CHAPTER 2

REVIEW OF LITERATURE

2.1 Liteature Survey of Heat Tranfer Correlation

The heat transfer performance of plate-type units can also be calculated from typical dimensionless heat transfer equations with the appropriate exponents and constant for each specific type of exchanger[3].

The general correlation for turbulent flow is

$$Nu = (const.)\ Re^{n}\ Pr^{m}\ (\mu/\mu_{w})^{x} \qquad (2.2.1)$$

Typical reported values[3] for plate and frame exchangers are:

const. = 0.15 to 0.4

n = 0.65 to 0.85

m = 0.30 to 0.45 (usually 0.333)

x = 0.05 to 0.2

For laminar flow, it seems that the following Sieder-Tate type relationship is applicable.

$$Nu = const.\ (Re\ Pr\ D_{e}/L)^{0.33}\ (\mu/\mu_{w})^{0.14} \qquad (2.2.2)$$

where const. = 1.86 to 4.50 depending on the geometry.

For a small plate heat exchanger, the following heat transfer correlation has been proposed [4].

Turbulent flow:

$$Nu = 0.2\ Re^{0.67}\ Pr^{0.4}\ (\mu/\mu_w)^{0.1} \qquad (2.2.3)$$

Laminar flow:

$$Nu = 1.68(\ Re\ Pr\ D_e/L\ )^{0.4}\ (\mu/\mu_w)^{0.1} \qquad (2.2.4)$$

In equations (2.2.3) and (2.2.2), the plate value of L should be in the range of 0.7-2 m and the value of D_e in the neighborhood of 4 mm. The typical length/width ratio for many plates is 2. The plate gap can be taken as one half the equivalent diameter De for budget estimation.

2.2 Theory of Heat Transfer

Heat transfer occurs between materials by reason of the temperature differance between them. The rate of heat transfer is equal to the product of a driving force and a thermal conductance. Driving forces and thermal conductances vary for the three modes of heat transfer[5].

1. Conduction.
2. Convection.
3. Radiation.

The plate heat exchanger involves only the conduction and convection modes.

Conduction

Conduction is primarily a molecular phenomena requiring a temperature gradient as a driving force. A quantitative expression relating a temperature gradient and the nature of the conducting medium to the rate of heat transfer is attributed to Fourier, who in 1822 presented the relation

$$Q_x/A = -k(dt/dx) \qquad (2.2.5)$$

where Q_x = the x-directional heat flow rate in watts.

A = the area normal to the direction of heat flow in m^2

dt/dx = the temperature gradient in the x direction in $^\circ K/m$

k = the thermal conductivity in $watt/m^\circ K$

Convection

Convection involves energy exchange between a bulk fluid and a surface or interface. Sir Issac Newton, in 1701,first expressed the basic rate equation for convection heat transfer as

$$Q = hA(T_{surf} - T_{fluid}) \qquad (2.2.6)$$

where Q = the rate of convective heat transfer in watt

A = the area normal to the direction of heat flow in m^2

$T_{surf} - T_{fluid}$ = the temperature driving force in K

h = convective heat transfer coefficient in W/m^2K

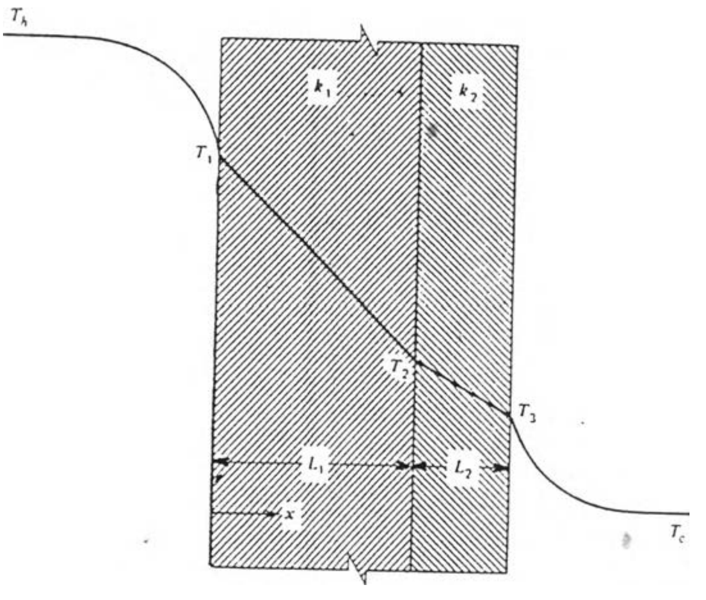


Figure 2.2.1 Heat transfer through a composite plane wall with convection at the surfaces.

Fig 2.2.1 show the situation in which a composite plane wall, consisting of two different materials, separates two gases at different temperatures. Denoting the hot and cool gas temperatures by T_h and T_c , respectively, the surface temperatures of the materials by T_1 and T_3, and the interface temperature between the two wall materials by T_2, we may write, for the heat transfer rate for each part of this process,

$$Q_{h-1} = h_h A(T_h - T_1) \quad (2.2.7)$$

$$Q_{1-2} = -k_1 A \, (dt/dx) \quad (2.2.8)$$

$$Q_{2-3} = -k_2 A \, (dt/dx) \quad (2.2.9)$$

$$Q_{3-c} = h_c A(T_3 - T_c) \quad (2.2.10)$$

For the steady-state case, the two conduction expressions will take the same form, the difference being in the limits of integration. Solving for Q_{1-2}, we obtain

$$Q_{1-2}\int_{0}^{L1} dx = -k_1 A\int_{T1}^{T2} dT \qquad (2.2.11)$$

$$Q_{1-2} = k_1 A(T_1 - T_2)/L_1 \qquad (2.2.12)$$

A similar analysis for Q_{2-3} will yield

$$Q_{2-3} = k_2 A(T_2 - T_3)/L_2 \qquad (2.2.13)$$

It may be noted here that for a plane wall at steady-state condition the temperature profile is linear.

Steady state also requires that all of the Q's be equal, thus the following string of equalities may be written

$$Q = h_h A(T_h - T_1) = k_1 A(T_1 - T_2)/L_1 \qquad (2.2.14)$$

$$= k_2 A(T_2 - T_3)/L_2 = h_c A(T_3 - T_c)$$

Each of these forms is sufficient to calculate the heat flow. Additional expressions may be obtained if each temperature difference is written in terms of Q as

$$T_h - T_1 = Q(1/(h_h A))$$

$$T_1 - T_2 = Q(L_1/(k_1 A))$$

$$T_2 - T_3 = Q(L_2/(k_2 A))$$

$$T_3 - T_c = Q(1/(h_c A))$$

If all of the temperature difference equations are added, the resulting expressing for Q is

$$Q = \frac{T_h - T_c}{\frac{1}{h_h A} + \frac{L_1}{k_1 A} + \frac{L_2}{k_2 A} + \frac{1}{h_c A}} \qquad (2.2.15)$$

Another common way of expressing the heat transfer rate when combined modes are involved is

$$Q = UA\triangle T \qquad (2.2.16)$$

where U is the overall heat transfer coefficient, having units of W/m^2 K

$$U = \frac{1}{\frac{1}{h_h} + \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{1}{h_c}} \qquad (2.2.17)$$

Temperature Distribution in Heat Exchangers

In a heat exchanger the fluid temperature changes as the fluids flow along the heat exchanger length. Fig.2.2.2 shows typical temperature distributions which may be observed in heat exchangers[6]. For the element of length dx with associated heat transfer area dA,

$$dQ = U(T_h - T_c)dA \qquad (2.2.18)$$

Also from the steady-flow energy equation for each fluid,

$$dQ = w_c di_c = -w_h di_h \qquad (2.2.19)$$

where w is the flow rate, i is the enthalpy, and the kinetic and potential energy changes of the fluids are neglected.

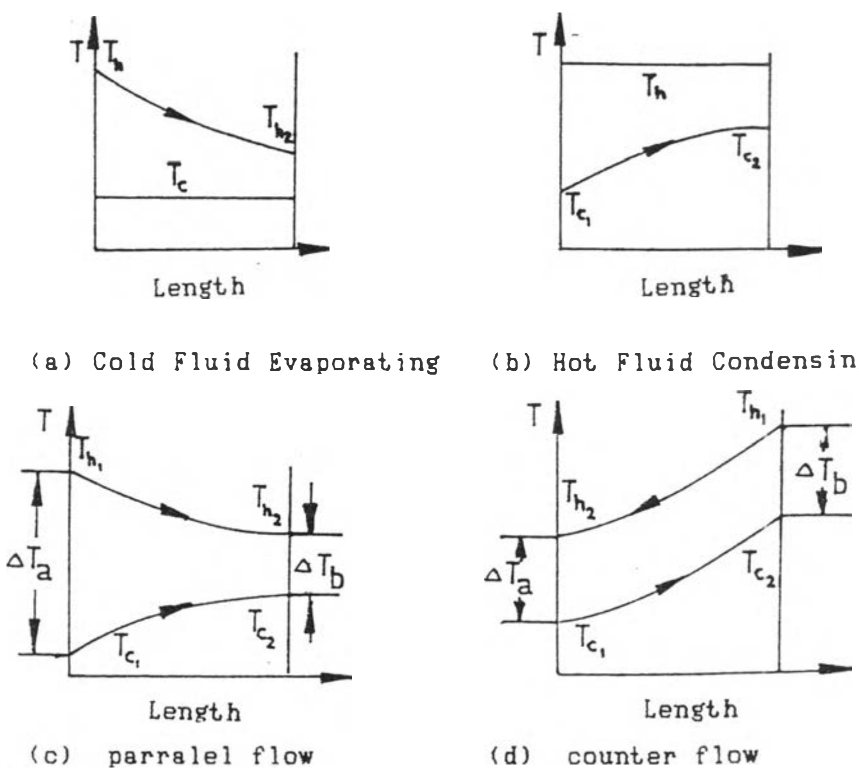


Figure 2.2.2 Axial Temperature Distribution in Heat Exchangers

For the cases of parallel flow and counterflow of fluids without phase change, Fig. 2.2.2c and Fig. 2.2.2d, the enthalpy change may be written as di = cdt; then

$$dQ = w_c c_c dT_c = \pm w_h c_h dT_h \qquad (2.2.20)$$

where the (+) sign refers to counterflow since dT_h/dx is positive and the (-) sign to parallel flow since dT_h/dx is negative.

Then from equation (2.2.20)

$$d(T_h - T_c) = dT_h - dT_c = dQ\left[\pm\frac{1}{w_h c_h} - \frac{1}{w_c c_c}\right] \qquad (2.2.21)$$

Substituting for dQ from equation (2.2.18)

$$\frac{d(T_h - T_c)}{(T_h - T_c)} = U\left[\pm\frac{1}{w_h c_h} - \frac{1}{w_c c_c}\right]dA \qquad (2.2.22)$$

which, when integrated with constant values of U, $w_h c_h$ and $w_c c_c$ between limits T_a and T_b, results in

$$\ln\frac{\triangle T_b}{\triangle T_a} = UA\left[\pm\frac{1}{w_h c_h} - \frac{1}{w_c c_c}\right] \qquad (2.2.23)$$

Similarly, integration of equation (2.2.20) results in

$$Q = w_c c_c (T_{c2} - T_{c1}) = w_h c_h (T_{h1} - T_{h2}) \qquad (2.2.24)$$

Solving this for $w_c c_c$ and $w_h c_h$ and substituting then in equation (2.2.23), we obtain

$$Q = UA\frac{\triangle T_a - \triangle T_b}{\ln(\triangle T_a/\triangle T_b)} \qquad (2.2.25)$$

for either parallel or counterflow. The quantity

$$\frac{\triangle T_a - \triangle T_b}{\ln(\triangle T_a / \triangle T_b)} \qquad (2.2.26)$$

is the logarithmic mean value, $\triangle T_{lm}$, of $\triangle T$ between $\triangle T_a$ and $\triangle T_b$, so $Q = UA\triangle T_{lm}$ [6].

It should be noted that $\triangle T$, for example is the difference in temperature of the fluids at a particular place in the heat exchanger, $\triangle T_a = (T_{h1} - T_{c1})$ for parallel flcw, but $\triangle T_a = (T_{h2} - T_{c1})$ for counterflow. In case the value of $\triangle T_a/\triangle T_b$ is between 1 to 2.5 or less than 1, the temperature difference in the heat exchanger can be estimated in terms of the following equation

$$\triangle T_{lm} = \frac{(T_{h1} + T_{h2}) - (T_{c2} + T_{c1})}{2} \qquad (2.2.27)$$

$\triangle T_{lm}$ from equation (2.2.23) is the arithmatic mean temperature difference [7].

The integration of equation (2.2.18) for the other flow arrangements results in a form of an integrated mean temperature difference $\triangle T_m$ such that

$$Q = AU\triangle T_m \qquad (2.2.28)$$

where $\triangle T_m$ is a complex function of T_{h1}, T_{h2}, T_{c1} and T_{c2}. Generally this function $\triangle T_m$ can be determined in

$$\triangle T_{lmc} = \frac{(T_{h2} - T_{c1}) - (T_{h1} - T_{c2})}{\ln\left[\dfrac{T_{h2} - T_{c1}}{T_{h1} - T_{c2}}\right]} \qquad (2.2.29)$$

$$C_c = \frac{w_c C_c}{w_h C_h} = \frac{T_{h1} - T_{h2}}{T_{c2} - T_{c1}} \qquad (2.2.30)$$

$$E_c = \frac{T_{c2} - T_{c1}}{T_{h1} - T_{c1}} \qquad (2.2.31)$$

These quantities may be interpreted as follows: $\triangle T_{lmc}$ is the logarithmic mean temperature difference for a counterflow arrangement with the same fluid inlet and outlet temperatures; C_h/C_c is the ratio of wC products of the two fluids; and E_c is called the effectiveness of the heat exchanger on the cold fluid side because it is a measure of the ratio of the heat actually transferred to the cold fluid to the heat which would be transferred if the same fluid were to be raised to the temperature of the hot inlet fluid.

2.3 Pressure Drop

Increasing the fluid flow rates in a plate heat exchanger increases the heat-transfer rate but it also the increases pressure drop. For practical reasons, related largely to other equipment commonly used with these exchanger units, a low pressure drop is preferred.

The overall pressure drop between two flat plates with turbulence promoters present can be expressed as [10]:

Total pressure drop = (lift) + (core friction) + (convergence, divergence, and reversal effects) + (momentum increase by change in velocity profile) (2.3.1)

The lift term g_x/g_c is a direct function of the vertical distance between the inlet and outlet and can be evaluated directly. Therefore, each pressure drop, $\triangle P$, has been corrected to exclude this term. The remaining three terms are of greater concern here, but are not easily separated. The core friction takes the form of the conventional Fanning equation and accounts for the loss caused by wall shear stress. In ducts with a uniform flow channel, such as a pipe, this term accounts for almost the entire pressure drop.

The convergence, divergence, and reversal effects can be divided into losses that occur due to expansions and contractions along the flow channel and those due to entrances, exits, and crossovers of the plates. The value of the term for these effects depends almost solely on the geometry of the channel, and in flow paths with turbulence promoters, it would be expected to account for an appreciable pressure drop [11,12].

The momentum term is of importance at the entrance to a tube. According to the boundary layer theory, turbulence promoters reduce the boundary layer but heat transfer is correspondingly increased. It thus appears that the boundary layer is reduced by turbulence promoters. Therefore, this momentum term would be significant in the plate heat exchanger.

Since the core pressure-drop terms cannot be evaluated individually, the pressure drop in a plate heat exchanger can always be calculated from a friction factor type equation

$$\triangle P = \frac{2fLG^2}{\rho D_e g_c} \qquad (2.3.2)$$

in which f is assumed to be a function only of the Reynolds numbers. As a general rule, all types of plate heat exchangers will operate in fully turbulent flow at Reynolds numbers over 1,000 and will be in laminar flow at Reynolds numbers below 10.

For a typical small plate heat exchanger with chevron corrugations, the transition regime would range approximately from Reynolds numbers of 10 to 150 [4] as presented in Fig. 2.3.1.

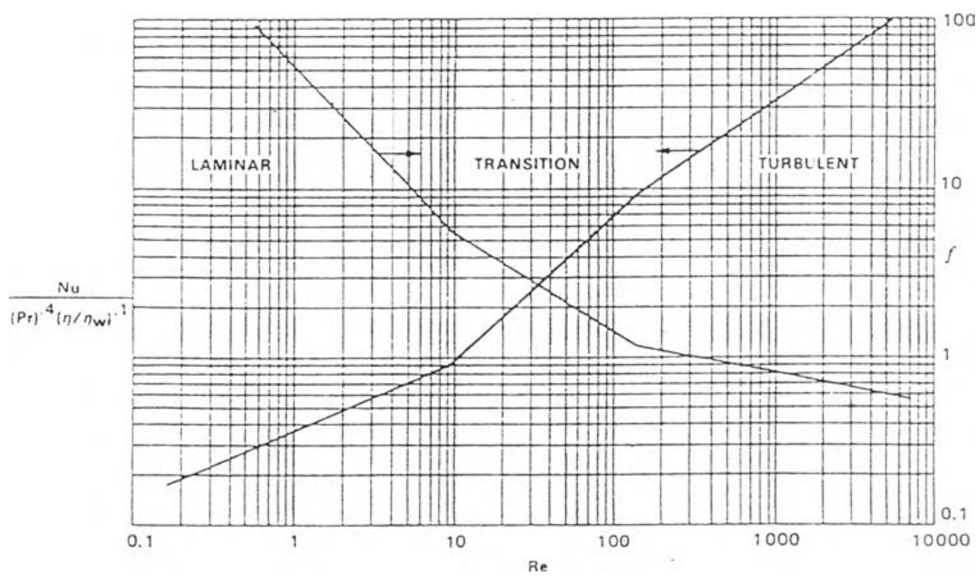


Figure 2.3.1 Performance characteristics of a small chevron-trough plate heat exchanger

The friction factor would be predicted from the following equations:

In turbulent flow: $f = 1.22/(Re^{0.382})$ (2.3.3)

In laminar flow: $f = 38/Re$ (2.3.4)

It is obvious that with plate heat exchangers the friction factors are much higher than for equivalent Reynolds numbers in tubes. However, it is found that velocities in plates are much lower and usually range from 0.1 to 3 m/s depending on the type of plate and application.

2.4 Fouling

Often termed the most unresolved problem in heat transfer, fouling is usually handled better in plate heat exchanger than in most other types of exchangers[13]. The induced high turbulence between the plates minimizes the build-up of most types of fouling. The high turbulence also enhances the effect of in-place cleaning methods. Should the extent of fouling require manual cleaning, this type of exchanger is easily opened, cleaned, and closed.

Fouling factors required in plate heat exchangers are small compared with those commonly used in shell and tube designs for six reasons [3].

1. High degree of turbulence maintains solid in suspension.

2. Heat transfer surfaces are smooth. For some types, a miror finish may be available.

3. No "dead space", where fluids can stagnate exist, as for exanple, near the shell-side baffles in the tubular unit.

4. Since the plate is necessarily of a material not subject to massive corrosion, deposits of corrosion products to which fouling can adhere are absent.

5. High film coefficients tend to lead to lower surface temperatures for the cold fluid.

6. Simplicity of cleaning. The small holdup volume and very high turbulence in a plate heat exchanger means that chemical cleaning methods are rapid and effective. If mechanical cleaning is required, all wetted parts within a plate heat exchanger are readily accessible.

Recommended fouling factors for plate heat exchangers are as follows [3]:

Fluid	Fouling factor (sq.m.)(°C.)(h.)/kcal. x 4.88 = (sq.ft.)(°F.)(hr.)/Btu.
Water	
Demineralized or distilled	0.00001
Towns (soft)	0.00002
Towns (hard) heating	0.00005
Cooling tower (treated)	0.00004
Sea (coastal) or estuary	0.00005
Sea (ocean)	0.00003
River, canal, borehole, etc.	0.00005
Engine jacket	0.00006
Oils, lubricating	0.00002 to 0.00005
Oils, vegetable	0.00002 to 0.00006
Solvents, organic	0.00001 to 0.00003
Steam	0.00001
Process fluids, general	0.00001 to 0.00006

Because heat transfer coefficients for plate exchangers are much higher than those for tubular units, it is recommended that values of not greater than one-fifth of the published tubular figures be used for plate exchangers.

2.5 Heat Transfer Method of Plate Heat Exchanger

The modern plate heat exchanger is rapidly becoming a widely used apparatus for heat transfer. The efficiency of heat transfer in plate heat exchangers is considerably higher than that in conventional heat exchangers. This higher efficiency is attributable to the pattern of the heat transfer plates which produces turbulence at low fluid velocities. The induced tubulence is produced by the plate pattern when the fluids flow in narrow streams (3 to 5 mm.) with many abrupt changes in direction and velocity. This turbulence, created by the shape of the plate pattern, reduces the liquid film resistance to heat transfer more efficiently than turbulence created by high flow rates and pressures in conventional exchangers. In addition to more efficient heat transfer, induced turbulence also compels even distribution of fluids in the exchanger.

Because of the nature of the flow patterns in a plate heat exchanger the conventional log-mean rate equation for the design of heat transfer equipment does not hold in most cases. Small exchangers with one or two thermal plates are the only configurations which may be designed with an uncorrected log-mean temperature difference. As shown in Fig 2.5.1, a fluid occupying one stream(b-c) may receive heat simultaneously from two other streams(a-b and c-d) flowing in opposite directions. Thus each stream, except for the two ends, is affected by two surrounding streams. It may also be seen that, as one adds plates to increase the heat transfer area, the flow patterns become more complex and very many configurations are possible.

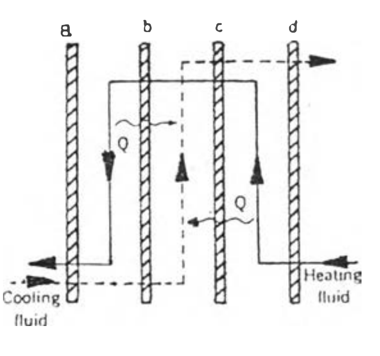
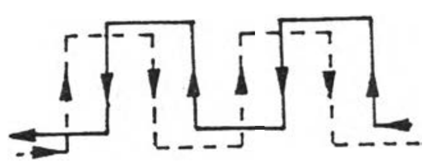
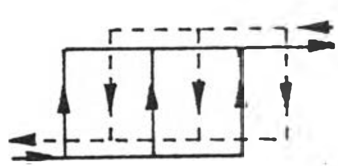


Figure 2.5.1 Mechanism of heat transfer in a plate heat exchanger

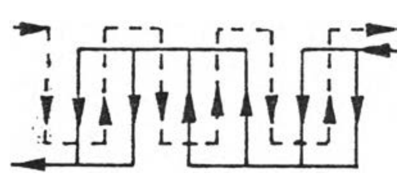
Of the many flow configurations possible, two basic pattern have thoroughly been investigated. The series flow configurations shown in Fig.(2.5.2 a) are those exchangers whose streams are continuous and change direction after each vertical pass. Fig (2.5.2 b) shows looped flow configurations whose streams divide into parallel flow channels and then reconverge to exit in a single stream. Complex flow patterns such as the one shown in Fig.(2.5.2 c) are developed from various combinations of series and looped flow patterns.



(a) Series flow pattern



(b) Looped flow pattern



(c) Complex flow pattern

Figure 2.5.2 Typical arrangements of plates and flow channels

A conventional heat exchanger design has several general assumptions. In order to validate the use of the design equation, the following conditions are imposed [14-18]:

1. The temperature and flow transients in the plate heat exchanger are negligible.

2. The heat losses to the surroundings are negligible.

3. The fluids exist only in the liquid phase within the exchanger.

4. The average overall coefficient of heat transfer is constant throughout.

5. Heat transfer resistances caused by fouling and channel wall thermal conductivity are negligible.

The plate heat exchanger satisfies these conditions quite favorably. The first condition has been substantiated by several experimenters. Steady state conditions have been found to exist by the constancy of multiple temperature measurements made simultaneously in both fluid streams. The edge channeling effect so pertinent in fluid flow characteristics does not affect the steady state thermal characteristics of the exchanger.

Since the ends of the exchanger are terminated by plates which provide dead air spaces, a situation closely approaching adiabatic walls exists. Heat losses from the edges of the plates are prevented by the gasketing arragement. Thus, condition 2 is well satisfied.

Regarding condition 3, the shape of the turbulence promoters and gasketing arrangement does not cause air voids to exist in the fluid streams.

The weakest assumed condition is the contancy of the average overall coefficient of heat transfer. The variation of the average overall coefficient depends upon the temperature difference of the inlet and outlet streams and is relatively constant while operating with small temperature differences.

For the purpose of engineering design calculations, the fourth condition approximates the actual situation.

The average overall coefficient is calculated from the individual film coefficients, the thermal conductivity, and the thickness of the heat transfer plates. Because of the dependency of the film coefficient on the physical properties of the fluid, the film and average overall coefficients are determined at the arithmetic average of the inlet and outlet stream temperatures. This is in agreement with the assumed condition of a constant average overall coefficient of heat transfer, U_{ave} [13].

A form of the Nusselt equation is used for the determination of the individual film coefficients. The equivalent diameter D_e for the plate heat exchanger is expressed as twice the width between plates. The expression comes from the conventional form, i.e. four times the flow area divided by the wetted perimeter, based on the fact that the channel width is very much smaller than the plate width.

NTU Rating

A heat exchange plate can be regarded as a module of known characteristics that can be defined by the NTU range covered by a single plate. For such a plate operating under unit flow ratio conditions [10,14],

$$NTU = \frac{\Delta T}{\Delta T_{lm}} = \frac{Ua}{wC_p} \qquad (2.5.1)$$

For n active plates, the total area is na. The total flow passages for each liquid are (n+1)/2. Hence,

$$NTU = \frac{2n}{n+1} \cdot \frac{Ua}{wC_p} \quad (2.5.2)$$

and when n is infinite,

$$NTU = \frac{2Ua}{wC_p} \quad (2.5.3)$$

Because plates are normally arranged in large packs, it is customary to define the NTU value of a plate by the expression given in equation (2.5.3) for $n=\infty$. NTU, also known as the performance factor, or thermal length θ , or temperature ratio TR, can be defined as the total temperature change for the process fluid, divided by the ΔT_{lm} for the exchanger.

Design Procedure

Basically, there are two approaches in plate exchanger design. One approaches uses LMTD correction factors; the other, heat-transfer effectiveness E, as a function of NTU [1].

The design procedures can be best illustrated by typical problems.

Given: 1. Hot fluid inlet and outlet temperatures and flow rates.

2. Cold fluid inlet temperature and flow rate.

3. Physical properties of the fluids.

4. Physical characteristics of the plates.

Required: The areas for both series- and parallel- flow exchangers.

The LMTD Approach

Considerations involved in the selection of flow patterns, including the number of passes for parallel flows, have already been discussed.

1. Calculate the heat load:

$$Q = (wc_p \triangle T)_h \qquad (2.5.4)$$

2. Calculate the cold fluid exit temperature:

$$T_{co} = T_{ci} + (Q/(wc_p)_c) \qquad (2.5.5)$$

3. Determine the physical properties of the fluids at the arithmetic average of their exchanger inlet and outlet temperatures.

4. Calculate LMTD:

$$LMTD = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln[(T_{hi} - T_{co})/(T_{ho} - T_{ci})]} \qquad (2.5.6)$$

5. Calculate NTU from equation (2.5.1).

6. Determine the LMTD correction factor [3] from Fig 2.5.3.

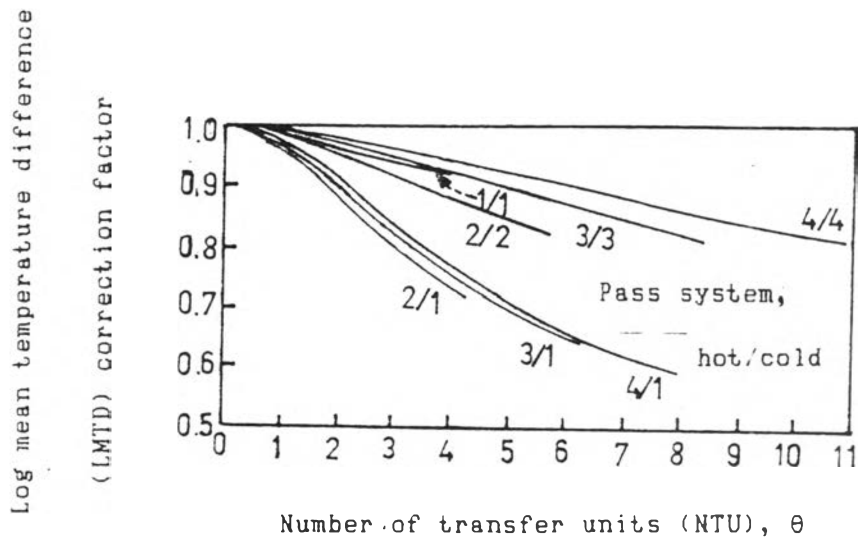


Figure 2.5.3 LMTD correction factors for multipass systems

7. Calculate the Reynolds number for each stream.

For (a) series flow (each fluid passes as one single stream through the channels):

$$Re = D_e G/\mu \qquad (2.5.7\ a)$$

For (b) parallel flow, assume the number of thermal plates and determine the number of substreams, n_c and n_h, into which each fluid is divided:

$$Re = D_e (G/n)/\mu \qquad (2.5.7\ b)$$

8. Calculate the heat transfer coefficients from heat transfer correlations.

9. Calculate the overall heat transfer coefficient:

$$U_{av} = \frac{1}{(1/h_h)+(x_p/k_p)+(1/h_c)+(d_{fh})+(d_{fc})} \qquad (2.5.8)$$

10. Calculate total heat transfer area:

$$A_t = Q/(U_{av} \text{ LMTD } F) \quad (2.5.9)$$

11. Calculate the number of thermal plates:

$$N = A_t/A_p \quad (2.5.10)$$

12. For parallel flow, from the number of thermal plates calculated in step 11, determine n for the hot and cold streams. If N is an odd number, n_h and n_c will be equal. If N is an even number, n_h and n_c will be unequal, and one of the fluids will have one more substream than the other.

13. Compare the values of n_c and n_h determined in step 12 with the corresponding values assumed in step 7(b). If the calculated values do not agree with the assumed values, repeat step 7(b) through 13, replacing the assumed values with the values calculated in step 12, until the values agree.

Steps 1 to 11 are common for both series and parallel flows. Steps 12 and 13 apply only to parallel flow.

Effectiveness-NTU Approach

The concepts of heat transfer effectiveness, NTU, and fluid heat capacity ratio are made use of in the development of a design method that can be applied to plate exchangers having different plate and flow configurations.

The procedure can be illustrated in the following steps

1. Calculate the heat load via equation (2.5.4).

2. Calculate the cold fluid exit temperature via equation (2.5.5).

3. Determine the physical properties of the individual fluids at the arithmetic average of their inlet and outlet temperatures.

4. Calculate the heat transfer effectiveness:

$$E = \frac{(wc_p)_h \; (T_{hi} - T_{ho})}{(wc_p)_{min} \; (T_{hi} - T_{ci})} \quad , \text{ if } (wc_p)_h > (wc_p)_c$$

or (2.5.11)

$$E = \frac{(wc_p)_c \; (T_{co} - T_{ci})}{(wc_p)_{min} \; (T_{hi} - T_{ci})} \quad , \text{ if } (wc_p)_c > (wc_p)_h$$

5. Calculate the heat capacity ratio, $(wc_p)_{min}/(wc_p)_{max}$.

6. Assume an exchanger containing an infinite number of channels, and select the flow pattern and configuration to find the required NTU using the appropriate E-NTU relationship (Figs. 2.5.4, 2.5.5, 2.5.6, 2.5.7).

7. Calculate the Reynolds number for each stream.
For (a) series flow, use equation (2.5.7 a). For (b) parallel flow, assume the number of substreams, n_c and n_h, into which each fluid is divided, according to the selected flow pattern, and use equation (2.5.7 b).

8. Calculate the heat transfer coefficients from applicable heat transfer correlations.

9. Calculate the overall heat transfer coefficient using

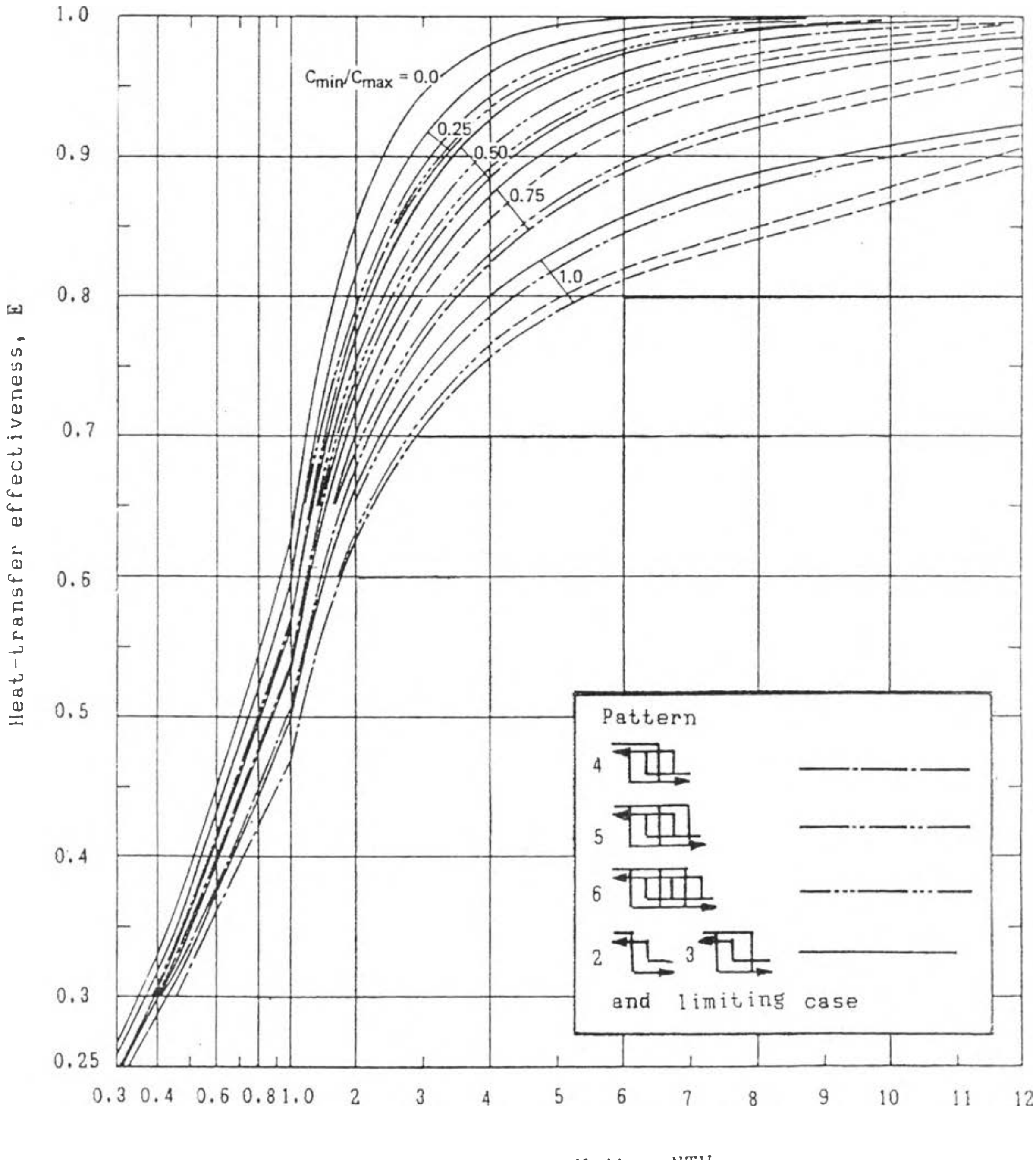


Figure 2.5.4 Countercurrent E-NTU relationships for loop patterns

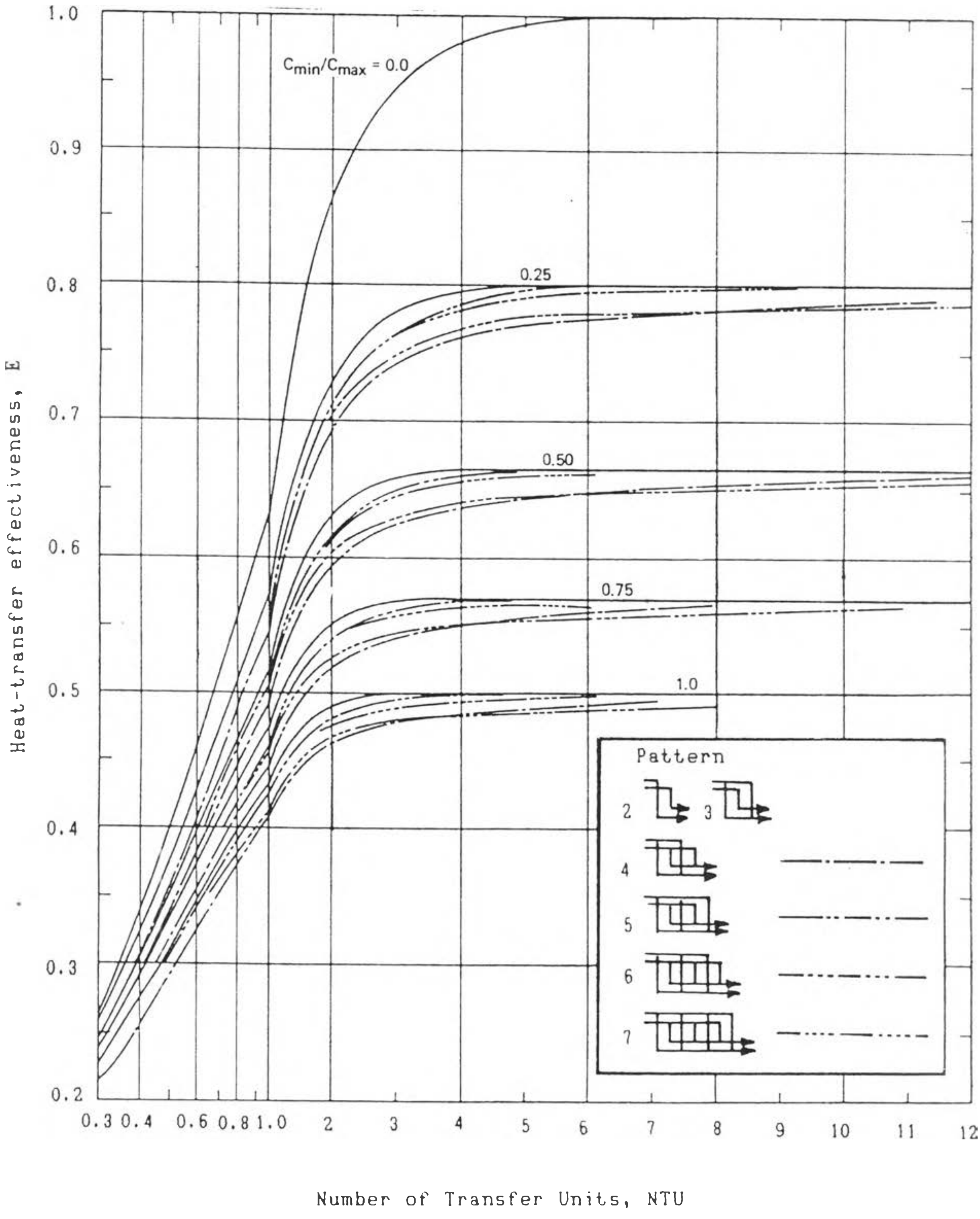


Figure 2.5.5 Cocurrent E-NTU relationships for loop patterns

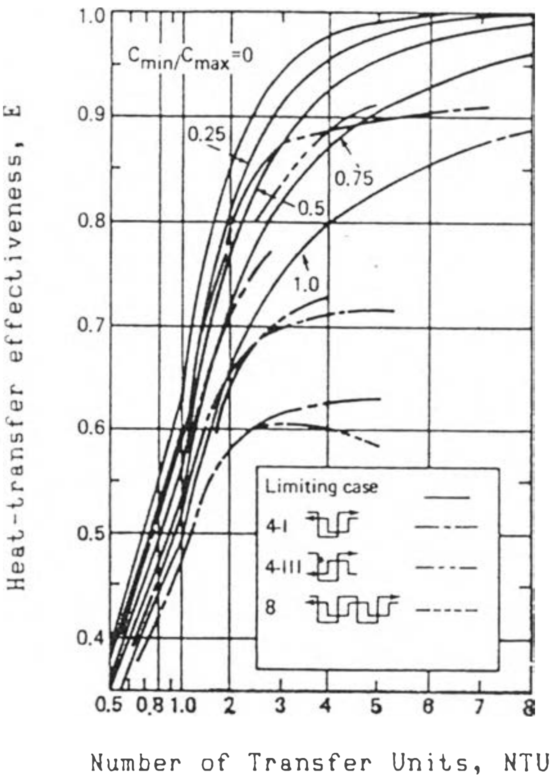


Figure 2.5.6 Series-series E-NTU relationships for 4 channels, types I and III, and 8 channels

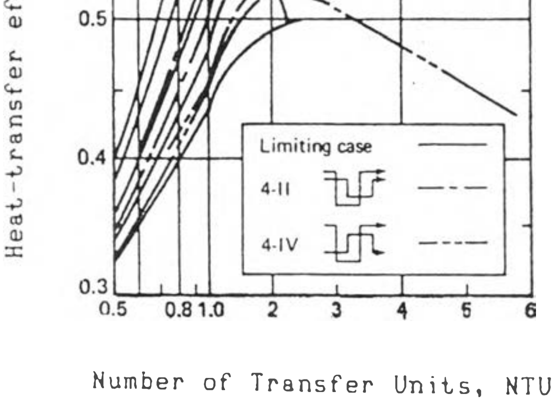


Figure 2.5.7 Series-series E-NTU relationships for 4 channels, types II and IV

equation (2.5.8).

10. Calculate the approximate number of thermal plates:

$$N = \frac{NTU(wc_p)_{min}}{U_{av} A_p} \qquad (2.5.12)$$

The number of channels is N+1 channels.

11. Calculate the pressure drop of each stream. If either of them is below its allowable value, the required thermal plates is N plates and the design calculation is completed. Otherwise, continue to step 12.

12. For (a) series flow, use a new flow pattern to find NTU from the appropriate curve (Figs. 2.5.6, 2.5.7) and proceed to steps 10 and 11. This is repeated until the design is satisfied or the flow patterns run out. In the latter case, change the characteristics of the plate and return to steps 6 through 11.

For (b) parallel flow, increase the channels by one to (n+1) to obtain a new flow pattern. Then the required NTU from the appropriate curve (Figs. 2.5.4, 2.5.5) and proceed to steps 7(b) through 11. This is repeated until the design is satisfied or the number of channels in each figure run out. In the latter case, change the characteristics of the plate and return to steps 6 through 11.

Steps 1 to 11 are common for both series and parallel flows. Steps 7(a) and 12(a) apply to series flow, and 7(b) and 12(b) to parallel flow.

In the two design approaches presented, the calculated heat load, stream temperatures and overall heat transfer coefficients are the same. Both design approaches achieve the same number of thermal plates.