Compact Heat Exchangers

THIRD EDITION

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Compact Heat Exchangers

Preface to the Third Edition

It is now 19 years since the second edition of *Compact Heat Exchangers* was published. In the intervening years manufacturing techniques have been developed to fabricate some of the older compact surface configurations using high-temperature materials, new surface configurations with superior flow characteristics have been manufactured, and the area per unit volume has been increased substantially. The research program which generated the original data on which this book is based was slowed down and eventually terminated, but a number of the newer surfaces were tested using essentially the same techniques as were used for the earlier work.

In the meantime new research on some of the theoretical solutions for flow in the simple geometries has rendered obsolete some of the solutions presented in the second edition. The same thing can be said of the solutions for the transient behavior of heat exchangers.

The availability of new data and more modern solutions suggested that the time was appropriate for a new edition. This edition does not differ radically from its predecessor but it does contain the basic test data for 11 new surface configurations, including some of the very compact ceramic matrices. In addition to modernization of the theoretical solutions and correlations for simple geometries, and the transient solutions, a number of other improvements will be found. Finally, the slow conversion (at least in the United States) to the Système Internationale (SI) system of units suggested that the time had come to make that conversion in *Compact Heat Exchangers*. Since the English system is apparently destined to disappear only slowly in the United States, it was decided to introduce a dual system of units in the new edition. So all dimensions are given in both systems, and the fluid properties in the Appendix are likewise presented in both systems.

A unique feature of *Compact Heat Exchangers* has always been that virtually all of the basic test data originate from a single research program under the supervision of the authors. There is thus no question about the comparability of the test results of one surface to another. In recent years additional data have been obtained by others, but the authors have chosen to maintain the original tradition so that there is almost complete internal consistency.

W. M. Kays

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Preface to the Second Edition

For many years the only generally available basic heat transfer and flow-friction data of sufficient accuracy for heat-exchanger design was for flow through and over banks of circular tubes. The need for small-size and lightweight heat exchangers in all varieties of powered vehicles from automobiles to spacecraft, as well as in a multitude of other applications, has resulted in the development of many heat transfer surfaces that are much more compact than can be practically realized with circular tubes. In addition, many of these surfaces possess other characteristics that are superior to circular tubes. However, lack of basic heat transfer and flow-friction design data, and a lack of understanding of the basic mechanisms involved, for a long period of time restricted their use to heat exchangers that could be developed by cut-and-try methods. It ultimately became apparent that rationally optimized heat-exchanger design, the development of new surfaces of superior characteristics, and the development of methods of fabrication of compact surfaces for high-temperature service could only take place after the basic characteristics of the already existing surfaces were known and understood.

Recognizing the need for such data the U.S. Navy Bureau of Ships initiated in 1945 a test program at the Naval Engineering Experiment Station, Annapolis, Maryland. In 1947, the Office of Naval Research, in cooperation with the Bureaus of Ships and Aeronautics, extended this work by establishing a similar program at Stanford University. Later the Atomic Energy Commission joined in support.

A number of manufacturers provided test cores for these investigations, and the authors acknowledge especially the cooperation of the Harrison Radiator Division of General Motors, Lockport, New York; the Modine Manufacturing Company, Racine, Wisconsin; The Trane Company, LaCrosse, Wisconsin; The AiResearch Manufacturing Company, Los Angeles, California; and The Ferrotherm Company, Cleveland, Ohio.

Most of the test cores were of low-temperature construction employing soldering or brazing techniques. However, the primary objective of this program was to investigate the effects of geometry on convective heat transfer and flow-friction performance, with the hope that the geometrical advantages would provide incentive for the development of high-temperature fabrication techniques and of new superior surfaces. Since the first publications of the results of the program, both kinds of developments have indeed occurred.

The American Society of Mechanical Engineers published the first results of the program in 1951 in a monograph entitled *Gas Turbine Plant Heat Exchangers—Basic Heat Transfer and Flow Friction Design Data,* by W. M. Kays, A. L. London, and D. W. Johnson. In 1955, *Compact Heat Exchangers,* by W. M. Kays and A. L. London, was published; it contained a considerable additional body of basic data from the test program, as well as data from other investigators. Following the publication of *Compact Heat Exchangers,* the test program was continued, and new test cores were obtained, some of which were developed directly as a result of the earlier work. This second edition of *Compact Heat Exchangers* contains all the new basic data that have been obtained, as well as extensive revisions and additions to the chapters on analytic solutions for flow in tubes, an extension of the chapter on heat exchanger design theory, and a new chapter on the transient behavior of heat exchangers; various other sections have been brought up to date in the light of more recent research. The basic data section has been expanded to include the characteristics of 25 new surfaces, and this section, reporting the characteristics of more than 90 surfaces, remains the real core of the book.

Although too numerous to name specifically, the authors take this opportunity to acknowledge the assistance over the past 15 years of the approximately 60 Stanford University mechanical engineering students who participated in various phases of the test program. Without their assistance this book could never have been written.

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Nomenclature

Most of the nomenclature is defined as it is introduced or else is obvious from the context of its use. However, it is summarized here for convenience.

Any consistent dimensioning system may be used. All the heat transfer and flow-friction parameters are presented in nondimensional form so that a shift to a preferred system of dimensions presents no complications.

Roman Letter Symbols

A	Exchanger total heat transfer area on one side
A_c	Exchanger minimum free-flow area, or pA_{fr} for matrix surfaces
A_f	Exchanger total fin area on one side
A_{fr}	Exchanger total frontal area
A_k	Cross-sectional area for longitudinal conduction
a	Plate thickness
a	Short side of a rectangular flow passage
b	Plate spacing
b	Long side of a rectangular flow passage
C	Flow-stream capacity rate (Wc_p)
C_c	Flow-stream capacity rate of cold-side fluid
C_h	Flow-stream capacity rate of hot-side fluid
C_L	Coupling-liquid capacity rate
$C_{\min}$	Minimum of C_c or C_h
$C_{\max}$	Maximum of C_c or C_h
C_r	Rotor capacity rate of a rotating periodic flow exchanger (rotor mass times specific heat times r/h)
C_r^*	Rotor capacity-rate ratio ($C_r/C_{\min}$), dimensionless
$\underline{C_r^*}$	($C_r^* \theta_r / \theta_{d,\min}$), dimensionless
$\overline{C}$	Fluid heat capacity within exchanger ($C\theta_d$)
$\overline{C}_{\min}$	$C\theta_d$ for minimum-capacity-rate fluid
$\overline{C}_w$	Wall total heat capacity (exchanger core mass times specific heat of core material)
$\overline{C}_w^*$	$\overline{C}_w/\overline{C}_{\min}$, dimensionless
c	Specific heat
c_p	Specific heat at constant pressure
c_v	Specific heat at constant volume

D	Inside diameter of a circular tube
D_h	Hydraulic diameter of any internal passage ($D_h = 4r_h = 4A_cL/A$)
d	Outside diameter of a tube in a tube bundle, crossed-rod matrix, or a pin in pin-fin surface
E	Friction power expended per unit of surface heat transfer area [see Eq. (1-2)]
F_G	Correction factor to log-mean rate equation, dimensionless
f	Mean friction factor, defined on the basis of mean surface shear stress [Eq. (1-6)]
f	Fuel-air ratio
f_x	Local friction factor, defined on the basis of local surface shear stress
f_{app}	Apparent mean friction factor [Eq. (6-6)]
G	Exchanger flow-stream mass velocity (W/A_c)
g_c	Proportionality factor in Newton's second law
H/C	Hydrogen-carbon ratio for hydrocarbon fuels
h	Unit conductance for thermal-convection heat transfer
K_c	Contraction loss coefficient for flow at heat exchanger entrance [Eq. (5-1)], dimensionless
K_d	Momentum flux correction factor, [Eq. (6-7)], dimensionless
K_e	Expansion loss coefficient for flow at heat exchanger exit [Eq. (5-2)], dimensionless
k	Unit thermal conductivity
L	Total heat-exchanger flow length; also flow length of uninterrupted fin
l	Fin length from root to center
M	Molecular weight
m	A fin effectiveness parameter $\sqrt{2h/k\delta}$, $\sqrt{4h/kd}$
m	Exponent in Eq. (4-2)
m_o	Slope of operating line (C_c/C_h), dimensionless
n	Number of passes in a multipass heat exchanger
n	Exponent in Eq. (4-1)
P	Pressure
p	Porosity of a matrix surface, dimensionless
q	Heat transfer rate
q''	Heat flux, heat transfer rate per unit of surface area
R	Universal gas constant
R	Heat transfer resistance
R_c	Resistance on the cold-fluid side of a heat exchanger
R_h	Resistance on the hot-fluid side of a heat exchanger
R^*	Heat transfer resistance ratio, (R on C_{min} side)/(R on C_{max} side)
r	A radial coordinate
r_h	Hydraulic radius (A_cL/A) (or $pA_{fr}L/A$ for matrix surfaces)
r_i	Inner radius of an annulus or inner radius of a circular fin
r_o	Outer radius of an annulus or outer radius of a circular fin
r^*	r_i/r_o
T	Absolute temperature
t	Temperature to any arbitrary scale
t_c	Cold-fluid-side temperature
t_h	Hot-fluid-side temperature
U	Unit overall thermal conductance
V	Velocity
V	Volume

v	Specific volume
W	Mass flow rate
X	Parameter in the log-mean rate equation approach to heat exchanger design
X	Parameter in Fig. 3-6
X_c	Specific-heat correction factor for humidity and products of combustion
X_d	Density correction factor for humidity and products of combustion
x	Axial flow coordinate
x^*	Axial flow coordinate (x/L), dimensionless
x_l	Longitudinal-tube pitch ratio in a circular tube bank (Fig. 7-5), dimensionless
x_t	Transverse-tube pitch ratio in a circular tube bank (Fig. 7-5), dimensionless
Y	Parameter in Fig. 3-6
Z	Parameter in the log-mean rate equation approach to heat exchanger design
Z	Influence coefficient for annulus heat transfer [Eqs. (6-1) and (6-2)]

Greek Letter Symbols

α	Ratio of total transfer area on one side of the exchanger to total volume of the exchanger For matrix surfaces $\alpha = A/A_{fr}L$ for either one side or both sides
α^*	Aspect ratio of a rectangular flow passage (b/a), dimensionless
β	Ratio of total heat transfer area on one side of a plate-fin heat exchanger to the volume between the plates on that side
Δ	Denotes difference
δ	Fin thickness
δ	Denotes difference
ε	Exchanger effectiveness, dimensionless [Eq. (2-6)]
ε_p	Effectiveness of one pass of a multipass heat exchanger, dimensionless
ε_f^*	Outlet-fluid temperature response to a step change in one of the fluid inlet temperatures, dimensionless
ε_W^*	Wall temperature response at the fluid outlet section to a step change in fluid inlet temperature, dimensionless
Γ	Parameter defined by either Eq. (2-15) or Eq. (2-19), dimensionless
Γ'	Parameter defined by Eq. (2-16), dimensionless
λ	Longitudinal conduction parameter defined by Eq. (2-25), dimensionless
η_f	Fin temperature effectiveness, dimensionless [Eq. (2-4)]
η_o	Total surface temperature effectiveness, dimensionless [Eq. (2-3)]
Φ	Indicates "function of"
σ	Ratio of free-flow area to frontal area, A_c/A_{fr}, dimensionless
μ	Viscosity coefficient
ρ	Density
ω	Absolute humidity
τ_0	Unit surface shear stress
θ	Time
θ	Angular position coordinate in a circular tube (see Fig. 6-1)
θ_d	Dwell time, exchanger residence time for a fluid (L/V)
$\theta_{d,\min}$	θ_d for the $C_{\min}$ fluid
$\theta_{d,\max}$	θ_d for the $C_{\max}$ fluid
θ_r	Rotor rotation period for a periodic-flow heat exchanger
θ^*	Generalized time parameter for a direct-transfer exchanger ($\theta/\theta_{d,\min}$), dimensionless

θ_d^*	Dwell time ratio ($\theta_{d,\min}/\theta_{d,\max}$), dimensionless
θ_R^*	Generalized time parameter for a periodic-flow exchanger (θ/θ_r)

Dimensionless Groupings

Re	Reynolds number ($4r_hG/\mu$), a flow modulus
Re_d	Reynolds number (dG/μ)
St	Stanton number (h/Gc_p), a heat transfer modulus
Nu	Nusselt number ($h4r_h/k$), a heat transfer modulus
Pr	Prandtl number ($\mu c_p/k$), a fluid properties modulus
N_{tu}	Number of heat transfer units of an exchanger, a heat transfer parameter ($AU/C_{\min}$); more formally defined by Eq. (2-7)
$C_{\min}/C_{\max}$	Flow-stream capacity-rate ratio [$(Wc_p)_{\min}/(Wc_p)_{\max}$]

Subscripts

a	Air side
av	Average
c	Cold-fluid side of heat exchanger
h	Hot-fluid side of heat exchanger
i	Refers to inner surface of an annular passage or inner radius of a circular fin
L	Coupling liquid in a liquid-coupled heat exchanger
m	Mean conditions, defined as used
o	Refers to conditions at surface, or specifically to inner surface of an annular passage or inner radius of a circular fin
p	Refers to one pass of a multipass heat exchanger
r	Matrix rotor
w	Wall; water side
x	Local conditions
∞	Conditions far downstream
ii	Conditions at inner surface of an annular passage when the inner surface alone is heated
oo	Conditions at outer surface of an annular passage when the outer surface alone is heated
1,2	Indicate different sides of the heat exchanger; inlet and outlet conditions
max	Maximum
min	Minimum
lma	Log mean average

The following symbols and systems of units are used:

	English	SI
Mass	lb_m (pound mass)	kg (kilogram)
Force	lb_f (pound force)	N (newton, kg/(m·s²))
Length	ft, in (foot, inch)	m (meter)
Time	s, h (second, hour)	s (second)
Thermal energy	Btu (British thermal unit)	J (joule, N·m)
Power	Btu/s, Btu/h, hp (horsepower)	W (watt, J/s)
Temperature	°F, °R (degree Fahrenheit, degree Rankine)	K, °C (kelvin, degree Celsius)
Pressure	$\mathrm{lb}_f/\mathrm{ft}^2$	Pa (pascal, $\mathrm{N/m^2}$)

Under the **English System**, the proportionality factor in Newton's second law becomes $g_c = 32.2\ \mathrm{lb}_m \cdot \mathrm{ft}/(\mathrm{lb}_f \cdot \mathrm{s}^2)$, and the universal gas constant becomes $R = 1{,}545\ \mathrm{ft} \cdot \mathrm{lb}_f/(\mathrm{lb}_m \cdot \mathrm{mol} \cdot \mathrm{R})$. In the SI, g_c has the value of unity and is dimensionless. The universal gas constant is $R = 8{,}314\ \mathrm{J}/(\mathrm{kmol} \cdot \mathrm{K})$.

Note the dimensions consistently employed for the following properties, coefficients, and variables:

	English	SI
Density ρ	$\mathrm{lb}_m/\mathrm{ft}^3$	$\mathrm{kg/m^3}$
Viscosity μ	$\mathrm{lb}_m/(\mathrm{h} \cdot \mathrm{ft})$	$\mathrm{Pa \cdot s}$, $\mathrm{N \cdot s/m^2}$
Thermal conductivity k	$\mathrm{Btu}/(\mathrm{h} \cdot \mathrm{ft}^2 \cdot {}^\circ\mathrm{F/ft})$	$\mathrm{W}/(\mathrm{m} \cdot \mathrm{k})$
Specific heat c_p	$\mathrm{Btu}/(\mathrm{lb}_m \cdot {}^\circ\mathrm{F})$	$\mathrm{kJ}/(\mathrm{kg} \cdot \mathrm{k})$
Heat transfer coefficient h	$\mathrm{Btu}/(\mathrm{h} \cdot \mathrm{ft}^2 \cdot {}^\circ\mathrm{F})$	$\mathrm{W}/(\mathrm{m}^2 \cdot \mathrm{k})$
Heat flux q''	$\mathrm{Btu}/(\mathrm{h} \cdot \mathrm{ft}^2)$	$\mathrm{W/m^2}$
Mass flow rate W	lb_m/h	$\mathrm{kg/s}$
Molecular weight M	$\mathrm{lb}_m/(\mathrm{lb}_m \cdot \mathrm{mole})$	$\mathrm{kg/kmol}$

Table A-15 in App. A contains a set of conversion factors for shifting from one system of units to another, including some of the commonly used archaic units.

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1

Introduction

The design of a heat exchanger involves a consideration of both the heat transfer rates between the fluids and the mechanical pumping power expended to overcome fluid friction and move the fluids through the heat exchanger. For a heat exchanger operating with high-density fluids, the friction-power expenditure is generally small relative to the heat transfer rate, with the result that the friction-power expenditure is seldom of controlling influence. However, for low-density fluids, such as gases, it is very easy to expend as much mechanical energy in overcoming friction power as is transferred as heat. And it should be remembered that in most thermal power systems mechanical energy is worth 4 to 10 times as much as its equivalent in heat.

It can be readily shown that for most flow passages that might be used for the heat transfer surfaces of an exchanger, the heat transfer rate per unit of surface area can be increased by increasing fluid-flow velocity, and this rate varies as something less than the first power of the velocity. The friction-power expenditure is also increased with flow velocity, but in this case the power varies by as much as the cube of the velocity and never less than the square. It is this behavior that allows the designer to match both heat transfer rate and friction (pressure-drop) specifications, and it is this behavior that dictates many of the characteristics of different classes of heat exchangers.

If the friction-power expenditure in a particular application tends to be high, the designer can reduce flow velocities by increasing the number of flow passages in the heat exchanger. This will also decrease the heat transfer rate per unit of surface area, but according to the above relations the reduction in heat transfer rate will be considerably less than the friction-power reduction. The loss of heat transfer rate is then made up by increasing the surface area (lengthening the tubes), which in turn also increases the friction-power expenditure, but only in the same proportion as the heat transfer surface area is increased.

In gas-flow heat exchangers the friction-power limitations generally force the designer to arrange for moderately low mass velocities. Low mass velocities, together with the low thermal conductivities of gases (low relative to most liquids), result in low heat transfer rates per unit of surface area. Thus large amounts of surface area become a typical characteristic of gas-flow heat exchangers. Gas-to-gas heat exchangers may require up to 10 times the surface area of condensers or evaporators or liquid-to-liquid heat exchangers in which the total heat transfer rates and pumping-power requirements are comparable. For example, a regenerator for a gas-turbine plant, if it is to be effective, requires several times as much heat transfer surface as the combined boiler and condenser in a steam power plant of comparable power capacity.

These considerations have led to the development of many ways to construct heat transfer surfaces for gas-flow applications in which the surface area density is large. Such surfaces will be referred to here as *compact heat transfer surfaces*. Several typical compact heat transfer surface arrangements are illustrated in Fig. 1-1.

Perhaps the simplest and most common surface arrangement for a two-fluid heat exchanger is the circular tube bundle shown in Fig. 1-1*a*. This arrangement, of course, has long been used for both high- and low-density fluids, but the only way in which surface area density can be substantially increased is to decrease the diameter of the tubes. Fabrication difficulties and cost place a rather severe limitation on what can be accomplished in this direction, and large heat exchangers with tube diameters of less than $\frac{1}{4}$ in. (0.006 m) are rare.

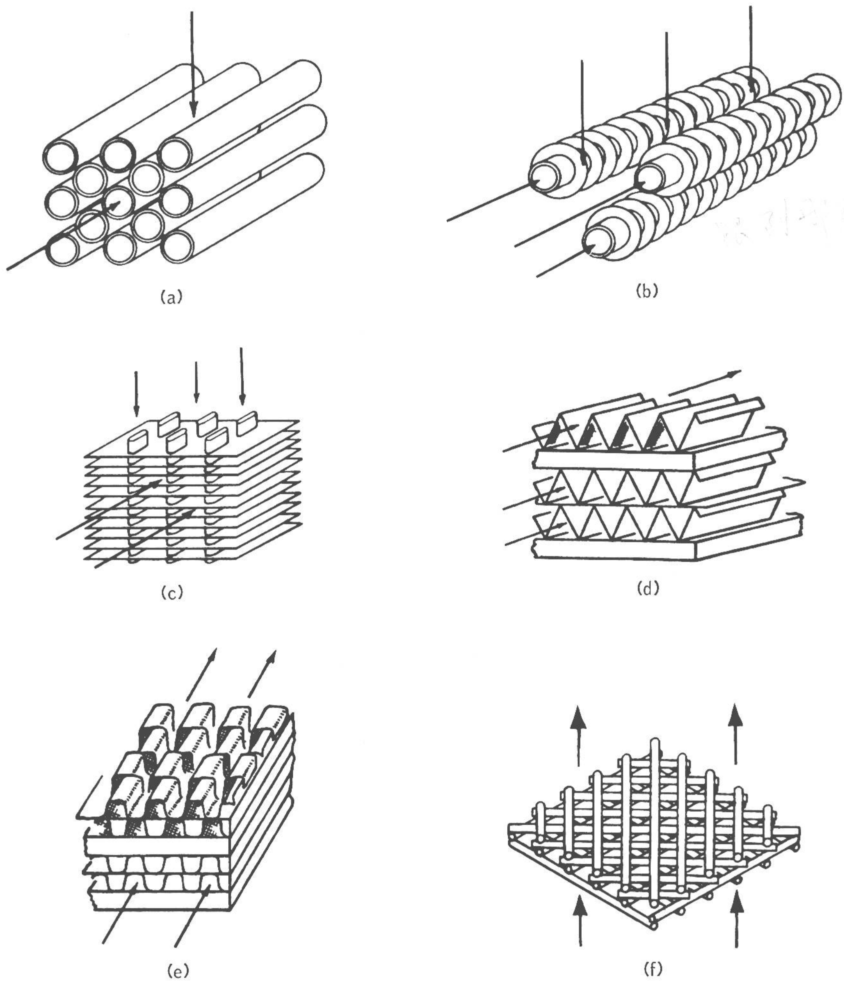
An effective way to increase surface area density is to make use of secondary surfaces, or fins, on one or both fluid sides of the surface. Figure 1-1*b* illustrates a finned circular tube surface in which circular fins have been attached to the outside of circular tubes. Such an arrangement is frequently used in gas-to-liquid heat exchangers where optimum design demands a maximum of surface area on the gas side. Fins *could* be used in a liquid-to-liquid heat exchanger, or on the liquid side of a gas-to-liquid heat exchanger, but here another difficulty arises. The low friction-power requirement characteristic of high-density fluids, together with the relatively high thermal conductivity of liquids, results in high convection heat transfer rates in any optimum design (high heat transfer coefficients). If fins are employed, the high heat transfer rates must be conducted along the fins, and the conduction resistance may destroy all or most of the advantage of the extra surface area gained (see the discussion of fin effectiveness in Chap. 2).

Another popular variation of the finned-tube arrangement is shown in Fig. 1-1*c*. Here the tubes are illustrated as flat, but they can also be circular.

In compact gas-to-gas heat exchangers, large area density is desirable on *both* fluid sides, and a method for accomplishing this objective with fins is illustrated by the plate-fin arrangement of Figs. 1-1*d* and *e*. The heat exchanger is built up as a sandwich of flat plates bonded to interconnecting fins. The two fluids are carried between alternate pairs of plates and can be arranged in either counterflow or crossflow, which provides an added degree of flexibility in this arrangement.

Figure 1-1*e* also illustrates another variation; the fins can be interrupted rather than continuous, an arrangement which alters the basic convection heat transfer and flow-friction characteristics in a manner that will be discussed presently.

Fig. 1-1 Some typical examples of compact heat exchanger surfaces.



In the periodic-flow-type heat exchanger, energy is transferred by convection and stored in a matrix, from which it is later given up to the other fluid. Figure 1-1*f* illustrates one such compact matrix, which could be built up of stacks of solid rods or stacks of wire screens. Matrices can also be constructed using stacks of plates and fins or simply packed bundles of tubes. Some of the most common matrices are made using glass ceramic materials.

An interesting and important feature of the compact heat transfer surfaces illustrated in Fig. 1-1 can be demonstrated if the heat transfer rate per unit of surface area is plotted as a function of the mechanical power expended to overcome fluid friction per unit of surface area. Such a plot for three different surfaces is shown in Fig. 1-2. The heat transfer rate for a unit of area and for one degree of temperature difference is merely the heat transfer coefficient h evaluated for some particular set of fluid properties from

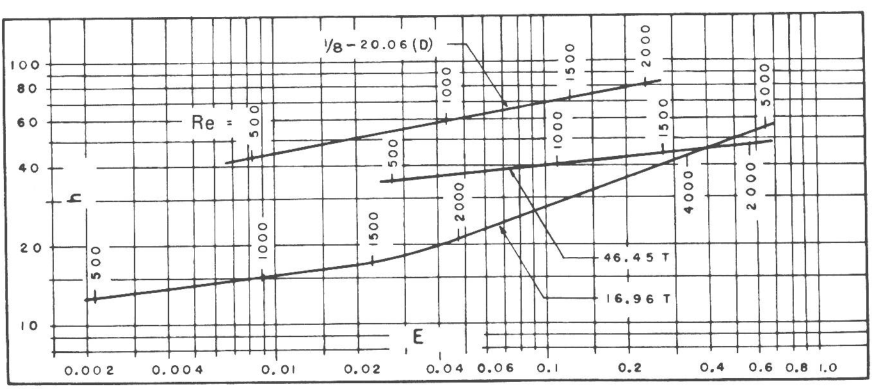
$$h = \frac{c_p \mu}{\mathrm{Pr}^{2/3}} \frac{1}{4r_h} (\mathrm{St}\,\mathrm{Pr}^{2/3})\mathrm{Re} \tag{1-1}$$

The friction power expended per unit of surface area can be readily evaluated as a function of the Reynolds number, the friction factor, and the specified fluid properties from

$$E = \frac{1}{2g_c} \frac{\mu^3}{\rho^2} \left(\frac{1}{4r_h}\right)^3 f\,\mathrm{Re}^3 \tag{1-2}$$

A plot of h versus E can be prepared once the basic convection heat transfer and friction characteristics are known as functions of the Reynolds number. Any particular surface arrangement is then represented by a single curve on a plot such as that in Fig. 1-2 [for fluid properties of air at 1 atm and 500°F (260°C)].

Fig. 1-2 A comparison of heat transfer and friction-power characteristics of three compact surfaces on a unit of surface area basis. Dimensions are in Btu/(h · ft² · °F) for h and in hp/ft² for E. Geometrical descriptions of the surfaces are provided in Chap. 9.



The interesting feature of this plot is the very wide difference in friction-power expenditure for a given heat flux for different surfaces, or conversely, the smaller difference in heat flux for a given friction-power expenditure. At the beginning of this chapter the important influence of friction-power expenditure on heat exchanger design was discussed, and in gas-flow heat exchangers it is the necessity to minimize friction power that forces the use of large amounts of surface area. This in turn has resulted in the development of more compact heat