TABLE 12.9 Geometrical data for typical corrugations (Marston Excelsior Ltd.)

Type	Height (mm)	Thickness (mm)	Pitch (fins/m)	a (m^2/m)	A_1(m^2/m)	A_2(m^2/m)	D_h(mm)
P, R, S, H	3.8	0.20	470	0.00326	1.81	3.41	2.5
P, R	5.1	0.20	550	0.004328	1.776	5.376	2.42
P, R	5.1	0.20	1020	0.00387	1.584	9.984	1.34
P, R, S, H	6.35	0.30	470	0.005175	1.712	5.712	2.79
P, R, H	8.9	0.46	590	0.00615	1.46	9.96	2.16
P, R	8.9	0.61	240	0.00707	1.712	3.912	5.04

In the above table,

a = free flow area per metre width of corrugation

A_1 = (primary surface area per metre width) x (metre length of corrugation)

A_2 = (secondary surface area per metre width) x (metre length of corrugation)

D_h = hydraulic mean diameter

i.e.

$$D_h = \frac{4 \cdot a}{\text{wetted perimeter}}$$

i.e.

$$D_h = \frac{4 \cdot a}{(A_1 + A_2)} \quad \text{(for areas specified above.)}$$

Kays and London have studied a large number of compact heat exchanger matrices and presented their experimental results in the form of generalised graphs. Heat transfer data is plotted as $St.Pr^{2/3}$ against Re, where, St = Stanton number = $h/(G.C_p)$, Pr = Prandtl number = $\mu.C_p/k$, and $Re = G.D_h/\mu$, G = mass velocity (= mass flow rate/Area of cross section), kg/(sm^2.)

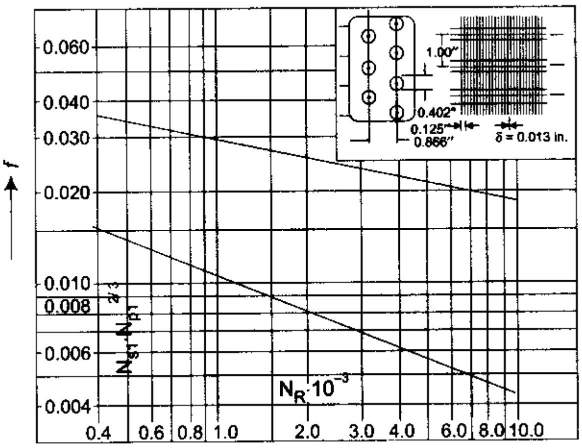
In the same graphs, friction factor, f, is also plotted against Re.

As an example, heat transfer and friction factor characteristics for a particular tube-fin matrix are shown in Fig.12.19.

Pressure drop in plate-fin heat exchangers:

Total pressure drop for the fluid flowing across the heat exchanger is given by:

$$\Delta P = \frac{G^2}{2 \cdot \rho_i} \cdot \left[(K_c + 1 - \sigma^2) + 2 \cdot \left(\frac{\rho_i}{\rho_o} - 1 \right) + f \cdot \frac{A}{A_{min}} \cdot \frac{\rho_i}{\rho_m} - (1 - K_e - \sigma^2) \cdot \frac{\rho_i}{\rho_o} \right] \quad \text{...(12.59)}$$



Tube outside diameter = 0.402 in.
Fin pitch = 8.0 per in.
Flow passage hydraulic diameter, $4r_h$ = 0.01192 ft.
Fin thickness = 0.013 in.
Free-flow area/frontal area, σ = 0.534
Heat transfer area/total volume, α = 179 ft^2/ft^3
Fin area/total area = 0.913
Note: Minimum free-flow area in spaces transverse to flow.

FIGURE 12.19 Heat transfer and friction factor for plate-finned circular tube matrix (Trane Company) where $$\sigma = \frac{A_{min}}{A_{fr}} = \frac{\text{minimum free flow area}}{\text{frontal area}}$$

$$\frac{A}{A_{min}} = \frac{4 \cdot L}{D_h} = \frac{\text{total heat transfer area}}{\text{minimum free flow area}}$$

$$G = \frac{\rho \cdot u \cdot A_{fr}}{A_{min}} = \frac{\rho \cdot u}{\sigma} = \text{mass velocity ...kg/sm}^2$$

K_c and K_e = flow contraction and expansion coefficients, respectively

ρ_i and ρ_o = density at inlet and exit, respectively

$$\frac{1}{\rho_m} = \frac{1}{2} \cdot \left(\frac{1}{\rho_i} + \frac{1}{\rho_o} \right).$$

In Eq. 12.59, on the RHS, the first term inside the square brackets represents the entrance contraction effect, second term—the flow acceleration, the third term—core friction and the fourth term—the exit expansion effect. Core friction drop is generally 90% of the total pressure drop. For liquids, entrance and exit losses are negligible. Values of K_c and K_e are given in Kays and London.

Pressure drop for finned-tube exchangers: Entrance and exit effects are included in the friction factor; therefore, $K_c = K_e = 0$. Then, total pressure drop across the tube bank is:

$$\Delta P = \frac{G^2}{2 \cdot \rho_i} \cdot \left[(1 + \sigma^2) \cdot \left(\frac{\rho_i}{\rho_o} - 1 \right) + f \cdot \frac{A}{A_{min}} \cdot \frac{\rho_i}{\rho_m} \right] \qquad \text{...(12.60)}$$

Here, first term on the RHS is the flow acceleration effect, and the second term is the core friction.

Fig. 12.20 shows a plate-fin exchanger for an ethylene plant and Fig. 12.21 shows another plate-fin exchanger for an air liquefier.

Regenerators:

Regenerators are extensively used in blast furnace stoves, open hearth furnaces, coke manufacture, glass production, for air pre-heating in power plants, in gas turbine systems and in cryogenic plants, in Stirling cycle air (or helium) liquefiers, in cryogenic mini-coolers used for cooling infrared detectors, etc. In a regenerator, hot and cold fluids flow alternately through the regenerator matrix. The matrix may be sand-lime bricks, metal packings, wire screen mesh, lead balls, etc., depending upon application. During the 'hot blow' hot fluid flows through the

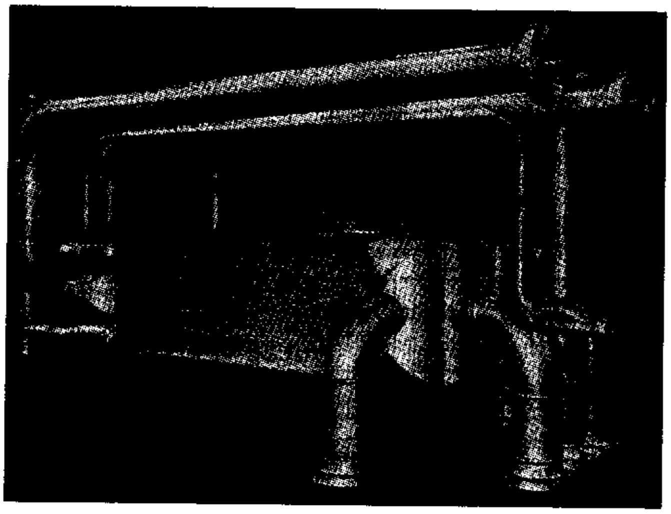


FIGURE 12.20 A heat exchanger assembly with associated pipe work for an ethylene plant (Marston Excelsior Ltd.)

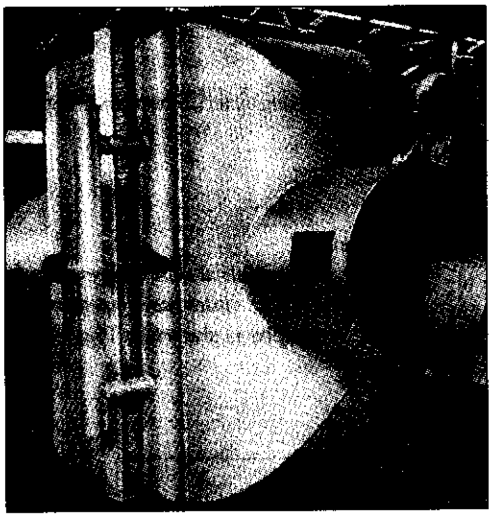


FIGURE 12.21 Air liquefier for a 400 Ton/day Oxygen plant using two, 763 mm × 763 mm blocks in parallel (Marston Excelsior Ltd.)

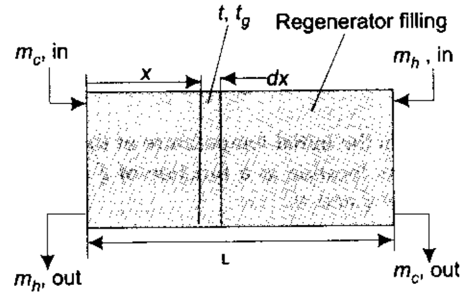


FIGURE 12.22 Periodic-flow heat exchanger (Regenerator)

matrix and the matrix absorbs the heat from the fluid; during the 'cold blow', cold fluid flows through the matrix and the matrix gives up the absorbed heat to the cold fluid, thus heating the fluid. Thus, suitable valving is necessary to alternately switch the hot and cold fluids through the regenerator. In a valved type of exchanger, generally, two identical matrices are provided such that when one matrix is being heated, the other is being cooled. Alternately, regenerator may be of rotary type, where a porous matrix is rotated around its axis cutting the hot and cold fluid lines, thus transferring heat from the hot to the cold fluid.

Analysis of a periodic flow HX is complicated since the matrix and gas temperatures vary with both position and time. A rough outline of the analysis is given below:

Refer to Fig. 12.22, which shows a regenerator diagrammatically. Hot fluid flows through the matrix during the hot blow and heats the matrix. Then, the flow is switched to effect the cold blow and the cold fluid flows through the matrix and gets heated up. Thus, in effect, heat is transferred from the hot fluid to the cold fluid.

We are interested in the gas and matrix temperatures at any location and at any time. These are obtained by writing an energy balance for an element of width dx, shown in the Fig. 12.22. Following notations are used:

M_s = Mass of solid (matrix filling) per unit length, kg/m
M = Mass flow rate of gas, kg/s
C_{ps} = specific heat of solid, J/(kgK)
C_{pg} = specific heat of gas, J/(kgK)
V = free volume per unit length
A = heat transfer area per unit length
ρ = density of gas, kg/m^3
L = length of matrix column
h = convective heat transfer coefficient between the gas and the matrix
t = solid temperature at a given location x
t_g = gas temperature at location x

Writing the heat balance:

Heat transferred by convection between the gas and the solid = Heat stored in the solid

i.e.

$$h \cdot A \cdot (t_g - t) \cdot dx \cdot d\tau = M_s \cdot C_{ps} \cdot dx \cdot \frac{dt}{d\tau} \cdot d\tau \quad ...(12.61)$$

Now, the heat transferred by convection is also equal to the heat stored in the gas contained in length dx *plus* the increase in the enthalpy of the gas as it passes through the element dx.

i.e.

$$h \cdot A \cdot (t_g - t) \cdot dx \cdot d\tau = \rho \cdot C_{pg} \cdot (V \cdot dx) \cdot \frac{dt_g}{d\tau} \cdot d\tau + M \cdot C_{pg} \cdot \left(\frac{dt_g}{dx} \cdot dx \right) \cdot d\tau \quad ...(12.62)$$

Above equations are simplified as:

$$\partial t/\partial\tau = \{(h.A)/(C_{ps}.M_s)\}.(t_g - t) \qquad ...(12.63)$$

$$\partial t_g/\partial x + (\rho.V/M).\partial t_g/\partial\tau = \{(h.A)/(C_{pg}.M)\}.(t - t_g) \qquad ...(12.64)$$

In most of the practical situations, the term $(\rho.V/M)$ is very small and is neglected. Then, making following substitutions

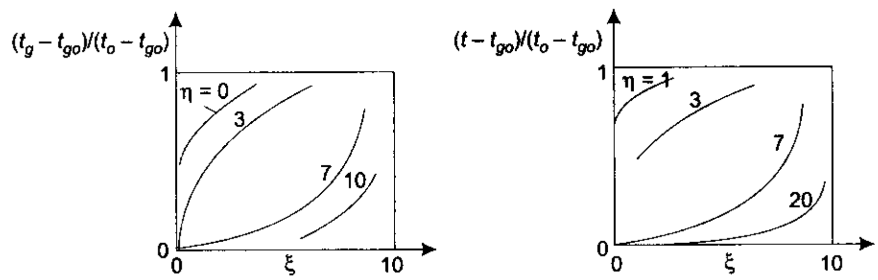
$$\xi = \frac{h \cdot A \cdot x}{C_{pg} \cdot M} \quad \text{and} \quad \eta = \frac{h \cdot A}{C_{ps} \cdot M_s} \cdot \tau$$

the resulting equations are solved with the following boundary conditions:

$t = t_o$ at $\eta = 0$ *(initial solid temperature for all ξ)*

and, $t_g = t_{go}$ at $\xi = 0$ *(inlet gas temperature for all η)*

The results are presented usually in graphical form and the nature of graphs is shown in Fig. 12.23 (a) and (b).



(a) To find gas temperature (b) To find solid temperature

FIGURE 12.23 Gas and solid temperature charts for a regenerator

In these graphs, t_{go} is the initial temperature of the gas, and t_o is the initial temperature of the solid.

Fig. 12.23 (a) presents the dimensionless gas temperature at any location as a function of ξ and η and Fig. 12.23 (b) shows the dimensionless solid temperature as a function of ξ and η.

Effectiveness–NTU relations for regenerator:

Effectiveness of a regenerator is presented as a function of three dimensionless parameters, as follows:

$$\varepsilon = f\left(\text{NTU}_{\text{mod}}, \frac{C_{\min}}{C_{\max}}, \frac{C_r}{C_{\min}}\right) \qquad ...(12.65)$$

where, NTU_{mod} = modified NTU, given by:

$$\text{NTU}_{\text{mod}} = \frac{1}{C_{\min}} \cdot \left[\frac{1}{\left(\frac{1}{h \cdot A}\right)_c + \left(\frac{1}{h \cdot A}\right)_h}\right] \qquad ...(12.66)$$

and, matrix capacity rate is equal to matrix mass rate times the specific heat of the solid.

For the rotary type of regenerator,

$$C_r = \left(\frac{Rev}{s}\right) \cdot (\text{matrix mass}) \cdot C_{ps} \text{ W/K} \qquad \textit{...for rotary type regenerator...}(12.67).$$

For the valved type of regenerator, total mass of both the identical matrices is used, multiplied by valve cycles/s, where period is the interval between 'valve–on-to–off–to–on'.

Kays and London have presented ε–NTU_{mod} graphs for different $C_r/C_{\min}$ ratios (ranging from 1 to infinity), for given $C_{\min}/C_{\max}$ ratios (ranging from 0.5 to 1). Table 12.10 is a sample table showing ε values for $C_{\min}/C_{\max}$ = 1. Fig. 12.24 presents this table in graphical form.

TABLE 12.10 Effectiveness of periodic flow HX (C_{min}/C_{max} = 1)

NTU_{mod}	C_r/C_{min} = 1	C_r/C_{min} = 1.5	C_r/C_{min} = 2	C_r/C_{min} = 5	C_r/C_{min} = Infinity
0	0	0	0	0	0
0.5	0.322	0.328	0.33	0.333	0.333
1	0.467	0.485	0.491	0.499	0.5
1.5	0.548	0.576	0.586	0.598	0.6
2	0.601	0.636	0.649	0.664	0.667
2.5	0.639	0.679	0.694	0.711	0.714
3	0.667	0.712	0.728	0.746	0.75
3.5	0.69	0.738	0.755	0.774	0.778
4	0.709	0.759	0.776	0.796	0.8
4.5	0.724	0.776	0.794	0.814	0.818
5	0.738	0.791	0.809	0.829	0.833
5.5	0.749	0.803	0.821	0.842	0.846
6	0.759	0.814	0.832	0.853	0.857
6.5	0.768	0.824	0.842	0.862	0.867
7	0.776	0.833	0.85	0.87	0.875
7.5	0.784	0.84	0.858	0.878	0.882
8	0.79	0.847	0.865	0.884	0.889
8.5	0.796	0.854	0.871	0.89	0.895
9	0.802	0.859	0.876	0.895	0.9
9.5	0.807	0.864	0.881	0.9	0.905
10	0.811	0.869	0.886	0.904	0.909

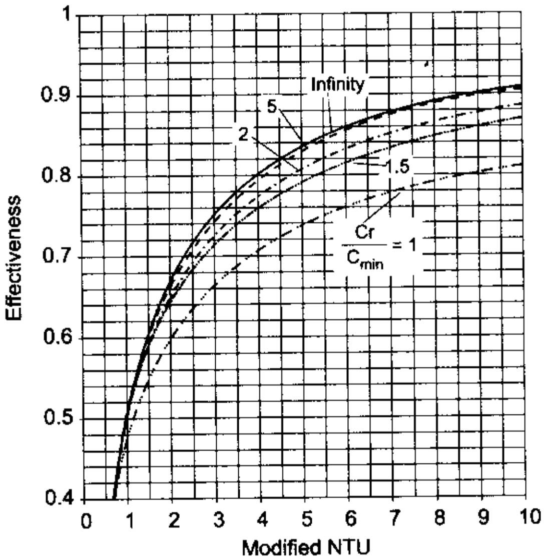


FIGURE 12.24 Effectiveness of a periodic-flow HX (regenerator) for C_{min}/C_{max} = 1

Higher NTU_{mod} ranges (of the order of 100 or more) are generally applicable to regenerators used in cryogenic applications; in such cases, since the effectiveness approaches unity asymptotically, for better clarity, graphs are plotted with $(1 - \varepsilon)$ vs. NTU_{mod}. (See Appendix at the end of chapter).

To calculate NTU_{mod} we need the heat transfer coefficients on the cold and hot fluid sides. We also need the heat transfer area. Heat transfer characteristics in terms of Colburn j factor vs. Reynolds number, and friction factor vs. Reynolds number are presented for many types of matrices by Kays and London. They also provide physical data such as hydraulic diameter, heat transfer area, porosity, etc., for those matrices. As an example, heat transfer characteristics and friction factor data for a randomly stacked, wire screen matrix (used typically, in gas turbine regenerators) are shown in Fig. 12.25 and Fig. 12.26, respectively.

Advantages of regenerators:

(i) High surface density, of the order of 3000 m^2/m^3 (for a 24 mesh screen matrix, typically used in gas turbine regenerators), can be packed into a given volume

(ii) Tends to be 'self-cleaning' because of periodic flow reversals

(iii) Cheaper on per unit heat transfer area basis.

Disadvantages:

(i) Some mixing of hot and cold fluids is unavoidable

(ii) Sealing between the fluids presents some problem if the pressure differential is large.

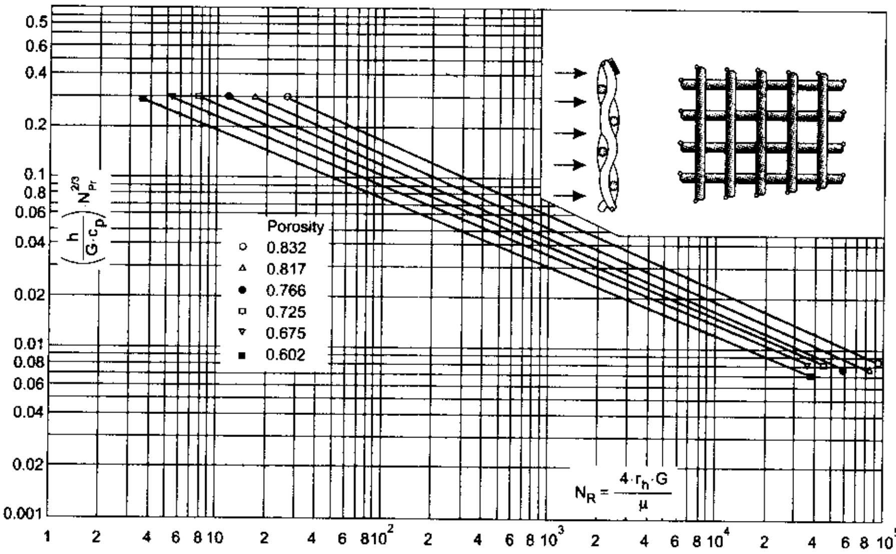


FIGURE 12.25 Colburn j factor vs. Reynolds number for a randomly sacked wire screen matrix

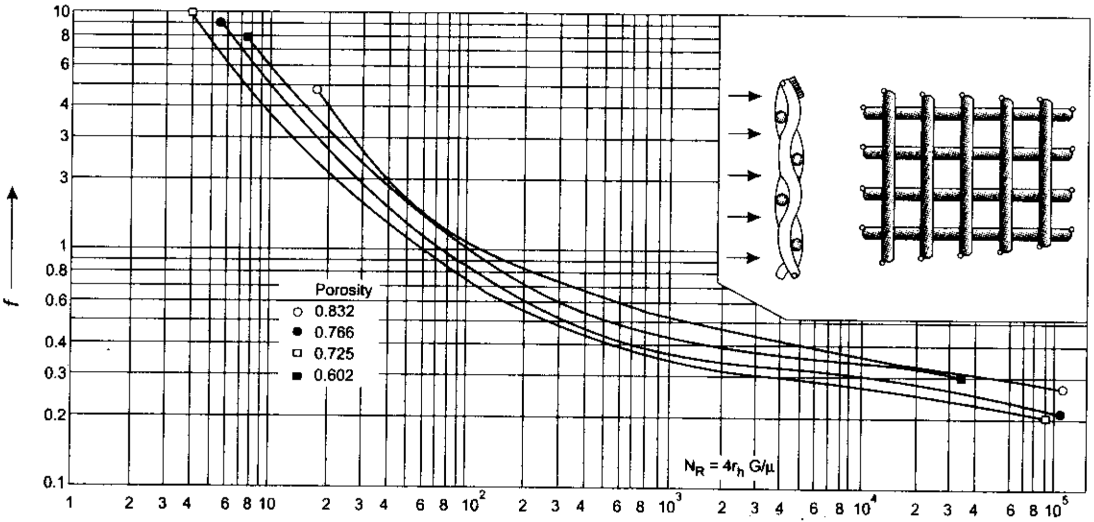


FIGURE 12.26 Friction factor vs. Reynolds number for a randomly stacked wire screen matrix

12.9 Hydro-mechanical Design of Heat Exchangers

So far, we studied thermal design aspects for a heat exchanger. But, from a practical point of view, the pressure drop that occurs when the fluid passes through the heat exchangers and the pumping power required to effect this flow, are also important. We shall only briefly mention about this aspect.

Obviously, flow of fluid through heat exchanger passages involves pressure drop. And, higher the viscosity of the fluid, higher the pressure drop. Total pressure drop in a heat exchanger section is calculated by summing up the following individual pressure drops:

(a) Pressure drops in straight passages and pipe bends, ΔP_f
(b) Pressure drops due to 'end effects', i.e. due to flow contraction and expansion at the ends, ΔP_e
(c) Pressure drops due to flow acceleration (in cases of gases in non-isothermal flow), ΔP_a, and
(d) Pressure drop due to self draught (due to buoyant forces) as a result of change in elevation of flow channels, ΔP_s.

(a) **Pressure drops in straight passages and bends** These are determined by Darcy formula, as explained in the chapter on convection.

$$\Delta P_f = f_D \cdot \frac{L}{D} \cdot \frac{\rho \cdot V^2}{2} \text{ N/m}^2 \qquad \textit{(Darcy formula,...(12.68))}$$

In Eq. 12.68, ρ is density of fluid (kg/m^3) and V is mean velocity of flow (m/s). The friction factor, f_D is determined depending on the Reynolds number, as explained in the chapter on Forced convection.

Effect of bends and valves in the flow lines is generally accounted for by adding an 'equivalent length' for each bend or valve, to the straight length. Equivalent lengths of a few fittings are shown in Table 12.11:

(b) **Pressure drops due to contractions and expansions** ΔP_e is a function of area ratio A_1/A_2, where A_1 is the smaller area.

$$\Delta P_e = f \cdot \frac{\rho \cdot V^2}{2} \text{ N/m}^2 \qquad \textit{(where } f \textit{ is } f_{cont} \textit{ or } f_{expn} \textit{...(12.69))}$$

where, V refers to velocity at smaller cross section.

Values of f_{cont} and f_{expn} are given in Table 12.12; these are shown graphically in Fig. 12.27.

(c) **Pressure drops due to flow acceleration** This pressure drop, in a channel of constant crosssection, is equal to twice the difference in velocity heads, i.e.

$$\Delta P_a = 2 \cdot \left(\frac{\rho_2 \cdot V_2^2}{2} - \frac{\rho_1 \cdot V_1^2}{2} \right) \text{ N/m}^2 \qquad ...(12.70)$$

TABLE 12.11 Equivalent lengths of fittings

Fitting	L_e/D (for turbulent flow only)
45° Elbow	15
90° Elbow (standard radius)	31
90° Elbow (medium radius)	26
90° Elbow (long sweep)	20
90° Square elbow	65
180° Close return bend	75
Swing check valve, open	77
Tee (as eL, entering run)	65
Tee (as eL, entering branch)	90
Couplings, unions	Negligible
Gate valve, open	7
Gate valve, 1/4 closed	40
Gate valve, 1/2 closed	190
Gate valve, 3/4 closed	840
Globe valve, open	340
Angle valve, open	170

TABLE 12.12 Sudden contraction and expansion coefficients for a tube

A_1/A_2	f_{cont}	f_{expn}
0	1	0.5
0.1	0.8	0.45
0.2	0.65	0.42
0.3	0.5	0.38
0.4	0.37	0.34
0.5	0.22	0.28
0.6	0.14	0.23
0.7	0.1	0.2
0.8	0.03	0.12
0.9	0.01	0.07
1	0	0

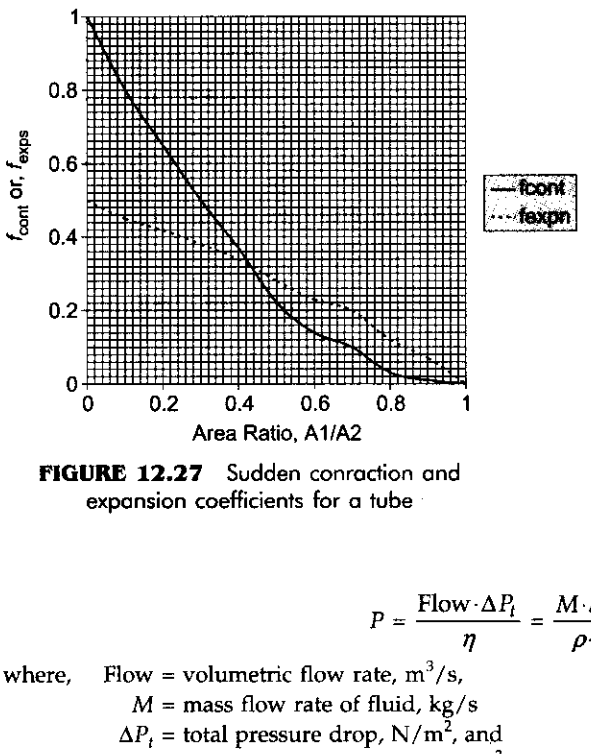


FIGURE 12.27 Sudden conraction and expansion coefficients for a tube

where, subscripts 1 and 2 refer to inlet and outlet, respectively.

(d) **Pressure drop due to self-draught** If the height of the vertical channel (flue) is 'h', ρ_o = density of cold fluid (say, air) and ρ = density of hot fluid (say, flue gas), then pressure drop due to self-draught is given by:

$$\Delta P_s = (\rho_o - \rho)\cdot g\cdot h \text{ N/m}^2 \quad \text{...(12.71)}$$

where, 'g' is the acceleration due to gravity.

ΔP_s is positive for the descending fluid and negative, if the fluid is ascending through the channel. ΔP_s is zero if the heat exchanger is not exposed to ambient air, but is connected in a closed system.

Then, the total pressure drop is given by the summation of all these pressure drops:

$$\Delta P_t = \Delta P_f + \Delta P_e + \Delta P_a + \Delta P_s \quad \text{...(12.72)}$$

Power required to originate fluid flow:

Once the total pressure drop in the system is determined, the power required to circulate the fluid through the system is easily calculated:

$$P = \frac{\text{Flow}\cdot\Delta P_t}{\eta} = \frac{M\cdot\Delta P_t}{\rho\cdot\eta}, \text{ W} \quad \text{...(12.73)}$$

where, Flow = volumetric flow rate, m^3/s,
M = mass flow rate of fluid, kg/s
ΔP_t = total pressure drop, N/m^2, and
ρ = density of liquid or gas, kg/m^3
η = efficiency of pump or fan.

12.10 Summary

Heat exchanger is one of the important pieces of process equipment, used extensively in research as well as industrial applications. Heat exchangers may be of recuperative, regenerative or direct contact type.

In this chapter, we focussed on the thermal design aspects of heat exchangers. First, the method of calculating the overall heat transfer coefficient was explained. Inclusion of fouling resistance is an important aspect of design and this was discussed next.

Calculation of logarithmic mean temperature difference, LMTD, between the two fluid streams exchanging heat, is an important step in the design. Procedure of calculating the LMTD for parallel and counter-flow heat exchangers was explained; for more complicated type of exchangers, such as cross flow or multi-pass shell-and-tube heat exchangers, mean temperature difference is calculated by multiplying the LMTD of a counter-flow HX by a correction factor. Correction factor graphs have been given for a few types of heat exchangers.

Problems in heat exchanger are mainly of two types: (i) design problems where one has to calculate the area of the HX, and (ii) performance problems where one has to calculate the outlet temperatures of both the fluids, given the inlet temperatures. LMTD approach is suitable for the first type of problems, whereas for the second type of problems, ε-NTU approach is recommended, since in this case LMTD approach would require a laborious iterative procedure. ε-NTU relations and graphs for a few important cases have been given. Further, operating-line/equilibrium-line method was also briefly explained.

Compact heat exchangers and regenerators are also used in a variety of applications. Brief mention has been made about these; however, their design is rather more involved and use of proprietary technical information from the suppliers' catalogues will be required. Finally, calculation of pressure drops and the necessary pumping power in a heat exchanger, has been explained.

Selection of heat exchangers for a particular application is a serious task for the engineer and the following aspects must be borne in mind while selecting a heat exchanger:

(i) required heat transfer rate
(ii) necessary pumping power
(iii) type of heat exchanger most suitable, depending upon the process
(iv) materials of construction and fabrication and testing procedures, with due consideration to operating temperatures and pressures
(v) size and weight, depending upon application
(vi) ease of maintenance and servicing
(vii) safety and reliability aspects, and
(viii) cost.

Questions

1. How are heat exchangers classified? Discuss briefly different types of heat exchangers. Why is counter-flow HX better than parallel-flow HX? [M.U.]
2. Draw temperature vs. length profiles for: (i) Condenser (ii) Evaporator (iii) Counter-flow HX with $C_h = C_c$ [M.U.]
3. What is overall heat transfer coefficient? What is its importance? Derive an expression for overall heat transfer coefficient for a tubular HX based on inner surface area. [M.U.]
4. Explain the terms: Fouling factor, Effectiveness, NTU and LMTD. [M.U.]
5. Write short notes on correction factor charts for cross-flow heat exchangers. [M.U.]
6. Starting from fundamentals, derive an expression for the mean temperature difference for counter-flow HX in terms of inlet and outlet temperatures of hot and cold fluids. [M.U.]
7. Derive an expression for the LMTD of a parallel-flow HX. State clearly the assumptions. [M.U.]
8. Derive an expression for the effectiveness of a counter-flow HX when capacity rate of hot fluid is more than that of cold fluid. Hence show that effectiveness of a condenser is given by:
$$\varepsilon = 1 - \exp(-NTU) \quad ...[M.U.]$$
9. Starting from basics, derive an equation for the effectiveness of a parallel-flow HX in terms of NTU and capacity ratio. Also, show that when capacity ratio is 1, effectiveness is given by:
$$\varepsilon = (½).\{1 - \exp(-2.NTU)\} \quad ...[M.U.]$$
10. Prove that for a counter-flow HX, when $C_{min}/C_{max} = 1$,
$$\varepsilon = NTU/(1 + NTU). \quad [M.U.]$$
11. Compare LMTD and ε-NTU methods of solving heat exchanger problems.
12. Using the operating-line/equilibrium-line method, derive an expression for the effectiveness of a counter-flow HX. Assume $C_c > C_h$.
13. Write a short note on compact heat exchangers and regenerators.

Problems

1. A copper pipe (k = 350 W/mK) of 17.5 mm ID and 20 mm OD conveys water and the oil flows through the annular passage between this pipe and a steel pipe. On the water side, the film coefficient is 4600 W/(m^{2}K) and fouling factor is 0.00034 m^{2}K/W. The corresponding values for the oil side are 1200 W/(m^{2}K) and 0.00086 m^{2}K/W. Calculate the overall heat transfer coefficient between the water and oil, based on outside surface area of inner pipe.

2. In a shell and tube counter-flow HX, water flows through a copper tube (20 mm ID, 23 mm OD), while oil flows through the shell. Water enters at 20°C and comes out at 30°C while oil enters at 75°C and comes out at 60°C. The water and oil side film coefficients. are: 4500 and 1250 W/(m^{2}K), respectively. Thermal conductivity of tube wall is 355 W/(mK). Fouling factors on water and oil sides are: 0.0004 and 0.001 m^{2}K/W, respectively. If the length of tube is 2.4 m, calculate the overall heat transfer coefficient and rate of heat transfer. [M.U.]
3. Saturated seam at 120°C is condensing on the outer surface of a single pass HX. The overall heat transfer coefficient is 1600 W/(m^{2}K). Determine the surface area of the HX required to heat 2000 kg/h of water from 20°C to 90°C. Also, determine the rate of condensation of steam in kg/h. Assume latent heat of steam to be 2195 kJ/kg. [M.U.]
4. A HX is required to cool 55,000 kg/h of alcohol from 66°C to 40°C using 40000 kg/h of water entering at 5°C. Calculate (i) the exit temperature of water (ii) heat transfer (iii) surface area required for: (a) parallel-flow type (b) counter-flow type of HX.
   Take overall heat transfer coefficient U = 580 W/(m^{2}K). C_p (alcohol) = 3760 J/(kgK) and C_p (water) = 4180 J/(kgK).
5. In a counter-flow double pipe HX, water flow rate is 1300 kg/h. and it enters at 15°C. It is heated by oil, C_p = 2 kJ/kgK; oil flow rate is 550 kg/h. Oil inlet temperature is 95°C. Overall U = 800 W/m^{2}K. Surface area of HX: 1.34 m^2. Table of NTU–ε is given as follows:

Capacity ratio, R	NTU	Effectiveness
0.202	3	0.93
0.202	4	0.96

   Find out ε, NTU and outlet temperatures. [M.U.]
6. In a gas turbine installation, a counter-flow HX, has hot exhaust gas outlet at 330°C, and air outlet at 460°C. For each element of HX, $(dQ/(dT.dX))$ is uniform and is equal to C and $C.L$ = 52.3 kW/K. Capacity rate for hot fluid = 21.76 kW/K and for cold fluid = 19.04 kW/K. Temperature variation along the length is linear for both fluids. Calculate temperatures at entry. [M.U.]
7. A one shell, 2-tube pass steam condenser, has 2000 tubes of 20 mm diameter, with cooling water entry at 20°C, flow rate 3000 kg/s; U = 6890 W/m^{2}K. Total heat to be transferred, $Q = 2.331 \times 10^8$ W. Steam condenses at 50°C. Determine tube length per pass using NTU method. Given that at 0.6 and 0.64 effectiveness, NTU is 0.78 and 0.82. [M.U.]
8. A water pre-heater of ID:3.2 cm, OD:3.52 cm, is heated by steam at 180°C. Water flows through pipe at a velocity of 1.2 m/s. 'h' on steam side:11,000 W/m^{2}K; water is heated from 25°C to 95°C. k of pipe material: 59 W/mK. Properties of water at 60°C are given. Calculate the length required. Use appropriate empirical relation. Data: $\mu = 4.62 \times 10^{-4}$ kg/ms; k = 0.653 W/mK; C_p = 4200 J/kgK. [M.U.]
9. Consider a HX for cooling oil entering at 180°C, by water entering at 25°C; mass flow rates of oil and water are: 2.5 and 1.2 kg/s, respectively. Area: 16 m^2. Specific heat data for oil and water and overall U are given:
   Data: C_p of oil = 1900 J/kgK; C_p of water = 4184 J/kgK; U = 285 W/m^{2}K.
   Calculate outlet temperature of oil and water for parallel and counter-flow HX. [M.U.]
10. In a shell-and-tube HX, 50 kg/min of furnace oil is heated from 10°C to 90°C. Steam at 120°C flows through the Shell and oil flows inside the tube. Tube size: 1.65 cm ID and 1.9 cm OD. Film coefficients on oil and steam sides are: 85 and 7420 W/(m^{2}K). Find the number of passes and number of tubes in each pass if the length of each tube is limited to 2.85 m. Velocity of oil is limited to 5 cm/s. Density and specific heat of oil are 900 kg/m^3 and 1970 J/(kgK), respectively. [M.U.]
11. Water at a rate of 4080 kg/h is heated from 35°C to 75°C by an oil of C_p = 1.9 kJ/(kgK). The HX is of counter-flow, double pipe design. The oil enters at 110°C and leaves at 75°C. Determine: (i) mass flow rate of oil (ii) area of HX necessary to handle this load, if overall heat transfer coefficient, U = 320 W/(m^{2}K). [M.U.]
12. A steam condenser, condensing at 70°C has to have a capacity of 100 kW. Water at 20°C is used and the outlet water temperature is limited to 45°C. If the overall heat transfer coefficient is 3100 W/(m^{2}K), determine the area required. If the inlet water temperature is increased to 30°C, determine the increased flow rate of water to maintain the same outlet temperature. [M.U.]
13. Water enters a counter-flow, double pipe HX at 38°C flowing at a rate of 0.76 kg/s. It is heated by oil (C_p = 1.88 kJ/kgK) flowing at a rate of 0.152 kg/s from an inlet temperature of 116°C. For an area of 1.3 m^2 and an overall heat transfer coefficient of 340 W/(m^{2}K), determine the total heat transfer rate. Take C_p for water = 4.170 kJ/kgK.
    Given: expression for effectiveness of double pipe, counter-current HX:

$$\varepsilon = \frac{1-\exp(-NTU(1-C))}{1-C\cdot\exp(-NTU(1-C))}$$ [M.U.]

14. A refrigerator is designed to cool 250 kg/h of hot liquid (C_p = 3350 J/kgK) at 120°C using a parallel-flow arrangement. 1000 kg/h of cooling water is available at a temperature of 10°C. If overall U = 1160 W/(m^{2}K) and the surface area of HX is 0.25 m^2, calculate the outlet temperatures of the fluids and also the effectiveness of the HX. [M.U.]
15. A steam condenser, condensing at 100°C has a capacity of 150 kW. Water at 25°C is used and the outlet water temperature is 35°C. If the overall heat transfer coefficient is 3000 W/(m^{2}K), determine the area required for the HX. [M.U.]
16. A parallel-flow HX has hot and cold water streams running through it and has following data: m_h = 10 kg/min, m_c = 25 kg/min, $C_{ph} = C_{pc}$ = 4.18 kJ/(kgK), T_{h1} = 70°C, T_{h2} = 50°C, T_{c1} = 25°C. Individual heat transfer coefficients on both sides are 60 W/(m^{2}K), each. Calculate: (i) area of HX (ii) exit temperatures of hot and cold fluids, if hot water flow rate is doubled. [M.U.]
17. Steam at atmospheric pressure enters the shell of a surface condenser in which water flows through a bundle of tubes of diameter = 25 mm at a rate of 0.05 kg/s. The inlet and outlet temperatures of water are 15°C and 70°C, respectively. Condensation of steam takes place on the outside surface of the tube. If U = 230 W/(m^{2}K), using NTU method, find: (i) effectiveness of the HX (ii) length of tube required, and (iii) rate of steam condensation. [M.U.]
18. An economiser in a boiler has water flowing inside the pipes and hot gases on the outside, flowing across the pipes. The flow rate of the gases is 2000 tonne/h and they are cooled from 390°C to 200°C and their specific heat is 1005 J/(kgC). Water is heated under high pressure from 100°C to 220°C. Assuming an overall heat transfer coefficient of 35 W/(m^{2}C), determine the area required. The correction factor, F = 0.8. [M.U.]
19. Hot oil is being cooled from 200°C to 130°C in a parallel-flow HX by water entering a 25°C and exiting at 60°C. Determine the outlet temperatures of both the streams if the HX is made counter-flow. [M.U.]
20. A shell and tube HX with two shell passes and 8 tube passes has ethyl alcohol (C_p = 2670 J/kgC) flowing inside the tubes, and water (C_p = 4190 J/kgC) flows through the shell. Ethyl alcohol enters at 25°C and leaves at 75°C with a flow rate of 2 kg/s whereas water enters at 95°C and leaves at 45°C. Overall heat transfer coefficient U = 850 W/(m^{2}K). Determine the surface area required for the HX.

Appendix

In this Appendix to Chapter 12, some more information on compact heat exchangers and regenerators is given.

Example A12.1. Air at 2 atm and 400 K flows at a rate of 5 kg/s, across a finned circular tube matrix shown in Fig. 12.19. Dimensions of the heat exchanger matrix are: 1 m (W) × 0.6 m (Deep) × 0.5 m (H), as shown in Fig. A12.1. Find: (a) the heat transfer coefficient (b) the friction factor, and (c) ratio of core friction pressure drop to the inlet pressure.

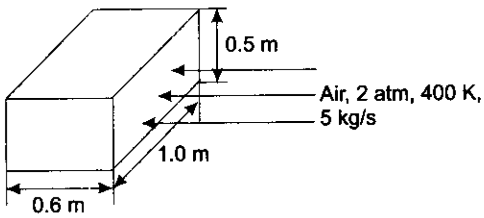


FIGURE A12.1 Configuration of compact heat exchanger for Example A12.1

Solution.

Data:

m = 5 kg/s ...*mass flow rate*

A_{fr} = 0.5 m^2 ...*frontal area*

L = 0.6 m ...*length of flow*

Physical properties of air at 2 atm and 400 K:

$\rho := 0.883\cdot 2$ kg/m^3 i.e. $\rho := 1.766$ kg/m^3 $\mu := 2.29 \times 10^{-5}$ kg/ms $C_p := 1013$ J/kgK $P_r := 0.703$

From Fig. 12.19, we have:

$$\sigma := 0.534 \text{ where, } \sigma = \frac{A_{min}}{A_{fr}}$$

and,
$$D_h := \frac{0.01192}{3.28} \text{ m}$$

i.e.
$$D_h := 3.634 \times 10^{-3} \text{ m}$$ (*hydraulic diameter*)

Then,

Mass velocity:

$$G = \frac{m}{A_{min}} \quad \text{and, } A_{min} = \sigma\cdot A_{fr}$$

i.e. $$G := \frac{m}{\sigma \cdot A_{fr}}$$

i.e. $$G = 18.727 \text{ kg/sm}^2$$ *(mass velocity)*

Reynolds number:

$$Re := \frac{G \cdot D_h}{\mu}$$

i.e. $$Re = 2.972 \times 10^3$$ *(Reynolds number)*

Then, from Fig. 12.19, for $Re = 2972$, we get:

$$\frac{h}{G \cdot C_p} \cdot Pr^{\frac{2}{3}} = 0.0069$$

(a) **And, heat transfer coefficient:**

$$h := \frac{0.0069 \cdot G \cdot C_p}{Pr^{\frac{2}{3}}}$$

i.e. $$h = 165.557 \text{ W/(m}^2\text{C)}$$ ***(heat transfer coefficient.)***

(b) **Friction factor:**

From Fig. 12.19, for $Re = 2972$, we get:

$$f = 0.024$$ ***(friction factor)***

(c) **Pressure drop:**

$$\Delta P_f = f \cdot \frac{G^2}{2 \cdot \rho} \cdot \frac{A}{A_{min}} \text{ N/m}^2$$ *(fricitional pressure drop)*

Now, $$\frac{A}{A_{min}} = \frac{4 \cdot L}{D_h}$$

i.e. $$\frac{A}{A_{min}} = 660.403$$

Therefore,

$$\Delta P_f := f \cdot \frac{G^2}{2 \cdot \rho} \cdot 660.403 \text{ N/m}^2$$ *(core friction pressure drop)*

i.e. $$\Delta P_f = 1.574 \times 10^3 \text{ N/m}^2$$ ***(core friction pressure drop.)***

And, $$\frac{\Delta P_f}{P} = \frac{1574}{2 \cdot (1.0132 \times 10^5)} = 0.78\%$$

i.e. frictional pressure drop is 0.78% of the inlet pressure.

Fig. A12.2, A12.3, and A12.4 show data for three more compact heat exchanger matrices, from Kays and London.

Fig. A12.5 shows data for crossed rod matrices, random stacking ($d = 0.375$ in.) used in regenerators (Kays and London):

Fig. A12.6 gives data for an infinite, randomly stacked sphere matrix, with porosity varying from 0.37 to 0.39. (Kays and London)

Data of Fig. A12.6 is given in tabular form in Table A12.1.

Analysis of regenerators by NTU method:

As explained earlier, in a regenerator, the same space is alternately occupied by the hot and cold fluids; regenerator matrix stores the 'heat' during the flow of hot fluid (i.e. during 'hot blow') and rejects this heat to the cold fluid during the flow of cold fluid through the regenerator matrix (i.e. during 'cold blow'). Temperatures of the gas as well as of the matrix solid are functions of both position and time. After sufficiently long time, some sort of steady state is reached, and the same temperature distribution is repeated in each cycle of operation.

An NTU analysis, similar to the one done for heat exchangers, proceeds as follows, with the assumption that the same mass flow rate of gas is maintained during both the hot and cold blows. (Ref: Cryogenic Systems by R.F. Barron).

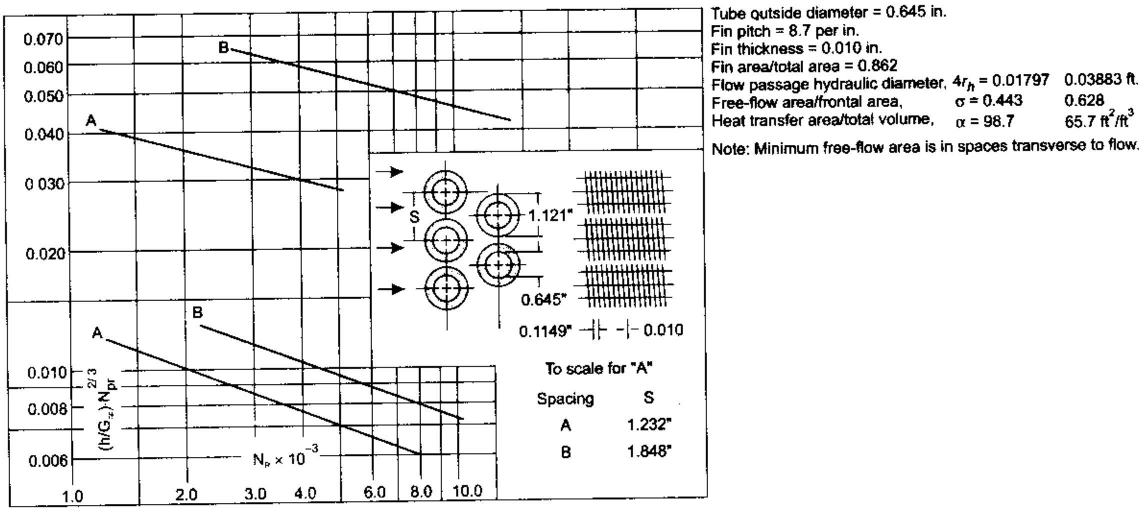


FIGURE A12.2 Heat transfer coefficient and friction factor data for finned circular tube matrix (surface CF 8.7–5/8J)

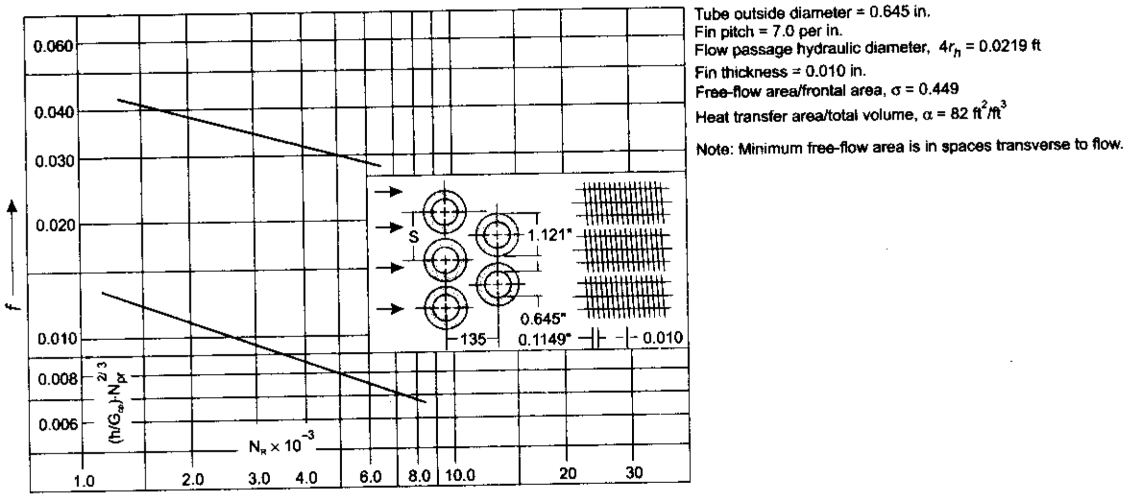
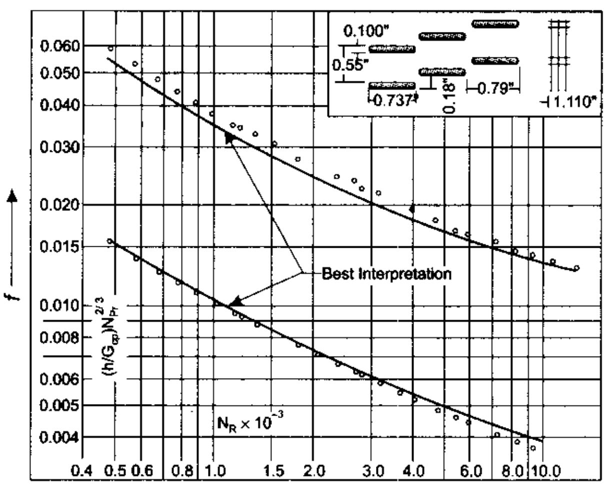


FIGURE A12.3 Heat transfer coefficient and friction factor data for finned circular tube matrix (surface CF–7.0–5/8J)

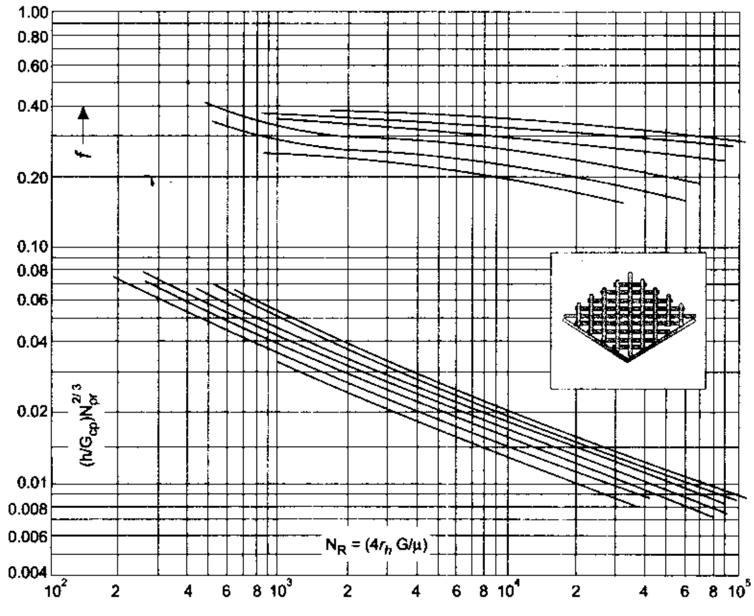
Following notations are used:

m_s = mass of solid (matrix filling) in regenerator, kg
m = mass flow rate of gas through regenerator, kg/s
c_s = specific heat of solid, J/(kgK)
c_p = specific heat of gas, J/(kgK)
L = length of regenerator, m
A_h = total heat transfer area of solid material in regenerator, m^2
ρ = density of gas, kg/m^3
T = temperature of gas at location x and time τ
T_s = temperature of solid material at location x and time τ



Fin pitch = 9.1 per in.
Flow passage hydraulic diameter, $4r_h$ = 0.01380 ft
Fin metal thickness = 0.004 in., copper
Free-flow area/frontal area, σ = 0.788
Total heat transfer area/total volume, α = 224 ft^2/ft^3
Fin area/total area = 0.813

FIGURE A12.4 Heat transfer coefficient and friction factor data for finned flat tube matrix (surface 9.–0.737-5)



		Porosity p	Transverse pitch X_t
I	×	0.832	4.675
II	△	0.817	4.292
III	●	0.766	3.356
IV	□	0.725	2.856
V	▽	0.675	2.417
VI	○	0.602	1.974
VII	■	0.500	1.571

FIGURE A12.5 Crossed rod matrix, random stacking (rod diameter d = 0.375 in.)

h_c = heat transfer coefficient between the solid material and gas, W/(m^2/C).

Now, applying the First law to a differential element of gas:

heat transferred to or from the gas = change in enthalpy of gas in a length dx, as it flows through the regenerator i.e.

$$h_c(T - T_s)(A_h/L)\,dx = -\,mc_p\,(\partial T/\partial x)\,dx \qquad ...(1)$$

Here, change in energy stored in the gas within the differential element is neglected. (This is true for cryogenic regenerators.)

Again, applying the First law to a differential element of solid material, we have:

heat transferred to or from the solid material = change in energy stored within the solid material

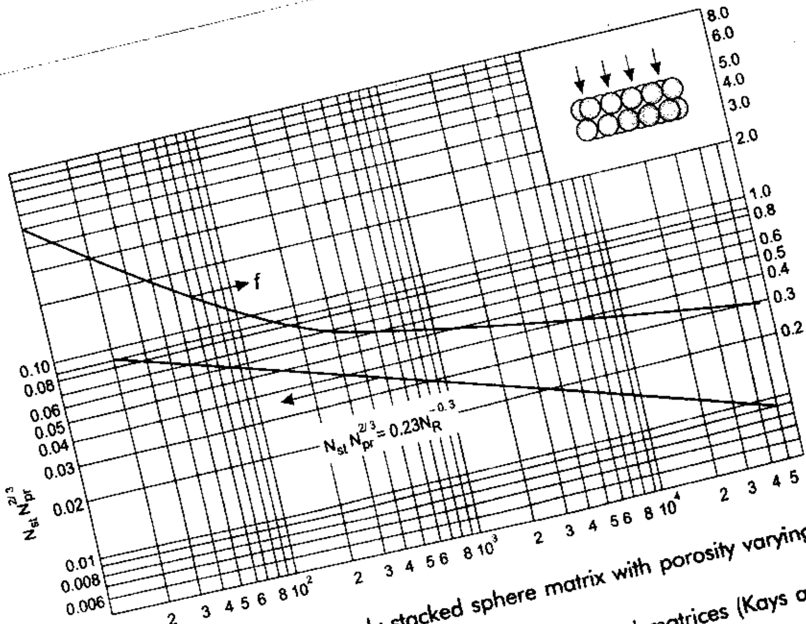


FIGURE A12.6 Data for an infinite, randomly stacked sphere matrix with porosity varying from 0.37 to 0.39

TABLE A12.1 Heat transfer and friction data for sphere bed matrices (Kays and London) (Random packing, p = 0.37 to 0.39)

Reynolds number, N_R	$N_{St} N_{Pr}^{2/3}$	f
50000	0.0089	0.30
20000	0.0118	0.34
10000	0.0144	0.37
5000	0.0178	0.41
2000	0.023	0.47
10000	0.029	0.52
500	0.0355	0.59
200	0.046	0.80
100	0.056	1.10
50	0.069	1.65
20	0.091	3.0
10	0.112	5.2

i.e.

$h_c(T - T_s)(A_h/L)\,dx = m_s (dx/L)\, c_s (\partial T_s/\partial \tau)$ ...(2)

Solving Eqs. 1 and 2, we get the partial differential equation for the temperature of gas flowing through the regenerator:

$[\partial^2 T/(\partial x.\partial \tau)] + [h_c A_h/(m\, c_p\, L)]\,(\partial T/\partial \tau) + [h_c\, A_h/(m_s c_s)]\,(\partial T/\partial x) = 0$ ...(3)

From Eq. (3), we observe that two important dimensionless quantities are involved in the analysis of a regenerator, i.e.

$N_{tu} = h_c\, A_h/(m\, c_p)$ = Number of heat transfer units,

and

$F_n = h_c\, A_h/(m_s\, c_s f)$ = Frequency number, where f = $1/P$ = frequency of switching the hot and cold stream, P = heating or cooling period.

$f.\tau$ = dimensionless time.

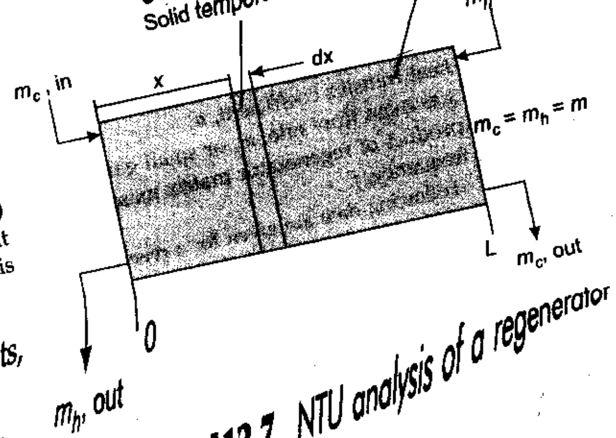


FIGURE A12.7 *NTU analysis of a regenerator*