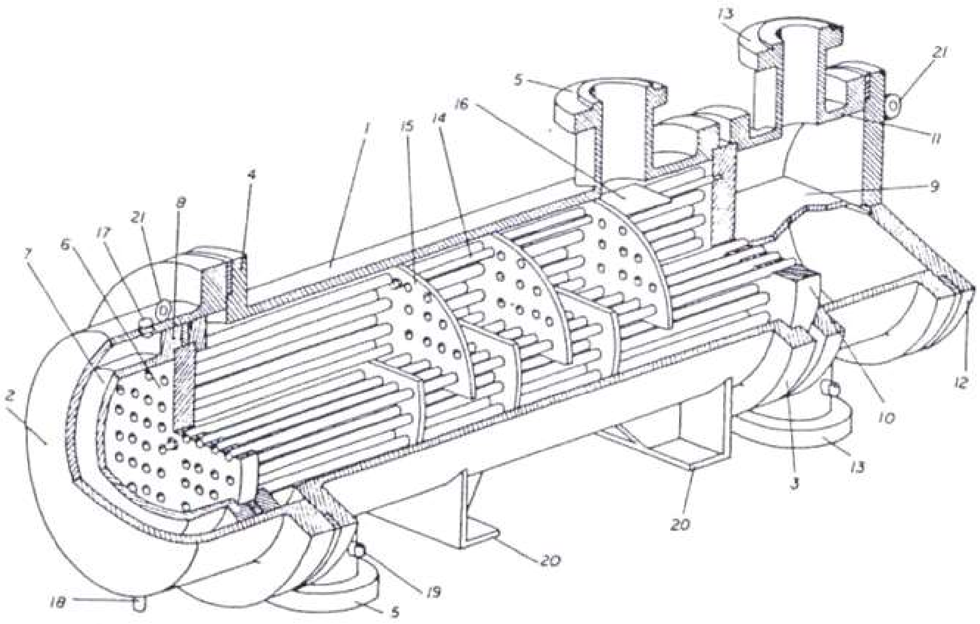
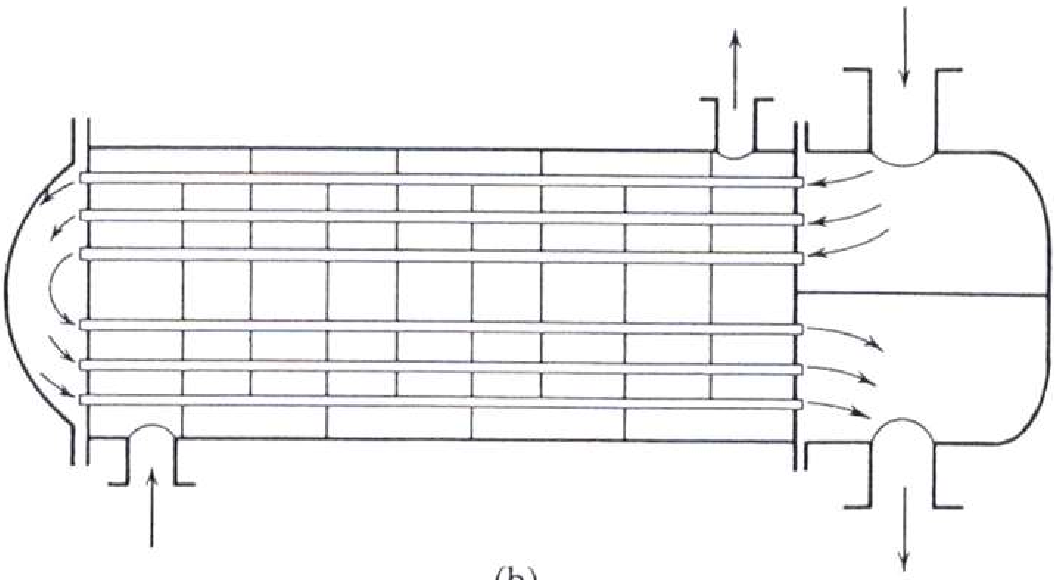


Figure 3 Double pipe heat exchanger (Welty et al, 1984).



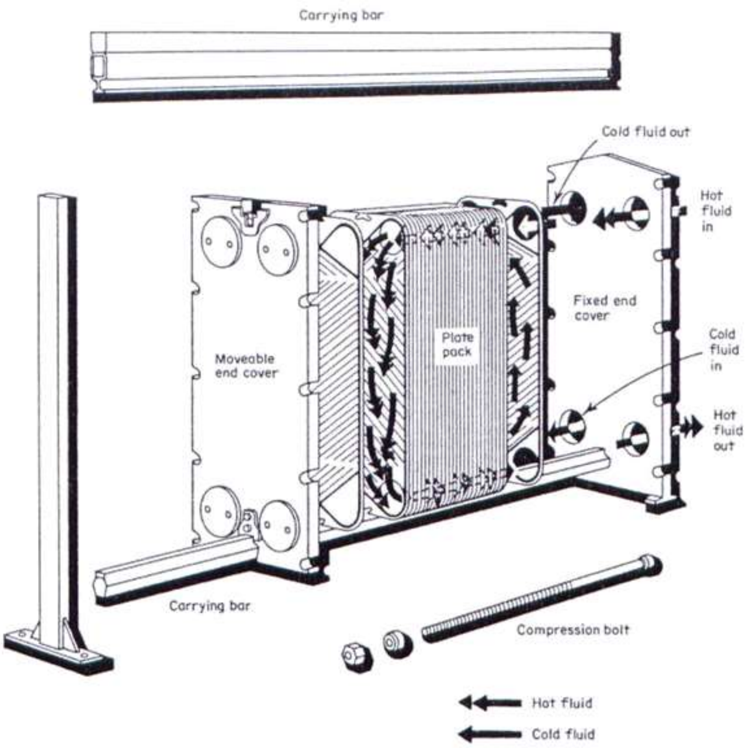
1. Shell
2. Shell Cover
3. Shell Channel
4. Shell Cover End Flange
5. Shell Nozzle
6. Floating Tubesheet
7. Floating Head
8. Floating Head Flange
9. Channel Partition
10. Stationary Tubesheet
11. Channel
12. Channel Cover
13. Channel Nozzle
14. Tie Rods and Spacers
15. Transverse Baffles or Support Plates
16. Impingement Baffle
17. Vent Connection
18. Drain Connection
19. Test Connection
20. Support Saddles

(a)

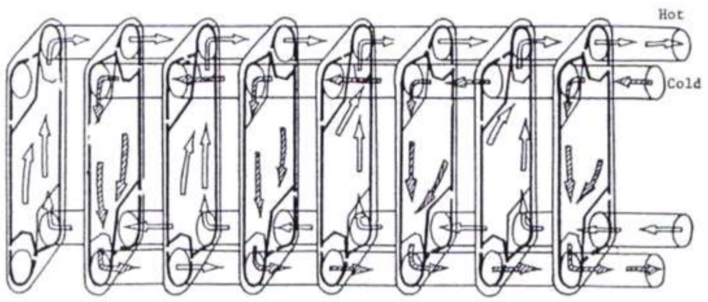


(b)

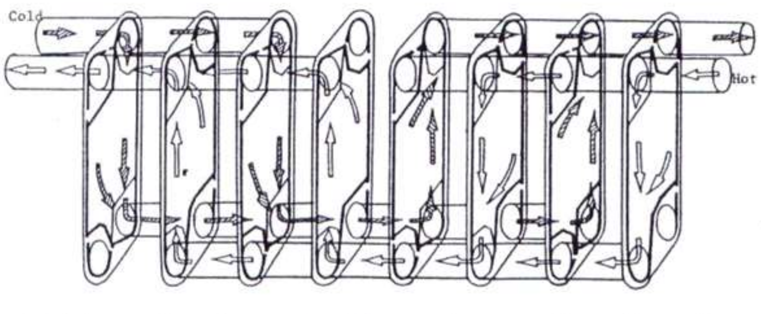
Figure 4 Shell and tube heat exchanger. (a) 3D representation (Azbel, 1984). (b) Longitudinal section (Welty et al, 1984).



(a)



Single pass arrangement, U-configuration.



Multipass (double pass) with equal passes, Z-configuration.

(b)

Figure 5 Plate heat exchanger. (a) 3D representation (Perry, 1984). (b) Flow arrangement (Palen, 1986).

1.2 *Installation of a Heat Exchanger*

Heat exchangers are usually part of a larger processing plant and their designer needs to consider not just the process fluid but also the service fluids available to the plant, e.g. hot water temperature and pressure. In many installations, it is also desirable to control the process temperature within tight bounds and the designer must consider the operation of this system.

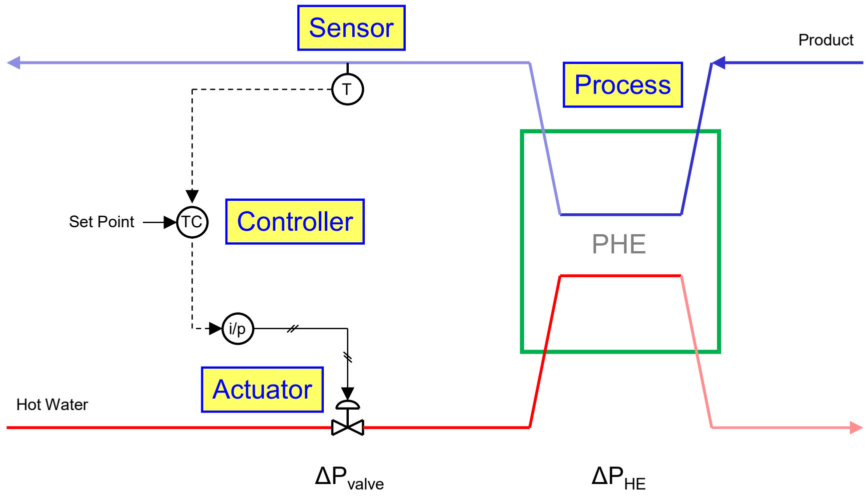


Figure 6 Typical heat exchanger control schematic

A simple heat exchanger control system attempts to control the temperature of the process fluid by manipulating a control valve in the services fluid line (Figure 6). As the control valve opens it reduces the pressure drop across the valve (ΔP_{valve}), which results in an increase in the pressure across the heat exchanger (ΔP_{HE}) and in the services flow rate. This changes the temperature profile along the heat exchanger, i.e. an increased services flow rate means that the temperature profile for that stream moves closer to horizontal thereby increasing ∇T_{lm} and hence, the amount of heat transferred (q). In practise, the heat exchanger must be sized for worst case scenario, i.e. highest required q, and for good control, the valve should be partially open under normal operating conditions, say ⅔ open.

2 GENERAL HEAT EXCHANGER DESIGN EQUATIONS

2.1 Heat Transfer

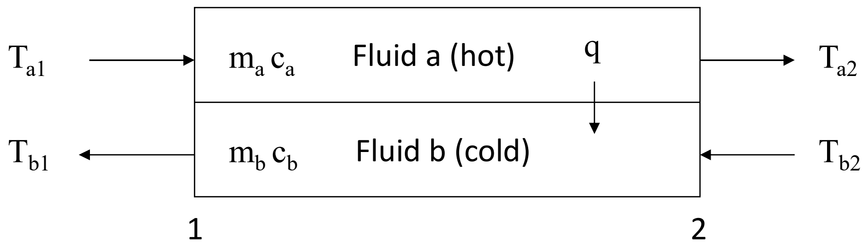


Figure 7 Diagrammatic representation of a continuous indirect heat exchanger

The variables that are involved in the exchanger's thermal (heat transfer) performance are:

U	=	overall heat transfer coefficient (W m^{-2} K^{-1})
A	=	area for heat transfer (m^2)
T_{a1}, T_{a2}	=	temperatures of fluid a (°C)
T_{b1}, T_{b2}	=	temperatures of fluid b (°C)
$C_a = m_a c_a$	=	capacity rate of fluid a (W K^{-1})
$C_b = m_b c_b$	=	capacity rate of fluid b (W K^{-1})
c_a, c_b	=	specific heats of fluids a and b (J kg^{-1} K^{-1})
m_a	=	mass flow rate of fluid a (kg s^{-1})
m_b	=	mass flow rate of fluid b (kg s^{-1})
Flow	=	parallel or countercurrent or crossflow or mixed or isothermal

The relationships between these variables provide the basis of the heat transfer aspects of heat exchanger design. The two fundamental relationships are the rate equation:

$$q = U\, A\, \nabla T_{tm} \qquad (1)$$

and the energy balance equation:

$$q = C_a\, \Delta T_a = C_b\, \Delta T_b \qquad (2)$$

where

q	=	rate of heat transfer between fluids a and b
∇T_{tm}	=	true mean temperature difference driving force between the two fluids
ΔT_a	=	temperature change in fluid a (°C)
ΔT_b	=	temperature change in fluid b (°C)

NB: in these notes we use ΔT to represent a temperature change (increase or decrease), while

∇T represents a temperature difference (i.e. a temperature driving force for heat transfer).

For countercurrent, concurrent or isothermal:

$$\nabla T_{tm} = \nabla T_{lm} \tag{3}$$

where

∇T_{lm} = logarithmic mean temperature difference (LMTD)

$$\nabla T_{lm} = \frac{\nabla T_1 - \nabla T_2}{\ln\left(\dfrac{\nabla T_1}{\nabla T_2}\right)} \tag{4}$$

where

∇T_1 = temperature difference between the fluids at end 1 of the heat exchanger

∇T_2 = temperature difference between the fluids at end 2 of the heat exchanger

For all other flow arrangements, e.g. mixed, crossflow, we need a correction factor:

$$\nabla T_{tm} = F_T \nabla T_{lm} \tag{5}$$

where

F_T = LMTD correction factor (dimensionless)

F_T is < 1, and its value depends on the precise nature of the flow arrangement. The evaluation of F_T is covered in Section 4.1.

Although in the general case U varies along the heat exchanger, it is usually assumed to be constant to simplify the design procedure. For a plane heat transfer surface (a slab) or for a thin-walled cylindrical heat transfer surface, 1/U, the overall resistance to heat transfer, is related to the individual resistances as follows:

$$\frac{1}{U} = \frac{1}{h_a} + \frac{x_w}{k_w} + \frac{1}{h_b} + R_a + R_b \qquad (m^2\ K\ W^{-1}) \tag{6}$$

where

$h_a\ h_b$ = film heat transfer coefficients on fluid a and b sides ($W\ m^{-2}\ K^{-1}$)

x_w = wall thickness (m)

k_w = thermal conductivity of wall ($W\ m^{-1}\ K^{-1}$)

$R_a\ R_b$ = fouling resistances ($m^2\ K\ W^{-1}$)

A more complex equation, which takes radius of curvature into account, is required for thick-walled cylindrical surfaces.

2.2 Heat Transfer Coefficient Correlations

For forced convection, the dimensionless Nusselt number is often expressed as an empirical function of the Reynolds and the Prandtl numbers. For example, the Sieder-Tate correlation for the turbulent flow of Newtonian liquids inside straight round tubes is given by:

$$\mathrm{Nu} = 0.027\ \mathrm{Re}^{0.8}\ \mathrm{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{7}$$

where Nu = Nusselt number (dimensionless)

Re = Reynolds number (dimensionless)

Pr = Prandtl number (dimensionless)

$$\frac{hD}{\lambda} = 0.027\left(\frac{Dv\rho}{\mu}\right)^{0.8}\left(\frac{c_p\mu}{\lambda}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{8}$$

where h = film heat transfer coefficient (W m^{-2} K^{-1})

D = tube diameter (m)

v = mean velocity of liquid (m s^{-1})

λ = liquid's thermal conductivity (W m^{-1} K^{-1})

ρ = liquid's density (kg m^{-3})

μ = liquid's viscosity at bulk mean temperature (Pa s)

μ_W = liquid's viscosity at tube wall temperature (Pa s)

c = liquid's specific heat (J kg^{-1} K^{-1})

The Sieder-Tate equation may require iterative solution as μ_W is often unknown at the start. In this case, the viscosity ratio is normally taken as 1 for the first iteration.

2.3 Pressure Drop Over a Heat Exchanger

The design of a heat exchanger involves a consideration not only of the heat transfer performance, but also of the mechanical pumping power required to move the fluids through the exchanger. So two further variables are of importance:

ΔP_a = pressure loss owing to viscous friction on fluid a side (Pa)

ΔP_b = pressure loss owing to viscous friction on fluid b side (Pa)

Pressure losses owing to net velocity changes and to net static head changes also occur on each side of a heat exchanger, and are easy to calculate. But it is the pressure loss owing to viscous friction that is of most significance in design because it is this pressure drop that is usually much the greatest and it is this pressure drop that is directly related to the heat transfer characteristics of an exchanger flow passage.

The frictional pressure loss (ΔP) can be found from:

$$\Delta P = 4f_F \frac{L}{D}\frac{\rho v^2}{2} \tag{9}$$

where f_F = Fanning friction factor

L = tube length (m)

The friction factor can be obtained from a Moody diagram or the appropriate equations. The same equation is often used for turbulent flow in non-circular ducts but replacing D with the hydraulic diameter (D_h):

$$D_h = \frac{4 \times \text{flow area}}{\text{wetted perimeter}} \tag{10}$$

2.4 Double Pipe Heat Exchanger Example

See examples 1a and 1b on the Stream site.

When tackling heat exchanger problems, it is recommended to begin by drawing a schematic of the equipment and filling in the known information. Then a suggested calculation strategy is to first ask whether all the inlet and outlet temperatures are specified or available from energy balance?

- NO $\rightarrow$ Use ε - NTU method
- YES $\rightarrow$ Is the flow regime pure countercurrent, cocurrent or isothermal?
  - YES $\rightarrow$ $F_T = 1$, use ∇T_{lm}
  - NO $\rightarrow$ Use F_T - LMTD method

Note F_T - LMTD and ε - NTU approaches are different ways of grouping the variables involved in heat transfer and the equations are closely related.

3 INTEGRATED APPROACH TO THE THERMAL AND HYDRAULIC DESIGN OF HEAT EXCHANGERS

Heat exchanger design problems are of two kinds:

1. **Rating** (performance determination) of an existing or fully-specified heat exchanger.
2. **Design** (sizing, or surface area determination) of a new heat exchanger, which afterwards is actually built.

3.1 Rating (performance determination)

Rating means calculating how an existing or fully-specified exchanger will perform under given conditions, using appropriate heat transfer and friction factor correlations (such as equations (8) and (9)) and equations relating all the heat transfer and pressure drop variables involved.

Input data are:

- the heat exchanger flow passage geometries and dimensions (including surface area),
- fouling factors (fouling resistances),
- fluid flow rates,
- fluid inlet temperatures, and
- the relevant physical properties of the fluids.

The outputs of the calculations are:

- the heat transfer rate that will be achieved (q),
- the fluid outlet temperatures ($T_{a,o}$ and $T_{b,o}$) and
- the frictional pressure drops on each side of the heat exchanger (ΔP_a and ΔP_b).

It can then be determined by inspecting the outputs whether or not the heat exchanger will cope with a desired duty.

3.2 Designing (sizing, or surface area determination)

Designing or sizing means determining the physical size (surface area) and flow configuration which a heat exchanger must have if it is to achieve a specified heat transfer rate (q) while at the same time fully utilizing, but not exceeding, the maximum allowable or available frictional pressure drop on each side.

The type of exchanger to be used for the duty (shell and tube, or plate, or some other type) is selected before detailed sizing calculations are carried out. The relationships used in the calculations are exactly the same as those used for rating.

Input data are:

- fluid flow rates,
- fluid inlet and outlet temperatures (and thus, from a heat balance, the required (q)),
- fouling factors,
- the maximum allowable pressure drops, and
- the relevant physical properties of the fluids.

Outputs are:

- the total surface area and
- the flow arrangement required to meet the specified duty.

In practice, because of the complexity of the relationships between flow passage geometry, heat transfer characteristics, friction power characteristics and flow arrangement, design calculations often comprise a series of ratings of successive assumed designs (Figure 8). Each assumed design is a modified version of the preceding one, the modification(s) being based on the results of rating the preceding design. Design or sizing is thus an **iterative** process, which continues until a design is arrived at that will perform according to the desired heat transfer and pressure drop specifications.

In the design modification step the results of the rating step are used to decide how the assumed design should be changed or modified in such a way that the new exchanger configuration will do a better job of meeting the specified heat transfer and pressure drop requirements. The design modification step is potentially complex because it must determine what is limiting the performance of the exchanger, and what can be done to remove that limitation without adversely affecting either the cost of the exchanger or the operating characteristics of the assumed design that are satisfactory.

If, for example, it is found that the heat exchanger is limited by the amount of heat it can transfer, then the configuration must be altered so as to increase either the film heat transfer coefficients or the surface area.

- A film heat transfer coefficient can be increased by decreasing the number of parallel flow passages for the fluid, thereby increasing fluid velocity. But note that unless the film heat transfer coefficient is **controlling** the overall coefficient (see equation (6)) then it is pointless to increase it by increasing velocity; a higher pressure drop will be incurred for nothing.
- Surface area can be increased by increasing flow passage length or by increasing the number of passes in series (each pass consisting of a number of parallel flow passages).

If the exchanger is limited by a very low ∇T_{tm}, the configuration may need to be altered so as to increase the degree of countercurrent flow. For fully countercurrent flow, $\nabla T_{tm} = \nabla T_{lm}$ = maximum ∇T_m possible (equation (3)).

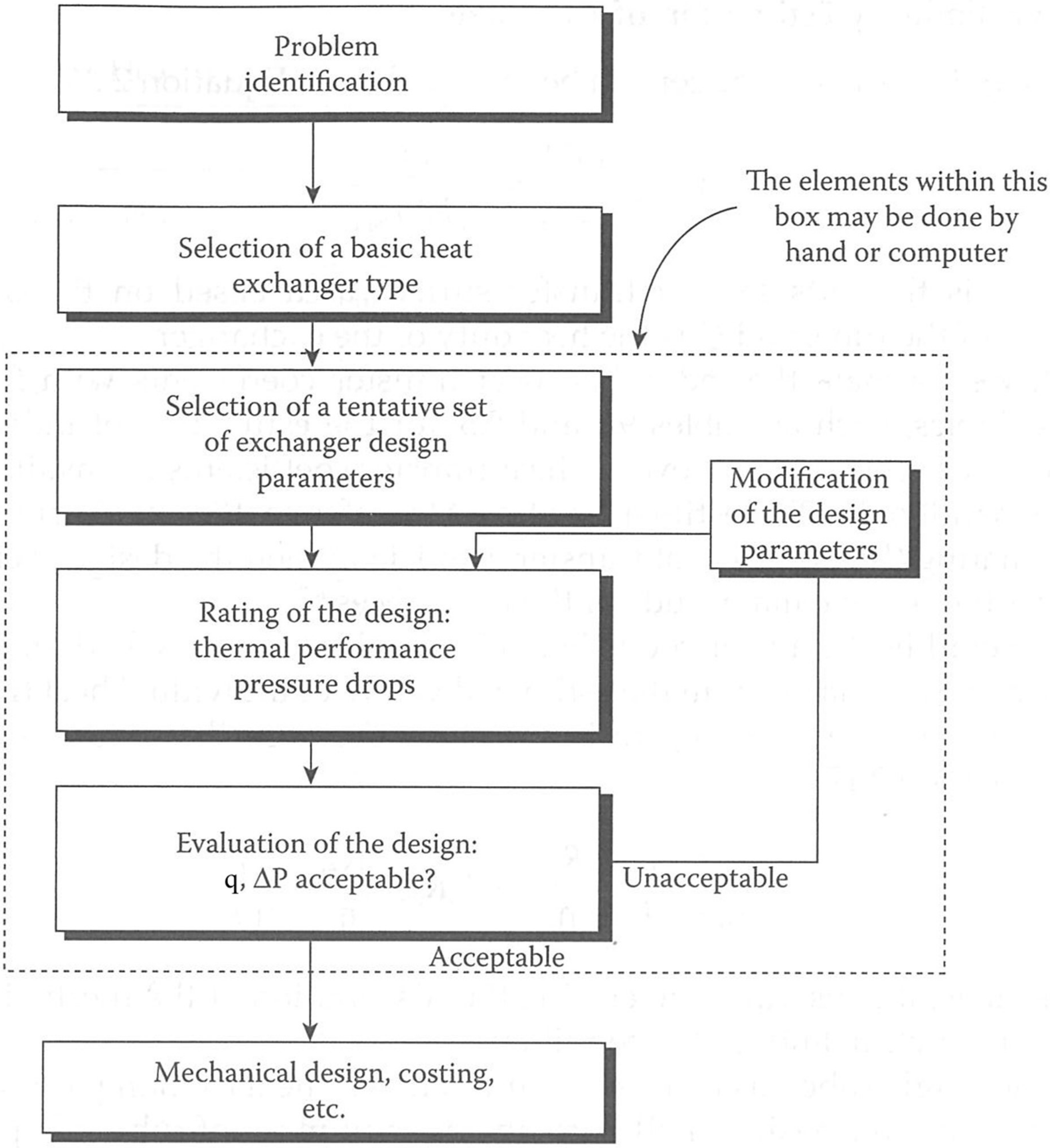


Figure 8 Typical Heat Exchanger Design Process

If the exchanger is limited by pressure drop, on one side or the other, the number of parallel flow passages per pass on the side concerned, or the duct equivalent diameter on that side, can be increased to reduce velocity.

3.3 Relationship Between ΔP and HT Performance

It can readily be shown that, for most flow passages that might be used to provide the heat transfer surfaces of an exchanger, the heat transfer rate per unit of surface area per unit temperature difference (= $q/A\nabla T_{tm} = h$) can be increased by increasing the fluid velocity, and that this rate varies as something less than the first power of the velocity, e.g. from the Sieder-

Tate correlation:

$$h\left(=\frac{q}{A\nabla T_{tm}}\right)\propto v^{0.8} \tag{11}$$

The friction power expenditure per unit of surface area also increases with velocity, but in this case the power varies by as much as the cube of the velocity and never less than the square. This can be shown by considering the frictional pressure loss per unit surface area:

$$\frac{\Delta P}{A}=\frac{\Delta P}{\pi DL}=4f_F\frac{\rho v^2}{2\pi D} \tag{12}$$

Friction power expenditure per unit surface area is given by

$$\frac{\Delta PQ}{A}=\frac{\Delta PQ}{\pi DL}=4f_F\frac{\rho v^2 Q}{2\pi D} \tag{13}$$

where Q = volumetric flow rate ($m^3\ s^{-1}$).

$$\begin{aligned} f_F &= fn\left(\frac{1}{Re}\right)^x \\ f_F &= fn\left(\frac{\mu}{Dv\rho}\right)^x \end{aligned} \tag{14}$$

where $x < 1$.

Therefore, since $Q \propto v$ (because $Q = \pi D^2 v/4$), it can be seen from equations (12) and (13) that friction power expenditure per unit surface area is:

$$\frac{\Delta PQ}{A}\propto v^{2\rightarrow 3} \tag{15}$$

It is this behaviour that allows the heat exchanger designer to match both heat transfer rate and frictional pressure drop specifications, and that dictates many of the characteristics of different types of heat exchanger.

If the friction power expenditure is (ΔPQ) too high, the heat exchanger designer can reduce fluid velocities by increasing the number of parallel flow passages in the exchanger. This will also decrease the heat transfer rate per unit of surface area, but because of the relationships given above (equations (11) and (15)) the reduction in heat transfer rate will be considerably less than the friction power reduction. The loss in heat transfer rate is then made up by increasing the surface area (by lengthening the flow passages). This in turn also increases the friction power expenditure, but only in the same proportion as the heat transfer surface area is increased, i.e. as the tube length is increased (see equation (13)).

The power expended to overcome fluid friction is the non-thermal price paid for heat transfer when a heat exchanger is operating, i.e. it constitutes the main running cost of the exchanger.

(The heat energy transferred must, of course, ultimately be paid for as well). If this running cost is to be kept low, the exchanger must have a large surface area, i.e. it must be relatively big, and therefore expensive; the physical size of the exchanger, which is directly proportional to its surface area, contributes most to the exchanger's capital cost. The higher the friction power expenditure, the smaller and cheaper the exchanger can be. It is therefore very important to maximize the use of any available pressure drop in the processing system of which the exchanger forms a part. By doing this, the size and capital cost of the exchanger can be minimized.

4 REVIEW OF THE THERMAL DESIGN OF CONTINUOUS HEAT EXCHANGERS

The variables involved in the thermal performance of a heat exchanger are listed in Section 2. These are too numerous to permit ready graphical or mathematical descriptions of all their possible relationships. However, they can be grouped judiciously into a smaller number of dimensionless groups which do allow this. There are two common approaches: the ε-NTU approach and the LMTD approach.

4.1 The LMTD Approach

The numerical value of F_T, the LMTD correction factor in equation (5), depends on the flow arrangement of the heat exchanger (i.e. on the degree of countercurrent flow that exists) and on the terminal fluid temperatures.

F_T can be found from published charts. A chart for a 1 shell pass/2 tube passes shell and tube exchanger is shown in Figure 9. The parameters R and S needed for reading the chart are defined as follows:

$$R = \frac{\Delta T_{min}}{\Delta T_{max}} \tag{16}$$

$$S = \frac{\Delta T_{max}}{\Delta T_{max\,possible}} \tag{17}$$

(The following pair of alternative definitions of R and S will give the same value of F_T from the chart in a given case: $R = \Delta T_{max}/\Delta T_{min}$, $S = \Delta T_{min}/\Delta T_{max\,possible}$)

There is a one to one correspondence between the LMTD and ε - NTU approaches. They are simply different ways of grouping the variables involved in the heat transfer performance of heat exchangers.

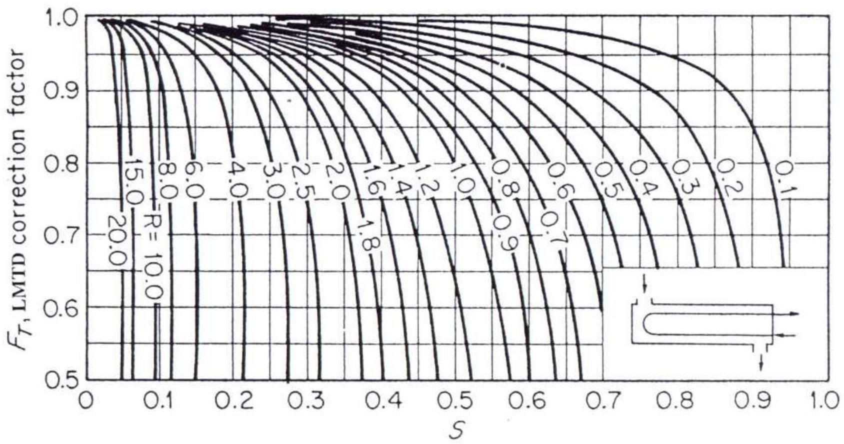


Figure 9 LMTD correction factors for a STHE (1 shell & 2 tube passes) (Perry, 1984)

4.2 *The ε-NTU approach*

Here, the variables are put into three dimensionless groups, each with a readily visualized physical significance: the exchanger effectiveness (ε), the capacity-rate ratio (C_R), and the number of exchanger heat transfer units (NTU).

4.2.1 Effectiveness (ε)

$$\varepsilon = \frac{q}{q_{max}} = \frac{C_a \Delta T_a}{C_{min}\left(T_{a,i} - T_{b,i}\right)} = \frac{C_b \Delta T_b}{C_{min}\left(T_{a,i} - T_{b,i}\right)} \qquad (18)$$

where C_{min} is the smaller of C_a and C_b, and ($T_{a,i}$ - $T_{b,i}$) is the maximum possible temperature change that could occur in the heat exchanger. It can be seen from equation (15) and equation (2) that this temperature change could occur only in the C_{min} liquid. So, from equation (15),

$$\varepsilon = \frac{C_{min} \Delta T_{max}}{C_{min}\left(T_{a,i} - T_{b,i}\right)} = \frac{\Delta T_{max}}{\Delta T_{max\,possible}} \qquad (19)$$

The effectiveness ε compares the actual heat transfer rate to the thermodynamically limited maximum possible heat transfer rate, which would be realized only in a countercurrent exchanger of infinite heat transfer area (for which ε would be 1.0). ε is the effectiveness of the exchanger from a thermodynamic point of view.

4.2.2 Capacity-rate ratio (*C_R*)

$$C_R = \frac{C_{min}}{C_{max}} \quad \left(= \frac{\Delta T_{min}}{\Delta T_{max}} \right) \qquad (20)$$

4.2.3 Number of transfer units (NTU)

$$NTU = \frac{UA}{C_{min}} = \frac{\Delta T_{max}}{\nabla T_{tm}} \qquad (21)$$

The NTU is a dimensionless expression of the "heat transfer size" of the heat exchanger. Its significance is discussed below in Section 4.1.5.

4.2.4 Relationships between ε, *C_R* and *NTU*

These three dimensionless groups are related as follows:

$$\varepsilon = \text{fn}\left(\text{NTU}, \text{CR}, \text{flow arrangement}\right) \qquad (22)$$

The function in equation (**22**) is displayed graphically in Figures 10, 11 & 12, for a pure countercurrent heat exchanger, a pure parallel flow heat exchanger and a shell and tube heat exchanger (with one shell pass and two tube passes). Mathematically, the functions for these kinds of exchanger are as follows:

Pure countercurrent:

$$\varepsilon = \frac{1 - e^{-NTU\,(1 - C_R)}}{1 - (C_R)\,e^{-NTU\,(1 - C_R)}} \tag{23}$$

Pure parallel flow:

$$\varepsilon = \frac{1 - e^{-NTU\,(1 + C_R)}}{1 + C_R} \tag{24}$$

Shell and tube (1 shell pass/2 tube passes):

$$\varepsilon = \frac{2}{1 + C_R + \sqrt{1 + C_R^2}\left(\dfrac{1 + e^{-\Gamma}}{1 - e^{-\Gamma}}\right)} \tag{25}$$

where $\Gamma = NTU\sqrt{1 + C_R^2}$

For an isothermal heat exchanger (which has a fluid evaporating or condensing on one side) C_R = 0 and the above equations all reduce to:

$$\varepsilon = 1 - e^{-NTU}$$

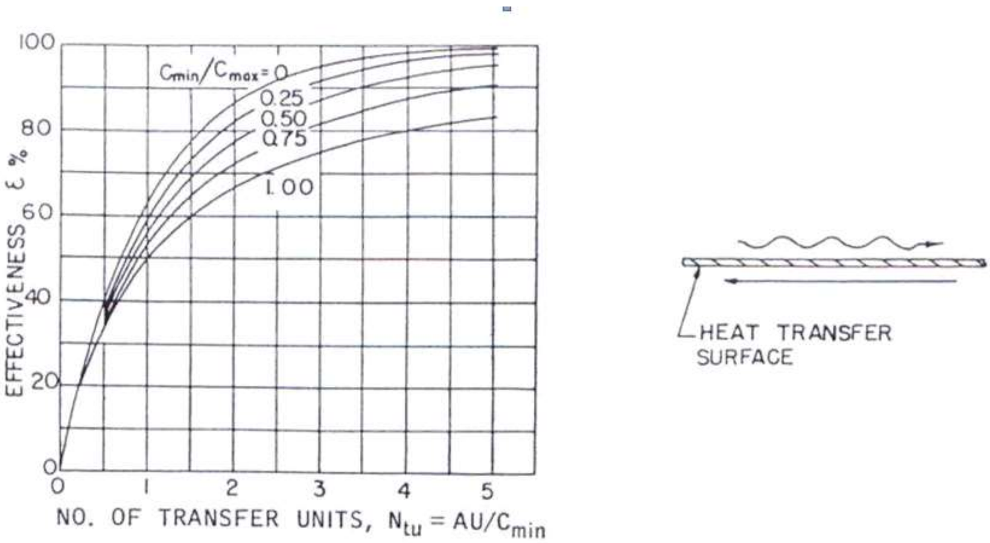


Figure 10 Concurrent heat exchanger (Kays & London, 1964).

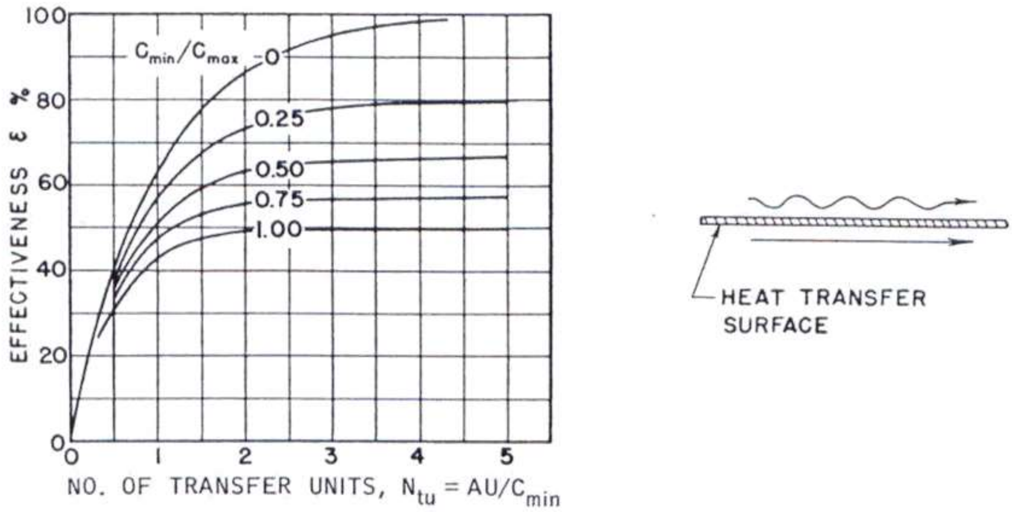


Figure 11 Parallel flow heat exchanger (Kays & London, 1964).

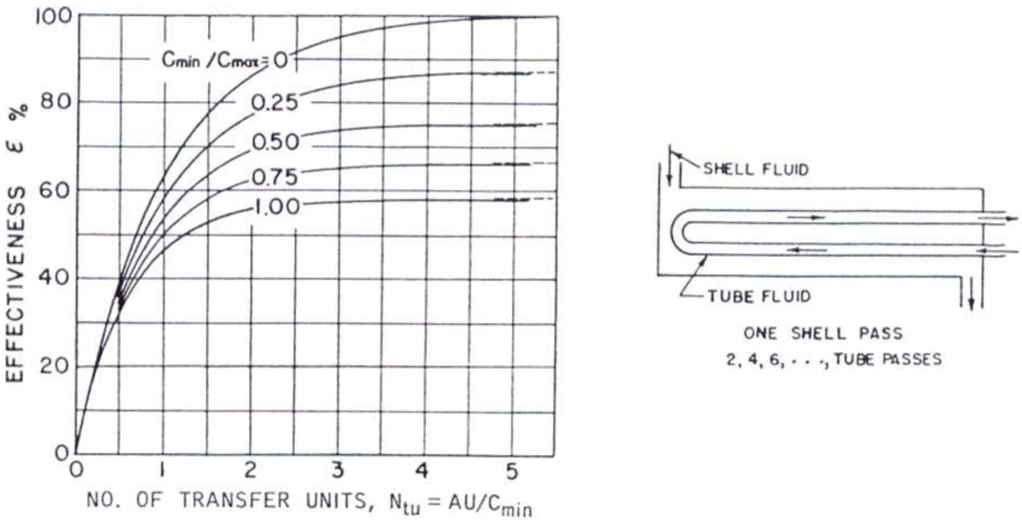


Figure 12 Shell and tube heat exchanger, 1-shell/2-tube passes (Kays & London, 1964).

4.2.5 Significance of NTU

As stated above, NTU, which is equal to UA/C_{min}, is a dimensionless expression of the "heat transfer size" of the heat exchanger. (A is a measure of the exchanger's physical size). For a given C_{min}, when NTU is small (small U and/or small A) the effectiveness ε of the exchanger is low.

5 RATING AND DESIGN OF SHELL AND TUBE HEAT EXCHANGERS

The tube side analysis of STHEs is normally straightforward and a similar approach to the dual pipe HE example can be used. However, the shell side has a mixed flow arrangement, which means that the true mean temperature difference (∇T_{tm}) is less than the log mean temperature difference (∇T_{lm}) and one needs to use the F_T-LMTD or the ε-NTU method.

5.1 Heat Transfer in a STHE

5.1.1 Thin-Walled Tubes

Assuming that the walls of the tubes are thin enough to enable their curvature to be ignored (r_2 / r_1 < 1.05), equation (6) can be used, in the following form:

$$\frac{1}{U}=\frac{1}{h_t}+\frac{r_o-r_i}{k_w}+\frac{1}{h_{ss}}+R \tag{26}$$

where h_t and h_{ss} are the tube side and shell side film heat transfer coefficients (W m^{-2} K^{-1})

r_i and r_o are the tube inside and outside radii (m)

k_w is the thermal conductivity of tube metal (W m^{-1} K^{-1})

R is the total fouling resistance (m^2 K W^{-1})

The heat transfer area can be calculated as a simple arithmetic average:

$$A=2\pi\left(\frac{r_o+r_i}{2}\right)LN \tag{27}$$

where L is the tube bundle length (m)

N is the total number of tubes

5.1.2 Thick-Walled Tubes

In most SHTEs, the thin-walled assumption does not hold and it is necessary to allow for the difference between the inside (A_i) and outside (A_o) areas of the tubes. A convenient way to do this is to determine the overall HTC (U_o) based on the outside area:

$$\frac{1}{A_oU_o}=\frac{1}{A_ih_t}+\frac{(r_o-r_i)}{A_{lm}k_w}+\frac{1}{A_oh_{ss}} \tag{29}$$

or more simply:

$$\frac{1}{U_o}=\frac{r_o}{r_ih_t}+\frac{r_o\ln(r_o/r_i)}{k_w}+\frac{1}{h_{ss}} \tag{28}$$

This value of the overall HTC should be used in conjunction with the outside area:

$$A_o = 2\pi r_o L N \tag{31}$$

Alternatively, it is possible to calculate an overall HTC based on the inside area or the log mean area so long as it is used in conjunction with the appropriate area when calculating the rate of heat transfer.

5.1.3 Tube Side

It is possible to use the Sieder-Tate equation (7) given earlier but Gnielinski's empirical equation below is considered better for predicting h_t for typical heat exchanger tubes, which possess some surface roughness.

$$Nu_t = \frac{\left(\frac{f_{F,t}}{2}\right)(Re_t - 1000)Pr_t}{1 + 12.7\sqrt{\frac{f_{F,t}}{2}}\left(Pr_t^{2/3} - 1\right)}\left(1 + \left(\frac{D}{L}\right)^{2/3}\right) \tag{292}$$

The tube side Fanning friction factor ($f_{F,t}$) can be found from the Moody plot or using Petukhov's empirical equation, which is suitable for turbulent Newtonian flow in tubes.

$$f_{F,t} = 0.25(1.82 \log Re_t - 1.64)^{-2} \tag{30}$$

The dimensionless numbers are calculated as below but taking care to use the appropriate characteristic length and velocity:

$$Nu_t = \frac{h_t D}{\lambda} \tag{34}$$

$$Re_t = \frac{D v_t \rho}{\mu} \tag{35}$$

$$Pr_t = \frac{c\mu}{\lambda} \tag{31}$$

where D = tube internal diameter (ID) (m)

v_t = tube side mean velocity (m s^{-1})

λ, ρ, c and μ are the thermal conductivity (W m^{-1} K^{-1}), density (kg m^{-3}), specific heat (J kg^{-1} K^{-1}) and viscosity (Pa s) of the tube side liquid, all evaluated at the liquid's bulk mean temperature, i.e. $(T_i + T_o) / 2$

$$v_t = \frac{Q_t}{\frac{\pi D^2}{4} N_{t/pass}} = \frac{4Q_t}{\pi D^2 N_{t/pass}} \tag{32}$$

where Q_t = total volumetric flow rate on tube side (m^3 s^{-1})

$N_{t/pass}$ = number of tubes per pass

5.1.4 Shell Side

The shell side heat transfer coefficient (h_{SS}) is predicted from:

$$Nu_{ss} = 0.36 Re_{ss}^{0.55} Pr_{ss}^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \tag{33}$$

The dimensionless numbers are calculated as below, again taking care to use the appropriate characteristic length and velocity:

$$Nu_{ss} = \frac{h_{ss} D_e}{\lambda} \tag{34}$$

$$Re_{ss} = \frac{D_e v_{ss} \rho}{\mu} \tag{35}$$

$$Pr_{ss} = \frac{c\mu}{\lambda} \tag{36}$$

where D_e = shell side equivalent diameter (m)

v_{ss} = shell side mean velocity (m s^{-1})

λ, ρ, c and μ are the physical properties of the shell side liquid evaluated at its bulk mean temperature

μ_w = viscosity evaluated at the mean wall temperature

D_e may be thought of as the hydraulic diameter of the gap between the tubes as defined in equation (10). For a triangular pitch tube bank:

$$D_e = \frac{8\left(0.433\, P^2 - \frac{\pi}{8} D_o^{\,2}\right)}{\pi D_o} \tag{37}$$

For a square pitch tube bank:

$$D_e = \frac{4\left(P^2 - \frac{\pi}{4} D_o^{\,2}\right)}{\pi D_o} \tag{38}$$

where P = tube pitch, measured between centres (m)

D_o = tube outside diameter (m)

The shell side velocity is given by:

$$v_{ss} = \frac{Q_{ss}}{a_{ss}} \tag{39}$$

where Q_{ss} = total shell side volumetric flow rate (m^3 s^{-1})

a_{ss} = shell side cross-section area for flow (m^2)

$$a_{ss} = \frac{D_{ss}(P - D_o)B}{P} \tag{40}$$

where D_{ss} = shell diameter (m)

B = baffle spacing (m)

5.2 Frictional Pressure Drops in a STHE

The frictional pressure loss on the tube side can be calculated from

$$\Delta P_t = 4f_{F,t}\frac{LN_p}{D}\frac{\rho v_t^2}{2} + 4N_p\frac{\rho v_t^2}{2} \tag{41}$$

where N_p = number of tube passes

The second term on the RHS of equation (46) represents four velocity heads per pass to allow for sudden expansions and contractions at the tube ends. $f_{F,t}$ can be read from the appropriate turbulent flow curve on a Moody chart or calculated from equation (33).

The frictional pressure loss on the shell side is given by

$$\Delta P_{ss} = 4f_{F,ss}\frac{LD_{ss}}{BD_e}\frac{\rho v_{ss}^2}{2} \tag{42}$$

where $f_{F,ss}$ can be read from the appropriate curve or calculated from:

$$f_{F,ss} = \frac{0.45}{(Re_{ss})^{0.195}} \tag{43}$$

where Re_{SS} is given by equation (40).

5.3 Design Modifications in a STHE

In shell and tube heat exchanger design, various options are available in the design modification step.

- If, for example, the exchanger is limited by the amount of heat it can transfer, it will be necessary to increase U or to increase A.
  - If h_t is controlling U, one can increase the number of tube passes (for the same number of tubes), thereby increasing tube side velocity.
  - If h_{SS} is controlling, one can increase shell side velocity by decreasing baffle spacing or decreasing baffle cut (baffle window).
  - To increase A, one can increase the length of the exchanger, or increase shell

diameter and the total number of tubes, or go to multiple shells in series or in parallel. Multiple shells in series will help if ∇T_{tm} is very small.

- If the exchanger is limited by excessive frictional pressure drop on the tube side
  - Decrease the number of tube passes or increase tube diameter.
  - Decrease tube length and increase the shell diameter and the total number of tubes.
- If the exchanger is limited by excessive shell side pressure drop
- Increase baffle spacing or baffle cut, or increase tube pitch.

An **existing** shell and tube exchanger can mainly only be rated, but there is some scope for design modification by altering the shell side baffle spacing. The tube bundle has to be temporarily withdrawn from the shell for this to be done.

Important note

The rating/design procedure for shell and tube heat exchangers exemplified above is somewhat idealised as far as the shell side is concerned. Strictly, the Nu_{ss} predicted by equation (38) and the ΔP_{ss} predicted by equation (47) should be corrected to allow for non-ideal flow on the shell side. Flow is non-ideal for three reasons:

1. Flow deviates from ideal cross flow through a tube bundle.
2. There is leakage flow between baffles and shell, and between tubes and baffles.
3. There is bypass flow between tube bundle and shell.

Design procedures which allow for these phenomena can be found in the literature. They will not be discussed here.

6 SIZING OF PLATE HEAT EXCHANGERS

6.1 Plate heat exchanger construction and flow arrangements

The overall design and flow principle of the plate heat exchanger (PHE) are shown in Figure 5. There are three fundamental ways in which plates can be arranged: single pass, multipass with equal passes and multipass with unequal passes.

6.1.1 Single pass

Each liquid flows through a set of parallel passages that make up a single pass (Figure 13). Flow is countercurrent. This arrangement is the best from the point of view of heat transfer effectiveness because the LMTD correction factor (F_T) is only slightly less than 1.

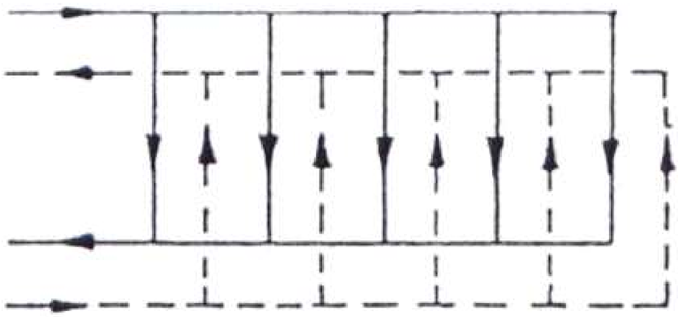


Figure 13 Single pass arrangement

Inlet and outlet connections can be in the Z-configuration or in the U-configuration (Figure 14). In the Z-configuration, inlet and outlet connections are made to both the head and the follower. In the U-arrangement connections are made to the head only. The Z-configuration gives more even distribution of flowing liquid among the parallel flow passages, while the U-configuration has the advantage that the plate pack can be disassembled (for cleaning, inspection or rearrangement) without the need to disturb any pipework.

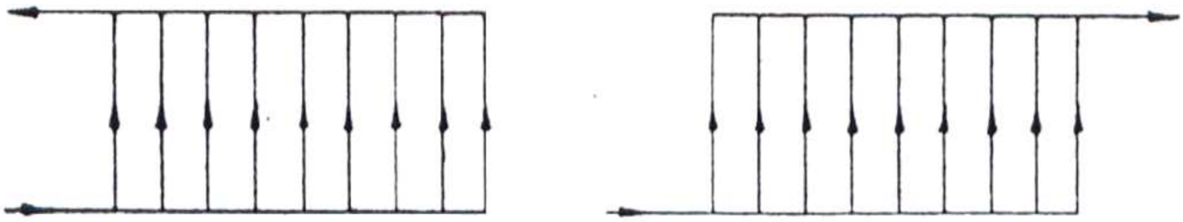


Figure 14 U and Z configurations

The U-configuration is possible only with the single pass arrangement. The Z-configuration is the only possible one for multipass arrangements.

6.1.2 Multipass with equal passes

When NTU values greater than those given by a single pass are required, and when there is a sufficient available pressure drop on each side, multipass arrangements can be obtained by blanking off the appropriate plate flow ports at x and y (Figure 15) to define the ends of the passes.

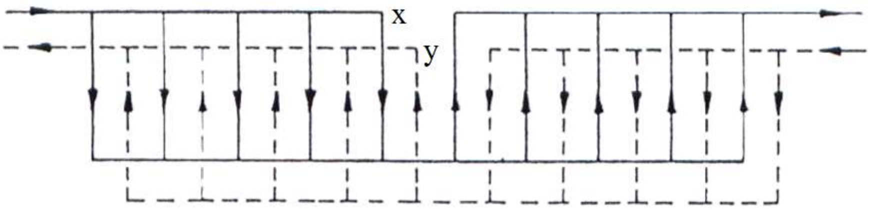


Figure 15 Multipass configuration with equal passes, a 2 pass/2 pass arrangement in this case.

Note that on each side of a PHE the flow channels within any given pass are arranged in parallel, but that passes themselves, when there are more than one, are arranged in series.

6.1.3 Multipass with unequal passes

When the flow ratio (Q_{max} (m^3/s)/Q_{min} (m^3/s)) is high, or there is some other reason for minimizing the pressure drop on one side, unequal passes can be used, with fewer passes on the low pressure drop side.

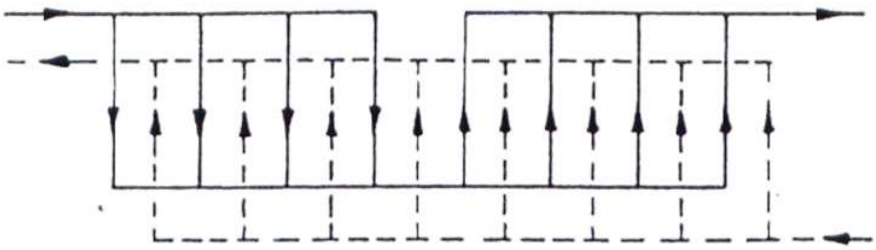


Figure 16 Multipass configuration with unequal passes, 2 pass/1 pass arrangement in this case.

6.1.4 Disadvantage of multipass arrangements

In multipass arrangements, especially when the passes on each side are unequal in number, there is a degree of parallel flow of the two liquids (see Figure 16). This reduces the effectiveness of the PHE for given NTU and C_R; looking at the situation from the LMTD point of view, it decreases the value of the correction factor F_T, i.e. makes it significantly less than 1. Also, only the Z-configuration is possible.

6.2 PHE Plate Design

The heat transfer and pressure drop characteristics of a plate (or, more exactly, of the flow channel formed between two identical plates) depends on plate aspect ratio (width/length) and on the shape and pattern of profilations. Many different plate designs are available from PHE manufacturers.

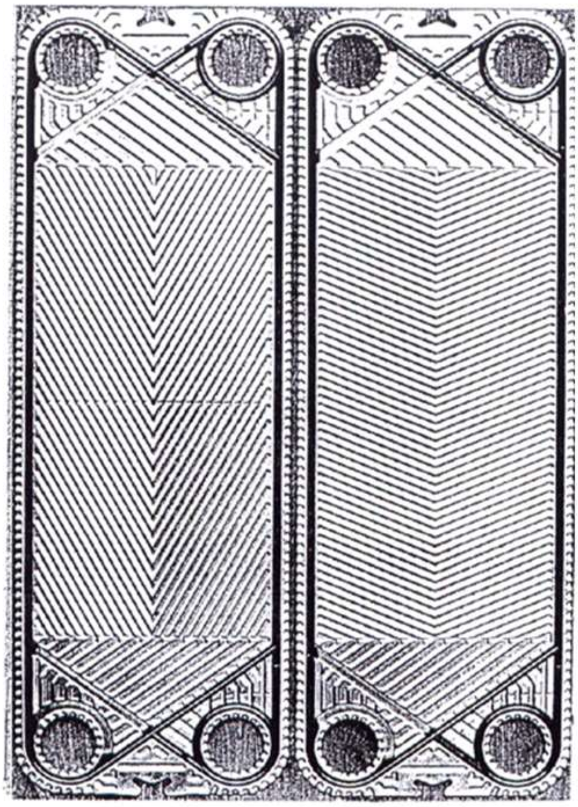
Plate dimensions depend on port size, which in turn depends on throughput. Port size governs plate width, because the plate must be wide enough to accommodate a pair of ports at each end. Plate length depends on plate width and aspect ratio.

Plate design is the business of the PHE manufacturer, and only the manufacturer will have detailed knowledge of the heat transfer performance and the pressure drop performance of a given design. This knowledge is, of course, proprietary.

6.2.1 "Hard" vs "Soft" Plates

The plates forming a flow passage can be either "hard" or "soft". ("Hardness" and "softness" have nothing to do with the mechanical properties of the plates (which are usually made of stainless steel), but refer to the heat transfer and pressure drop characteristics of the flow channel). A hard plate is long and narrow and/or has a profilation pattern causing much flow disturbance. Such a plate gives a high film heat transfer coefficient and thus high U and thus high NTU, but also incurs a relatively high frictional pressure drop. A "soft" plate has exactly the opposite characteristics.

SOFT plate
60° angle



HARD plate
120° angle

Figure 17 Soft and hard plate designs

Hard plates are used for difficult duties like regenerative heat recovery, where ∇T_{tm} is low because of close temperature approach at each end of the heat exchanger; a low ∇T_{tm} means that a high NTU is required for a given desired temperature change, ΔT. Soft plates are used for easier duties where ∇T_{tm} is large.

In PHE flow passages generally, high heat transfer coefficients can be attained at relatively moderate velocities, so high NTU duties can be fulfilled in a comparatively small number of passes (each pass achieving a proportion of the total NTU required). Thus overall frictional pressure losses are relatively low. (Note that when a liquid flows through several passes in series, the total frictional pressure loss is the sum of the frictional pressure losses in the individual passes.) For water-water duties, most PHEs have 'specific pressure drops' in the range 20-100 kPa per NTU.

6.2.2 Port Pressure Losses

Plates are designed to minimize frictional pressure losses in the port manifolds of the plate pack; such pressure losses are wasted from the point of view of heat transfer.

6.2.3 Fouling

PHEs have advantages over most other types of heat exchanger with regard to fouling. The flow disturbance and high degree of turbulence caused by plate profilations inhibit the build-up of fouling on the plate surfaces. The high turbulence also enhances the efficacy of in-place cleaning. Should the extent of fouling necessitate manual cleaning, the PHE is easily opened, cleaned and closed again.

6.3 PHE selection

Assessment of the suitability of the PHE (as against other types of exchanger) for a particular duty must be based on the following ranges of parameters, which are covered by the available world-wide range of plates operating with a single pass at unity flow rate ratio ($m_a/m_b = 1$):

Individual plate area:	up to about 2.5 m^2
NTU per pass:	0.3 - 3.5 approximately
Maximum operating pressure:	2000 kPa approximately
Maximum viscosity:	5 Pa s
Port diameter:	up to 400 mm
Maximum total flow rate:	2500 $m^3\ h^{-1}$

A quick assessment of PHE suitability can be made by comparing the above figures with the actual flow rate and duty NTU required. The duty NTU is ($\Delta T_{reqd}/\nabla T_{tm}$). NTUs greater than 3.5 can be achieved by using more than one pass. However, NTUs in excess of 12 are normally avoided because the increase in ΔT obtained rarely justifies the large amount of additional heat transfer area (i.e. the large number of additional plates) required. The duty NTU can be reduced (for the same ΔT_{reqd}) by increasing ∇T_{tm}, which can be achieved by changing the inlet temperature of the service liquid.

6.4 PHE Sizing

Once a PHE has been selected as being suitable for a particular heat transfer job, it must be sized. Sizing means determining the total number of plates (n) of given size, aspect ratio and profilation pattern and the number of passes (H) that will be needed to meet the specified heat transfer and pressure drop requirements. Economic sizing demands complete utilization of the available pressure drops while meeting the specified thermal performance.

PHE manufacturers have large and complex computer programs for carrying out the iterative sizing procedure but such programs and the empirical data they are based upon are strictly proprietary. However, there is an approximate sizing procedure available in the open literature

(Heat Exchanger Design Handbook, 1983) that can be used to determine the heat transfer area (and thus the total number of plates and a budget PHE capital cost). This procedure provides a means for the process technologist to determine if a PHE is a feasible selection for a particular heat transfer job, or to determine if a quote from a PHE supplier seems technically and economically reasonable or not.

This approximate method is limited to turbulent flow (viscosity < 4 x 10^{-3} Pa s) and volumetric flow rate ratios less than 3:1. It makes use of the three curves shown below which were determined experimentally for water/water at unity flow rate ratio.

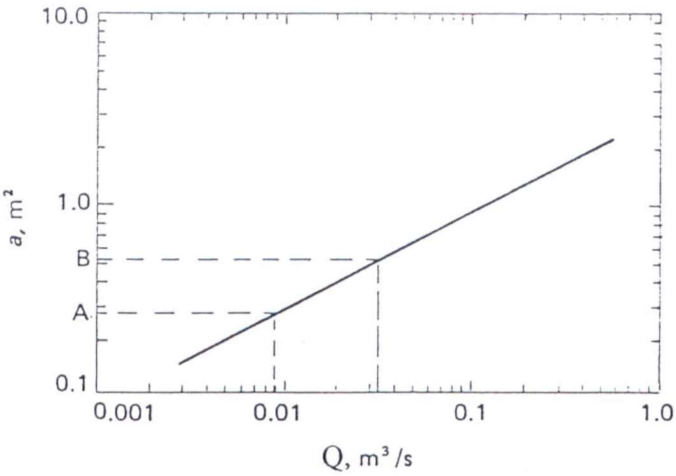


Figure 18 PHE curve 1, heat transfer area of individual plate (a) versus total volumetric flow rate Q (HEDH, 1983).

Curve 1 (Figure 18) reflects the fact that as total volumetric flow rate increases, the plate port diameter and thus the size of the plate have to increase as well. This curve is the origin for the equation (47) used later.

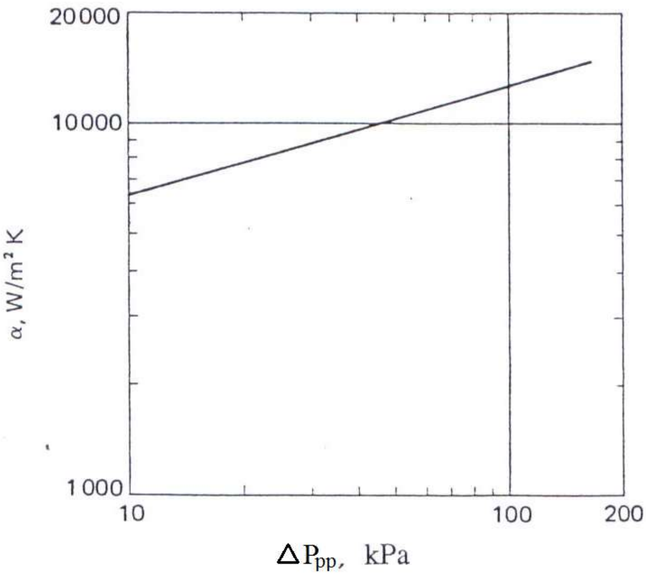


Figure 19 PHE curve 2, film heat transfer coefficient (α) versus pressure drop for a single flow passage (ΔP_{pp}) (HEDH, 1983).

Curves 2 & 3 (Figures 19 & 20) reflect the fact that as flow rate through a passage increases, so do the pressure drop across that passage as well as the heat transfer coefficient. Curve 2 is the origin for the equation (52) used later, and curve 3 is the origin of equation (54).

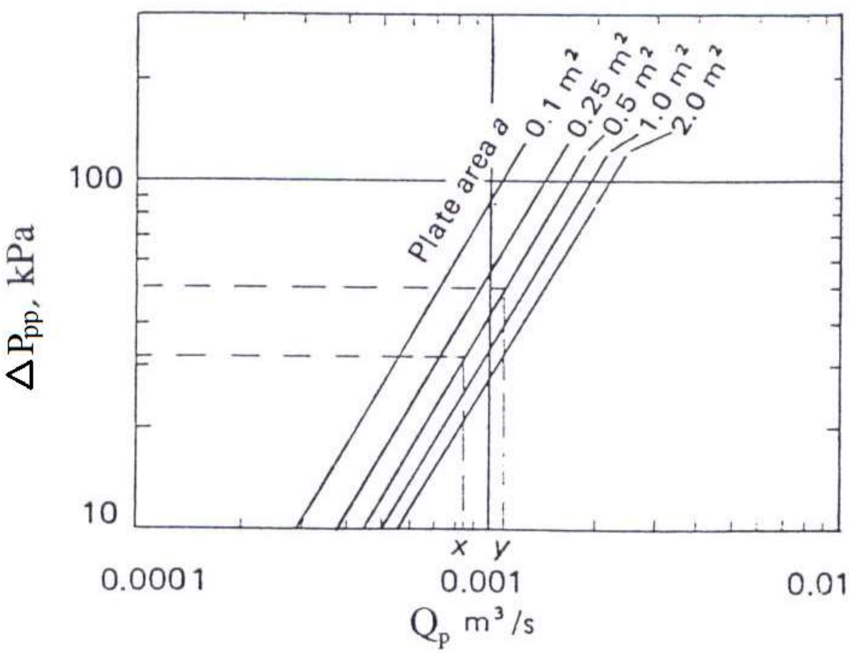


Figure 20 PHE curve 3, pressure drop for a single flow passage (ΔP_{pp}) versus the flow rate through that passage (Q_p) and the plate size (HEDH, 1983).

6.5 Approximate Method for PHE Design

The approximate method can be summarized as:

- Select a plate size (a, m^2) based on volumetric flow rate
- Iterative procedure:
    1. Guess number of passes (H)
    2. Calculate the number of parallel channels per pass that fully utilise the ΔP available for each pass
    3. Calculate total number of plates (n) and compare trial heat transfer rate (q_T) with the required heat rate (q_R)
        - If $q_T \approx q_R$ then stop
        - If $q_T << q_R$ then increase H and try again
        - If $q_T >> q_R$ then decrease H and try again

Step 1

Obtain the $\Delta P_{available}$ on each side of the PHE and use the energy balance equation to find any missing volumetric flow rates, i.e. Q_a and Q_b :

$$q_R = m_a c_a \Delta T_a = m_b c_b \Delta T_b \tag{44}$$

Step 2

As total volumetric flow rate increase, the plate port diameter and plate size has to increase. Calculate the heat transfer area of an individual plate (a) and select the next largest plate size available.

$$a = 2.68\, Q_{max}^{\;0.485} \tag{45}$$

where Q_{max} = the larger of Q_a and Q_b.

Step 3

Guess number of passes (H), e.g. H = 3, and calculate the pressure drop per pass (ΔP_{pp}). Note that the correlation below gives a value with units kPa.

$$\Delta P_{pp} = \frac{\Delta P_{available}}{H} \tag{46}$$

Step 4

Calculate the heat transfer coefficient (α) of fluid 'a' and 'b'. Note that these coefficients include an allowance for the conduction through the plate wall.

$$\alpha_{a/b} = 3057\left(\Delta P_{pp}\right)^{0.308} \tag{47}$$

Step 5

Calculate the overall heat transfer coefficient (U_T):

$$\frac{1}{U_T} = \frac{1}{\alpha_a} + \frac{1}{\alpha_b} + R \tag{48}$$

Step 6

Calculate the volumetric flow rate per flow passage (Q_p) from the pressure drop per pass (ΔP_{pp}) and the selected the heat transfer area of an individual plate (a):

$$\mathbf{Q_{p\ a/b} = 0.000127\ (a)^{0.224}\ (\Delta P_{pp\ a/b})^{0.584}} \tag{49}$$

Step 7

Using the higher of the Q_p values calculated in step 6 and the higher of the two total flow rates (Q_{max}), calculate the number of plates required 'n':

$$n = \frac{2HQ_{max}}{Q_{p\ max}} \tag{50}$$

where Q_p max is the higher of $Q_{p\,a}$ and $Q_{p\,b}$.

Step 8

Calculate log mean temperature difference:

$$\nabla T_{lm} = \frac{\nabla T_1 - \nabla T_2}{\ln\left(\frac{\nabla T_1}{\nabla T_2}\right)} \tag{51}$$

Step 9

Determine the trial heat transferred q_T:

$$q_T = U_T n a \nabla T_{lm} \tag{52}$$

Step 10

Compare q_T with the required heat transferred q_R. Repeat steps 3 to 9 with different H values until agreement is reached (q_T and q_R within ± 10%). At that stage, the total heat transfer surface area = n a

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