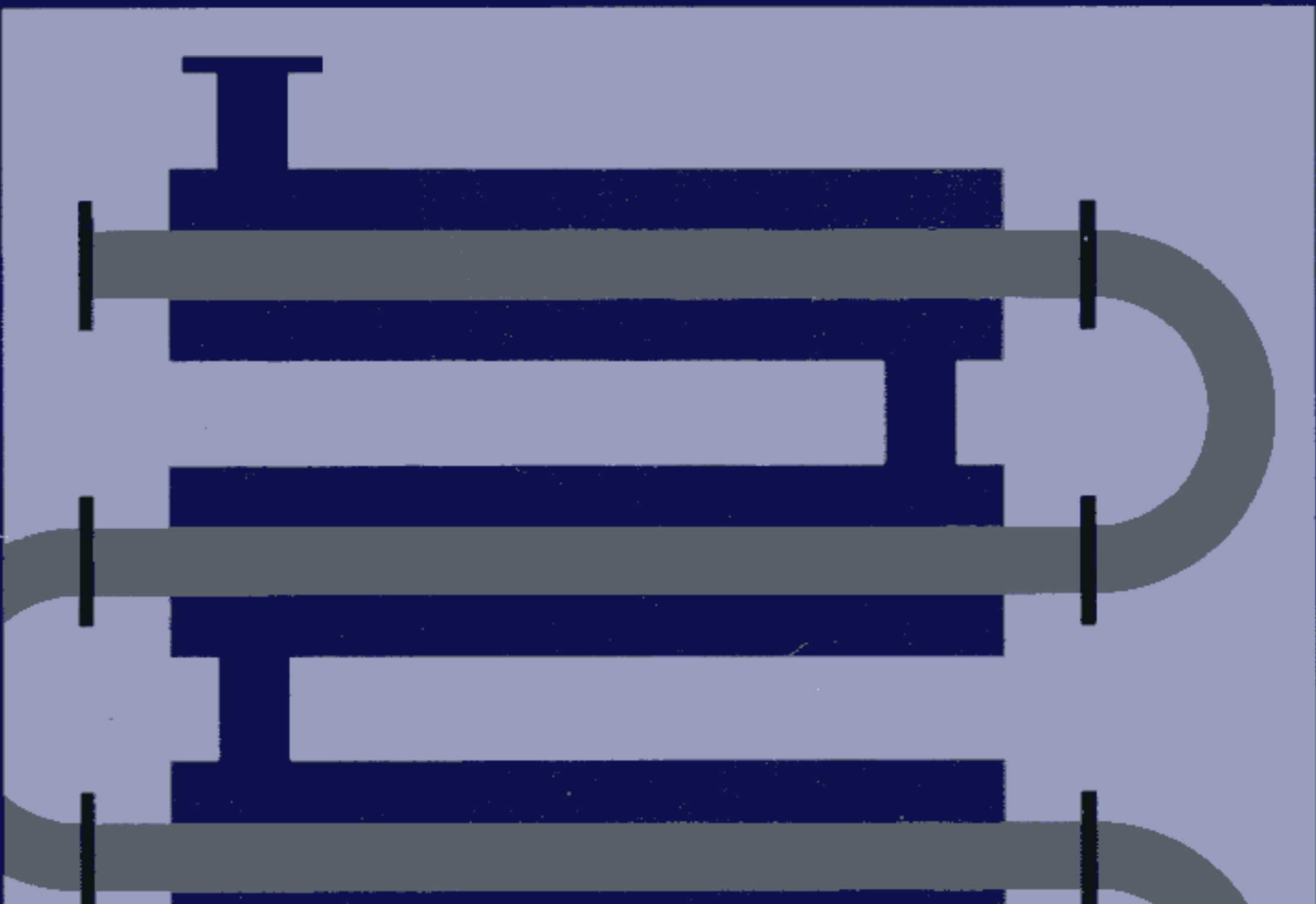
Engineering Design Guides 18

INTRODUCTION TO HEAT TRANSFER

D Butterworth



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Contents

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Notation

Symbol	Designation	Units (SI)
A	Heat-transfer area for small portion of surface	m^2
A_F	Fin surface area	m^2
A_T	Total heat-transfer area in heat exchanger (often referred to tube o.d.)	m^2
A_o	Heat-transfer area referred to tube o.d.	m^2
A_{req}	Heat-transfer area which would be required to achieve the heat load	m^2
C	Specific heat capacity of fluid	J/kg K
C_G	Gas-phase specific heat capacity	J/kg K
C_L	Liquid-phase specific heat capacity	J/kg K
D_i	Tube internal diameter (i.d.)	m
D_l	Outside diameter of lagging	m
D_o	Tube external diameter (o.d.)	m
D_w	Mean diameter of tube wall given by eqn (40b)	m
E	Parameter given by eqn (18)	—
F	Prandtl number correction factor for natural convection	—
F_B	Bypass correction factor for heat transfer	—
F_B'	Bypass correction factor for pressure drop	—
F_L	Leakage correction factor for heat transfer	—
F_L'	Leakage correction factor for pressure drop	—
F_N	Heat-transfer correction factor for number of rows crossed	—
F_T	MTD-value correction factor for non-counter-current flow	—
F_W	Heat-transfer correction factor for windows	—
F_c	Convective-boiling correction factor	—
G	Mass flow per unit area in uniform channel	kg/s m^2
Gr	Grashof number (see Table 2)	—
Gz	Graetz number (see Table 2)	—
H	Enthalpy, shell-side specific enthalpy	J/kg
H_F	Fin height	m
H_c	Baffle overlap height or height of cross-flow zone	m
H_{cut}	Baffle cutdown height	m
H_{in}	Shell-side inlet enthalpy	J/kg
H_{out}	Shell-side outlet enthalpy	J/kg
H_1, H_2	Enthalpy shown on Fig. 34	J/kg
J	Parameter given by eqn (37)	—
K	Number of velocity heads lost in headers and/or tube entry	—
K_W	Number of velocity heads lost for flow through window	—
L	Tube length	m
N	Number of tubes in a vertical row	—
N_c	Number of rows of tubes crossed	—
N_p	Number of tube-side passes	—
N_t	Number of tubes in a transverse row	—
Nu	Nusselt number (see Table 2)	—
P	Channel perimeter: parameter defined by eqn (58)	—
Pr	Prandtl number (see Table 2)	—
Pr_L	Prandtl number for the liquid phase	—
Q	Heat load in small area A of exchanger surface	W
Q_T	Total heat load for exchanger	W
R	Parameter defined by eqn (59)	—
R_W	Ratio of tubes in windows to tubes in cross-flow zones	—
Re	Reynolds number (see Table 2)	—
Re_L	Reynolds number of the liquid flow alone; eqn (31)	—
Re_c	Cross-flow Reynolds number defined by eqn (23)	—
Re_g	Cross-flow Reynolds number defined on basis of gaps between tubes; eqn (67)	—
S	Flow area	m^2
S_L	Total flow area for leakage in one baffle	m^2
S_W	Flow area through baffle window	m^2
S_c	Nucleation suppression factor in Chen (1966) correlation	—
S_m	Minimum cross-flow area given by eqn (25); for circular bundles it is taken near the equator	m^2
S_{sb}	Leakage flow area for gap between shell and baffle	m^2
S_{tb}	Leakage flow area for gaps between tubes and baffle	m^2
T	Bulk temperature, shell-side bulk temperature	K
T_R	Reference temperature for enthalpy	K
T_{cold}	Cold-side temperature	K
T_{hot}	Hot-side temperature	K
T_{in}	Shell-side inlet temperature	K
T_{out}	Shell-side outlet temperature	K
T_s	Saturation temperature	K
T_w	Temperature of fluid adjacent to the wall	K
U	Local overall heat-transfer coefficient	W/m^2K
$\bar{U}$	Average overall heat-transfer coefficient	W/m^2K

Symbol	Designation	Units (SI)
V_G	Gas-phase superficial velocity	m/s
V_L	Liquid-phase superficial velocity	m/s
W	Stream mass flowrate, shell-side mass flowrate	kg/s
W_G	Gas mass flowrate	kg/s
W_L	Liquid mass flowrate	kg/s
X_{tt}	Two-phase flow parameter defined by eqn (33)	—
Y	Tube pitch	m
Y_F	Fin pitch $(y_F+\delta_F)$	m
Y_b	Baffle pitch	m
Y_l	Longitudinal pitch (see Figs 4 and 5)	m
Y_t	Transverse pitch (see Figs 4 and 5)	m
Z	Parameter given by $\{\mu_L{}^2/\rho_L(\rho_L-\rho_G)g\}^{1/3}$	m
a	Constant in eqn (22) and Table 3	—
f	Friction factor for flow in tubes	—
f_c	Friction factor for flow across tube banks	—
g	Gravitational acceleration	m/s^2(9·81)
h	Enthalpy of tube-side fluid	J/kg
h_{in}	Tube-side inlet enthalpy	J/kg
h_{out}	Tube-side outlet enthalpy	J/kg
k	Thermal conductivity	W/m K
k_L	Liquid thermal conductivity	W/m K
k_l	Lagging thermal conductivity	W/m K
k_w	Wall thermal conductivity	W/m K
l	Characteristic length in dimensionless groups (see Table 1)	m
m	Index in eqn (22) given in Table 3	—
p	Pressure	Pa
p_s	Saturation pressure	Pa
p_w	Saturation pressure corresponding to wall temperature	Pa
q	Heat flux	W/m^2
q_{crit}	Critical heat flux	W/m^2
r	Fouling resistance	K m^2/W
r_F	Fouling resistance on fin surface	K m^2/W
r_i	Fouling resistance inside tubes	K m^2/W
r_o	Fouling resistance outside tubes	K m^2/W
r_w	Resistance of tube wall	K m^2/W
r_1	Fouling resistance on one side of a plate	K m^2/W
r_2	Fouling resistance on the other side of a plate	K m^2/W
t	Tube-side bulk temperature	K
t_{in}	Tube-side inlet temperature	K
t_{out}	Tube-side outlet temperature	K
u	Fluid mean velocity	m/s
u_m	Cross-flow velocity calculated at the minimum area (for circular tube bundles, calculated at the equator of the bundles)	m/s
u_W	Velocity of fluid through the baffle window	m/s
w	Tube-side stream mass flowrate	kg/s
y_F	Distance between fins	m
y_w	Wall thickness	m
α	Heat-transfer coefficient	W/m^2 K
α_{DB}	Heat-transfer coefficient calculated from Dittus–Boelter method	W/m^2 K

Symbol	Designation	Units (SI)
α_F	Heat-transfer coefficient referred to fin surface area	W/m^2 K
α_{FZ}	Heat-transfer coefficient given by Forster-Zuber (1955) pool boiling correlation	W/m^2 K
α_I	Heat-transfer coefficient for cross flow to an ideal tube bank	W/m^2 K
α_N	Condensing coefficient for N tubes in a vertical row	W/m^2 K
α_{Nu}	Condensing coefficient on a horizontal tube as given by the Nusselt (1916) equation	W/m^2 K
α_w	Wall heat-transfer coefficient	W/m^2 K
α_{con}	Convective component of boiling heat-transfer coefficient	W/m^2 K
α_i	Heat-transfer coefficient inside tube	W/m^2 K
α_l	Heat-transfer coefficient on the outside of lagging	W/m^2 K
α_{nuc}	Nucleate component of convective boiling heat-transfer coefficient	W/m^2 K
α_o	Heat-transfer coefficient outside tube	W/m^2 K
α_1	Coefficient on one side of a plate	W/m^2 K
α_2	Coefficient on the other side of a plate	W/m^2 K
β	Coefficient of cubical expansion	1/K
Γ	Condensation rate per unit perimeter of tube	kg/s m
Γ'	Condensation rate per unit length of tube	kg/s m
ΔA	Heat-transfer area calculated for a zone	m^2
ΔH	Change in enthalpy of shell-side fluid over a zone	J/kg
Δp	Pressure drop	Pa
Δp_c	Pressure drop in cross-flow region of baffle space	Pa
Δp_I	Pressure drop for cross-flow in an ideal bundle	Pa
δ	Thickness of equivalent laminar film	m
δ_F	Fin thickness	m
η	Fin effectiveness	—
θ	Mean temperature difference (MTD)	K
θ_{ln}	Logarithmic mean temperature difference (LMTD)	K
θ_{cc}	Mean temperature difference for counter-current flow	K
λ	Latent heat of vaporization	J/kg
μ	Viscosity	N s/m^2
μ_G	Gas-phase viscosity	N s/m^2
μ_L	Liquid-phase viscosity	N s/m^2
ρ	Density	kg/m^3
ρ_G	Gas-phase density	kg/m^3
ρ_L	Liquid-phase density	kg/m^3
σ	Surface tension	N/m
ω	Flow width per transverse pitch (see Fig. 5)	m

Introduction

In engineering it is often found necessary to transfer heat from hot to cold fluids by means of heat exchangers. There is a wide variety of equipment available for this purpose, although in this Engineering Design Guide discussion is restricted to the more common types (other heat exchangers being mentioned only in passing). The very important aspect of removing heat from a primary source, such as a fired heater or the fuel elements of a nuclear reactor, also falls outside the scope of this guide. Although sufficient information is provided to enable the reader to understand and deal with simple heat-transfer problems, this text must to a large extent serve as an introduction to the more specialized books to which reference is made.

Heat transfer is not an easy subject and time must be taken to master the basic concepts before attempting even the simplest design. Preferably, this guide should be read through methodically and the worked examples examined carefully. However those readers who do not wish to proceed right through the book but none the less have a specific heat-transfer problem to solve may like to adopt the following procedure.

1. Read carefully from *Conduction, convection, and radiation,* (p. 3) to *Natural convection on the outside of a horizontal tube* (p. 7).

2. Read all the material under the headings *Fouling and scale formation* (p. 15) and *Overall coefficients* (p. 15).

3. Scan through all the remaining material under the main heading **Basic concepts in heat transfer** (pp. 3–16) and read any additional text bearing on your particular problem.

4. Scan through the text under the main heading **Types of heat exchanger** (pp. 16–27) and read any relevant items.

5. Read carefully all the text under the main heading **The elements of design** (pp. 27–34).

Basic concepts in heat transfer

Conduction, convection and radiation

Heat travels from a hot body (or fluid) to a cold body (or fluid) and there must therefore be a temperature difference to enable the heat to flow. This elementary fact can easily be forgotten in design. In practice it means that it is impossible to heat a fluid up to exactly the same temperature as the heating fluid or, similarly, to cool a fluid down to exactly the same temperature as the cooling fluid. The last small fraction of heat transfer would require an exchanger of infinite size because the temperature difference would have fallen to zero.

Heat can be transferred from the hot fluid to the cold fluid by three processes which can operate together or separately. These processes are *conduction, convection,* and *radiation.*

Heat *conduction* is the transport of heat either through a solid body or through layers of fluid without there being any movement of hot material in the direction of heat flow (except perhaps on the molecular scale). The heat is transferred from molecule to molecule in the body by vibration of the molecules or, in the case of electrical conductors, by the movement of electrons from molecule to molecule. The ability to conduct heat is characterized by a quantity known as the thermal conductivity: i.e. a substance with a high thermal conductivity conducts heat well. Typical values of thermal conductivity for metals range from 15 W/m K for stainless steel up to 390 W/m K for copper.

For a flat metal wall separating two fluids (see Fig. 1) the heat flow through the wall is given by

$$Q_T = \frac{k_w}{y_w} A_T (T_1 - T_2), \tag{1}$$

where

Q_T is the total quantity of heat transferred per unit time,

k_w is the thermal conductivity of the wall material,

y_w is the wall thickness,

A_T is the total area of the wall, and

T_1 and T_2 are the temperatures of the hot and cold surfaces of the wall respectively.

Eqn (1) assumes that both T_1 and T_2 are constant over their respective surfaces.

Another term, the heat flux, q, is defined as the heat transferred per unit area:

$$q = Q/A, \tag{2}$$

where Q is the heat flowing through a small area A on the surface of the wall. For constant heat flux situations of course,

$$q = Q_T/A_T. \tag{3}$$

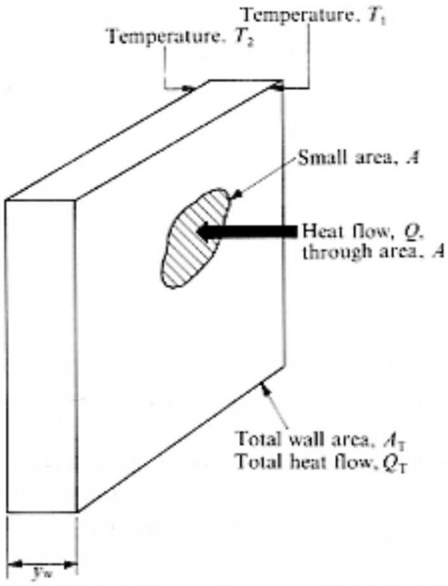


Fig. 1 Heat flow through a wall

Wall thermal conductance, α_W, and wall thermal resistance, r_w, can be defined from:

$$\alpha_w = 1/r_w = k_w/y_w. \tag{4}$$

α_w is also called the wall *heat-transfer coefficient*.

Heat convection is a process by which heat is transferred through a fluid by motion of the fluid. *Forced convection* occurs when the fluid is forced past a surface by an external force such as a pump, whereas with *natural convection* the fluid flow is generated by the heat-transfer process itself. Natural convection can be brought about by density differences between hot and cold parts of the fluid.

Heat transfer within a fluid in a heat exchanger is usually achieved by means of a combination of heat conduction and convection. The heat-convection effects are usually greatly improved by a process known as *turbulence*. A dye stream in a slowly flowing fluid will be seen to follow a steady path either straight or gently curved; this condition is known as a laminar flow. At higher fluid velocities, the dye stream will become jagged and will dance about. Also, it will tend to disappear a short distance from the point of injection since it will have become completely mixed with the full flow. This motion of the dye trace is caused by the chaotic movement of the fluid itself, and it is this movement which is called turbulence. Turbulent flow is generally useful in an exchanger since it gives better heat transfer. However, it also gives rise to a higher pressure drop through the heat exchanger, which has to be paid for by higher pumping costs.

Close to the wall of a channel, the turbulence tends to be suppressed, and near to the wall heat transfer occurs mainly as a result of conduction. Further from the wall, the convection due to turbulence can become much more important than the conduction. In both laminar and turbulent flows, the heat-transfer effects are represented by means of the heat-transfer coefficient which is defined as follows:

$$\alpha = q/(T_w - T), \tag{5}$$

where q is the heat flux into the fluid, T_w is the temperature of the adjacent wall, and T is the *bulk* temperature of the fluid. The bulk temperature at a given point in the exchanger is the temperature of the fluid if it were completely mixed. That this definition of α is similar to that given in eqn (4) for a tube wall becomes clear if eqn (5) is rewritten in the same form as eqn (1), which is perfectly possible if T_w, T, and α are constant. Thus,

$$Q_T = \alpha A_T (T_w - T). \tag{6}$$

By analogy between heat transfer in a wall and in a fluid, the idea of a laminar film of fluid which will have the same effect as the more complicated flow has been considered and, using this convention, eqn (6) may be rewritten as

$$Q_T = \frac{k}{\delta} A_T (T_w - T), \tag{7}$$

where k is the fluid conductivity and δ is the thickness of the equivalent laminar film. At one time, it was thought that such a well-defined film did exist and hence the heat-transfer coefficients referring to each fluid stream in an exchanger are still often called 'film coefficients'. Since this idea is, at best, only an approximation to the truth, equations of the form of (6) are recommended rather than those of the form of (7). Typical values of thermal conductivity for fluids are in the range 0·03 to 0·7 W/m K for liquids and 0·02 to 0·3 W/m K for gases.

Radiation is the transport of energy between two bodies which are not in contact without the assistance of an intermediate heat carrier, and it is usually important only in high-temperature situations such as furnaces. Radiation does not form part of the subject matter of this Engineering Design Guide, and reference should be made (1) to general books on heat transfer (such as McAdams, 1954, and Fishenden and Saunders, 1950), and then (2) to more specialized studies, for example Gray and Muller (1974).

Single-phase heat-transfer coefficients

Inside uniform cross-sectional channels. It will by this time have become obvious that one of the most important quantities in heat transfer is the heat-transfer coefficient. This must be calculated for both hot- and cold-fluid streams and the two coefficients combined in a way which will be described under the heading *Overall coefficients* (p. 15). To recapitulate, the stream coefficients (sometimes known as 'film coefficients') are defined by eqn (5).

Irrespective of the type of heat exchanger being considered the coefficient for single-phase flow may usually be written as

$$Nu = f_1(Re, Pr), \tag{8}$$

where Nu is the Nusselt number, Re the Reynolds number and Pr the Prandtl number.

$$Nu = \alpha l/k, \tag{9}$$

$$Re = \rho u l/\mu, \text{ and} \tag{10}$$

$$Pr = C\mu/k, \tag{11}$$

where l is a *characteristic length*, ρ the fluid density, u the fluid mean velocity, μ the fluid viscosity, and C the fluid specific heat. It is often helpful to remember that Prandtl numbers for gases are usually about 0·7 and for low-viscosity liquids about 1 to 10. Prandtl numbers are higher than 10 for viscous liquids.

Eqn (8) states only that the Nusselt number, Nu, is a function of Reynolds number, Re, and Prandtl number, Pr. The precise functional relationship and the definition of l depend on the geometry in question. For flow in a channel of uniform cross-section, a simple but reasonably accurate functional form is that of Dittus and Boelter (1930):

$$Nu = 0{\cdot}023\, Re^{0{\cdot}8} Pr^{0{\cdot}4}. \tag{12}$$

Here the characteristic length term to use is

$$l = 4S/P, \tag{13}$$

where S = the flow area, and P = the perimeter of the channel. Thus, for flow in a tube of internal diameter (i.d.) D_i,

$$l = 4 \times \frac{\pi D_i^2/4}{\pi D_i} = D_i. \tag{14}$$

Some characteristic lengths for other channels are given in Table 1. These characteristic lengths for uniform cross-section channels are often known as *hydraulic mean diameters*. The method may be applied to flow *parallel* to a bank (bundle) of tubes.

Eqn (12) can be written in the following alternative form

$$St = 0{\cdot}023\, Re^{-0{\cdot}2}\, Pr^{-0{\cdot}6}, \tag{15}$$

where St the Stanton number is given by

$$St = \frac{\cancel{k}\alpha}{\rho u C} = \frac{Nu}{Re\, Pr}. \tag{16}$$

Eqn (15) has been improved by the Engineering Sciences Data Unit (1967) and the result can be written as follows:

$$St = E\, Re^{-0{\cdot}205} Pr^{-0{\cdot}505}, \tag{17}$$

where

$$E = 0{\cdot}0225 \exp\{-0{\cdot}0225 (\ln Pr)^2\}. \tag{18}$$

E is plotted in Fig. 2 as a function of Prandtl number.

Four useful dimensionless groups, Nu, Re, Pr, and St, have been introduced. These are summarized in Table 2 together with an indication of the physical significance of each one.

Eqns (12) and (15) are recommended for quick preliminary calculations, whereas eqns (17) and (18) are better for the more precise final calculations. These equations all assume that the flow is turbulent, which occurs with certainty only if the Reynolds number is greater than 4000. They further assume that the ratio of channel length to characteristic length is greater than about 20 (so that entrance effects become unimportant), and that the physical properties are constant throughout the channel. Corrections for property variations and entrance effects are given by the Engineering Sciences Data Unit (1968b).

For low Reynolds numbers—less than about 2000—the flow in a uniform channel will remain laminar. Provided that natural convection effects remain small, the average coefficient may be represented approximately by

$$Nu = 1{\cdot}75\, Gz^{1/3} (\text{for } Gz > 9), \tag{19a}$$

and

$$Nu = 3{\cdot}66\ (\text{for } Gz \leqslant 9), \tag{19b}$$

where Gz is the Graetz number, given by

$$Gz = \frac{WC}{kL} = \frac{\pi}{4} Re\, Pr \frac{l}{L}; \tag{20}$$

W is the mass flowrate and L the channel length.

Since natural convection may be of great importance in laminar flows, it is important only to use eqn

Table 1. Characteristic lengths or hydraulic mean diameters to use in eqn (12) for heat transfer in uniform channels

Geometry	l
Flow on the inside of tube of i.d. D_i	D_i
Flow between parallel plates which are a distance b apart	$2b$
Flow in rectangular channel with sides of length a and b	$\frac{2ab}{a+b}$
Flow in annular passage whose outer tube has i.d. D and inner tube o.d. d	$D-d$
Flow in semicircular passage of diameter D over curved portion	$0{\cdot}611D$
Flow parallel to a equilateral triangular array of tubes of o.d. D_0 and pitch Y	$\frac{3{\cdot}46\,Y^2-\pi D_0^2}{\pi D_0}$
Flow parallel to a square ~~rectangular~~ array of tubes of o.d. D_0 and pitch Y	$\frac{4Y^2-\pi D_0^2}{\pi D_0}$

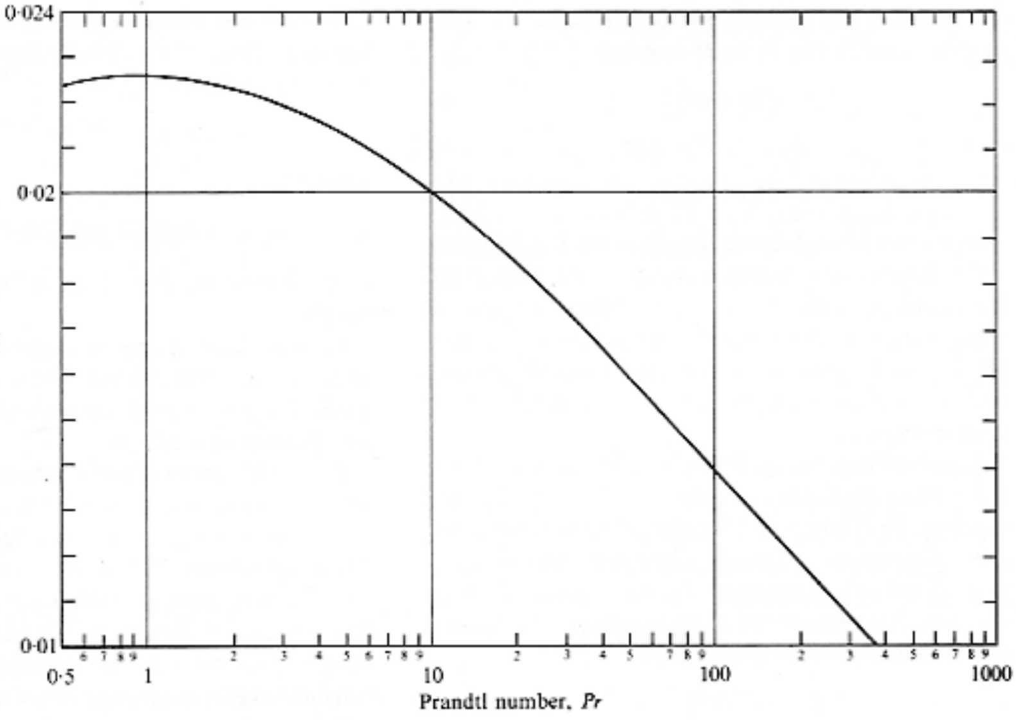


Fig. 2 Parameter E in the Engineering Sciences Data Unit (1967) correlation

(19) when

$$Gr\,Pr\frac{l}{L} \leqslant 10^3 \tag{21a}$$

if the tube is vertical, or

$$Gr\,Pr\frac{l}{L} \leqslant 10^4 \tag{21b}$$

if it is horizontal. Gr is the Grashof number given by

$$Gr = \frac{\beta g l^3 \rho^2 |T_w - T|}{\mu^2},$$

where β is the coefficient of cubical expansion of the fluid. When natural convection effects become important, the reader is advised to use the methods given by the Engineering Sciences Data Unit (1968a). The hydraulic mean diameter concept is less accurate for laminar flow than for turbulent flow and hence eqn (19) is most suitable for circular tubes.

For the Reynolds number range 2000 to 4000—known as the *transitional* range—heat-transfer coefficients are difficult to predict. It is therefore advisable to avoid this region in design if possible, but, if not, the coefficients for Reynolds numbers of 2000 (eqn 19) and 4000 (eqn 15 or 17) should be calculated and—to be on the safe side—the lower of these two coefficients used.

Heat-transfer coefficients for water flowing in tubes are typically about 1000–5000 W/m^2 K. For hydrocarbon and other organic fluids, coefficients are usually somewhat lower, being about 300–3000 W/m^2 K. For viscous fluids or for laminar flow, much lower coefficients are obtained of, say, 20–1000 W/m^2 K. For gases, coefficients are usually very low, being of the order of 10–300 W/m^2 K.

Examples 1 and 2 in the Appendix (p. 35) illustrate the use of these methods to calculate heat-transfer coefficients for flow in uniform channels.

Natural convection on the outside of a horizontal tube. It is not intended in this Engineering Design Guide to deal extensively with natural convection. However, there is one particular situation involving natural convection which merits special

Table 2. Summary of some dimensionless groups used in fluid flow and heat transfer

Group	Symbol		Physical significance
Nusselt	Nu	$\alpha l/k$, l/δ	Ratio of characteristic length to equivalent laminar film thickness
Reynolds	Re	$\rho u l/\mu$	Ratio of inertial to viscous forces
Prandtl	Pr	$C\mu/k$	Ratio of diffusivity of heat to that of momentum
Stanton	St	$\alpha/\rho u C$	Ratio of heat flow through wall to heat flow along channel
Graetz	Gz	WC/kL $\frac{\pi}{4}RePr\frac{l}{L}$	
Grashof	Gr	$\frac{\beta g l^3 \rho^2 \lvert T_w - T\rvert}{\mu^2}$	$\frac{\text{(Buoyancy forces)} \times \text{(inertial forces)}}{\text{(viscous forces)}^2}$

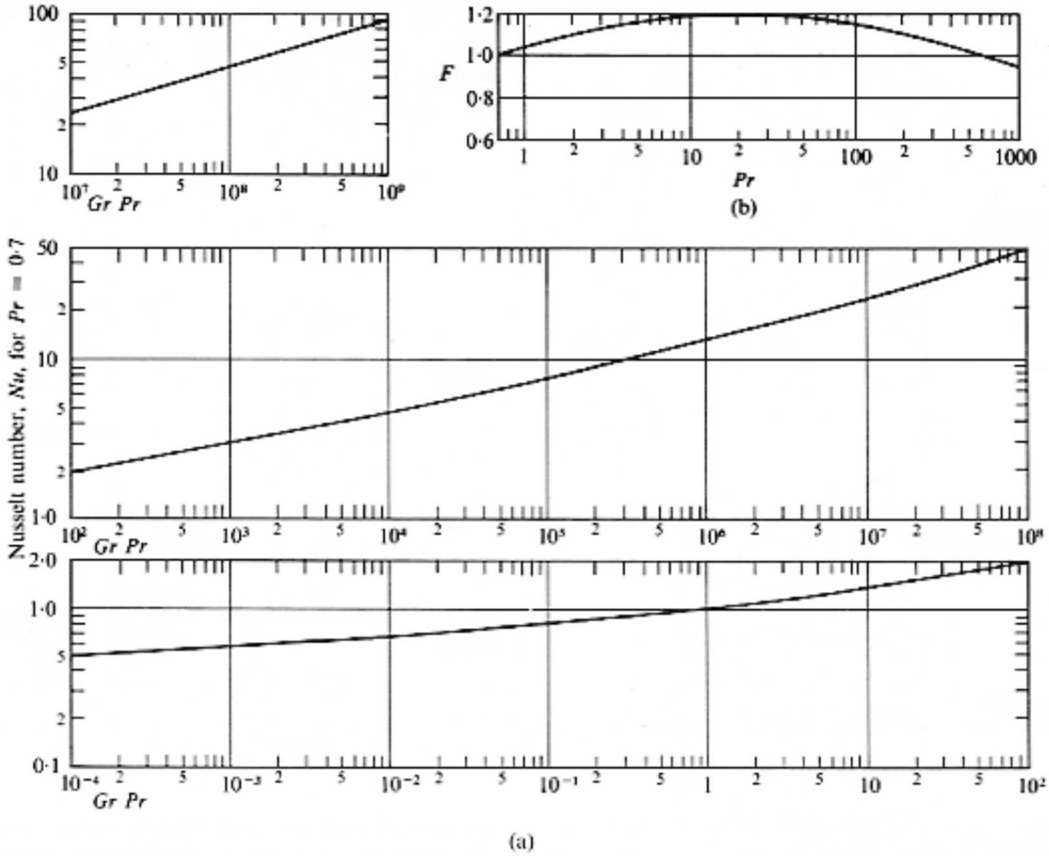


Fig. 3 Engineering Sciences Data Unit (1969) correlation for natural convection to a horizontal cylinder: (a) *Nu* plotted as a function of *Gr Pr*, (b) correction factor, *F*, for *Nu* if $Pr \neq 0{\cdot}7$ (multiply *Nu* by *F* if $Pr = 0{\cdot}7$)

consideration here, namely convection on the outside of a horizontal tube. This is especially significant because it is often necessary to calculate heat losses from horizontal pipes.

The Nusselt number for this situation is given as a function of the Prandtl number and the Grashof number, as shown in the ESDU correlation given in Fig. 3†. Example 3 (p. 36) illustrates the use of this figure. The characteristic length to be used in the Grashof and Nusselt numbers is the tube outside diameter.

Typical natural convection coefficients for gases are 0·5 to 20 W/m^2 K and for liquids 120 to 1000 W/m^2 K. For gases the heat transfer due to natural convection is usually of about the same order as the radiation heat transfer, and thus the coefficients obtained in practice are higher than those predicted using Fig. 3.

Flow across banks of smooth tubes. For flow across tube banks it is sometimes difficult in practice to ensure that all the fluid flows in the anticipated manner. For example, there is a marked tendency for fluid to evade the bundle by flowing around the sides, and certain types of heat exchanger allow fluid to avoid the bundle in a number of other ways (p. 17 ff.). This section however is concerned with flow over 'ideal' bundles; that is, ones which cannot be avoided by the flow.

With banks of tubes , it is important to consider whether the rows are staggered or not. Fig. 4 shows an *in-line* tube bank and a *staggered* bank. In the in-line bank, there are clear paths for the fluid to flow straight through the bundle whereas in the staggered bank the fluid tends to take a more tortuous path. Fig. 4 shows two important distances:

(1) Y_l, the longitudinal pitch which is the distance between successive rows in the bundle, and
(2) Y_t, the transverse pitch which is the distance between adjacent tube centre lines in a row.

It frequently occurs that tubes are arranged on a pitch which is either an equilateral triangle or square as shown in Fig. 5. In this figure, the pitch (Y) may be defined as the length of the side of the square or the triangle which forms the layout. The figure also gives the commonly used names and symbols for these pitch arrangements as defined by TEMA‡ (1968).

Heat transfer for flow over ideal tube bundles has been dealt with in detail by the Engineering Sciences Data Unit (1973), which presents the following equation for the flow of constant-property fluids and for the range of geometries of major practical interest:

$$Nu = a\,Re_c^m\,Pr^{0{\cdot}34}F_N. \qquad (22)$$

†As an alternative method the following equation may be employed:

$$Nu = \exp\{0{\cdot}0545 + 0{\cdot}0922 \ln Pr - 0{\cdot}0147 (\ln Pr)^2 + 0{\cdot}118 \ln Gr\,Pr + 0{\cdot}00485 (\ln Gr\,Pr)^2\}.$$

‡Tubular Exchanger Manufacturers' Association.

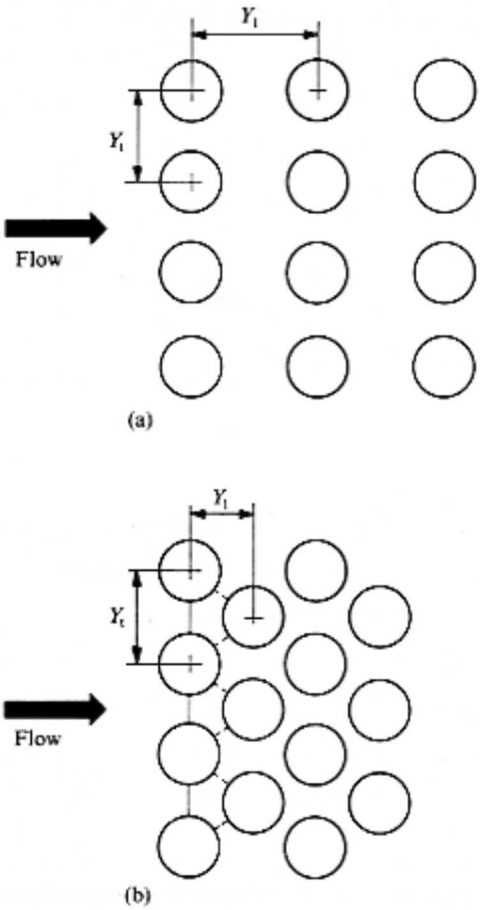


Fig. 4 Types of tube bank for cross flow: (a) in-line, (b) staggered

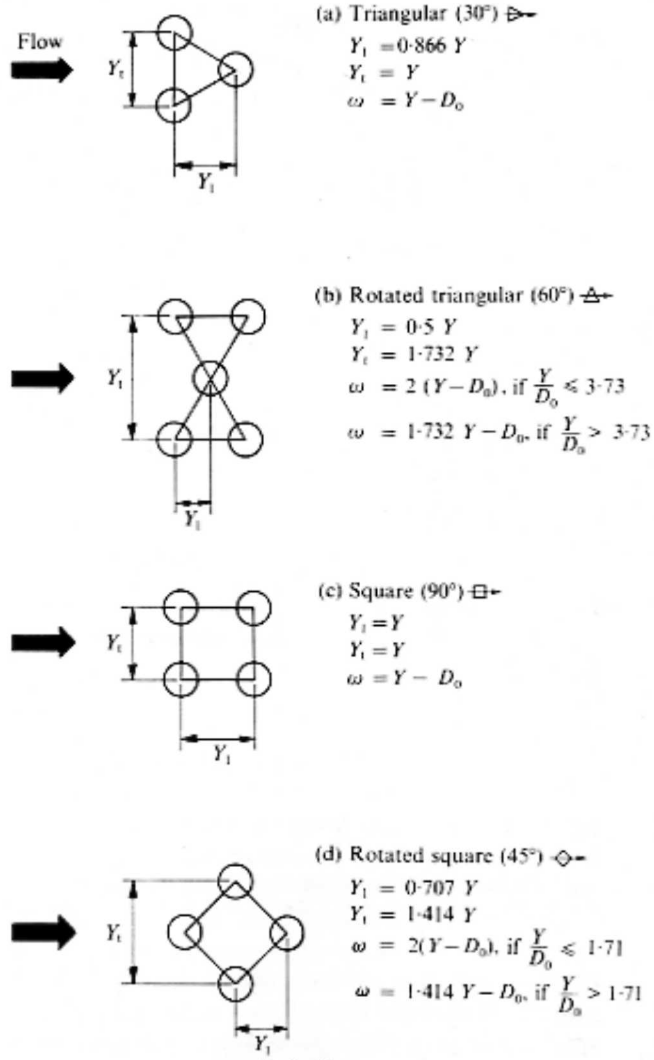


Fig. 5 Triangular and ~~rectangular~~ square tube banks: (a) triangular (30°), (b) rotated triangular (60°), (c) square (90°), (d) rotated square (45°)

In eqn (22) the length term in the Nusselt number is the tube external diameter (D_o). The Reynolds number for cross flow, Re_c, is defined as follows:

$$Re_c = \frac{\rho u_m D_o}{\mu}. \tag{23}$$

The velocity u_m in eqn (23) is calculated at the minimum cross-flow area, S_m, in the bundle, that is

$$u_m = \frac{W}{S_m \rho}, \tag{24}$$

where W is the mass flowrate. For staggered arrangements, S_m is not always easy to calculate but is found from taking the least of the two areas defined by the dotted or full line in Fig. 4. The minimum cross-flow areas for the square and equilateral triangle pitches are given by

$$S_m = L \omega N_t, \tag{25}$$

where N_t is the number of tubes in a row, L is the length of tube experiencing the cross flow, and ω depends on the arrangement and pitch as shown in Fig. 5. The values of a and m in eqn (22) depend on the tube arrangement and Reynolds number as shown in Table 3. To give an alternative representation, $Nu/Pr^{0{\cdot}34}$ is plotted against Re_c in Fig. 6. The factor F_N depends on the number of rows of tubes crossed and is to account for the turbulence generated as the fluid flows through the bundle (Fig. 7).

The way in which eqns (22–25) are employed in practice is illustrated by Example 4 (p. 36).

Further information given in Engineering Sciences Data Unit (1973) includes correction factors for fluid-property variations and for the flow to the bundle not being perpendicular to the tubes.

Flow over finned tubes. Heat-transfer coefficients for gases are usually very low. It is therefore advantageous to pack as much heat-transfer area as possible into a small volume of heat exchanger. One common way of doing this is to include transverse fins on the outside of the tubes. An amazing variety of fin types have been developed for different purposes

Table 3. Values of *a* and *m* to use in eqn (22)

Range of Re_c	In-line		Staggered	
	a	m	a	m
10 to 3×10^2	0·742	0·431	1·309	0·360
3×10^2 to 2×10^5	0·211	0·651	0·273	0·635
2×10^5 to 2×10^6	0·1[illegible]6	0·700	0·124	0·700

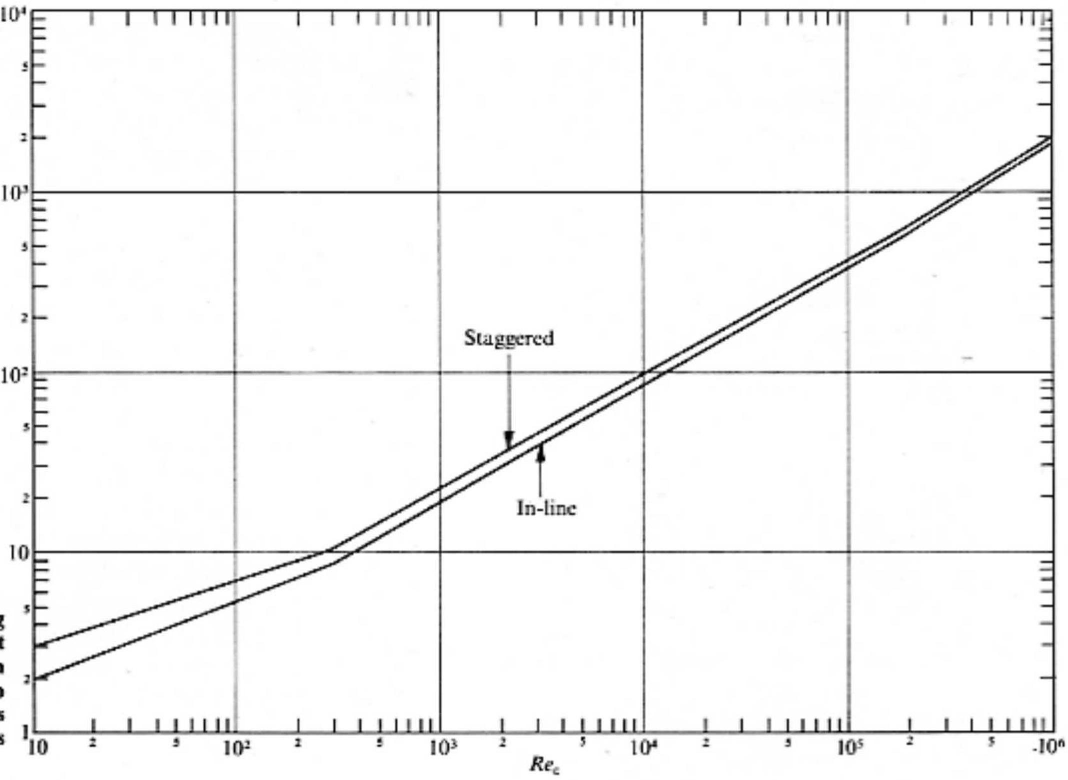
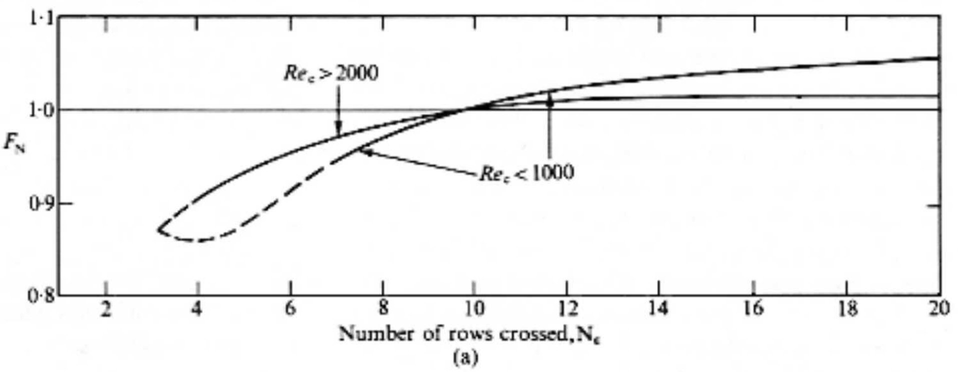


Fig 6 Engineering Sciences Data Unit (1973) correlation for cross flow to ideal tube banks (with ten rows crossed)

and by different manufacturers, and the information here is limited to those commonly used for air-cooled heat exchangers.

The fins, which frequently are circular and made of aluminium, are often produced by winding a strip of metal spirally onto a tube. This strip can be secured to the tube by locking into a groove on the tube surface or by tightly winding an L-shaped strip as shown in Fig. 8. These groove and L-type constructions are often referred to respectively as G- and L-fins. It is also possible to extrude fins on an aluminium tube and attach this tube to the outside of another tube as shown in Fig. 8. This has an advantage over G- and L-fins in certain corrosive environments since the inner tube is completely coated with aluminium. The disadvantage is,



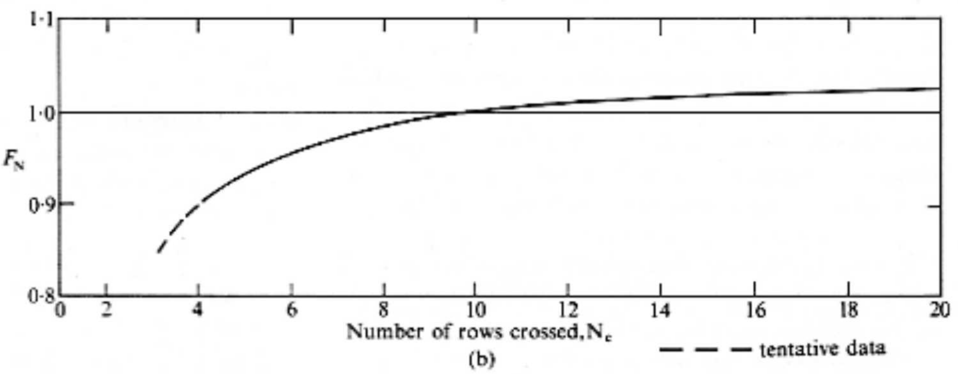


Fig. 7 Engineering Sciences Data Unit (1973) correlation for cross flow to ideal tube banks—correction factor for number of rows crossed: (a) in-line banks, (b) staggered banks

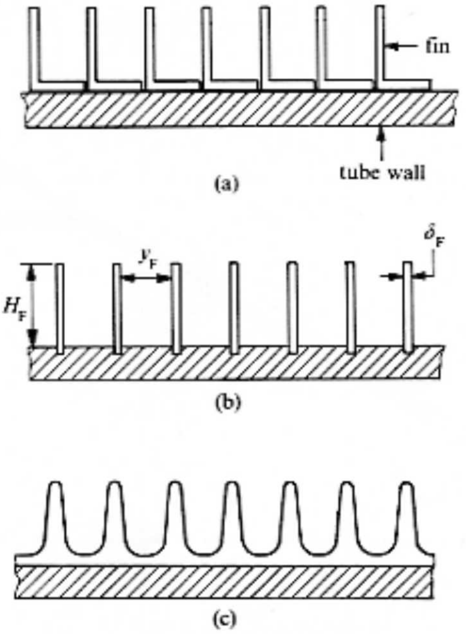


Fig. 8 Fin types: (a) L-fins, (b) G-fins, (c) extruded fins

however, that it is not possible to have as many fins per unit length of tube with extruded fins as with G- and L-fins. Typical fin spacings are 2·3–3·6 mm (7–11 fins per inch) and typical heights are 12·5–15·9 mm (0·5–0·625 in). Ratios of fin area to area of bare tube are usually between about 15:1 and 20:1.

In the air-conditioning industry, it is often the practice to have fins which are common to all tubes in the bundle as distinct from having each tube separately finned. This is effected by pressing out aluminium (or copper) plates with holes for the tubes and with other perforations or corrugations to improve heat transfer. The tubes are then threaded through these plates and expanded into them by forcing an expanding plug through the tubes. This type of finning is frequently called plate finning.†

A particular problem with finned tubes is that of calculating the thermal resistance in the fins. This problem can be overcome by introducing a quantity known as the *fin effectiveness*. The use of this quantity is discussed on p. 16. The heat-transfer coefficient for the surface of plain circular fins can be predicted approximately using the correlation of Briggs and Young (1963):

$$Nu = 0{\cdot}134\, Re_c^{0{\cdot}681} Pr^{0{\cdot}33} \left(\frac{y_F}{H_F}\right)^{0{\cdot}2} \left(\frac{y_F}{\delta_F}\right)^{0{\cdot}1134}, \quad (26)$$

where y_F is the distance between fins, H_F is the fin height, and δ_F is the fin thickness. These dimensions are illustrated in Fig. 8. The length term in the Nusselt number is the tube o.d. and the Reynolds number—given by eqn (23)—is the cross-flow Reynolds number assuming that the fins do not exist. The use of eqn (26) is illustrated in Example 5 (p. 36). General correlations of the eqn (26) type tend not to predict very accurately the coefficients for the wide range of finned-tube types, and it is, therefore, recommended that data supplied by the manufacturer should be used whenever possible.

Flow between special plates. In one form of exchanger the fluid streams pass between plates containing various sorts of corrugations to give rigidity and to improve heat transfer. Two examples of the plates for this type of exchanger, which is discussed further on p. 19, are shown in Fig. 9. There are many different types of plate, and in view of their number and the fact that much of the information is confidential it is difficult to give general heat-transfer correlations for all of them. All that can really be said is that the coefficients in these exchangers are often much higher than those obtained for a plain tube. This point is illustrated in Fig. 10 which gives information supplied by plate exchanger manufacturers (Alfa Laval, 1969, and A.P.V., 1974).

Boiling coefficients

Coefficients for boiling heat transfer are much more difficult to estimate than those for single-phase heat transfer. This is because of the much greater complexity of boiling phenomena involving gas and liquid mixtures. The subject of boiling heat transfer can be divided into the topics of *pool boiling* and *forced-convective boiling*, corresponding roughly to the natural and forced-convective heat transfer previously described for single-phase heat transfer. Pool boiling occurs when heat is being transferred to a pool of liquid where the motion in the liquid is due only to the agitation of bubbles produced in the boiling and to local density differences in the liquid. In forced-convective boiling, the fluid is being driven through the equipment by external forces, for example by a pump.

Since boiling is a very complex topic, only a superficial treatment of it can be given in this Engineering Design Guide. Detailed books on the subject are those of Collier (1972) and Hewitt and Hall Taylor (1970).

Pool boiling. Consider a vessel containing a liquid that has been heated up to its saturation temperature (boiling-point temperature for the system pressure). If this fluid is heated further by means of a single horizontal tube, the heat-transfer coefficient varies with the temperature difference as illustrated in Fig. 11, where T_w is the temperature of the tube wall (adjacent to the liquid) and T_s is the saturation temperature. The different regions shown on this figure have the following features.

1. *Single-phase natural-convection region.* For low values of wall superheat, $T_w - T_s$, no vapour bubbles form and hence the heat transfer is purely by single-phase natural convection. The

†There is a special type of plate exchanger used in the cryogenic industry which is known as a plate-fin exchanger (it is actually a plate exchanger with finning between the plates). Care must be taken not to confuse these special plate-fin exchangers with exchangers with fins formed from continuous plates.

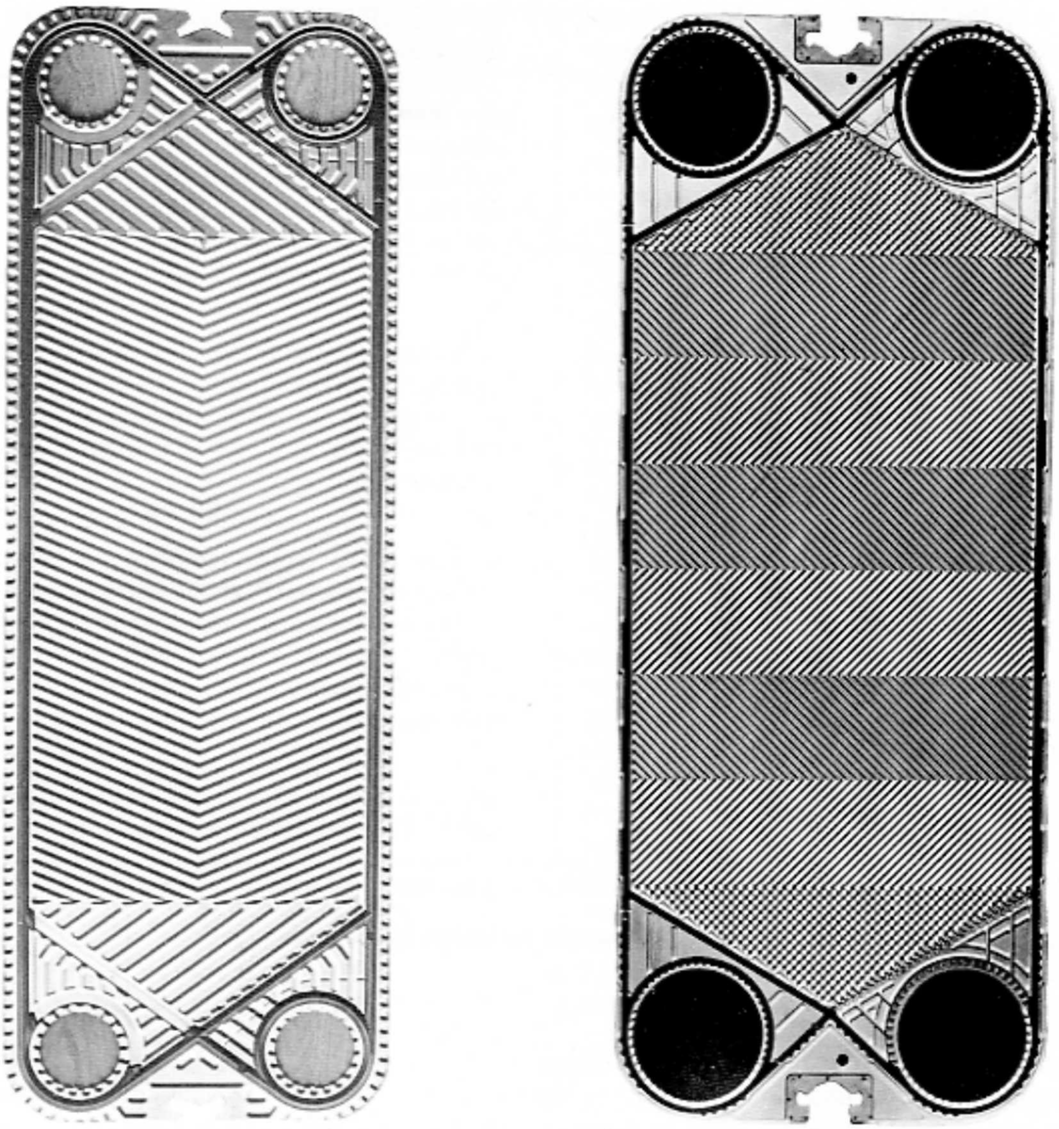


Fig. 9 Examples of plates used in commercial plate exchangers (courtesy of A.P.V Co. Ltd. and Alfa-Laval Co. Ltd.)

reason why no boiling occurs is that a bubble must be in superheated liquid before it can grow. This is because the vapour pressure inside the bubble must be increased by a small value of superheat in order to overcome the surface tension forces which are tending to collapse it.

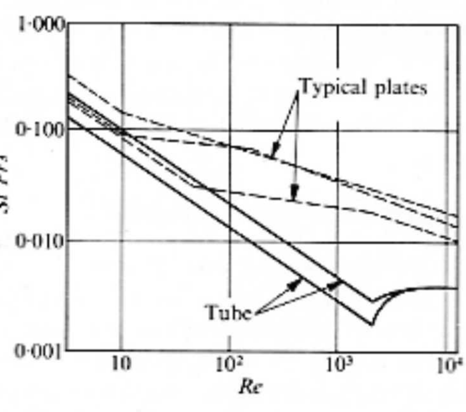


Fig. 10 Comparison of coefficients in plate and tubular exchangers (courtesy of Alfa-Laval Co. Ltd. and A.P.V. Co. Ltd.)

2. *Nucleate-boiling region.* For higher wall superheats, the bubbles can grow and break away from the surface. The agitation caused by these bubbles together with other effects caused by bubble growth, results in a large increase in heat-transfer coefficient.

3. *Transition-boiling region.* With increasing wall superheats it is difficult for the liquid to reach the surface owing to the vapour flow away from it. Hence dry patches begin to form on the surface which causes the heat-transfer coefficient to fall.

4. *Film-boiling region.* At very high wall superheats, no liquid can reach the surface and hence a continuous film of vapour is formed adjacent to it. The liquid is vaporized by heat which reaches it by conduction across the vapour film and also by radiation. The phenomenon of film boiling is well known to anyone who has spilt water onto the hot plate of a cooker when the plate has been heated to a high temperature. The water forms into droplets which skate about the

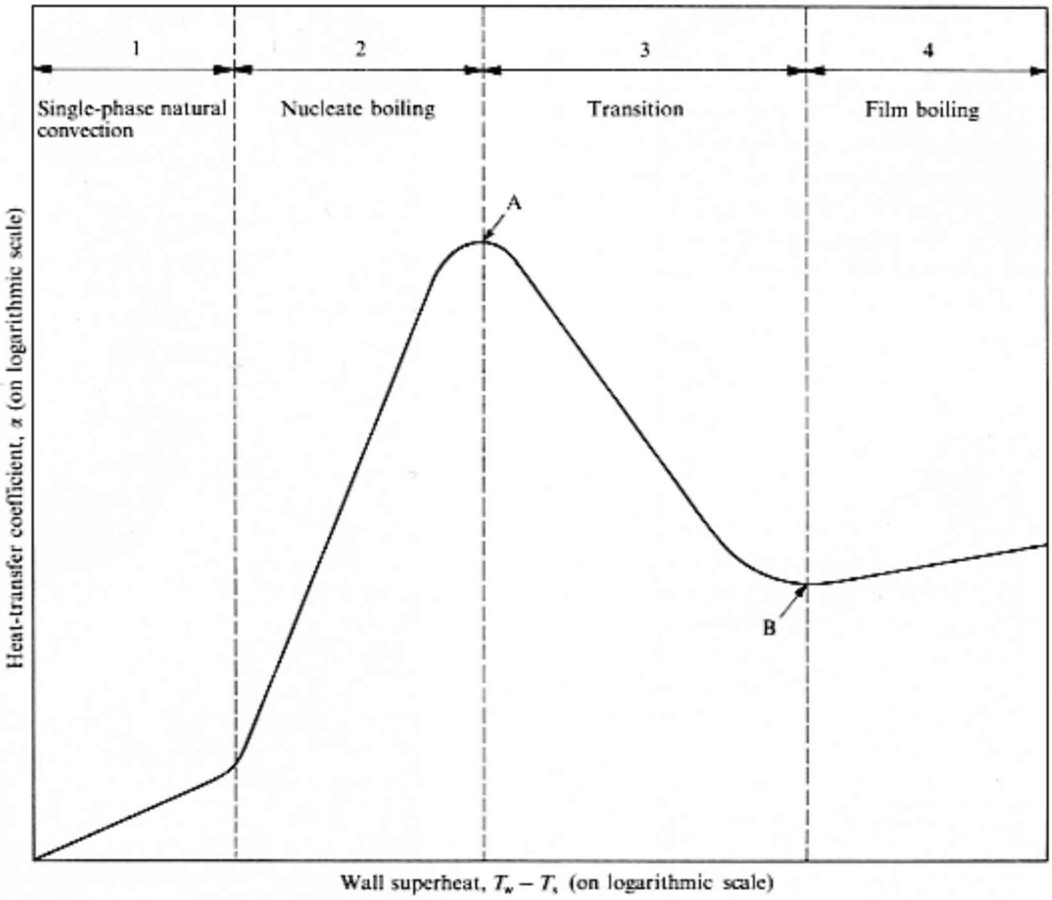


Fig. 11 Heat-transfer mechanisms in pool boiling

surface supported by the vapour film. These droplets evaporate very slowly because of the low heat-transfer coefficient.

It is possible for boiling to occur when the bulk of the liquid in the pool is below saturation. This is known as *subcooled* boiling (in contrast to the *saturation* boiling described previously) and for it to take place, the wall temperature must be above the saturated value. In subcooled boiling, the bubbles which break away from the surface tend to collapse as they rise through the cold liquid.

The point A (Fig. 11) corresponding to the maximum heat-transfer coefficient is referred to by a variety of names, for example *burnout point, critical point, departure from nucleate boiling* (DNB), etc. The point B is called the *Leidenfrost point* and the corresponding temperature the *Leidenfrost temperature*.

For design purposes, the nucleate-boiling region is the most important. Correlations for this region tend to be somewhat unreliable because of the complexity of the boiling process. In addition, the bubble nucleation properties of a surface depend on the nature of the surface, that is the wall material, its cleanliness, whether it is polished or rough, and so on. However, a useful but approximate correlation for saturated pool boiling is that of Forster and Zuber (1955):

$$\alpha = 0{\cdot}00122\left[\frac{k_L^{0\cdot79} C_L^{0\cdot45} \rho_L^{0\cdot49}}{\sigma^{0\cdot5} \mu_L^{0\cdot29} (\lambda \rho_G)^{0\cdot24}}\right] \times (T_w - T_s)^{0\cdot24} (p_w - p_s)^{0\cdot75}, \qquad (27)$$

where the subscript L refers to the liquid phase, the subscript G to the gas (vapour) phase, λ is the latent heat of vaporization, and σ is the surface tension. The pressures p_w and p_s are the saturation pressures corresponding to T_w and T_s, respectively. Eqn (27) is for boiling single components; for mixtures, the heat-transfer coefficients can be much lower than those predicted by this equation. Although any coherent system of units can be used in eqn (27), it is suggested that the SI set is the most convenient. To estimate the maximum value of the heat-transfer coefficient (point A in Fig. 11), which is of importance in design, the following theoretical equation proposed by Zuber, Tribus, and Westwater (1961) can be used.

$$q_{crit} = 0{\cdot}131\lambda\{\sigma g(\rho_L - \rho_G)\rho_G^2\}^{0\cdot25}, \qquad (28)$$

where g is the gravitational acceleration.

Eqns (27) and (28) apply to the case of a single tube or piece of heat-transfer surface inside a large pool of liquid. For boiling on the outside of bundles of tubes, the situation is far more complex on account of interactions occurring between the tubes, such as vapour from one tube affecting higher tubes in the bundle. Provided, however, that the heat flux is well below the critical value (as predicted from eqn (28)) eqn (27) is likely to be on the safe side.

Heat-transfer coefficients for pool boiling can range from about 100 W/m^2 K for boiling heavy oils up to about 10 000 W/m^2 K (or even higher) for boiling water. Typical critical heat flux values are 0·3–1 MW/m^2 for water and 0·1–0·4 MW/m^2 for organics. Example 6 (p. 37) illustrates the use of eqns (27) and (28).

Convective boiling. The mechanisms occurring in convective boiling are quite different from those occurring in pool boiling. These mechanisms are explained for upward flow in a vertical tube with reference to Fig. 12. The following conditions occur in different parts of the tube.

1. *Subcooled-single-phase region*. It is assumed here that the liquid enters the tube below saturation and must be heated to saturation before boiling occurs. Thus the tube has a single-phase heating region whose heat-transfer coefficient is given by the normal single-phase equations such as eqn (12).
2. *Subcooled-boiling region*. The liquid near the wall exceeds saturation before the bulk of the liquid reaches this condition. When the wall superheat is sufficient to produce bubbles, subcooled nucleate boiling can occur.
3. *Saturated-boiling region*. When the bulk of the liquid has been heated to saturation, saturated boiling occurs accompanied by net generation of vapour. The liquid and vapour flowing together can give rise to a number of different flow patterns. For long tubes, the final flow pattern reached in the saturated boiling region is known as annular flow and is characterized by films of liquid on the wall with droplets of liquid flowing in the high-velocity gas core.
4. *Dry-wall region*. This region, which is known by various names such as *liquid-deficient region*, *post-burnout* region, etc., occurs after the liquid films have been evaporated away. There may still, however, be a considerable quantity of liquid flowing in the form of droplets. The heat-transfer coefficient usually increases along the tube from the subcooled single-phase region through to the end of the saturated-boiling region. However, in the dry-wall region the heat transfer drops dramatically since the heat is transferred to the gas phase which has a low thermal conductivity.

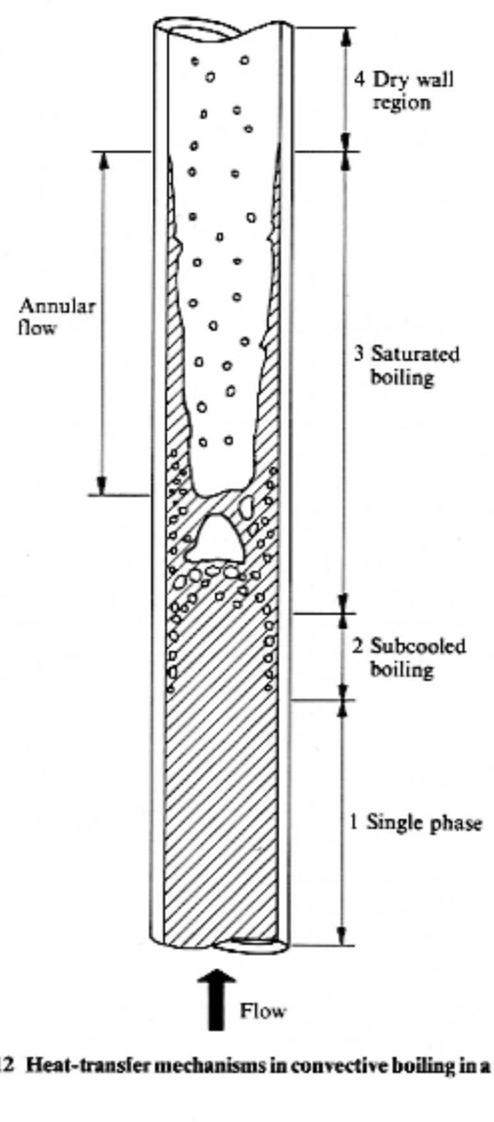


Fig. 12 Heat-transfer mechanisms in convective boiling in a tube

The saturated-boiling region is the most important region encountered in design. Chen (1966) has shown that the coefficient in this region is made up of 2 components, namely a nucleate-boiling component and a convective component. Thus,

$$\alpha = \alpha_{nuc} + \alpha_{con}. \tag{29}$$

Chen showed that the convective component, α_{con}, can be estimated using a modified Dittus–Boelter equation:

$$\alpha_{con} = F_c \times \alpha_{DB}, \tag{30}$$

where α_{DB} is the coefficient derived from eqn (12) if the Reynolds number is calculated on the assumption that the liquid alone is flowing in the pipe. Thus,

$$Re_L = \frac{(1-x)GD_i}{\mu_L}, \tag{31}$$

where G is the total mass velocity in the tube (total flow/flow area) and x is the mass fraction of vapour. These are defined as follows:

$$x = \frac{W_G}{W_L + W_G} \quad \text{and} \quad G = \frac{W_L + W_G}{S}. \tag{32}$$

The two-phase convective-correction factor F_c is given by Fig. 13, where $1/X_{tt}$ is a special two-phase parameter given by

$$\frac{1}{X_{tt}} = \left(\frac{x}{1-x}\right)^{0\cdot9} \left(\frac{\rho_L}{\rho_G}\right)^{0\cdot5} \left(\frac{\mu_G}{\mu_L}\right)^{0\cdot1}. \tag{33}$$

Eqn (30) gives an increase in coefficient with vapour fraction (except for vapour fractions greater than 0·8).

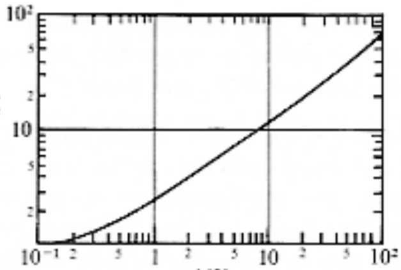


Fig. 13 Chen (1966) correlation—convective component correction factor

Chen showed that the nucleate-boiling component, α_{nuc}, is given by a modified form of Forster-Zuber (1955) equation:

$$\alpha_{nuc} = \alpha_{FZ} \times S_c, \tag{34}$$

where α_{FZ} is the coefficient given by eqn (27). S_c is known as the suppression factor and accounts for the fact that nucleate boiling is more difficult in convective conditions than in pool boiling. This factor is given by Fig. 14. With increasing quality, α_{nuc} decreases but this decrease is more than compensated by the increase in α_{con}. Example 7 illustrates the use of the Chen method.

Because of the sharp deterioration of heat-transfer coefficient in the dry-wall region, it is obviously important to be able to predict where the transition to this region from saturated boiling will occur and, if possible, to design in order to avoid it. These correlations, frequently referred to as burnout correlations and unfortunately known only for water and a few Freons, are reviewed by Macbeth (1968). A recent theory has, however, been developed by Whalley, Hutchinson, and Hewitt (1974) for calculating burnout for any fluid.

So far, only convective boiling in a vertical tube has been described. The Chen correlation may however be applied with some caution to a horizontal tube, and it is also useful for flow in uniform cross-section channels such as annuli. In this latter case D_i is replaced by the hydraulic mean diameter given in Table 1. There is a lack of information concerning cross flow to tube bundles, but it is tentatively suggested that the Chen method is again used but with α_{DB} being replaced by the cross-flow coefficient calculated from eqn (22). Once again, Re_c is calculated on the assumption that the liquid is flowing alone at that point in the bundle. This Reynolds number is also used in evaluating S_c from Fig. 14.

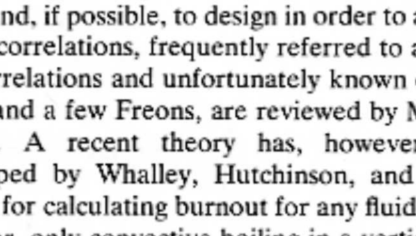


Fig. 14 Chen (1966) correlation—nucleate-boiling suppression factor

One of the difficulties in using convective-boiling correlations is that the coefficients vary through the exchanger. Step-by-step calculation methods must therefore be used in design calculations (see *The elements of design* (p. 27 ff.)).

Condensing coefficients

When condensation occurs in practical equipment the condensed liquid, or condensate, usually forms continuous films on the cooled surfaces. However, tubes can be coated in a particular way to produce a form of condensation, known as dropwise condensation, in which droplets of condensate form and coalesce until they are large enough to roll off the surface. Dropwise condensation has been studied extensively in the laboratory because it offers the potential of very high coefficients, although it has, as yet, no practical application. All the information given here is therefore for filmwise condensation.

Condensation outside horizontal tubes. For a single horizontal tube, the heat-transfer coefficient is given by the Nusselt (1961) relationship:

$$\alpha = 0{\cdot}951 k_L \left\{ \frac{\rho_L(\rho_L - \rho_G)g}{\mu_L \Gamma'} \right\}^{\frac{1}{3}} \tag{35}$$

where Γ' is the mass flowrate of condensate produced per unit length of tube.

In a bundle of tubes, the coefficient of the lower tubes is reduced owing to condensate falling from those higher in the bundle. If there are N tubes in a vertical row, the average coefficient, α_N, is given approximately by the Kern (1956) relationship,

$$\alpha_N = \alpha_{Nu} N^{-\frac{2}{3}}, \tag{36}$$

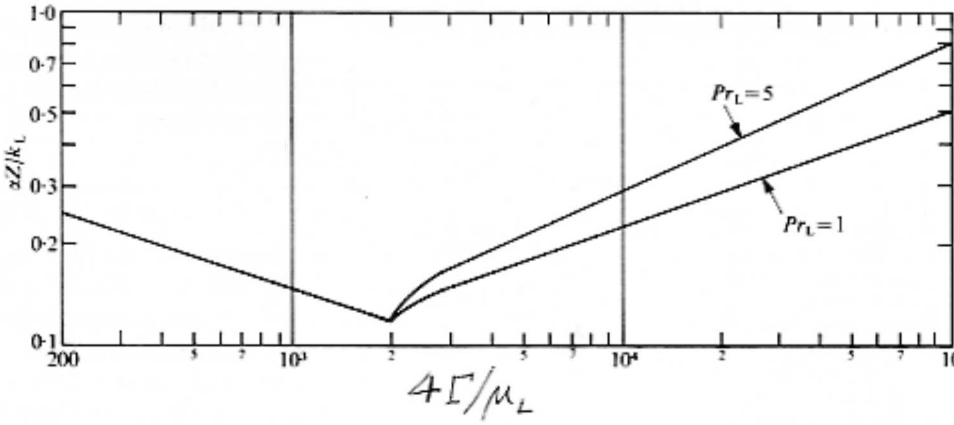
where α_{Nu} is the coefficient given by the Nusselt method in eqn (35). If there are high vapour velocities through the bundle, the coefficients calculated would normally, but not always, be low. Example 8 (p. 38) illustrates the use of eqns (35) and (36).

Condensation outside vertical tubes. The coefficient is given by Fig. 15 where Γ is the flowrate of the condensate produced per unit perimeter of tube. This calculation is illustrated in Example 9 (p. 38). Horizontal or downward vapour velocity can increase the coefficients above those predicted by Fig. 15.

Condensation inside horizontal tubes. Depending on the gas-flow condition, the condensate may (1) run down to the bottom of the tube before flowing out, or (2) form a uniform film (around the tube perimeter) that is dragged along by the vapour flow. In case (1), which is known as *stratifying* flow, the coefficient may be calculated by using eqn (35) and then multiplying the answer obtained by 0·8. In case (2), known as *annular* flow, the coefficient can be estimated using the method of Boyko and Kruzhilin

Fig. 15 Heat-transfer coefficients for condensation on a vertical surface with no vapour shear (Nusselt 1916 and Colburn 1934);

$$Z=\left\{\frac{\mu_L^2}{\rho_L(\rho_L-\rho_G)g}\right\}^{\frac{1}{3}}.$$



(1967) which is

$$\alpha=\frac{\alpha_L}{2}(J_{in}^{\frac{1}{2}}+J_{out}^{\frac{1}{2}}), \tag{37a}$$

where

$$J=1+\left(\frac{\rho_L}{\rho_G}-1\right)x. \tag{37b}$$

x is the quality or vapour mass fraction, J_{in} and J_{out} are the inlet and outlet valves of J, and α_L is the coefficient obtained in the tube for single-phase flow if all the liquid were condensed.

If there is doubt as to which type of flow is occurring in a horizontal tube, both annular and stratifying methods may be computed and the *higher* coefficient selected. This type of calculation is illustrated in Example 10 (p. 38).

Condensation inside vertical tubes. For vertical tubes with vapour downflow the coefficient should be calculated both from Fig. 15 and from eqn (37), and the higher of the two results used. For upflow of vapour, Fig. 15 alone is used but care must be taken to ensure that the tube does not flood, that is that the upward vapour flow does not prevent the condensate draining back down the tube. For low-viscosity liquids, this is ensured by meeting the following condition (Hewitt and Hall Taylor, 1970):

$$V_G^{\frac{1}{2}}\rho_G^{\frac{1}{4}}+V_L^{\frac{1}{2}}\rho_L^{\frac{1}{4}}<0{\cdot}6\{gD_i(\rho_L-\rho_G)\}^{\frac{1}{4}}, \tag{38}$$

where V_G and V_L are the velocities which the liquid and vapour would assume if either were flowing alone in the tube. These velocities must be calculated at the bottom of the tube where each has its highest value.

Problems of incondensible gases or multicomponents. The methods of calculating the condensing heat-transfer coefficient described previously take account only of the thermal resistance of the condensate film. Other thermal resistances can also be important if incondensible gas is present in the vapour or if a mixture of vapours is being condensed. These additional resistances cause a lowering of the calculated value of the coefficients. A simple explanation of a satisfactory way of dealing with this type of problem, which is outside the scope of this Engineering Design Guide, is given by Bell and Ghaly (1973).

It must be appreciated that incondensible gases can have a severe effect on the performance of a condenser and that provision must always be made for removing any such gases by means of a vent line, which exhausts gas–vapour mixture from the cold end of the condenser.

Fouling and scale formation

It is unlikely that heat exchangers in normal operation will remain clean for any length of time, and the dirt which collects on the heat-transfer surface introduces an extra thermal resistance. Resistances for fouling are usually expressed as a thermal resistance, r, rather than as a coefficient (the resistance is the reciprocal of the coefficient (p. 4)). Fouling resistances are often known as fouling factors, which is a somewhat misleading term. Even more misleading is that they are often quoted without units.

As fouling resistances tend to increase with time because of dirt build-up, the design should be made on the basis of the *expected worst fouling* assumed to have been encountered after the exchanger had been running for some time. Fouling resistances are extremely difficult to estimate and are usually obtained from past experience; some typical examples for exchangers with plain tubes are given in Table 4. Lower values of fouling resistance are frequently assumed in plate heat exchangers, and this resistance is often ignored on the air side of air-cooled exchangers of the type described on p. 20. Under clean conditions, the fouling resistances are of course zero.

Overall coefficients

It has been shown (p. 8) that, when transferring heat from a wall of constant temperature T_w to a fluid of constant temperature T, the heat load is given by

$$Q_T=\alpha A_T(T_w-T). \tag{6}$$

This equation is not usually very useful in design since the aim is to transfer heat from a fluid of temperature T_{hot} to a fluid of temperature T_{cold}. To facilitate this calculation we define an *overall* heat-transfer coefficient, U, as follows:

$$Q_T=UA_T(T_{hot}-T_{cold}). \tag{39}$$

Table 4. Typical fouling resistances (TEMA (1968))

Fluid	Fouling resistance (m^2 K/W)
Water	
Sea water 50 °C or less	0·0001
Sea water 50 °C or more	0·0002
Cooling water (treated) 50 °C or less	0·0002
Cooling water (treated) 50 °C or more	0·0003
Cooling water (untreated) 50 °C or less	0·0005
Cooling water (untreated) 50 °C or more	0·0008
Liquids	
Fuel oil	0·0008
Refrigerant liquids	0·0002
Molten heat transfer salts	0·0001
Crude oils	0·0003–0·0010
Gases and vapours	
Steam (oil free)	0·0001
Steam (oil bearing)	0·0002
Refrigerant vapours	0·0003
Compressed air	0·0003
Natural gas	0·0002

This equation must be modified as described on p. 27 when U, T_{hot}, and T_{cold} vary through the exchanger.

For tubular exchangers, A_T is usually taken as the external area of the tube, A_o. For a tube, the overall coefficient is given from the other coefficients and resistances as follows:

$$\frac{1}{U}=\frac{1}{\alpha_o}+r_o+\left(\frac{1}{\alpha_i}+r_i\right)\frac{D_o}{D_i}+\frac{y_w}{k_w}\frac{D_o}{D_w}, \quad (40a)$$

where

$$D_w=\frac{D_o-D_i}{\ln(D_o/D_i)}\simeq\tfrac{1}{2}(D_o+D_i). \quad (40b)$$

α_o and α_i are the coefficients and r_o and r_i the fouling resistances outside and inside the tubes respectively. The last term in eqn (40a) is the tube wall resistance. Eqn (40a) is simply a sum of a series of thermal resistances with a correction for the different areas for heat flow. When the areas on the two sides are the same, as say in plate heat exchangers, eqn (40) takes the following form:

$$\frac{1}{U}=\frac{1}{\alpha_1}+\frac{1}{\alpha_2}+r_1+r_2+\frac{y_w}{k_w}. \quad (41)$$

For finned tubes, eqn (40) can be used as it stands, provided that the tube outside coefficient (together with the fouling resistance) is referred to the tube o.d. as follows:

$$\frac{1}{\alpha_o}+r_o=\frac{1}{\eta}\left(\frac{1}{\alpha_F}+r_F\right)\frac{A_o}{A_F}, \quad (42)$$

where the subscript o means referred to the tube o.d. and the subscript F means referred to the fin area. A_o and A_F are the bare tube external area and the fin area, respectively, and η is the fin effectiveness, which is less than unity and is typically about 0·95 for copper and aluminium fins in normal gas-phase heat-transfer situations. For further details on fin effectiveness, the reader is referred to Kern and Kraus (1972).

Examples 11 and 12 (p. 39) illustrate how to calculate overall coefficients from the stream coefficients and fouling resistances.

Types of heat exchanger

Of the many different heat exchangers, some of the more important types are briefly described in this section. Additional specialized exchangers are reviewed by Murray (1972, 1974). Wherever possible in this Engineering Design Guide, methods are provided for calculating the overall coefficients for each type of equipment. In some instances, this simply involves reference to *Basic concepts in heat transfer*, p. 3 ff., while in others the information given in the following pages concerning the adaptation of these methods to the particular exchanger type is necessary. Where it is not possible to deal with an exchanger type in sufficient detail to estimate design data, references to other sources of information are given.

Double-pipe exchanger

This is one of the simplest designs of heat exchanger and consists of two concentric pipes. One stream flows inside the inner tube while the other flows in the annular space between the tubes. This type of exchanger is illustrated in Fig. 16. Although very simple and cheap to construct, this exchanger suffers from the disadvantage that the amount of space it occupies is generally fairly high compared with other types.

The overall coefficient for the double-pipe exchanger is calculated by eqn (40) where the

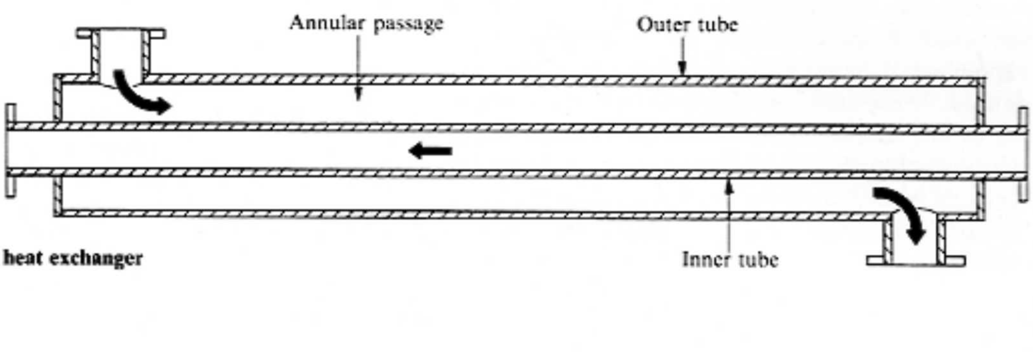


Fig. 16 Double-pipe heat exchanger

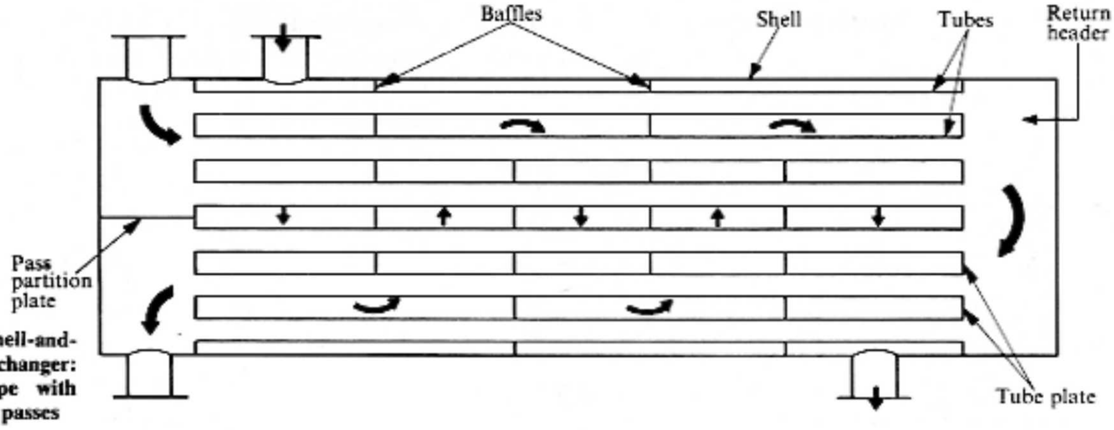


Fig. 17 Shell-and-tube heat exchanger: TEMA E-type with two tube-side passes

methods given under the following headings are used to obtain the inside and outside coefficients.

	Inside tube coefficient:	Outside tube coefficient (inside annulus)
Condensing†		
(i) vertical	*Condensation inside vertical tubes* (p. 15)	*Condensation outside vertical tubes* (p. 14)
(ii) horizontal	*Condensation inside horizontal tubes* (p. 14)	*Condensation outside horizontal tubes* (p. 14)
	Inside and outside tube coefficients	
Single phase	*Inside uniform cross-sectional channels* (p. 4)	
Boiling	*Convective boiling* (p. 13)	

Shell-and-tube exchanger

This type is very commonly used in chemical process applications. The exchanger is illustrated in Fig. 17 and consists of a large cylinder or shell, inside which there is a bundle of tubes. One fluid stream flows on the outside of the tubes (or on the shell side) while the other flows inside the tubes (or on the tube side). The detailed mechanical features of shell-and-tube exchangers are not discussed here but are dealt with in TEMA (1968).

† Incondensible gases or multicomponent effects are ignored (see p. 15).

The tube-side stream can be made to flow once through the bundle through a single set of tubes, or can be diverted back and forth across the bundle through different sets of tubes. In the exchanger shown in Fig. 17—known as a two-tube-pass exchanger—the stream flows the length of the exchanger twice; the fluid leaving the first pass enters a special chamber known as a header before entering the second pass. Instead of a header, U-tubes may be used as illustrated in Fig. 18. The headers at both ends of the exchanger can have partitions which give rise to multiple passes. Typical arrangements of passes are shown in Fig. 19.

The flow of the shell-side fluid stream can be made to follow a simple or complicated path through the exchanger in order to cope with different heat-transfer problems. The simplest type of shell-side flow occurs in an *E-type* exchanger as designated by TEMA. In this type, the flow enters one end of the shell and leaves at the other as shown in Figs 17 and 18. More complicated shell-flow types are shown in Fig. 20. The E type would normally be preferred in design to the other types because of its cheapness and greater simplicity. It is the most usual type for single phase on the shell side. For condensing on the shell side, the *J-type* exchanger is frequently encountered since it has some advantage when dealing with large volumes of vapour. The *K-type* exchanger is known as a kettle boiler and is specifically for pool-boiling applications (for the calculation of the shell-side coefficient in pool boiling see p. 10).

The shell-side flow path in the exchanger is frequently deflected by a series of cross baffles whose

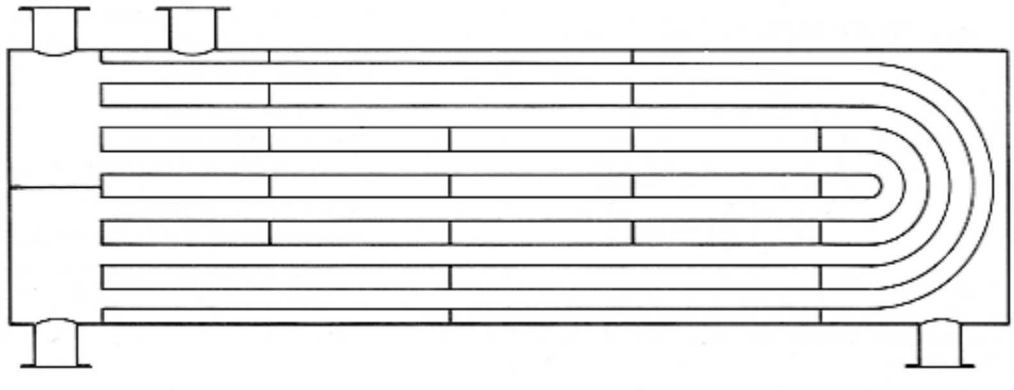
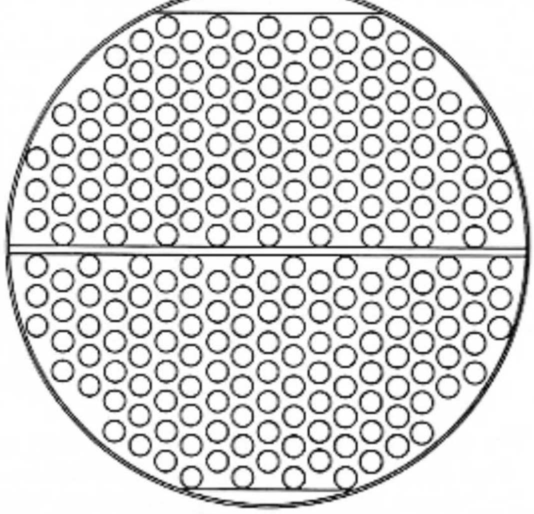
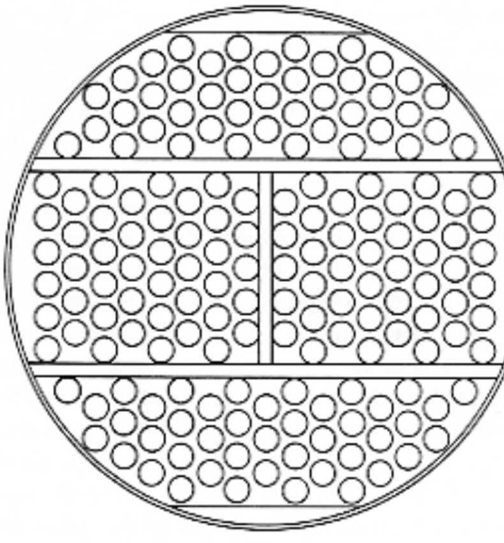


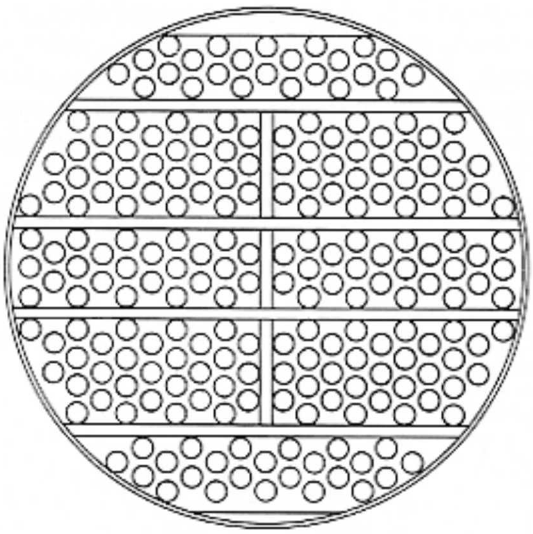
Fig. 18 Shell-and-tube heat exchanger with U-tubes: TEMA E-type with two tube-side passes



(a)



(b)



(c)

Fig. 19 Various arrangements for different numbers of tube-side passes in shell-and-tube exchangers: (a) two pass, (b) four pass, (c) eight pass

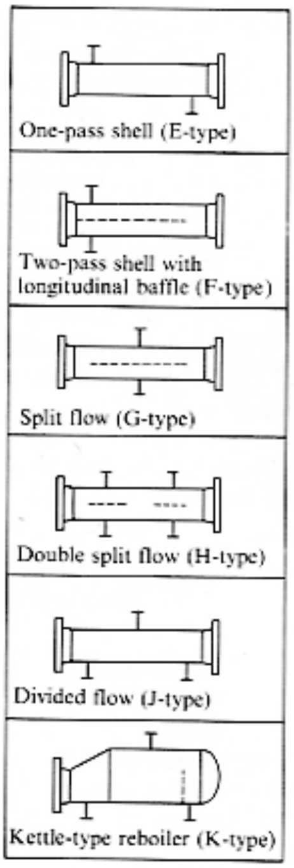


Fig. 20 Various shell types with type letters defined by TEMA (1968)

purpose is to direct the flow in the bundle at right angles to the tubes and also to provide support for them. The positioning of these baffles for a single-phase application is shown in Fig. 17. Two types of baffle are most frequently met with: (1) single segmental, and (2) double segmental. Both types are illustrated in Fig. 21, which also shows disc-and-doughnut baffles—a type frequently encountered in the literature, but not very often used. The term *per cent baffle cut*, which is often used to describe the design of baffles, is the height H_{cut} in Fig. 21 given as a percentage of the baffle diameter. The direction of the baffle cut is usually horizontal, but in shell-side condensers it is vertical to allow the condensate to drop out of the bundle.

Although the function of the cross baffles is to induce cross flow, they are not absolutely efficient in this respect on account of various gaps which must be left in order that the exchanger can be constructed. For example, the holes in the baffles through which the tubes pass must be slightly larger than the tube o.d. and the baffle diameter must be less than the shell i.d. This allows some of the stream flow to leak through the gaps. Further, the tube bundle does not completely fill the shell and hence some of the stream can bypass the bundle by flowing around the edge. In addition the flow has to turn round after it has passed between the baffles and it thus proceeds parallel to the tube as it flows through the baffle windows. The zones and leakage gaps in a baffle space are shown in Fig. 22.

The overall coefficient in a shell-and-tube exchanger is calculated from eqn (40) where the shell- and tube-side coefficients required in this equation are calculated by the methods described previously. The appropriate headings and pages are as follows:

	To calculate shell-side coefficient:	To calculate tube-side coefficient:
Single phase		
baffled†	*Flow across banks of smooth tubes* (p. 7)	*Inside uniform cross-sectional channels* (p. 4)
unbaffled	*Inside uniform cross-sectional channels* (p. 4)	*Inside uniform cross-sectional channels* (p. 4)
Boiling		
baffled†	*Convective boiling* (p. 13)	*Convective boiling* (p. 13)
kettle	*Pool boiling* (p. 10)	(not relevant)
Condensing‡		
vertical	*Condensation outside vertical tubes* (p. 14)	*Condensation inside vertical tubes* (p. 15)
horizontal	*Condensation outside horizontal tubes* (p. 14)	*Condensation inside horizontal tubes* (p. 14)

Because of the possibility of fluid evading the bundle, it is necessary to correct the single-phase cross-flow coefficients given under the heading *Flow across banks of smooth tubes* (p. 7) by the methods described here. These methods apply to the most common types of baffle, which are single segmental. The ideal cross-flow heat-transfer coefficient, α_I, is first calculated using eqn (22); assuming that all the fluid crosses the bundle, the area S_m required in using this equation is taken as that near the equator of the bundle. This coefficient can then be corrected for bypass, leakage and window effects by the methods proposed by Bell (1963):

$$\alpha_o = \alpha_I F_B F_L F_W, \qquad (43)$$

where F_B, F_L, and F_W are the bypass, leakage, and window correction factors respectively. These correction factors are given in Fig. 23. In this figure, S_B is the bypass flow area and S_L the total leakage flow area given by

$$S_L = S_{tb} + S_{sb}, \qquad (44)$$

where S_{tb} is the tube-to-baffle leakage flow area and S_{sb} the shell-to-baffle leakage flow area for a single baffle. These areas are illustrated in Fig. 22. R_W is the ratio of the number of tubes in both window zones to the total number of tubes. Special strips, known as sealing strips, may be placed in the bypass region to reduce the bypass flow, thus bringing F_B closer to unity than is shown in Fig. 23.

For shell-side convective-boiling situations, the convective component of the coefficient, α_{con}, must be corrected for leakage and bypass effects, but the nucleate-boiling component is not corrected. Typical stream heat-transfer coefficients for tubular exchangers are given in Table 5.

Plate exchanger

In this type of exchanger the two fluid streams are separated by plates which are held together by placing them in a press or frame and sealing them around the edge by means of gaskets. The operation

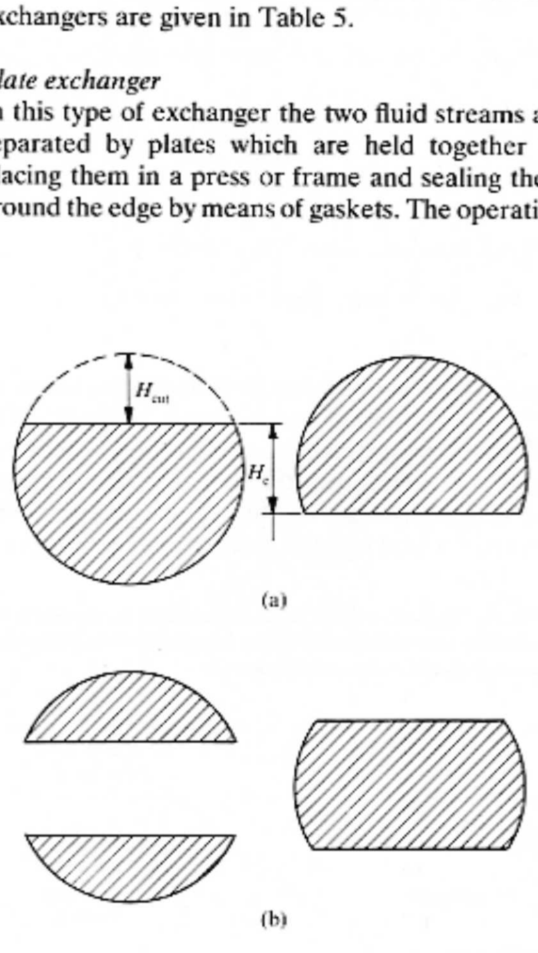


Fig. 21 Baffle types in shell-and-tube exchangers (the baffles in the left-hand column alternate along the shell with those in the right-hand column): (a) single segmental, (b) double segmental, (c) disc and doughnut

†The shell-side coefficient must be corrected for bypass and leakage effects.

‡Incondensible gases and multicomponent effects are ignored (see p. 15).

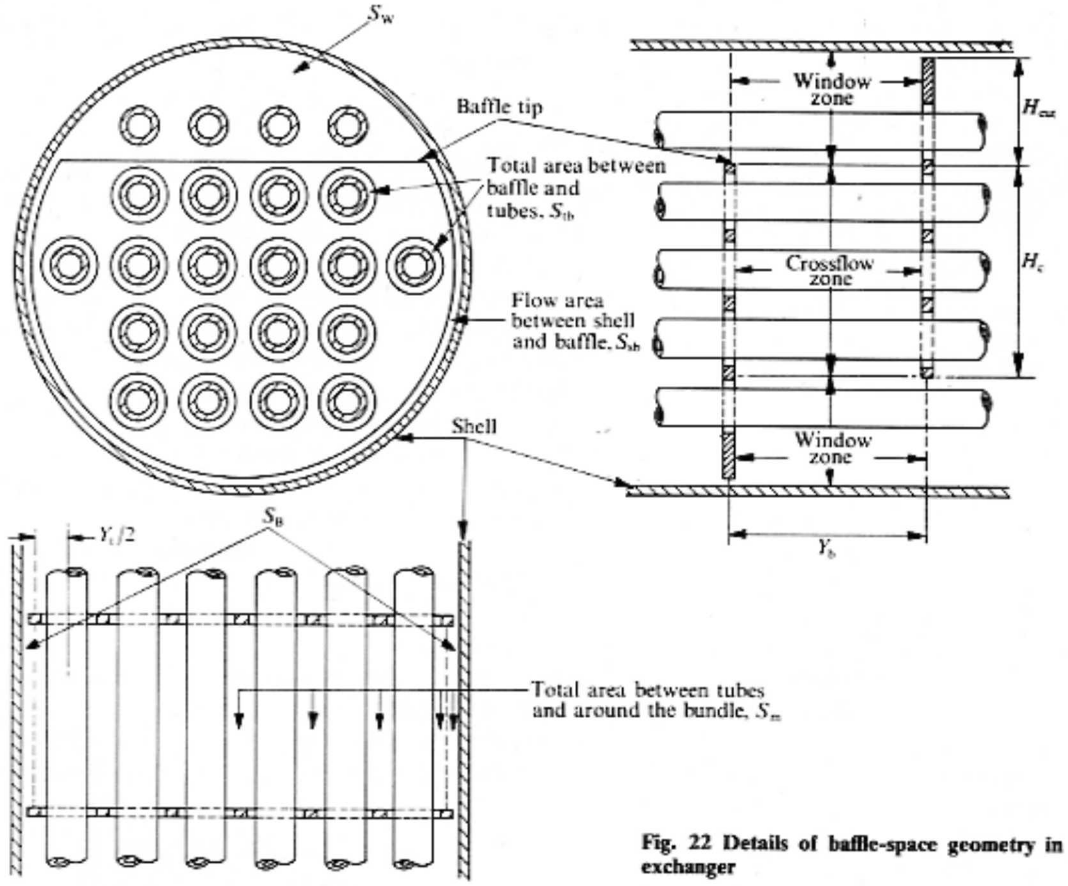


Fig. 22 Details of baffle-space geometry in a shell-and-tube exchanger

of this type of plate exchanger is illustrated in Fig. 24 and a photograph of one is shown in Fig. 25. The plates are corrugated in various patterns as shown in Fig. 9, both to improve the heat-transfer coefficient and to increase the rigidity of the plates. Some remarks on heat transfer in these plate passages are given under the heading *Flow between special plates* p. 10. The overall coefficient is calculated from the coefficients on each side by using eqn 41.

The combination of the narrow gap and the turbulence-inducing corrugated surfaces in plate heat exchangers gives rise to high coefficients at low liquid velocities and moderate pressure losses. Overall coefficients can be as high as two to three times those found in tubular exchangers for the same duty, and with some plates turbulence can be achieved at Reynolds numbers as low as 15. In consequence, plate heat exchangers are suitable for use with high-viscosity liquids, and have the additional advantage that they can be easily opened up for cleaning and inspection. One disadvantage with plate heat exchangers is that the pressure loadings which the plates can stand, combined with difficulties with the gasket, normally limit these exchangers to pressures below 25 bar and temperatures below 250 °C. The choice of gasket material becomes more limited at high temperatures, which therefore restricts the number of different fluids that can be handled under these conditions.

Table 5. Typical values of heat-transfer coefficients for the inside and outside of tubes (applicable to shell-and-tube and double pipe exchangers)

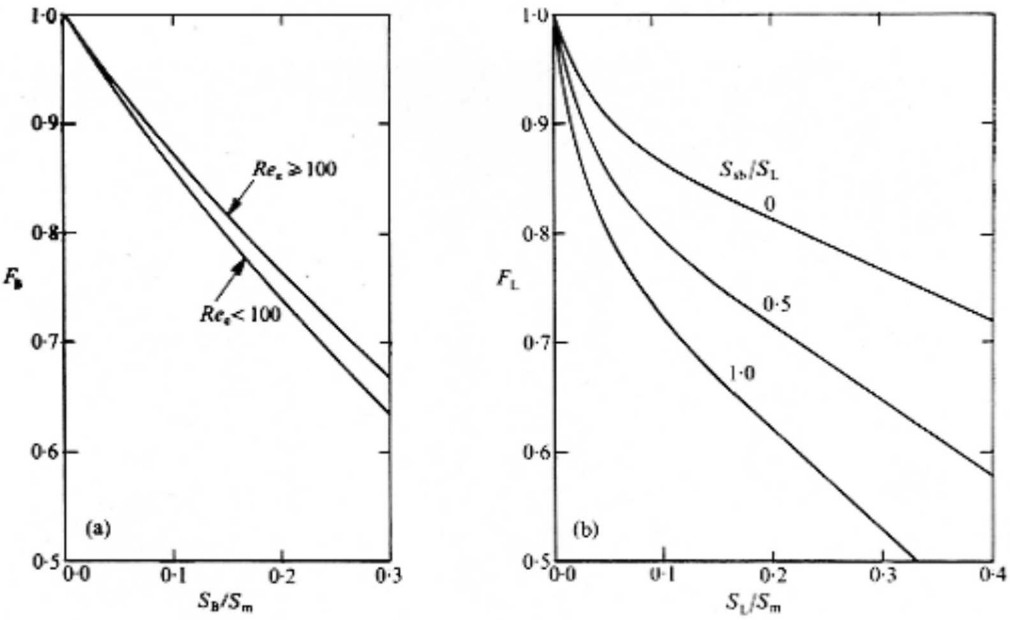
	Coefficient (W/m^2 K)
Single phase	
Water	1000–8000
Gases	10–300
Organic solvents	300–3000
Oils	50–800
Condensing†	
Steam	5000–15 000
Organic solvents	900–3000
Light oils	1000–2500
Heavy oils	100–300
Ammonia	3000–6000
Boiling‡	
Water	4000–12 000
Organic solvents	500–3000
Ammonia	1000–2500
Light oils	900–2000
Heavy oils	60–300

†These coefficients do not apply where there are incondensible gases present. Where incondensible gases are present, the coefficients can be much lower.

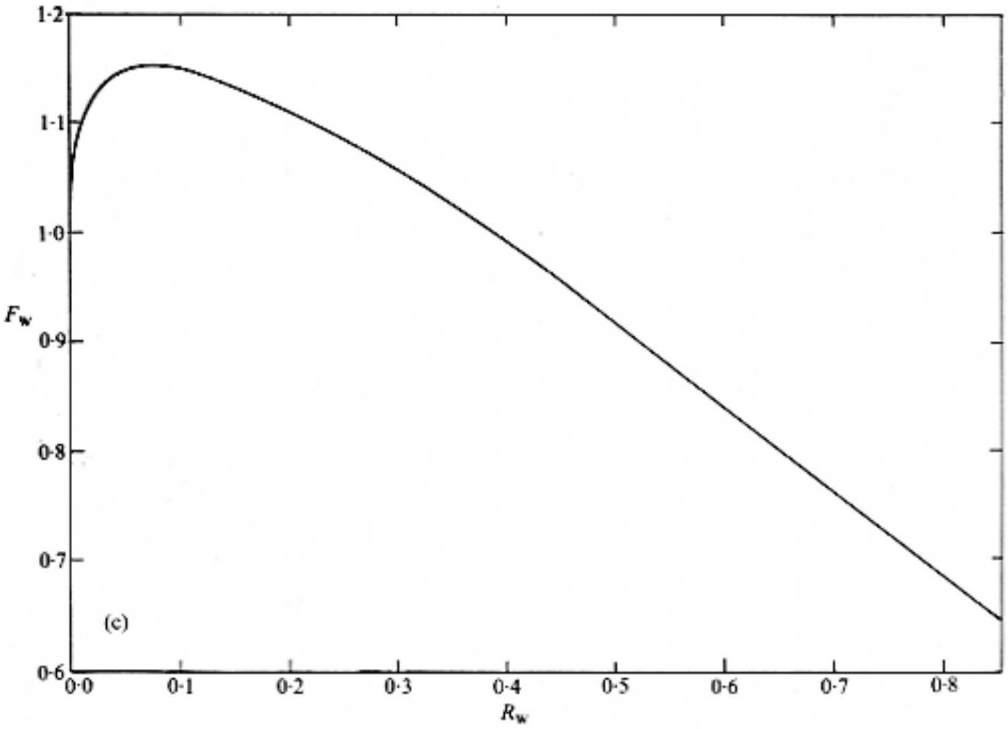
‡For convective boiling, coefficients can be obtained which are higher than those given here.

Air-cooled, finned-tube exchangers and radiators

This type consists of a rectangular bundle of tubes which are usually a few rows in depth. Hot fluid in each tube is either cooled or condensed by blowing or sucking air across the bundle by means of a large fan or fans. Because the air-heat-transfer coefficients are very low, it is usual to use finning outside the tubes in order to increase the heat-transfer



(a) (b)



(c)

Fig. 23 Bell (1963) correction factors for shell-side heat-transfer coefficients: (a) bypass, (b) leakage, (c) window

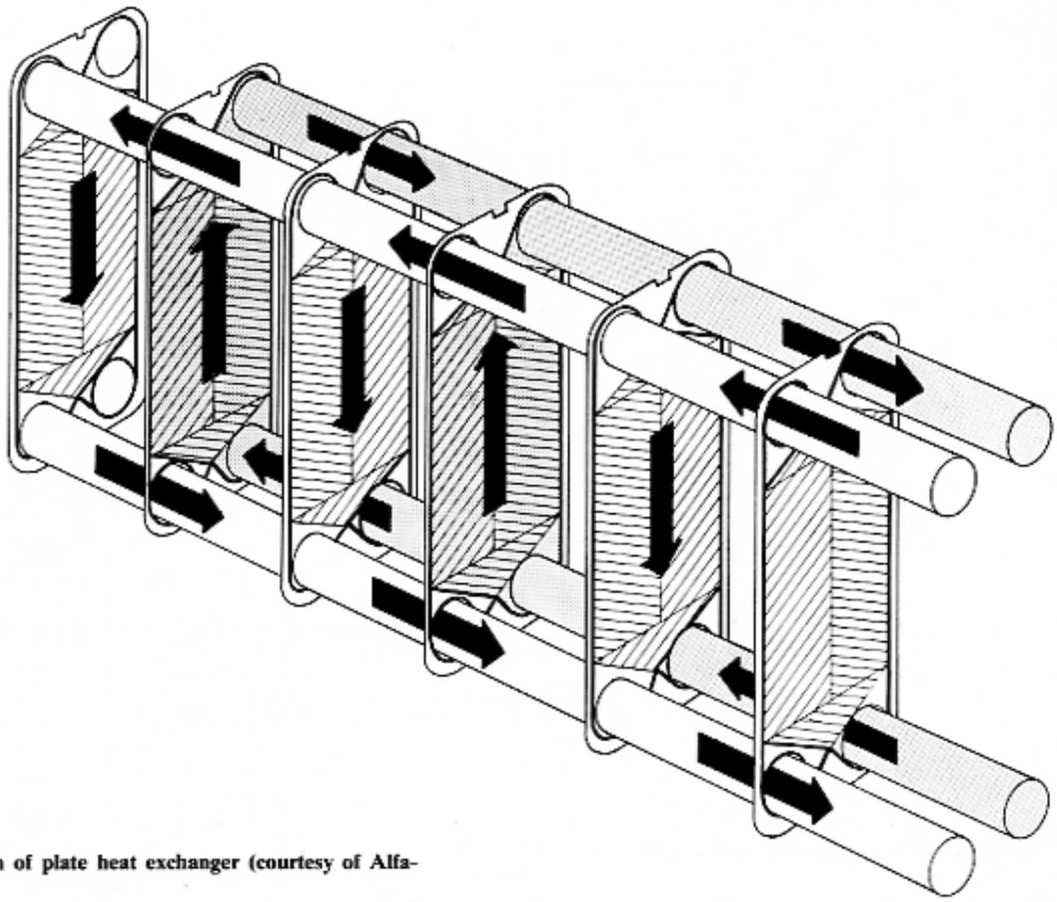


Fig. 24 Operation of plate heat exchanger (courtesy of Alfa-Laval Co. Ltd.)

surface area on the air side. The tube rows are normally staggered in order to increase the air-side coefficients.

A typical air-cooled exchanger is illustrated in Fig. 26. The bundle here consists of four tube rows and these are arranged with two tube passes, but a variety of tube-pass arrangements can be obtained by changing the header layout. The exchanger illustrated is a forced-draft unit since the air is blown across the bundle by means of the two fans. In an induced-draft exchanger, the fans are placed above the bundle and the air sucked upwards. The air is almost invariably passed upwards to avoid the recirculation of hot air. A photograph of a typical air-cooled heat exchanger is given in Fig. 27.

The overall coefficient in air-cooled exchangers is calculated using eqn (40), after correcting the fin-surface coefficient to the bare tube o.d. using eqn (42). The fin-surface coefficient can be estimated approximately using the method described under the heading *Flow over finned tubes* p. 8. Information on calculating the tube-side coefficients is given under the headings *Inside uniform cross-sectional channels* (p. 4) for single phase, and *Condensation inside horizontal tubes* (p. 14) for condensation of a single vapour.

Air-cooled exchangers are used where waste heat must be rejected under circumstances where little cooling water is available. Their disadvantage is that the fans are rather noisy and thus there may be difficulties if they are sited close to residential areas. When designing this type of exchanger, care must be taken that the exchanger will still fulfil the required duty on hot days, and that the fluid in the tubes does not freeze in cold weather.

A smaller version of the air-cooled heat exchanger is the type of radiator used in air-conditioning units and vehicle cooling systems. The variability of finning arrangements in such units is even greater than in the larger air-cooled units discussed previously. It is therefore impossible to provide equations for predicting their gas-side coefficients in this Engineering Design Guide. However, these coefficients will usually be similar to those obtained in air-cooled exchangers generally, for example 20–120 $W/m^2 K$. These are the coefficients on the surface of the fin which must be corrected to the tube o.d. using eqn (42) before calculating the overall coefficients from eqn (40).

Agitated vessels

Agitated vessels are used for the batchwise heating or cooling of liquids which may, in addition, be undergoing chemical reaction. Each consists of a vessel in which the process liquid is stirred or agitated by means of some type of impeller, such as the paddle type shown in Fig. 28. Other impellers used in different applications are shown in Fig. 29. In the vessel shown in Fig. 28, vertical baffles are included so that the fluid in the vessel is mixed rather than

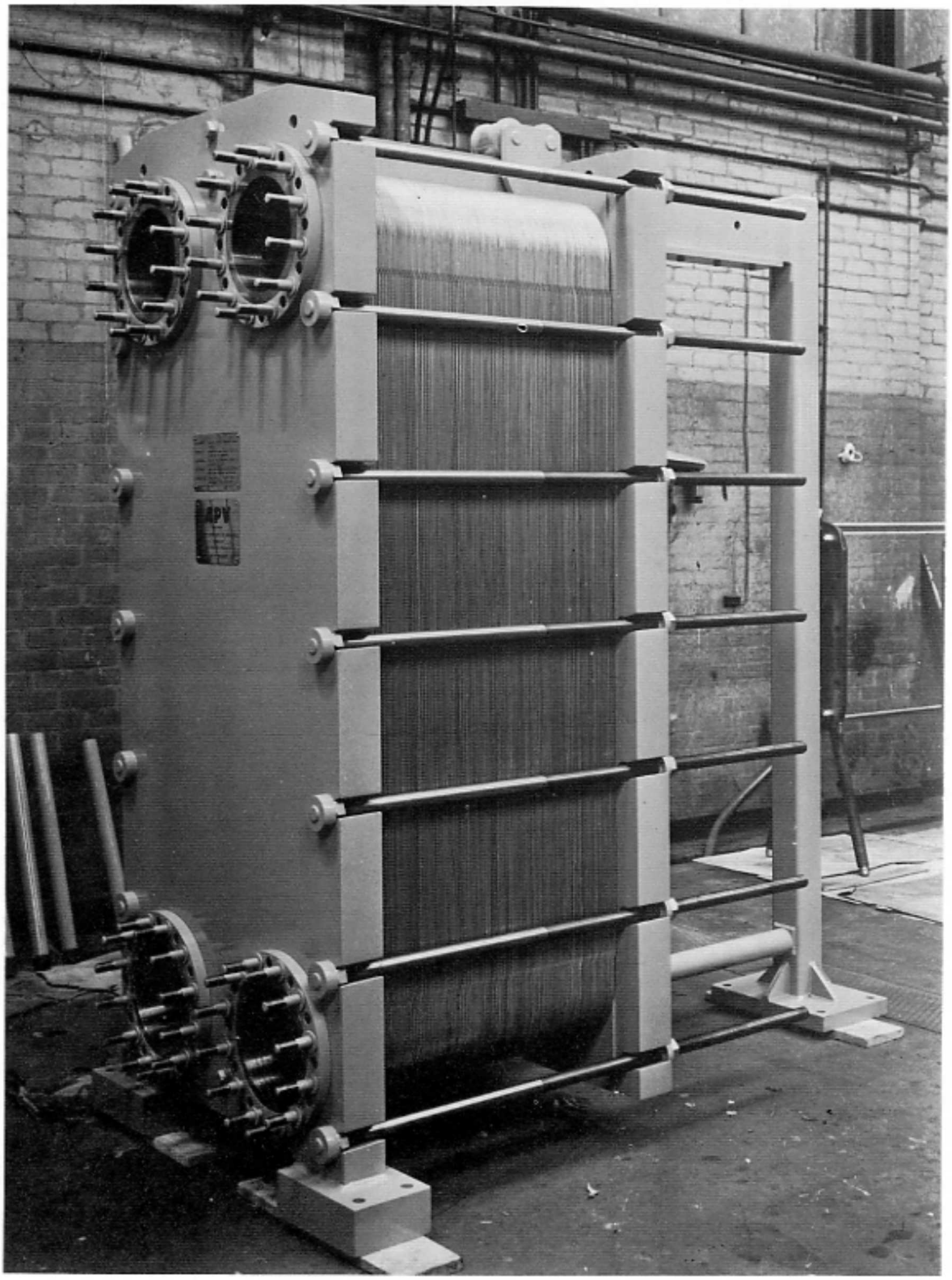


Fig. 25 Plate heat exchanger (courtesy of A.P.V. Co. Ltd.)

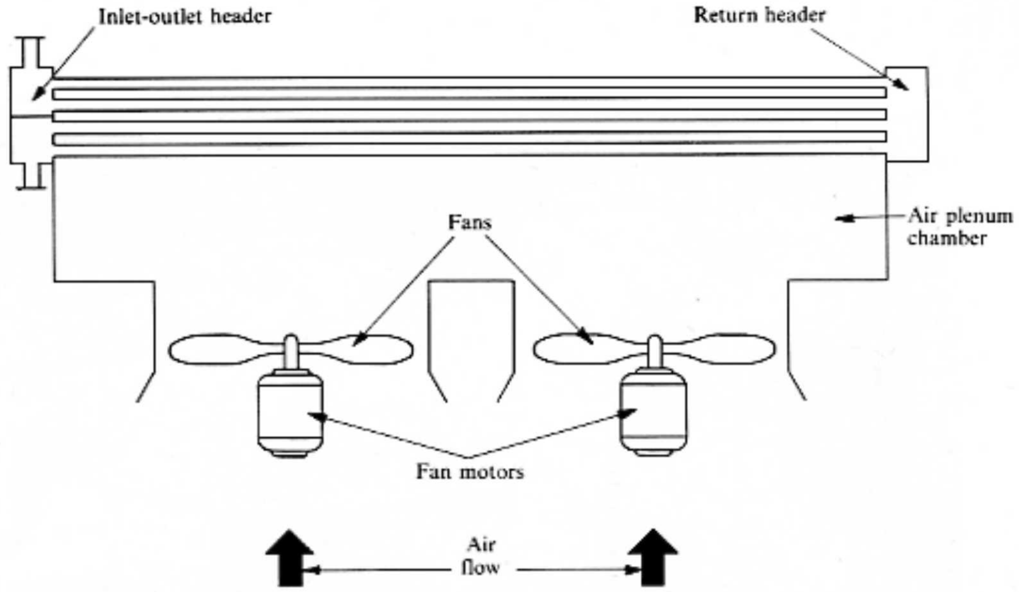


Fig. 26 Operation of air-cooled heat exchanger

simply rotated. Heating (process steam) or cooling (cooling water) is achieved by the use of a jacket; alternative methods are to include a coil inside the vessel or a half-tube welded on its outside.

It is beyond the scope of this Engineering Design Guide to deal in detail with the calculation of heat-transfer coefficients in this type of equipment. A useful review of this subject from which some design information can be obtained is that of Edwards and Wilkinson (1972).

Direct-contact heat exchangers

In the types of heat exchangers discussed previously, there are dividing walls between the two fluid streams. In certain situations, however, it is convenient to remove the intervening wall, thus eliminating the thermal resistance both of the wall itself and of any fouling layers which might have formed on it. An important advantage of these direct-contact heat exchangers is that they are usually of extremely simple construction and are therefore cheap to build; out of the many different types three are described in the following paragraphs.

The spray condenser shown in Fig. 30 is often very useful for low-pressure applications (say less than 40 mbar) since, if a shell-and-tube condenser were used, it would be necessary to leave many tubes out of the bundle (especially those close to the vapour inlet nozzle) to avoid erosion from the low-pressure, high-velocity vapour. In a spray condenser, the spray nozzles are often angled to give good coverage of the vessel volume as illustrated in Fig. 30, which also shows that if an adequate supply of the spray liquid is not available, it can be obtained by passing some of the liquid through a secondary exchanger. If dirty fluids are being used, there is a tendency for the spray nozzles to block. In these circumstances, a baffle column may be used as illustrated in Fig. 31.

Although it is true that direct-contact condensation can be achieved by passing vapour into a pool of liquid, there are certain dangers caused by the large volumes of vapour injected into a cold liquid collapsing violently and causing damage to the vessel.

Another commonly used type of direct-contact exchanger is a gas cooler or vapour desuperheater (Fig. 32), in which cold liquid is sprayed into the hot-gas stream. The heat removed from the gas goes into both heating and evaporation of the spray droplets. Direct-contact heat exchangers are reviewed by Fair (1972a, 1972b) and these reviews give simple design equations.

Heat pipes

Heat pipes constitute a specific type of heat-transfer device different in nature from the equipment described previously, which is concerned with transferring heat from one *fluid* to another. Heat pipes, on the other hand, are concerned with transferring heat from one *place* to another.

The operation of a heat pipe is illustrated in Fig. 33. Liquid is vaporized at the hot end of the heat pipe and the vapour then flows towards the cold end, where it condenses before flowing back to the hot end by means of capillary action in the wick, which consists of, say, a wire mesh. A variety of working fluids at various pressures may be used in a heat pipe in order to achieve a wide range of working temperatures. The effective thermal conductivity of a heat pipe is typically 500 times that of an equivalent copper pipe. Further information on heat pipes can be found in Chisholm (1971).

A pipe in free air

When a hot fluid flows in a pipe, heat will be lost to the atmosphere in spite of any precautions taken. Although such heat losses may occasionally represent a beneficial way of losing waste heat, in most

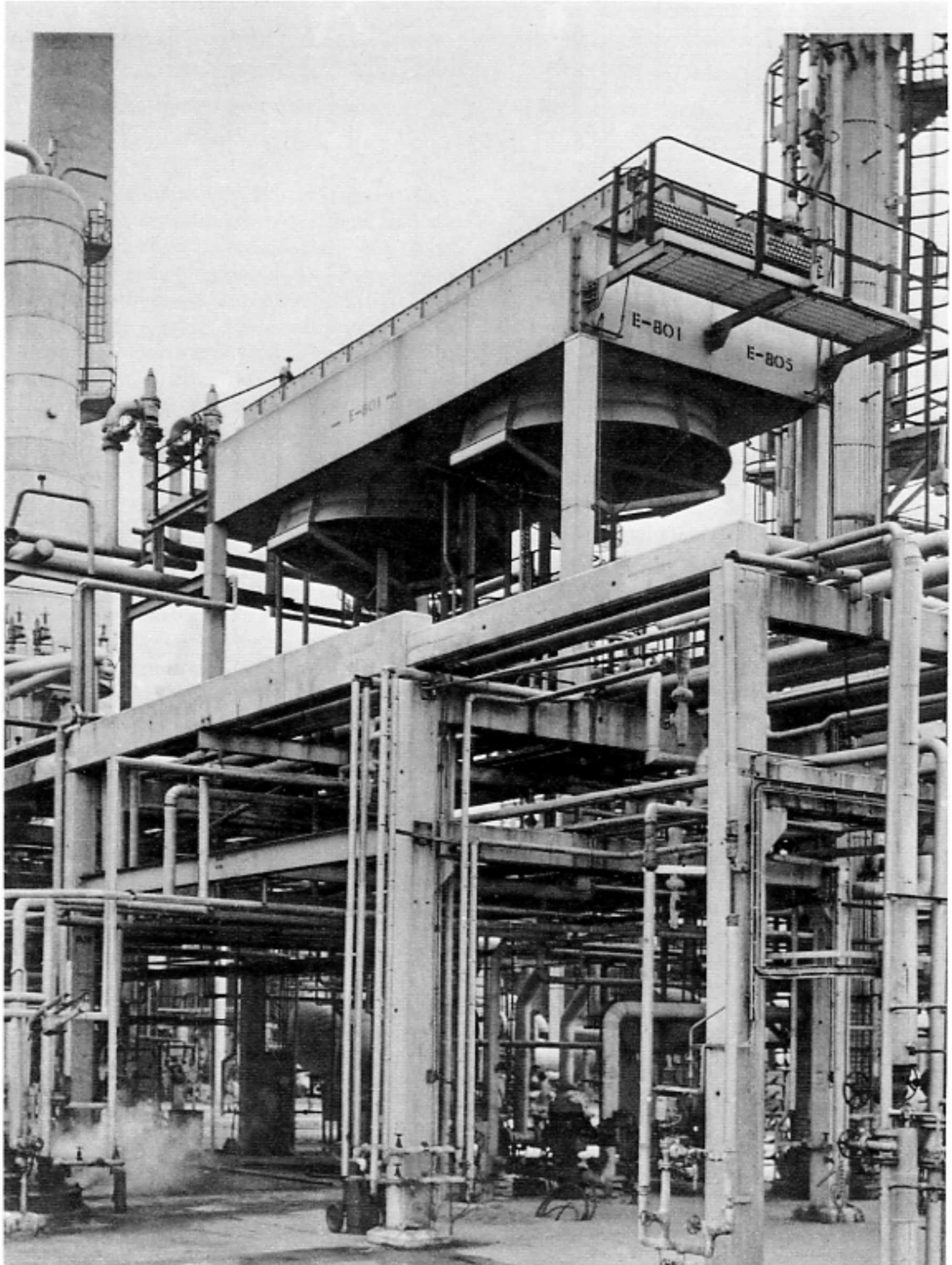


Fig. 27 Typical air-cooled heat exchanger (courtesy of G.E.A. Airexchangers Ltd. and Shell U.K. Ltd.)

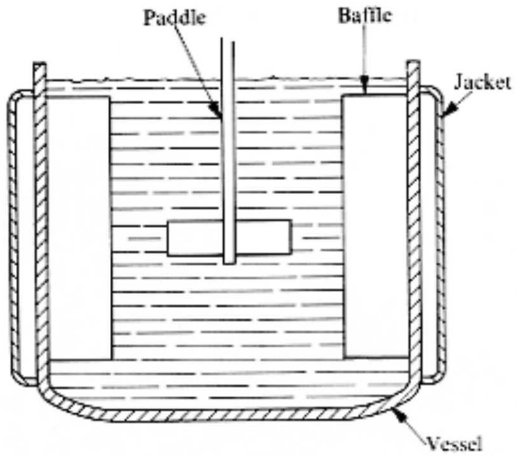


Fig. 28 Agitated vessel

cases, it is desirable to minimize them. The overall coefficient for this situation may be calculated by eqn (40), where the outside coefficient can be calculated using the method described under the heading *Natural convection on the outside of a horizontal tube* (p. 6). Usually, in these circumstances, the overall coefficient is almost identical to the outside coefficient, which is very small compared with the other coefficients.

When it is desired to avoid heat losses from a pipe, it is usual to lag or insulate the pipe. For lagged pipes, the overall coefficient (referred to the tube o.d.) is calculated from

$$\frac{1}{U}=\frac{1}{\alpha_l}\frac{D_o}{D_l}+\left(\frac{y_l}{k_l}\right)\frac{2D_o}{D_o+D_l}, \tag{45}$$

where α_l is the heat-transfer coefficient on the surface of the lagging, D_o the tube o.d., D_l the o.d. of the lagging, y_l the lagging thickness, and k_l the

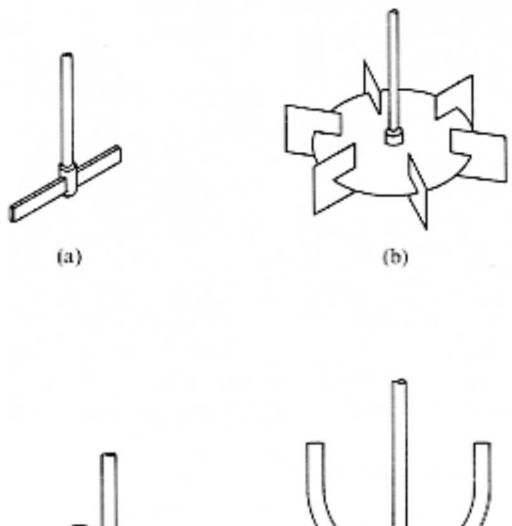


Fig. 29 Various types of agitators (impellers) in agitated vessels: (a) paddle, (b) turbine, (c) propeller, (d) anchor

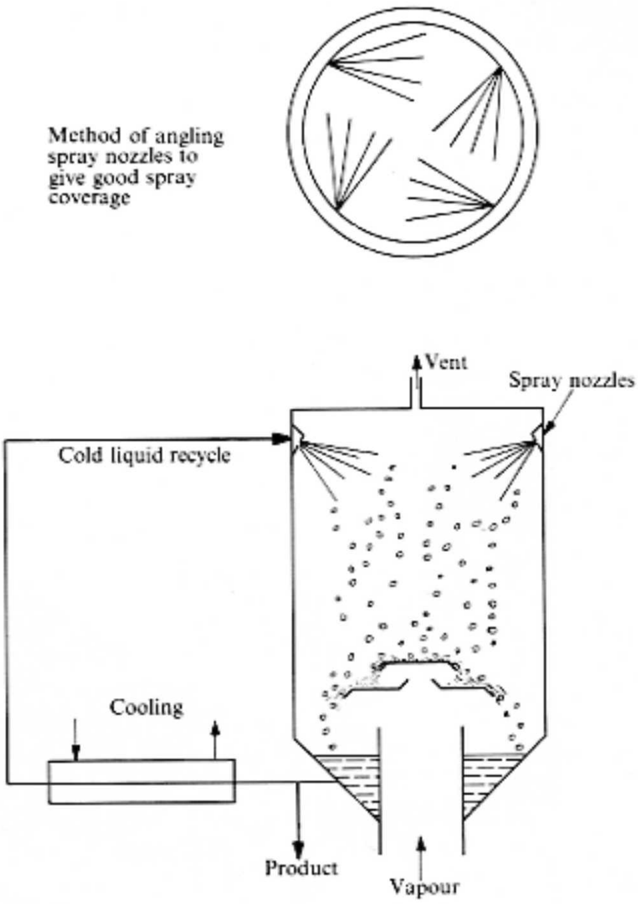


Fig. 30 Spray condenser showing method of angling the spray nozzles to give good spray coverage, and recycling of the product

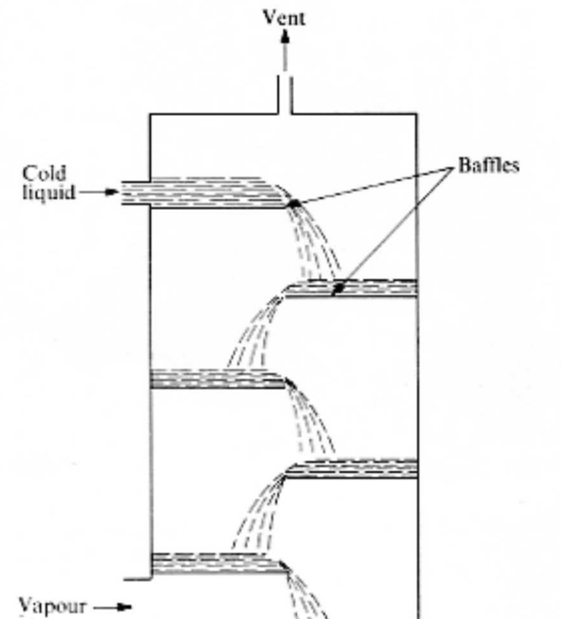


Fig. 31 Baffle column as direct-contact condenser

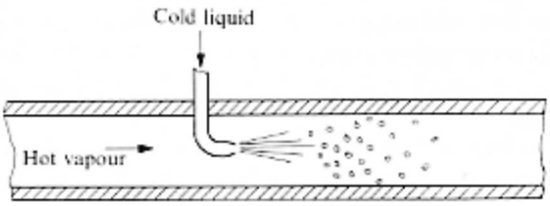


Fig. 32 Co-current vapour desuperheater

lagging thermal conductivity. Eqn (45) ignores fouling resistances, the tube-wall resistance, and the coefficient inside the tube because these resistances will usually be negligible compared with the lagging resistance and the resistance representing the coefficient outside the lagging. The coefficient α_l can be calculated for a horizontal pipe using the method described on p. 6 with the characteristic-length term being taken as the lagging o.d., D_l. Some values of thermal conductivities for lagging materials are given in Table 6.

The elements of design

The first step in design is to decide on the type of equipment to be used and the material of construction. The latter is governed by the fluid to be handled and its temperature. The type of equipment is mainly dictated by the duty (for example, boiling, condensing, cooling, etc.) and the pressure. However, it is often found that many different kinds of heat exchanger will deal with the same duty. In these circumstances, detailed designs of all the relevant types must be carried out and their advantages and disadvantages considered carefully before the final choice is made. The costs of the different types are obviously a major consideration. Some organizations or industries may traditionally use a particular type of equipment, and there may thus be some grounds for staying with a familiar type rather than changing to a slightly cheaper one with which the plant operators are unfamiliar.

Once the type of heat exchanger has been chosen, the problem is to select the details of the equipment, such as tube length, number of tubes, etc., which will meet the required heat duty, Q_T. If the overall heat-transfer coefficient, U, the hot-stream temperature, and the cold-stream temperature are constant, the heat load which the exchanger will achieve is given by eqn (39) which is restated here:

$$Q_T = UA_T(T_{hot} - T_{cold}). \qquad (39)$$

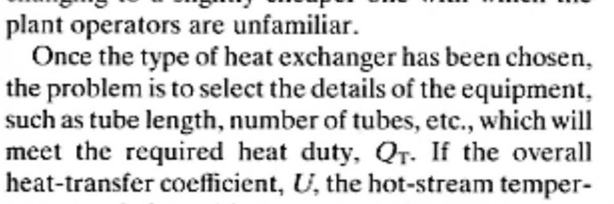


Fig. 33 Heat pipe

Table 6. Thermal conductivities of some insulating materials (W/m K)

Substance	Temperature (°C) 0	100	200
Polystyrene foam	0·003		
Pulverized cork	0·036	0·046	
Silk	0·052		
Cotton	0·057		
Laminated asbestos felt		0·088	0·10
Burned infusorial earth pipe covering (density 200 kg/m^3)	0·074	0·092	0·11
Asbestos (density 580 kg/m^3)	0·16	0·19	0·21

It is, however, most unusual that U, T_{hot}, and T_{cold} will remain constant through the exchanger and hence a modified form of eqn (39) is used in design as follows:

$$Q_T = \bar{U}A_T\theta, \qquad (46)$$

where $\bar{U}$ is the *mean* coefficient for the exchanger and θ is its mean temperature difference (MTD). Both these quantities must be averaged in a very special way, as described under the headings *The counter-flow exchanger* (p. 28) and *Temperature-correction factors for non-counter-current flow* (p. 31).

The first step in solving eqn (46) is to evaluate θ as it is not usually necessary to consider the details of the geometry in order to do this. Then a shape and size of exchanger must be guessed and, from this, $\bar{U}$ and A_T calculated. Eqn (46) can then be solved for Q_T, and the predicted and desired heat loads compared. Different geometries can be tried in a systematic way until one is found which meets the duty.

In the choice of detailed geometry, care has to be taken that mechanical constraints are met and that the equipment will fit into the space available. (This may, for example, limit the length of the exchanger.) An example of a mechanical constraint in the design of baffled shell-and-tube exchangers is that the baffles must not be so far apart that the tubes are not adequately supported. In the case of shell-and-tube exchangers, many of the mechanical constraints have been standardized (for example TEMA (1968)).

Usually, the design must avoid excessive pressure drop in the two fluid streams. Some brief information on pressure-drop calculation is given on p. 31.

Temperature–enthalpy curves

Before calculating the MTD-value it is first necessary to derive the temperature–enthalpy curve for each stream. The enthalpy of a fluid is the heat content per unit mass. For a liquid with constant specific heat, C_L, the enthalpy H is given by

$$H = C_L(T - T_R), \qquad (47)$$

where T_R is an arbitrary reference temperature which is often taken as 0 °C. The enthalpies of a number of important fluids have been tabulated as a function of temperature (and pressure), for example, water and refrigerant fluids (Mayhew and Rogers (1972)).

For single-phase situations, the temperature–enthalpy curves are nearly linear, as indicated by eqn (47). For systems with phase change (boiling or condensation) the curve is more complicated, as illustrated in Fig. 34. Consider a liquid being heated up, boiled, and then superheated: the curve in the liquid-heating region is given by eqn (47) until the boiling or saturation temperature, T_s, for that pressure is obtained; the curve is then flat over the vaporizing region since the enthalpy increases without any change in temperature (the width of this flat region is the latent heat of vaporization, as shown); at position H_2, all the liquid has been vaporized, and as the heat input is increased, the temperature rises again as shown in Fig. 34. This final part of the curve is given by

$$H = H_2 + C_G(T - T_s), \quad (48)$$

provided that the gas (vapour) specific heat, C_G, is constant. For a condensing situation, the curve is the same shape but it is followed from right to left rather than left to right, and the regions are known as vapour desuperheating, condensing, and liquid subcooling. Fig. 34 applies to a single component stream. If there is a mixture of components in the stream, the vaporizing and condensing regions are not flat, and calculating the temperature–enthalpy curves is rather complex. Kern (1958) shows how such a curve is calculated for a mixture of hydrocarbons.

When a fluid stream is passed through a heat exchanger, its enthalpy is changed from H_{in} to H_{out}. Depending on the application, H_{in} and H_{out} can be on any part of the stream temperature–enthalpy curve. The heat load for the hot stream of the exchanger is given by

$$Q_T = W(H_{in} - H_{out}), \quad (49)$$

where W is the mass flow for the stream. For the cold stream, the heat load is given by

$$Q_T = W(H_{out} - H_{in}). \quad (50)$$

These two heat loads must, of course, be the same before the design can proceed.

The counter-flow exchanger

The double-pipe exchanger shown in Fig. 35, in which the two fluid streams are flowing counter to one another is known as a *counter-flow* exchanger. It is desirable to achieve counter flow as this gives the highest value of mean temperature difference, θ. Fig. 36 shows how this situation can be achieved in E- and F-type shell-and-tube exchangers. A counter-flow plate exchanger is shown in Fig. 24. Counter flow cannot be obtained in air-cooled heat exchangers of the type described on p. 20 and it is difficult to obtain in direct-contact exchangers. The concept does not apply to agitated vessels.

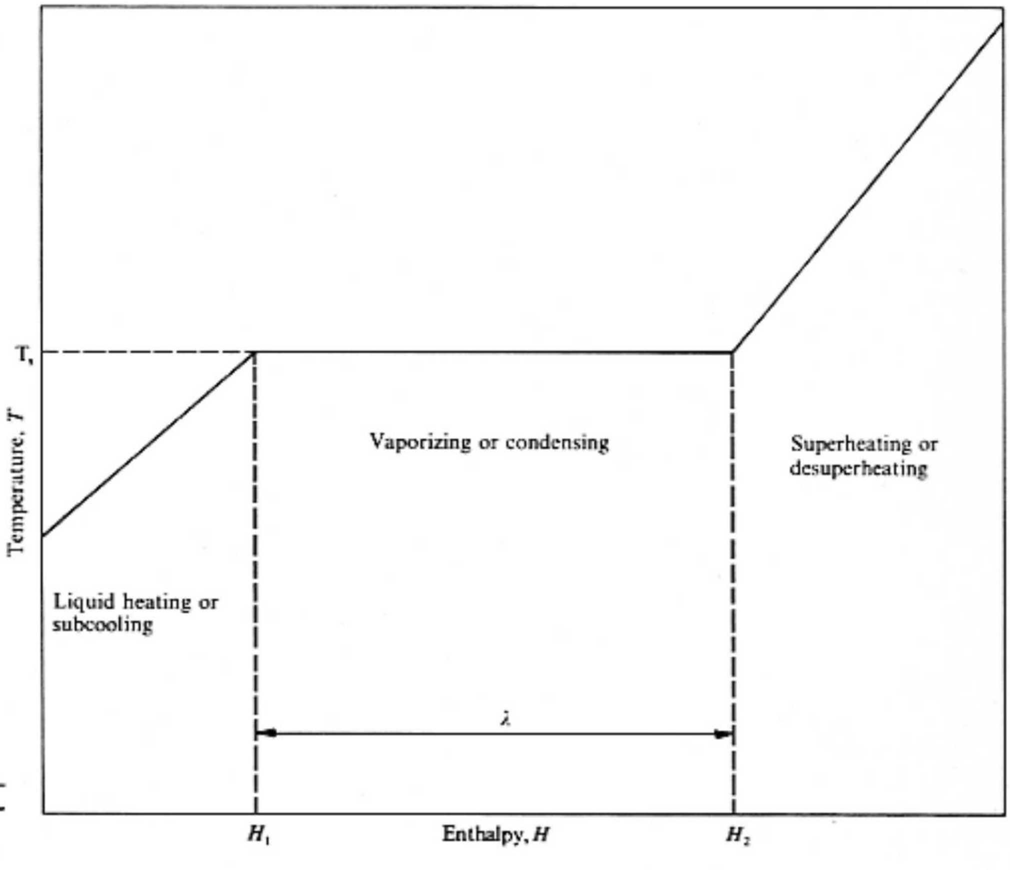


Fig. 34 Temperature–enthalpy curve for condensation or boiling

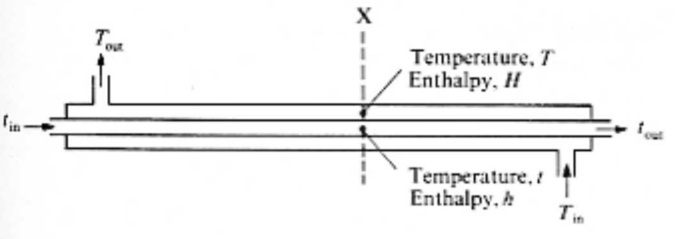


Fig. 35 Double-pipe exchanger as a counter-flow exchanger

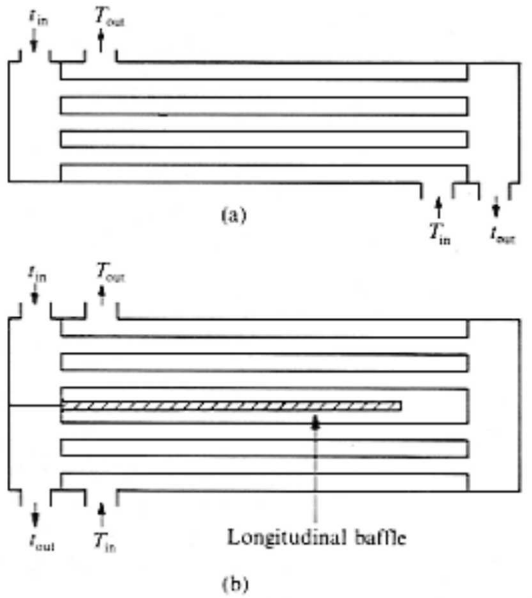


Fig. 36 Two methods of obtaining counter-current flow in shell-and-tube exchangers: (a) E-type shell with one tube pass, (b) F-type shell with two tube passes

To distinguish the two streams in the following equations, capital letters are used for one stream and lower case for the other. Hence the temperature, enthalpy, and flowrate for one stream are T, H, and W, respectively, while for the other they are t, h, and w. The convention with shell-and-tube exchangers (which is followed in this Engineering Design Guide) is that the capital letters are used for the shell side.

At some position along the flow path of one stream (say position X on Fig. 35) the enthalpy of one stream can be derived from that of the other as follows:

$$h = h_{in} + \frac{W}{w}(H - H_{out}). \tag{51}$$

Fig. 37 shows the T–H curve for a condensing stream which has been plotted by the methods dealt with under the heading *Temperature–enthalpy curves*. On this same figure, the temperature of the other stream has been plotted by (1) choosing values of H and calculating the corresponding values of h from eqn (51), (2) using the t–h curve for the stream to obtain values of t for corresponding values of h, and (3) plotting the value of t which corresponds with each H on Fig. 37 to give the t-curve shown. In this case the t-curve is a straight line, corresponding, say to a single-phase, liquid-cooling situation.

It sometimes happens that at some point the t-curve may cross the T-curve as is illustrated by the broken line on Fig. 37. An exchanger can *never* be

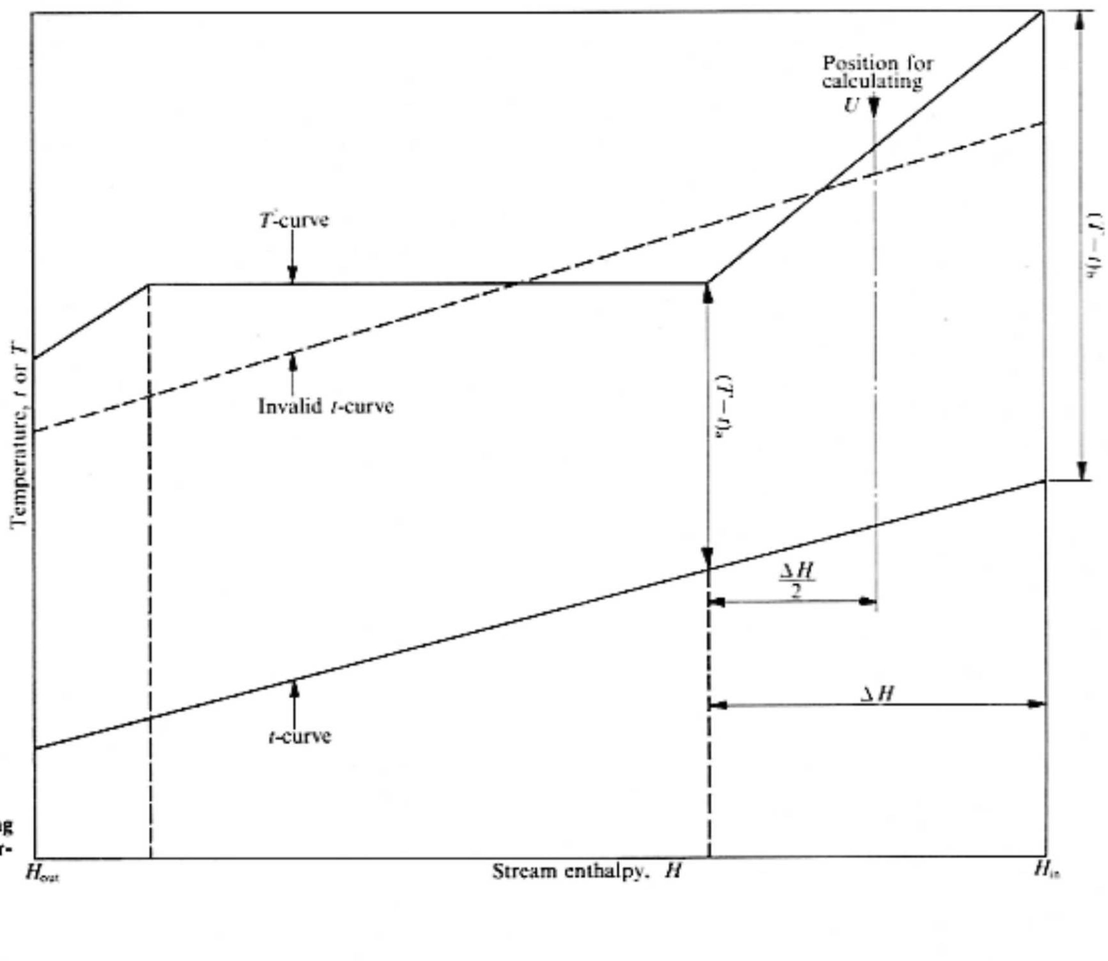


Fig. 37 Operating diagram for counter-flow exchanger

made to operate under these conditions and hence, when this occurs, the inlet and outlet conditions and/or flowrates of the streams must be changed in order to avoid the temperature cross.

To calculate the mean temperature difference (MTD), the diagram must be divided into zones using the vertical lines indicated in Fig. 37. The zones are chosen so that both the t- and T-curves are linear across each zone. In the case of multi-component boiling and condensing, there may be no straight portions to the curve. In these circumstances, zones must be chosen such that a straight-line representation of the curve in a given zone does not deviate far from the actual curve. Fig. 38 shows how the zones may be chosen for a more complicated example involving non-linear temperature–enthalpy curves.

In each zone, the enthalpy change, ΔH, and the logarithmic mean temperature difference (LMTD) for the zone, $\theta_{\ln}$, can be calculated, where

$$\theta_{\ln} = \frac{(T-t)_b - (T-t)_a}{\ln[(T-t)_b/(T-t)_a]}. \qquad (52)$$

$(T-t)_b$ and $(T-t)_a$ are the temperature differences between the streams at the two ends of the zone. When these two temperature differences are very similar, the arithmetic mean may be used for $\theta_{\ln}$.

For the special case where both temperature–enthalpy curves are linear across the exchanger, a value of $\theta_{\ln}$ for the whole exchanger can be determined and this is the required MTD-value to use in eqn (46). This is usually the procedure to follow if both streams are single phase. If, however, the curves are non-linear, the MTD-value must be calculated from the following equation:

$$\frac{1}{\theta} = \frac{1}{H_{in} - H_{out}} \left\{ \frac{\Delta H_1}{\theta_{\ln 1}} + \frac{\Delta H_2}{\theta_{\ln 2}} + \ldots \right\}, \qquad (53)$$

where ΔH_1, ΔH_2 etc. are the enthalpy changes in each zone and $\theta_{\ln 1}$ and $\theta_{\ln 2}$ are the corresponding zone LMTD-values. Example 13 and 14 (p. 40) illustrate the calculation of MTD-values.

Eqn (46) can easily be evaluated (after calculating the mean temperature difference by the method given) provided that the overall heat-transfer coefficient does not vary throughout the exchanger. If, however, the overall coefficient varies greatly, it becomes much more difficult to determine its mean value ($\bar{U}$) for use in this equation. In these circumstances the effective heat-transfer area for each zone, ΔA, must be calculated from

$$\Delta A = \frac{W \times \Delta H}{U\theta_{\ln}}, \qquad (54)$$

where H, U and $\theta_{\ln}$ are all calculated for the zone being considered. (If U varies over the zone, it can be calculated at the middle as illustrated in Fig. 37.) Once this has been done, the areas of the separate zones are added together to give the required total heat-transfer area, A_{req}:

$$A_{req} = \Delta A_1 + \Delta A_2 + \ldots, \qquad (55)$$

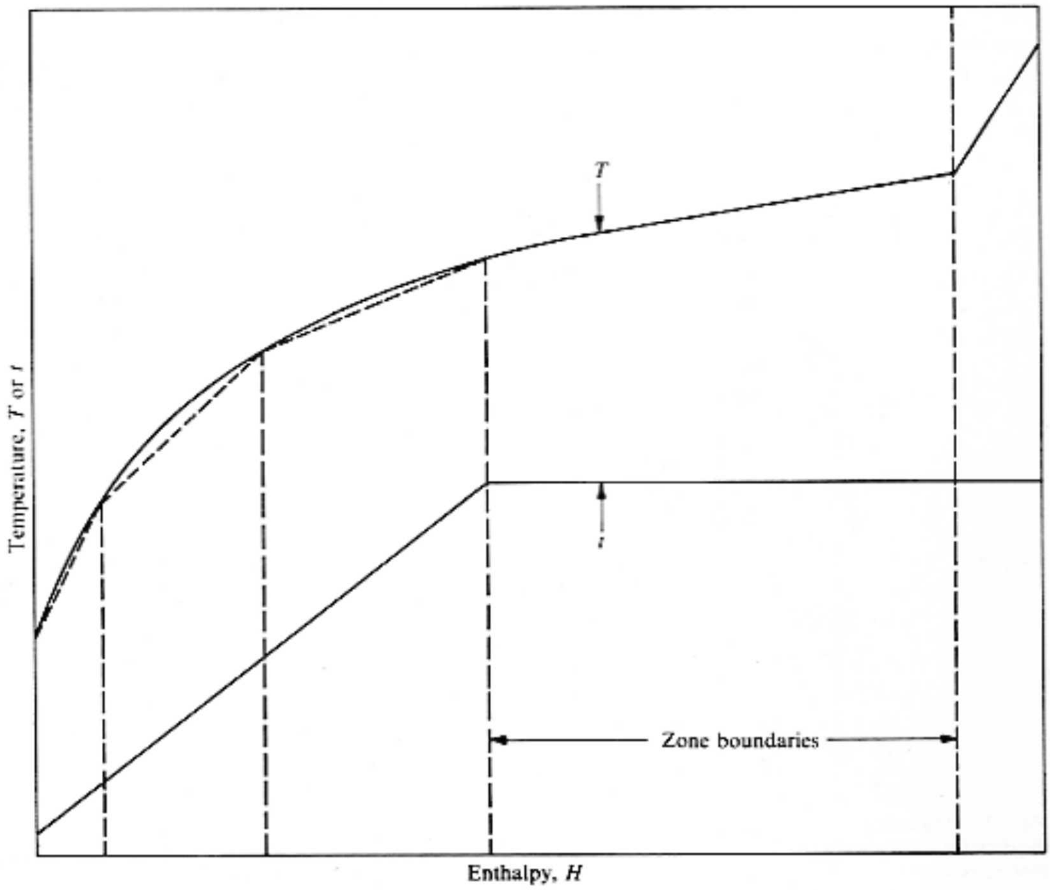


Fig. 38 Selection of linear zones for counter-flow exchanger

where ΔA_1, ΔA_2, etc. are the areas for each zone as determined from eqn (54). The average overall coefficient is calculated from

$$\bar{U} = (\Delta A_1 U_1 + \Delta A_2 U_2 + \ldots)/A_{\text{req}}. \qquad (56)$$

The calculation of heat-transfer area and average overall heat-transfer coefficient is illustrated in Example 14.

Temperature-correction factors for non-counter-current flow

Pure counter-current flow in an exchanger gives an increased MTD-value, θ, which can be expressed as follows:

$$\theta = \theta_{\text{cc}} F_{\text{T}}, \qquad (57)$$

where θ_{cc} is the MTD-value for a counter-current exchanger (calculated as described under that heading) and F_{T} is a temperature-difference correction factor whose value is less than unity.

Examples of non-counter-current-flow exchangers include shell-and-tube types with multiple passes, J- and G-type shell-and-tube exchangers, and cross-flow, air-cooled types. More heat-transfer area must be provided in non-counter-current-flow types of exchanger to compensate for the lowering of the MTD-value. However, by changing to non-counter-current flow the overall coefficients may be increased sufficiently to more than compensate for the loss in temperature difference. For example, a single-pass shell-and-tube exchanger may have very low tube-side coefficients, but these can be increased by increasing the number of tube-side passes, which increases the tube-side velocity.

Theoretical curves giving correction factors, F_{T}, have been derived for a variety of circumstances and some are given in Figs. 39, 40, and 41. These curves assume constant heat-transfer coefficients and linear temperature–enthalpy curves for both streams. Figs 39 and 40 are for shell-and-tube exchangers, and Fig. 41 is suitable for a cross-flow air-cooled exchanger with a single tube-side pass. Before using these curves, two parameters P and R must be calculated, which are as follows:

$$P = \frac{t_{\text{out}} - t_{\text{in}}}{T_{\text{in}} - t_{\text{in}}}, \qquad (58)$$

and

$$R = \frac{T_{\text{in}} - T_{\text{out}}}{t_{\text{out}} - t_{\text{in}}}. \qquad (59)$$

Further curves are given by Bowman, Mueller, and Nagle (1940), and TEMA (1968). Example 15 (p. 41) illustrates the use of these curves.

These correction-factor curves do *not* apply when the coefficient is not constant, or when the temperature–enthalpy curves are non-linear. However, they *can* be used in these situations, although the results obtained for the corrected MTD-values will be less accurate. For shell-and-tube exchangers, methods allowing for non-linear curves and non-constant coefficients have been developed by Butterworth (1973) and Emerson (1973). These involve calculations that are quite tedious, often requiring the use of computers (p. 34).

It is important that heat exchangers should be designed with F_{T} corrections not less than about 0·8 since, below this value, their performance will be extremely sensitive to small changes in stream temperature.

Pressure-drop considerations

When designing heat exchangers, it is necessary to calculate the pressure drops for both streams, since only a limited amount of pressure drop may be available to drive the fluid through the exchanger.

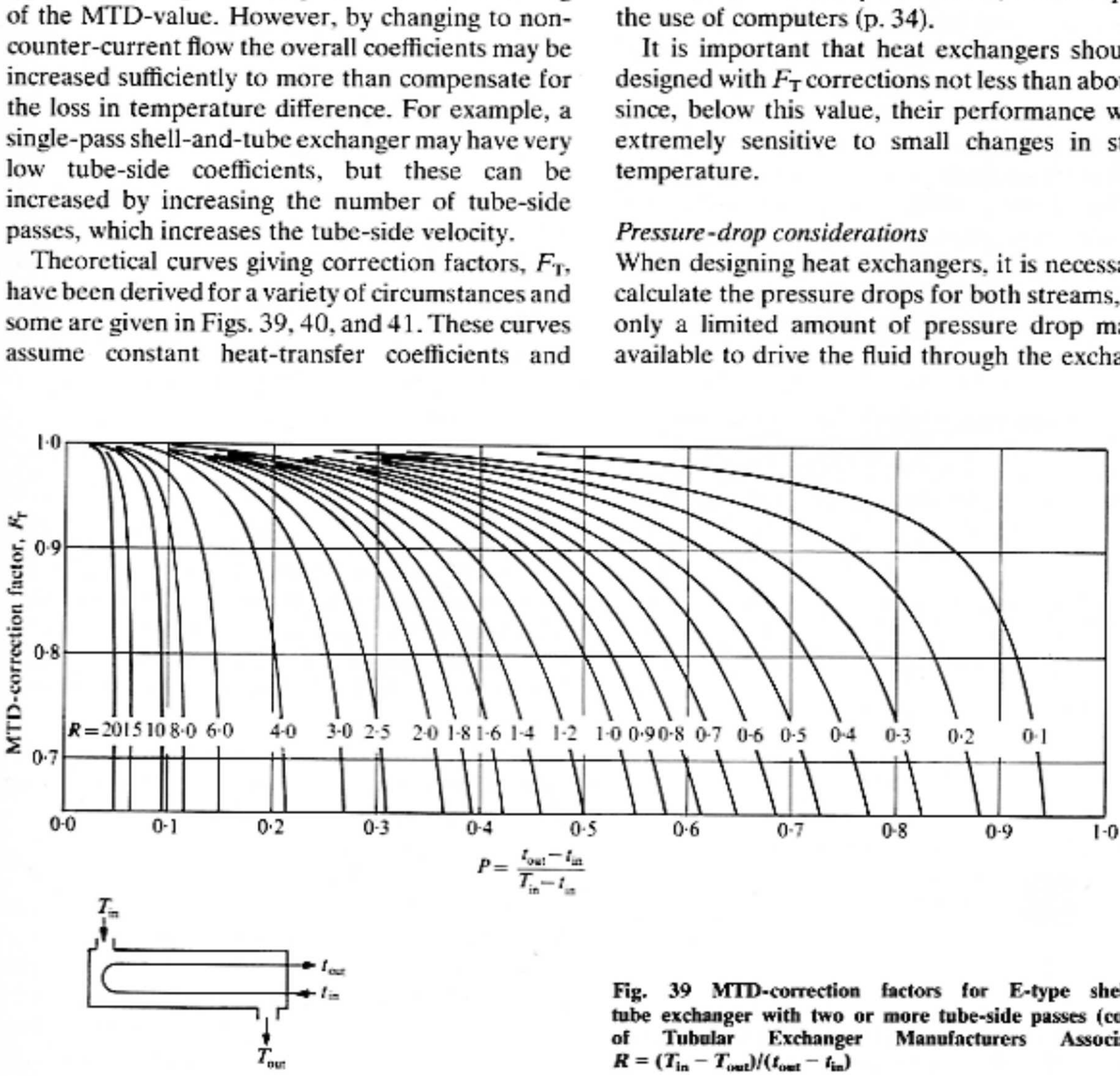
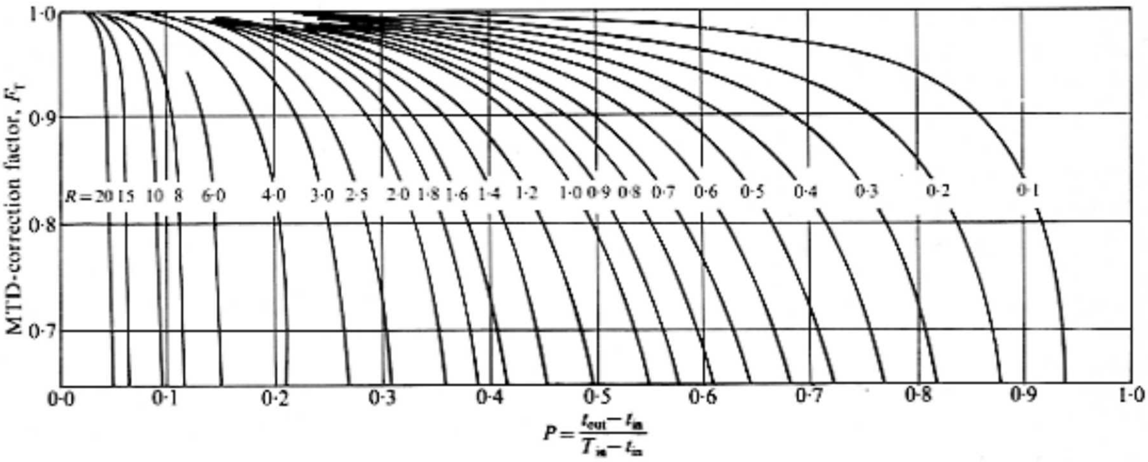


Fig. 39 MTD-correction factors for E-type shell-and-tube exchanger with two or more tube-side passes (courtesy of Tubular Exchanger Manufacturers Association); $R = (T_{in} - T_{out})/(t_{out} - t_{in})$



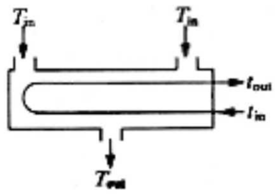


Fig. 40 MTD-correction factors for J-type, shell-and-tube exchanger with two or more tube-side passes (courtesy of Tubular Exchanger Manufacturers Association); $R = (T_{in} - T_{out})/(t_{out} - t_{in})$

Some information is given here on single-phase pressure-drop calculations for flow in tubes and on the shell side of shell-and-tube heat exchangers. For flow across finned tubes or between the plates of plate heat exchangers, the use of manufacturers' data is preferred. The calculation of two-phase pressure drop can be rather complicated and the reader is referred to books on two-phase flow for information on pressure drops in tubes (for example, Hewitt and Hall Taylor, 1970, and Collier, 1972). For shell-side, two-phase pressure drops, reference should be made to the papers by Grant (1973a, 1973b). Allowance must be made for the pressure drops at the exchanger inlet and outlet.

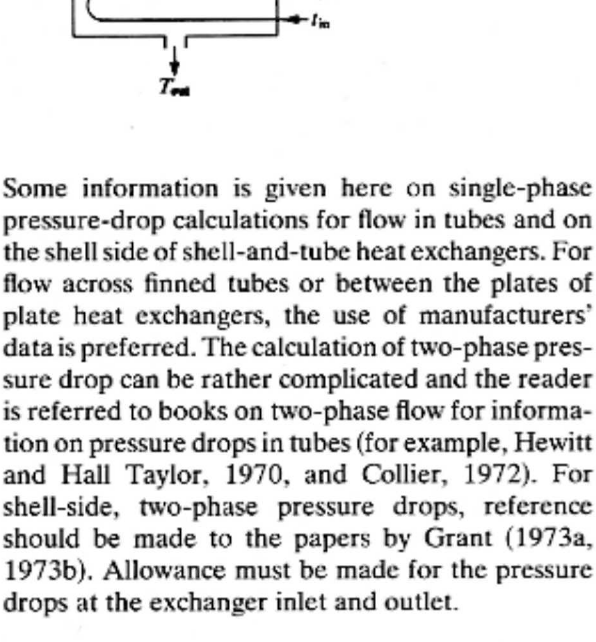


Fig. 41 MTD-correction factors suitable for single-pass air-cooled exchanger; $R = (T_{in} - T_{out})/(t_{out} - t_{in})$

Tube-side pressure drop. The pressure drop Δp for flow in a tube of length L and inside diameter D_i is given by the following equation:

$$\Delta p = 2fL\rho u^2/D_i. \tag{60}$$

where f is a friction factor. For low Reynolds numbers (<2000) the friction factor is given by

$$f = 16/Re. \tag{61}$$

For smooth pipes in turbulent flow (that is $Re > 4000$), the friction factor is given by Engineering Sciences Data Unit (1966) as

$$f = \{3{\cdot}6 \log_{10}(Re/7)\}^{-2}. \tag{62}$$

For commercially rough tubes of the type used in heat exchangers, Kern (1950) recommends the following equation for f:

$$f = 0{\cdot}0035 + \frac{0{\cdot}264}{Re^{0{\cdot}42}}. \tag{63}$$

For Reynolds numbers $2000 < Re < 4000$, the friction factor is difficult to predict but a value of 0·01 is recommended for smooth tubes and one of 0·012 for rough tubes. Eqns (60)–(63) apply to flow in non-circular channels provided that the hydraulic diameter given in Table 1 (p. 5) is used in place of the tube i.d.

In addition to the pressure drop for flow along the tube, there is usually a pressure drop on entry to the tube which is expressed as follows:

$$\Delta p = \tfrac{1}{2}K\rho u^2, \tag{64}$$

where K is known as the *number of velocity heads lost* and is about 1·3 for a pipe inlet.

It is somewhat difficult to estimate the pressure loss due to the fluid leaving a tube, changing direction in a header, and entering the next pass. The loss due to the header can be given by eqn (64) where K

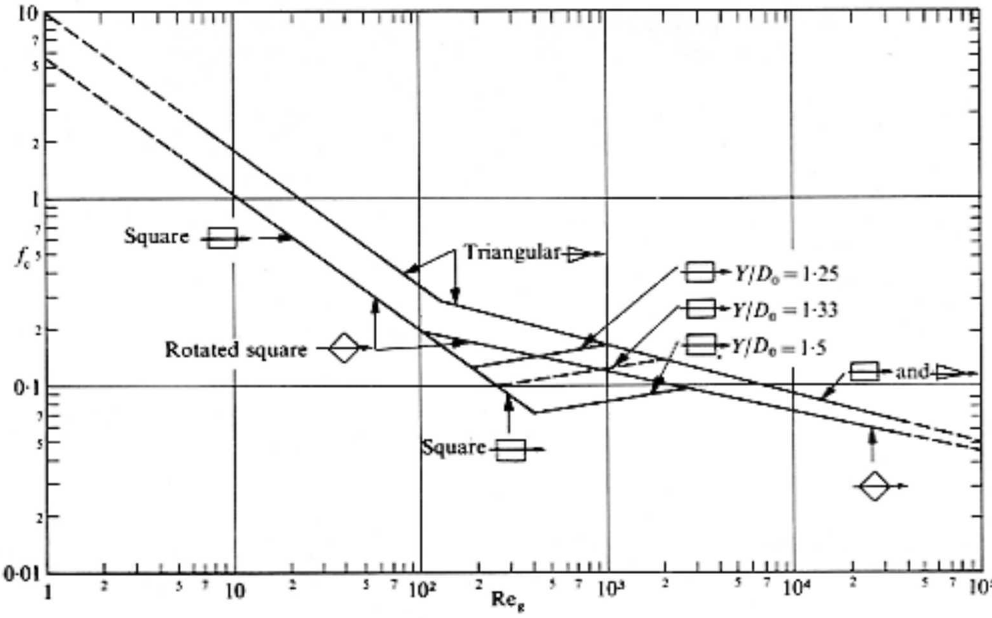


Fig. 42 Moore (1974) correlation for cross-flow pressure drop in ideal banks

is about 0·5. Probably a good rule is to combine the header pressure drop and the tube-entry pressure drop by using the following equation:

$$\Delta p = \tfrac{1}{2} K N_p \rho u^2, \tag{65}$$

where $K \simeq 1{\cdot}8$ and N_p is the number of tube passes.

Shell-side pressure drop. The pressure drop for flow across an ideal bundle is given by

$$\Delta p_1 = 2 N_c f_c \rho u_m^2, \tag{66}$$

where N_c is the number of rows crossed and f_c is a cross-flow friction factor. Fig. 42 was derived by Moore (1974) and gives this friction factor plotted as a function of a special cross-flow Reynolds number whose length term is the gap between the tubes:

$$Re_g = (Y - D_o)\rho u_m / \mu. \tag{67}$$

The broken curves on Fig. 42 indicate where the data have been extrapolated. There are insufficient data for rotated triangular tube bundles to enable a complete curve to be drawn, but such data as are available suggest that the friction factors for these rotated bundles are about 0·75 those of triangular bundles.

Eqns (66) and (67) can be used for calculating the pressure drop over each cross-flow zone (Fig. 22, p. 20) in the exchanger, although the resultant pressure drop must be corrected for leakage and bypass. Bell (1963) has proposed correction factors for this similar to those he introduced to correct the ideal bundle heat-transfer coefficients (p. 19). Thus, the pressure drop for cross flow in a given baffle is given by

$$\Delta p_c = \Delta p_1 F_B' F_L', \tag{68}$$

where F_B' is the bypass correction factor and F_L' the leakage correction factor. These factors are given in Fig. 43.

It is also necessary to calculate the window pressure drop in each baffle space. The combined pressure drop for flow in the two window zones (Fig. 22, p. 20) is given, according to the method of Ishigai, Nishikawa, Nakayama, and Fukuda (1967), by

$$\Delta p_W = \tfrac{1}{2} K_W \rho u_W^2, \tag{69}$$

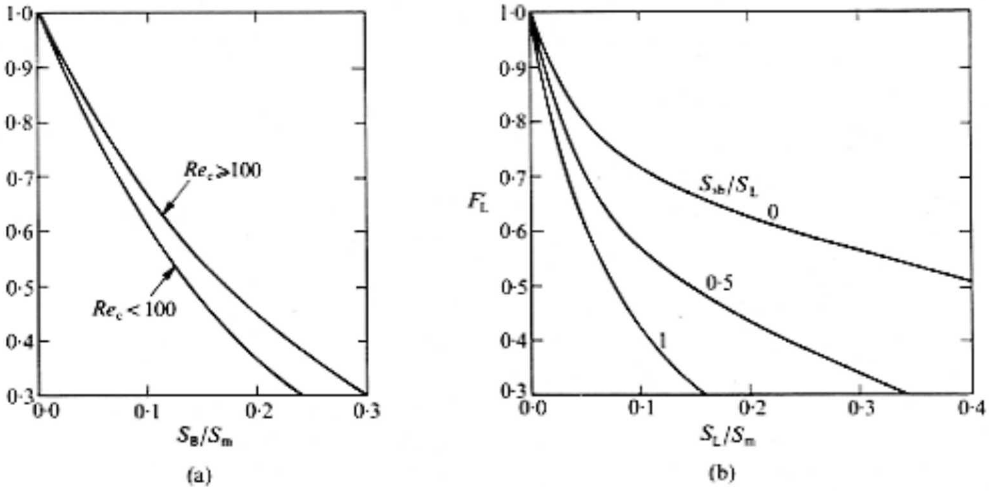


Fig. 43 Bell correction factors for shell-side pressure drop: (a) bypass, (b) leakage

where u_W, the velocity of fluid flowing through the baffle window, is given by

$$u_W = W/\rho S_W, \qquad (70)$$

and K_W by

$$K_W = 2{\cdot}5 - 3{\cdot}5\frac{S_L}{S_W} + 0{\cdot}2\frac{H_c}{Y_b}. \qquad (71)$$

In eqn (71) H_c is the height of the baffle overlap or cross-flow zone (Fig. 22) and Y_b is the baffle pitch.

An example of a design calculation
Example 16 illustrates the design of a relatively simple type of equipment: the double pipe exchanger. Although this is a fairly trivial example, it does serve to illustrate the overall design process. The worked examples given elsewhere in this Engineering Design Guide show how to carry out a wide variety of more complicated detail calculations, which can be used to build up more sophisticated design routines when necessary.

Use of computers in design
It is probable that many readers of this Engineering Design Guide will carry out numerous design calculations using existing computer programs. This is inevitable when one considers both the complexity and the number of the calculations that are required in design. Although some of this complexity has been illustrated by the examples given in the preceding pages, there is a tendency to use even more complex methods in computer programs because of the power of these programs to deal rapidly with such calculations. Computers are extremely valuable in taking the drudgery out of design but there are some pitfalls; the following check list shows how to avoid these.

1. *Understand the design problem.* Computers are not capable of initiative and it is difficult for programmers to allow for every eventuality. You, therefore, should have sufficient understanding of the design problem to be able to check when things are going wrong.

2. *Make sure the data is absolutely correct.* Make sure that you understand what every input item means and that it is entered correctly. Programs often require data items in a rigid format, for example a data item must be in the correct place on a card. Seemingly trivial input errors can be interpreted quite incorrectly by the computer and may lead to nonsensical designs, or to failure of the program.

3. *Check the output.* Because of errors which may be introduced as a result of items 1 or 2, or because of other possible malfunctioning of the program, it is essential to check the output carefully. Check that the answers are (1) sensible, and (2) self-consistent.

4. *Make sure that any program being used is kept up to date and that you are using current documentation.* Programs are rarely static, as they are continually being changed both to extend their range and to avoid difficulties that may have been found while using them. Make sure (1) that the copy of the program which you are using has been up-dated, and (2) that the documentation corresponds with the current version of the program.

Physical Properties
By following the calculations in the appendix, the reader will have become aware that a large amount of physical property data is required before any design is possible, and collecting this can often be the most difficult and tedious part of the design. Some useful references on physical property data are therefore given here.

Mayhew and Rogers (1972) provides useful information on the properties of steam and water, and also gives data on a few other selected fluids. Detailed properties of steam and water can be obtained from steam tables (for example Edward Arnold, 1970). Probably the most important property in any heat-transfer calculation is thermal conductivity. The Engineering Sciences Data Unit has produced many items on thermal conductivity and these may be found in their annual index. The Unit has also produced items covering other important fluid properties (for example viscosity). Further data on thermal conductivity can be obtained from Jamieson, Irving, and Tudhope (1974). The properties of hydrocarbons are given by the American Petroleum Institute (1970). Unfortunately, these properties are given in British Imperial units, whereas this Engineering Design Guide is written entirely in SI units. However, converting data to SI units before the calculation is a fairly simple operation.

Many heat-transfer textbooks contain a section on physical properties. Two such books with particularly useful sections are Fraas and Ozisik (1965) and Rohsenow and Hartnett (1973). Quite often the necessary properties cannot be found anywhere. In such cases, they can be estimated using a variety of theoretical and empirical calculation methods. The textbook by Reid and Sherwood (1966) deals with this topic in some detail.

Conclusion and further reading

It is hoped that reading this Engineering Design Guide has provided an insight into the various aspects of the thermal design of heat exchangers. Since it is not possible to deal with the subject fully in the space available, the following books (full details of which are to be found in the bibliography) are suggested for further reading:

Fraas and Ozisik (1965)
Kern (1950)
Rohsenow and Hartnett (1973), and
Ludwig (1965).

Despite its brevity considering the large amount of relevant subject matter available, this Engineering Design Guide is up to date at the time of publication. The information in the books listed, therefore, should be used to *supplement* rather than *replace* the material provided in these pages.

Appendix. Worked Examples

Example 1. To find the heat-transfer coefficient for single-phase flow in a tube

It is required to calculate the heat-transfer coefficient for Freon 12 flowing at two different flowrates inside a 0·0203 m i.d. tube (for example a 1 in o.d., 14 B.W.G. tube) which is 3·0 m long. Mass flowrates, W, are (1) 0·12 and (2) 0·006 kg/s.

Freon 12 properties from Mayhew and Rogers (1972) are $C = 976$ J/kg K, $\rho = 1304$ kg/m^3, $\mu = 2{\cdot}54 \times 10^{-4}$ N s/m^2, and $k = 0{\cdot}071$ W/m K.

(1) *Calculation for the flowrate of* 0·12 *kg/s*
To calculate the tube flow area, S:

$$S = \frac{\pi D_i^2}{4} = \frac{\pi \times (0{\cdot}0203)^2}{4} = 3{\cdot}24 \times 10^{-4}\ \text{m}^2.$$

To calculate the fluid mean velocity, u:

$$u = \frac{W}{\rho S} = \frac{0{\cdot}12}{1304 \times 3{\cdot}24 \times 10^{-4}} = 0{\cdot}284\ \text{m/s}.$$

From eqn (10) where $l = D_i$ (see Table 1),

$$Re = \frac{\rho u l}{\mu} = \frac{1304 \times 0{\cdot}284 \times 0{\cdot}0203}{2{\cdot}54 \times 10^{-4}} = 29\,600.$$

From eqn (11),

$$Pr = \frac{C\mu}{k} = \frac{976 \times 2{\cdot}54 \times 10^{-4}}{0{\cdot}071} = 3{\cdot}49.$$

From eqn (18) or Fig. 2,

$$E = 0{\cdot}0217.$$

From eqn (17),

$$St = 0{\cdot}0217\,(29\,600)^{-0{\cdot}205}\,(3{\cdot}49)^{-0{\cdot}505}$$
$$= 1{\cdot}40 \times 10^{-3}.$$

Finally from eqn (16) the heat-transfer coefficient, α, may be calculated as follows:

$$\alpha = \rho u C\, St = 1304 \times 0{\cdot}284 \times 976 \times 1{\cdot}40 \times 10^{-3}$$
$$= 506\ \text{W/m}^2\,\text{K}.$$

Note. The use of eqn (15) instead of (17) gives $St = 1{\cdot}39 \times 10^{-3}$ and hence $\alpha = 502$ W/m^2 K.

(2) *Calculation for a flowrate of* 0·006 *kg/s*
Repeating the calculation of Re with one twentieth of the previous flowrate gives $Re = 1480$. Since Re is less than 2000, the flow is laminar and eqn (19) must be used instead of eqn (17).
From eqn (20),

$$Gz = \frac{WC}{kL} = \frac{0{\cdot}006 \times 976}{0{\cdot}071 \times 3{\cdot}0} = 27{\cdot}5.$$

From eqn (19a),

$$Nu = 1{\cdot}75\, Gz^{1/3} = 1{\cdot}75\,(27{\cdot}5)^{1/3} = 5{\cdot}28.$$

From eqn (9) with $l = D_i$,

$$\alpha = \frac{k\, Nu}{l} = \frac{0{\cdot}071 \times 5{\cdot}28}{0{\cdot}0203} = 18{\cdot}5\ \text{W/m}^2\,\text{K}.$$

Example 2. To find the heat-transfer coefficient for single-phase flow in an annular passage

This example enables the heat-transfer coefficient for the flow of water in an annular passage formed by a 0·0254 m (1 in) o.d. tube inside a 0·0560 m i.d. (2·5 in o.d., 9 B.W.G.) tube to be calculated. The water mass flowrate is 0·9 kg/s and its temperature is 80 °C.

Water properties at 80 °C are $C = 4198$ J/kg K, $\rho = 972$ kg/m^3, $\mu = 3{\cdot}55 \times 10^{-4}$ N s/m^2, and $k = 0{\cdot}670$ W/m K.

Calculation
From Table 1, the hydraulic mean diameter,

$$l = D - d = 0{\cdot}0560 - 0{\cdot}0254 = 0{\cdot}0306\ \text{m},$$

where D is the i.d. of the outer tube and d the o.d. of the inner tube. To calculate the channel flow area, S:

$$S = \frac{\pi}{4}(D^2 - d^2) = \frac{\pi}{4}\{(0{\cdot}0560)^2 - (0{\cdot}0254)^2\}$$
$$= 1{\cdot}96 \times 10^{-3}\text{m}^2.$$

The fluid mean velocity,

$$u = \frac{W}{\rho \cancel{A} S} = \frac{0{\cdot}9}{972 \times 1{\cdot}96 \times 10^{-3}} = 0{\cdot}472\ m/s.$$

From eqn (10),

$$Re = \frac{\rho u l}{\mu} = \frac{972 \times 0{\cdot}472 \times 0{\cdot}0306}{3{\cdot}55 \times 10^{-4}} = 39\,500.$$

From eqn (11),

$$Pr = \frac{C\mu}{k} = \frac{4198 \times 3{\cdot}55 \times 10^{-4}}{0{\cdot}670} = 2{\cdot}22.$$

From eqn (18) or Fig. 2,

$$E = 0{\cdot}0222.$$

Hence from eqn (17),

$$St = 0{\cdot}0222\,(39\,500)^{-0{\cdot}205}\,(2{\cdot}22)^{-0{\cdot}505}$$
$$= 1{\cdot}69 \times 10^{-3}.$$

Finally from eqn (16)

$$\alpha = \rho u C\ St = 972 \times 0{\cdot}472 \times 4198 \times 1{\cdot}69 \times 10^{-3}$$
$$= 3250\ \text{W/m}^2\,\text{K}.$$

Example 3. To find the heat-transfer coefficient for natural convection on the outside of a horizontal tube

To calculate the natural convection coefficient for a 0·102 m (4 in) o.d. horizontal tube standing in still air the following procedure is adopted. The tube surface temperature, T_w, is 100 °C and the air temperature, T, is 10 °C.

The air properties used in the calculation are $C = 1004$ J/kg K, $\rho = 1{\cdot}23$ kg/m^3, $\mu = 1{\cdot}80 \times 10^{-5}$ N s/m^2, $k = 0{\cdot}0253$ W/m K, and $\beta = 3{\cdot}56 \times 10^{-3}$/°C.

Calculation
From Table 2

$$Pr = \frac{C\mu}{k} = \frac{1004 \times 1{\cdot}80 \times 10^{-5}}{0{\cdot}0253} = 0{\cdot}71.$$

From Table 2, where l is the tube o.d.,

$$Gr = \frac{\beta g l^3 \rho^2 |T_w - T|}{\mu^2}$$

$$= \frac{3{\cdot}56 \times 10^{-3} \times 9{\cdot}81 \times (0{\cdot}102)^3\,(1{\cdot}23)^2\,(100-10)}{(1{\cdot}80 \times 10^{-5})^2}$$

$$= 1{\cdot}56 \times 10^7.$$

Hence,

$$Gr\,Pr = 0{\cdot}71 \times 1{\cdot}56 \times 10^7 = 1{\cdot}11 \times 10^7.$$

From Fig. 3

$$Nu = 25.$$

The heat transfer coefficient, α, can now be calculated from eqn (9) with l as the tube o.d. as follows:

$$\alpha = \frac{k\,Nu}{l} = \frac{0{\cdot}0253 \times 25}{0{\cdot}102} = 6{\cdot}2\ \text{W/m}^2\,\text{K}.$$

Example 4. To find the heat-transfer coefficient for cross flow to an ideal bundle

The following procedure enables the heat-transfer coefficient for cross flow of water to a bundle consisting of 8 rows with 15 tubes in each row to be calculated. The o.d. of each tube, D_o, is 0·0254 m (1 in) and they are on a 0·0318 m (1·25 in) square pitch. The tube length exposed to the cross flow is 0·203 m (8 in). The water temperature is 80 °C and its flowrate is 10 kg/s.

The water properties are the same as those in Example 2, that is $C = 4198$ J/kg K, $\rho = 972$ kg/m^3, $\mu = 3{\cdot}55 \times 10^{-4}$ N s/m^2, and $k = 0{\cdot}670$ W/m K. The liquid Prandtl number, Pr, (as calculated in Example 2) is 2·22.

Calculation
Calculate ω from Fig. 5(c) as follows:

$$\omega = Y - D_o = 0{\cdot}0318 - 0{\cdot}0254 = 0{\cdot}0064\ \text{m}.$$

From eqn (25),

$$S_m = L\omega N_t = 0{\cdot}203 \times 0{\cdot}0064 \times 15 = 0{\cdot}0195\ \text{m}^2.$$

From eqn (24)

$$u_m = \frac{W}{S_m \rho} = \frac{10}{0{\cdot}0195 \times 972} = 0{\cdot}527\ \text{m/s}.$$

From eqn (23)

$$Re_c = \frac{\rho u_m D_o}{\mu} = \frac{972 \times 0{\cdot}527 \times 0{\cdot}0254}{3{\cdot}55 \times 10^{-4}} = 36\,650.$$

Select a and m from Table 3:

$$a = 0{\cdot}211,$$

and

$$m = 0{\cdot}651.$$

Determine F_N from Fig. 7 with $N_c = 8$:

$$F_N = 0{\cdot}98.$$

From eqn (22)†,

$$Nu = 0{\cdot}211\,(36\,650)^{0{\cdot}651}\,(2{\cdot}22)^{0{\cdot}34}\,0{\cdot}98 = 254.$$

From the definition of a Nusselt number, α can now be calculated as follows:

$$\alpha = \frac{k}{D_o} Nu = \frac{0{\cdot}670}{0{\cdot}0254} \times 254 = 6700\ \text{W/m}^2\,\text{K}.$$

Example 5. To find the heat-transfer coefficient for cross flow to finned tubes

The following calculation gives the heat-transfer coefficient on the surface of the fins for the cross flow of air to a finned-tube bundle. The bundle consists of 0·0254 m o.d. tubes on a transverse pitch of 0·0572 m. The fin height is 0·0158 m, the fin thickness is $3{\cdot}81 \times 10^{-4}$ m, and the distance between fins is 0·00254 m. The air approach velocity is 5 m/s.

The air properties used are $\rho = 1{\cdot}18$ kg/m^3, $\mu = 1{\cdot}85 \times 10^{-5}$ N s/m^2, $k = 0{\cdot}0262$ W/m K and $C = 1005$ J/kg K.

Calculation

$$\frac{\text{Face flow area}}{\text{min. flow area}} = \frac{Y_t}{Y_t - D_o} = \frac{0{\cdot}0572}{0{\cdot}0572 - 0{\cdot}0254} = 1{\cdot}8.$$

The velocity at minimum flow area, u_m, is therefore,

$$u_m = (\text{approach velocity}) \times 1{\cdot}8 = 5 \times 1{\cdot}8 = 9{\cdot}0\ \text{m/s}.$$

From eqn (23),

$$Re_c = \frac{\rho u_m D_o}{\mu} = \frac{1{\cdot}18 \times 9{\cdot}0 \times 0{\cdot}0254}{1{\cdot}85 \times 10^{-5}} = 14\,600.$$

The Prandtl number,

$$Pr = \frac{C\mu}{k} = \frac{1005 \times 1{\cdot}85 \times 10^{-5}}{0{\cdot}0262} = 0{\cdot}71.$$

†Note. As an alternative method, Nu can be determined from Fig. 6 and then corrected by multiplying by F_N.

From eqn (26), the Nusselt number,

$$Nu = 0{\cdot}134\ (14\ 600)^{0{\cdot}681}\ (0{\cdot}71)^{0{\cdot}33} \left(\frac{0{\cdot}00254}{0{\cdot}0158}\right)^{0{\cdot}2} \times \left(\frac{0{\cdot}002\ 54}{3{\cdot}81\times 10^{-4}}\right)^{0{\cdot}1134}$$

$$= 70{\cdot}1.$$

Hence, the heat-transfer coefficient,

$$\alpha = Nu\frac{k}{D_o} = 70{\cdot}1\,\frac{0{\cdot}0262}{0{\cdot}0254} = 72{\cdot}3\ \mathrm{W/m^2\,K}.$$

Example 6. To find the heat-transfer coefficient and critical heat flux in pool boiling

This example shows how to (1) calculate the heat-transfer coefficient for the pool boiling of water at atmospheric pressure with a wall superheat of 4·8 °C, and (2) check that the critical heat flux is not exceeded under these conditions.

The properties of water and steam used in the calculation are $C_L = 4218$ J/kg K, $\mu_L = 2{\cdot}83\times 10^{-4}$ N s/m^2, $k_L = 0{\cdot}681$ W/m K, $\rho_L = 958$ kg/m^3, $\rho_G = 0{\cdot}598$ kg/m^3, $\sigma = 0{\cdot}0588$ N/m, and $\lambda = 2{\cdot}257\times 10^6$ J/kg.

Calculation
Boiling is occurring at atmospheric pressure, hence

$$p_s = 1{\cdot}013\times 10^5\ \mathrm{N/m^2}.$$

The corresponding saturation temperature is,

$$T_s = 100\ °\mathrm{C}.$$

The wall temperature is therefore,

$$T_w = 100 + 4{\cdot}8 = 104{\cdot}8\ °\mathrm{C},$$

and the corresponding pressure from the saturation curve is

$$p_w = 1{\cdot}200\times 10^5\ \mathrm{N/m^2}.$$

Substituting this information, together with the property data into eqn (27)

$$\alpha = 0{\cdot}00122 \times \left\{\frac{(0{\cdot}681)^{0{\cdot}79}\ (4218)^{0{\cdot}45}\ (958)^{0{\cdot}49}}{(0{\cdot}0588)^{0{\cdot}5}\ (2{\cdot}83\times 10^{-4})^{0{\cdot}29} \times (2{\cdot}257\times 10^6\times 0{\cdot}598)^{0{\cdot}24}}\right\} \times (104{\cdot}8-100)^{0{\cdot}24}\ (1{\cdot}200\times 10^5 - 1{\cdot}013 \times 10^5)^{0{\cdot}75}$$

$$= 3860\ \mathrm{W/m^2\,K}$$

To check whether the heat flux is exceeded
The heat flux, q, with this coefficient is

$$q = \alpha(T_w - T_s) = 3860(104{\cdot}8-100) = 1{\cdot}85\times 10^4\ \mathrm{W/m^2}.$$

To calculate the critical heat flux from eqn (28),

$$q_{crit} = 0{\cdot}131\ (2{\cdot}257\times 10^6) \times\{0{\cdot}0588\times 9{\cdot}81\ (958-0{\cdot}598)(0{\cdot}598)^2\}^{0{\cdot}25}$$

$$= 1{\cdot}11\times 10^6\ \mathrm{W/m^2}.$$

Hence, the heat flux is well below the critical value.

Example 7. To find the heat-transfer coefficient for forced-convective boiling

The following calculation gives the local heat-transfer coefficient for the forced-convective boiling of water in a 0·0203 m i.d. tube. The water is at atmospheric pressure and the wall superheat is 4·8 °C: (these conditions are the same as in Example 6). The total flowrate in the tube is 0·1 kg/s and 7 per cent of the flow is in vapour form at the point being considered.

Except for the gas-phase viscosity, which is $\mu_G = 1{\cdot}21\times 10^{-5}$ N s/m^2, the required physical properties are given in Example 6.

Calculation
The tube flow area, S, is $3{\cdot}24\times 10^{-4}$ m^2 (see Example 1). To calculate G, the mass velocity or mass flow per unit area,

$$G = \frac{W}{S} = \frac{0{\cdot}1}{3{\cdot}24\times 10^{-4}} = 309\ \mathrm{kg/m^2\,s}.$$

A vapour percentage of 7 is equivalent to a vapour *fraction*, x, of 0·07. Thus from eqn (31),

$$Re_L = \frac{(1-x)GD_i}{\mu_L} = \frac{(1-0{\cdot}07)\ 309\times 0{\cdot}0203}{2{\cdot}83\times 10^{-4}}$$

$$= 20\ 600.$$

To calculate the Prandtl number for the liquid (see Table 2, p. 6),

$$Pr_L = \frac{C_L\mu_L}{k_L} = \frac{4218\times 2{\cdot}83\times 10^{-4}}{0{\cdot}681} = 1{\cdot}75.$$

To calculate α_{DB} using eqn (12)

$$Nu = 0{\cdot}023(20\ 600)^{0{\cdot}8}(1{\cdot}75)^{0{\cdot}4} = 81{\cdot}3,$$

from which

$$\alpha_{DB} = \frac{k_L}{D_i}Nu = \frac{0{\cdot}681}{0{\cdot}0203}\times 81{\cdot}3 = 2730\ \mathrm{W/m^2\,K}.$$

From eqn (33),

$$\frac{1}{X_{tt}} = \left(\frac{0{\cdot}07}{1-0{\cdot}07}\right)^{0{\cdot}9}\left(\frac{958}{0{\cdot}598}\right)^{0{\cdot}5}\left(\frac{1{\cdot}21\times 10^{-5}}{2{\cdot}83\times 10^{-4}}\right)^{0{\cdot}1} = 2{\cdot}85,$$

giving (from Fig. 13),

$$F_c = 5{\cdot}2.$$

Thus from eqn (30),

$$\alpha_{con} = F_c \times \alpha_{DB} = 5{\cdot}2 \times 2730 = 14\,200 \text{ W/m}^2\text{ K}.$$

$$Re_L F_c^{1{\cdot}25} = 1{\cdot}62 \times 10^5.$$

Thus (from Fig. 14)

$$S_c = 0{\cdot}27.$$

From Example 6,

$$\alpha_{FZ} = 3860 \text{ W/m}^2\text{ K},$$

giving (from eqn (34)),

$$\alpha_{nuc} = \alpha_{FZ} \times S_c = 3860 \times 0{\cdot}27 = 1040 \text{ W/m}^2\text{ K}.$$

The final coefficient (from eqn (29)) is as follows:

$$\alpha = \alpha_{nuc} + \alpha_{con} = 14\,200 + 1040 = 15\,200 \text{ W/m}^2\text{ K}.$$

Example 8. To find the heat-transfer coefficient for condensation on horizontal tubes

In this example the heat-transfer coefficient for condensing steam at 3·61 bar on the outside of a bundle of horizontal tubes is calculated. The bundle consists of 158 tubes of 0·0254 m o.d. and length 3·66 m, the number of tubes in an average vertical row is 12, and steam is condensed at the rate of 2·0 kg/s.

The properties of water and steam used are $\mu_L = 1{\cdot}94 \times 19^{-4}$ N s/m^2, $\rho_L = 926$ kg/m^3, $\rho_G = 1{\cdot}97$ kg/m^3, and $k_L = 0{\cdot}688$ W/m K.

Calculation

$$\Gamma' = \frac{\text{vapour condensed}}{(\text{number of tubes}) \times (\text{tube length})}$$

$$= \frac{2{\cdot}0}{158 \times 3{\cdot}66} = 0{\cdot}003\,46 \text{ kg/s m}.$$

From eqn (35),

$$\alpha = 0{\cdot}951 \times 0{\cdot}688 \left[\frac{926\,(926 - 1{\cdot}97)\,9{\cdot}81}{1{\cdot}94 \times 10^{-4} \times 0{\cdot}00346}\right]^{1/3}$$

$$= 15\,200 \text{ W/m}^2\text{ K}.$$

To correct α for the number of tubes in a vertical row using eqn (36),

$$\alpha_N = 15\,200\,(12)^{-2/9} = 8750 \text{ W/m}^2\text{ K}.$$

Example 9. To find the heat-transfer coefficient for condensation on the outside of a vertical tube

This is a repetition of the problem given in Example 8 with the tubes vertical instead of horizontal. In addition to the other properties required for this calculation and given in Example 8, the liquid heat, $C_L = 4282$ J/kg K.

Calculation

$$\Gamma = \frac{\text{vapour condensed}}{(\text{number of tubes}) \times (\text{wetted tube perimeter})},$$

where (for this particular calculation)

$$\text{wetted tube perimeter} = \pi D_o.$$

Hence,

$$\Gamma = \frac{2{\cdot}0}{158 \times \pi \times 0{\cdot}0254} = 0{\cdot}157 \text{ kg/s m},$$

and

$$\frac{4\Gamma}{\mu_L} = \frac{4 \times 0{\cdot}157}{1{\cdot}94 \times 10^{-4}} = 3240.$$

The condensate Prandtl number,

$$Pr_L = \frac{C_L \mu_L}{k_L} = \frac{4282 \times 1{\cdot}94 \times 10^{-4}}{0{\cdot}688} = 1{\cdot}21 \approx 1{\cdot}0.$$

From Fig. 15 ($Pr_L = 1$),

$$\frac{\alpha Z}{k_L} = 0{\cdot}15.$$

The parameter Z may be calculated as follows:

$$Z = \left\{\frac{\mu_L^2}{\rho_L(\rho_L - \rho_G)g}\right\}^{1/3}$$

$$= \left\{\frac{(1{\cdot}94 \times 10^{-4})^2}{926\,(926 - 1{\cdot}94)\,9{\cdot}81}\right\}^{1/3}$$

$$= 1{\cdot}65 \times 10^{-5} \text{ m}.$$

Hence to calculate the coefficient α:

$$\alpha = 0{\cdot}15 \frac{k_L}{Z} = 0{\cdot}15 \frac{0{\cdot}688}{1{\cdot}65 \times 10^{-5}} = 6250 \text{ W/m}^2\text{ K}.$$

Example 10. To find the heat-transfer coefficient for condensation inside a horizontal tube

This example is a further repetition of Examples 8 and 9, but with the condensation occurring *inside* a horizontal tube. Data is the same as in Examples 8 and 9.

Calculation for stratifying flow

To determine the required coefficient, (1) determine the coefficient for condensation outside a single tube and (2) multiply this by 0·8 (p. 14). The outside coefficient from Example 8 is 15 200 W/m^2 K, and thus the required coefficient,

$$\alpha = 15\,200 \times 0{\cdot}8 = 12\,200 \text{ W/m}^2\text{ K}.$$

Calculation for annular flow

The liquid velocity if all the fluid in the tube were condensed is given by

$$u = \frac{W}{\rho_L S\,(\text{number of tubes})}$$

$$= \frac{2{\cdot}0}{926 \times 3{\cdot}24 \times 10^{-4} \times 158}$$

$$= 0{\cdot}0422 \text{ m/s}.$$

The liquid Reynolds number if all the fluid in the tube were condensed is therefore given by

$$Re_L = \frac{\rho_L u D_i}{\mu_L} = \frac{926 \times 0{\cdot}0422 \times 0{\cdot}0203}{1{\cdot}94 \times 10^{-4}} = 4090.$$

The Nusselt number can now be obtained by substituting this value of Re_L and the value of Pr_L from Example 9 into the following equation:

$$Nu = 0{\cdot}023\, Re_L^{\,0{\cdot}8}\, Pr_L^{\,0{\cdot}4} \qquad (12)$$

$$= 0{\cdot}023(4090)^{0{\cdot}8}\,(1{\cdot}21)^{0{\cdot}4} = 19{\cdot}2.$$

Hence.

$$\alpha_L = Nu \frac{k_L}{D_i} = 19{\cdot}2 \frac{0{\cdot}688}{0{\cdot}0203} = 652\ \mathrm{W/m^2K}.$$

The mass fraction of vapour, x, varies from 1·0 (all vapour) at inlet to 0 (all liquid) at outlet. Hence, the J values from eqn (37b) are

$$J_{in} = 1 + \left(\frac{\rho_L}{\rho_G} - 1\right) = \frac{\rho_L}{\rho_G} = \frac{926}{1{\cdot}97} = 470,$$

and

$$J_{out} = 1.$$

Thus from eqn (37a)

$$\alpha = \frac{\alpha_L}{2}(J_{in}^{1/2} + J_{out}^{1/2})$$

$$= \frac{652}{2}[(470)^{1/2} + 1]$$

$$= 7390\ \mathrm{W/m^2\,K}.$$

Selecting the higher coefficient
The higher of the two coefficients is selected and used in any design. Thus, in this case,

$$\alpha = 12\,200\ \mathrm{W/m^2\,K}.$$

Example 11. To find the overall coefficient for a plain tube

A tube of 0·0254 m o.d. and 0·0203 m i.d. has a coefficient of 3250 $\mathrm{W/m^2\,K}$ on the outside and a coefficient of 502 $\mathrm{W/m^2\,K}$ on the inside (these are the results obtained in Examples 1 and 2). The tube is of mild steel which has a thermal conductivity of 45 W/m K. The following calculation gives the overall coefficient if the outside and inside fouling resistances are 0·0003 and 0·0002 $\mathrm{m^2\,K/W}$, respectively (see Table 4, p. 16).

Calculation
From eqn (40b),

$$D_w = \tfrac{1}{2}(D_o + D_i) = \tfrac{1}{2}(0{\cdot}0254 + 0{\cdot}0203) = 0{\cdot}0229\ \mathrm{m}.$$

The wall thickness,

$$y_w = \tfrac{1}{2}(D_o - D_i) = \tfrac{1}{2}(0{\cdot}0254 - 0{\cdot}0203) = 0{\cdot}0025\ \mathrm{m}.$$

To find $1/U$ from eqn (40a):

$$\frac{1}{U} = \frac{1}{3250} + 0{\cdot}0003$$

$$+ \left(\frac{1}{502} + 0{\cdot}0002\right)\frac{0{\cdot}0254}{0{\cdot}0203}$$

$$+ \left(\frac{0{\cdot}0025}{45}\right)\left(\frac{0{\cdot}0254}{0{\cdot}0229}\right)$$

$$= 3{\cdot}41 \times 10^{-3}\ \mathrm{m^2\,K/W}.$$

Hence, the overall coefficient,

$$U = \frac{1}{3{\cdot}41 \times 10^{-3}} = 293\ \mathrm{W/m^2\,K}.$$

Example 12. To find the overall coefficient for a finned tube

Consider the finned tube described in Example 5 including the air-side conditions. The following example shows how to determine the overall coefficient for the same tube if the i.d. is 0·0203 m and the conditions inside the tube are the same as those given for Example 10. Clean conditions on both sides are assumed. The tube is mild steel and the fins are of aluminium. The fin effectiveness may be taken as 0·95.

Calculation
The fin pitch,

$$Y_F = y_F + \delta_F = 0{\cdot}002\,54 + 3{\cdot}81 \times 10^{-4} = 0{\cdot}002\,92\ \mathrm{m}.$$

The tube outside area per fin pitch is, therefore,

$$\pi D_0 Y_F = \pi \times 0{\cdot}0254 \times 0{\cdot}002\,92$$

$$= 2{\cdot}33 \times 10^{-4}\ \mathrm{m^2}.$$

The total area of a fin is given by:

$$\frac{\pi}{2}[(D_0 + 2H_F)^2 - D_0^2]$$

$$= \frac{\pi}{2}[(0{\cdot}0254 + 2 \times 0{\cdot}0158)^2 - (0{\cdot}0254)^2]$$

$$= 4{\cdot}09 \times 10^{-3}\ \mathrm{m^2}.$$

Thus

$$\frac{A_0}{A_F} = \frac{2{\cdot}33 \times 10^{-4}}{4{\cdot}09 \times 10^{-3}} = 0{\cdot}057.$$

Hence, from eqn (42) and neglecting fouling resistances,

$$\frac{1}{\alpha_0} = \frac{1}{\eta}\frac{1}{\alpha_F}\frac{A_0}{A_F} = \left(\frac{1}{0{\cdot}95}\right)\left(\frac{1}{72{\cdot}3}\right)0{\cdot}057$$

$$= 8{\cdot}30 \times 10^{-4}$$

Eqn (40a), for zero fouling resistances, is

$$\frac{1}{U}=\frac{1}{\alpha_0}+\frac{1}{\alpha_i}\frac{D_0}{D_i}+\frac{y_w}{k_w}\frac{D_0}{D_w}$$

$$=8{\cdot}30\times10^{-4}+\left(\frac{1}{12\,200}\right)\left(\frac{0{\cdot}0254}{0{\cdot}0203}\right)+\left(\frac{0{\cdot}0025}{45}\right)$$

$$\times\left(\frac{0{\cdot}0254}{0{\cdot}0229}\right)$$

$$=9{\cdot}94\times10^{-4}.$$

Hence the overall coefficient referred to the tube o.d. is

$$U=1006\ \mathrm{W/m^2\,K}.$$

Example 13. To find the MTD-value for a counter-flow, single-phase exchanger

This calculation shows how to find the MTD-value for a counter-flow exchanger in which one stream is heated from 30–95 °C by another stream which is cooled from 125 °C down to 90 °C. Both streams are single phase with linear temperature–enthalpy curves.

Calculation

The temperature difference at each of the two ends of the exchanger is given by:

$$(T-t)_b=125-95=30\ ^\circ\mathrm{C},\ \text{and}$$

$$(T-t)_a=90-30=60\ ^\circ\mathrm{C}.$$

The logarithmic mean temperature difference, θ_{ln}, is the MTD-value required for this example, and this may be obtained from eqn (52) as follows:

$$\theta_{ln}=\frac{(T-t)_b-(T-t)_a}{\ln\dfrac{(T-t)_b}{(T-t)_a}}=\frac{(T-t)_a-(T-t)_b}{\ln\dfrac{(T-t)_a}{(T-t)_b}}$$

$$=\frac{60-30}{\ln(60/30)}$$

$$=43{\cdot}2\ ^\circ\mathrm{C}.$$

Example 14. To find the MTD-value and overall heat-transfer coefficient for a counter-flow condenser which has desuperheating and condensing zones

Vapour is condensed in a counter-flow exchanger using a single-phase coolant. The vapour is cooled from 150 °C down to 120 °C before it starts to condense. The change in enthalpy in the desuperheating zone is 27 000 J/kg. The vapour condenses isothermally undergoing an enthalpy change of 150 000 J/kg. The coolant enters at 30 °C and leaves at 110 °C. The overall coefficient in the condensing zone is 1400 W/m^2 K and in the desuperheating zone it is 90 W/m^2 K. The vapour inlet flow is 1·6 kg/s. This example shows how to calculate the MTD-value, the average overall coefficient, and the heat-transfer area.

Calculation

It is first necessary to calculate the coolant temperature at the end of the desuperheating zone. This temperature is easily obtained by proportioning the coolant temperature change according to the hot-side enthalpy change on the two zones.

The coolant temperature change in condensing zone

$$=(110-30)\frac{150\,000}{150\,000+27\,000}=67{\cdot}8\ ^\circ\mathrm{C},$$

therefore, the coolant temperature at end of desuperheating zone

$$=30+67{\cdot}8=97{\cdot}8\ ^\circ\mathrm{C}.$$

It is helpful to produce the following layout showing the adjacent temperatures in the two streams.

		Condensing $\Delta H_1=$ 150 000 J/kg		*Desuperheating* $\Delta H_2=$ 27 000 J/kg	
Hot	120 °C		120 °C		150° C
Cold	30° C		97·8 °C		110 °C
		$U_1=1400$ W/m^2 K		$U_2=90$ W/m^2 K	

To calculate the LMTD-value, θ_{ln}, for each zone using eqn (52),

$$\theta_{ln1}=\frac{(120-30)-(120-97{\cdot}8)}{\ln\dfrac{120-30}{120-97{\cdot}8}}=48{\cdot}4\ ^\circ\mathrm{C},\ \text{and}$$

$$\theta_{ln2}=\frac{(150-110)-(120-97{\cdot}8)}{\ln\dfrac{150-110}{120-97{\cdot}8}}=30{\cdot}2\ ^\circ\mathrm{C}.$$

To calculate the MTD-value, θ, from eqn (53),

$$\frac{1}{\theta}=\frac{1}{177\,000}\left(\frac{150\,000}{48{\cdot}4}+\frac{27\,000}{30{\cdot}2}\right)=0{\cdot}0226/^\circ\mathrm{C},$$

therefore,

$$\theta=44{\cdot}3\ ^\circ\mathrm{C}.$$

To calculate the area required for each zone from eqn (54):

$$\Delta A_1=\frac{1{\cdot}6\times150\,000}{1400\times48{\cdot}4}=3{\cdot}54\ \mathrm{m^2},\ \text{and}$$

$$\Delta A_2=\frac{1{\cdot}6\times27\,000}{90\times30{\cdot}2}=15{\cdot}89\ m^2.$$

To calculate the total required heat transfer area from eqn (55),

$$A_{req}=3{\cdot}54+15{\cdot}89=19{\cdot}4\ \mathrm{m^2}.$$

To calculate the average overall coefficient, $\bar{U}$, from eqn (56),

$$\bar{U}=\frac{1}{19{\cdot}4}(3{\cdot}54\times1400+15{\cdot}89\times90)$$

$$=329\ \mathrm{W/m^2\ K}.$$

In order to check the answers:

(1) calculate Q_T from eqn (49),

$$Q_T=1{\cdot}6\times177\,000=283\,200\ \mathrm{W};\ \text{and}$$

(2) calculate Q_T from eqn (46),

$$Q_T=\bar{U}A_T\theta=329\times19{\cdot}4\times44{\cdot}3=282\,700\ W.$$

Apart from rounded errors in the arithmetic, these two results are the same.

Example 15. To correct the MTD-value in Example 13 for non-counter-current flow

This example shows how to correct the MTD-value found in Example 13 if the heat transfer takes place in (1), an E-type and (2) a J-type shell-and-tube exchanger with two tube-side passes; the hot fluid is on the shell side.

Calculation
Restating the problem in the required nomenclature,

$T_{in}=125\ °\mathrm{C}$, $T_{out}=90\ °\mathrm{C}$, $t_{in}=30\ °\mathrm{C}$, $t_{out}=95\ °\mathrm{C}$.

From eqns (58) and (59) respectively,

$$P=\frac{95-30}{125-30}=0{\cdot}684,$$

and

$$R=\frac{125-90}{95-30}=0{\cdot}538.$$

(1) *E-type exchanger.* From Fig. 39 for the E-type exchanger,

$$F_T=0{\cdot}73.$$

Hence, from eqn (57),

$$\theta=43{\cdot}2\times0{\cdot}73=31{\cdot}5\ °\mathrm{C}.$$

(2) *J-type exchanger.* From Fig. 40 for the J-type exchanger

$$F_T=0{\cdot}715.$$

Therefore,

$$\theta=43{\cdot}2\times0{\cdot}715=30{\cdot}9\ °\mathrm{C}.$$

Note. Both these cases involve F_T values of less than 0·8. Such values are not desirable in design since they lead to exchangers that are sensitive to changes in operating conditions. This problem can be avoided by going to a pure counter-flow exchanger but this is not always possible. An alternative is to use two exchangers in series with counter-current flow between the exchangers. Hence the hot stream might be cooled from 125 °C to 107·5 °C in the first exchanger and from 107·5 °C down to 90 °C in the second, while the cold fluid would be heated from 30 °C up to 62·5 °C in the second exchanger and from 62·5 °C to 95 °C in the first (see Fig. 44). In these two exchangers, R is the same and is equal to the value for the single exchanger (that is 0·538). The P-values are 0·52 and 0·42 leading to F_T-values of 0·93 and 0·96, respectively, for the E-type exchanger.

Example 16. The design of a double-pipe exchanger

It is desired to heat Freon 12 from 20 °C up to 65 °C in a counter-flow, double-pipe exchanger. Hot water at 80 °C is available as the heating fluid. The heat exchanger is formed of two lengths of mild steel tubing with outer and inner diameters of 0·0254 m and 0·0203 m for one tube, and 0·0635 m and 0·0560 m for the other. The Freon flows in the inner tube. The Freon flowrate is 0·12 kg/s and the flow of water available is 0·9 kg/s. The following calculation gives the lengths of tubing required and the pressure drops in the two streams.

Calculation of length
Many of the calculations and fluid properties for this problem have already appeared in Examples 1, 2, and 11, and these will therefore be used here when required.
To calculate the heat load required to raise the Freon temperature,

$$Q_T=WC\times(\text{temp. rise})$$

$$=0{\cdot}12\times976\times(65-20)$$

$$=5270{\cdot}4\ \mathrm{W}.$$

To calculate the water temperature drop,

$$\text{temp. drop}=\frac{Q_T}{WC}=\frac{5270{\cdot}4}{0{\cdot}9\times4198}=1{\cdot}4\ °\mathrm{C}.$$

The temperature conditions are, therefore, as follows

Hot	80 °C	78·6 °C
Cold	65 °C	20 °C

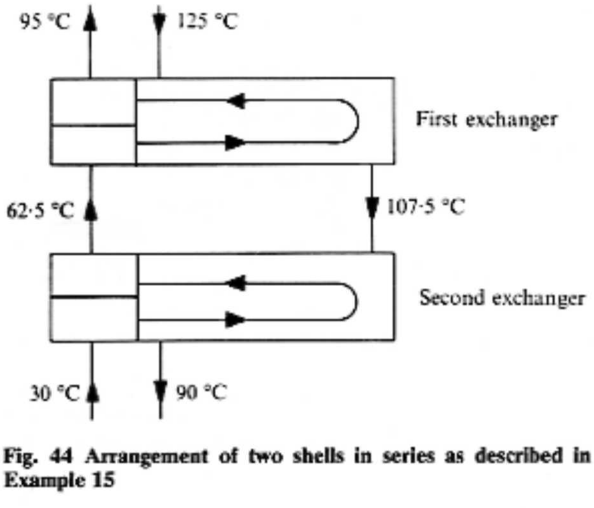


Fig. 44 Arrangement of two shells in series as described in Example 15

To calculate the MTD-value, which in this example is the logarithmic mean temperature difference,

$$\theta_{\ln} = \frac{(78\cdot6-20)-(80-65)}{\ln(78\cdot6-20)/(80-65)} = 32\cdot0\ ^\circ\text{C}.$$

The overall coefficient is the same as that given in Example 11.

$$U = 293\ \text{W/m}^2\ \text{K}.$$

To calculate the heat transfer area based on the inner tube o.d.,

$$A = \frac{Q_T}{U\theta_{\ln}} = \frac{5270\cdot4}{293\times32\cdot0} = 0\cdot562\ \text{m}^2.$$

To calculate the tube length (D_0 is the o.d. of the inner tube),

$$L = \frac{A}{\pi D_0} = \frac{0\cdot562}{\pi\times0\cdot0254} = 7\cdot04\ \text{m}.$$

The tube length is rounded up to 8 m for convenience of manufacture and to allow a safety margin in the calculation. Hence

$$L = 8\cdot0\ \text{m}.$$

(The length could have been rounded up to, say, 25 ft if the manufacture were in British Imperial rather than metric units).

To calculate the Freon-side pressure drop
From Example 1, $Re = 29\,600$ and $u = 0\cdot284$ m/s.
To calculate the friction factor, f, from eqn (63),

$$f = 0\cdot0035 + \frac{0\cdot264}{(29\,600)^{0\cdot42}} = 0\cdot0070.$$

To calculate the pressure drop from eqn (60),

$$\Delta p = \frac{2fL\rho u^2}{D_i} = \frac{2\times0\cdot0070\times8\cdot0\times1304\times(0\cdot284)^2}{0\cdot0203}$$

$$= 580\ \text{Pa},$$

which is an extremely small value.

To calculate the water-side pressure drop
From Example 2, $Re = 39\,500$, $u = 0\cdot472$ and $l = 0\cdot0306$ m.
To calculate f from eqn (63),

$$f = 0\cdot0035 + \frac{0\cdot264}{(39\,500)^{0\cdot42}} = 0\cdot0066.$$

To calculate Δp from equation (60),

$$\Delta p = \frac{2\times0\cdot0066\times8\cdot0\times972\times(0\cdot472)^2}{0\cdot0306} = \cancel{93\cdot4\ \text{Pa}}.$$

747 Pa

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British standards

BS 874: 1973		Methods for determining thermal properties, with definitions of thermal insulating terms.
BS 1588: 1969		The use of thermal insulating materials in the termperature range 95 °C to 230 °C.
BS 2972: 1975		Methods of test for inorganic thermal insulating materials.
BS 3274: 1960		Tubular heat exchangers for general purposes.
BS 3708: 1969		The use of thermal insulating materials between 230 °C and 650 °C.
BS 3958		Thermal insulating materials.
	Part 1: 1970	85 per cent magnesia preformed insulation.
	Part 2: 1970	Calcium silicate preformed insulation.
	Part 3: 1967	Metal mesh faced mineral wool mats and mattresses.
	Part 4: 1968	Bonded preformed mineral wool pipe sections.
	Part 5: 1969	Bonded mineral wool slabs (for use at temperatures above 50 °C).
	Part 6: 1972	Finishing materials; hard setting composition, self-setting cement and gypsum plaster.
BS 4856		Methods for testing and rating fan coil units—unit heaters and unit coolers.
	Part 1: 1972	Thermal and volumetric performance for heating duties without additional ducting.
	Part 2: 1975	Thermal and volumetric performance for cooling duties: without additional ducting.

British Standards may be obtained from BSI, Sales Department, 101 Pentonville Road, London N1 9ND.

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