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Heat exchanger theory

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Heat Exchanger Design Handbook

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1.1

Description of heat exchanger types

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1.1.0

Structure of the section

D. Brian Spalding

■ The purpose of Sec. 1.1 is to provide the words that enable the various kinds of heat exchanger to be **usefully** distinguished; and the purpose is fulfilled by classifying heat exchanger equipment from six distinct points of view.

Section 1.1.1 adopts the viewpoint of flow configuration. Here the major subdivisions are counterflow, parallel flow, cross flow, and so on. Section 1.1.2 draws attention to the distinctions that are afforded by the different ways in which the streams passing through a heat exchanger can interact, namely, by heat transfer alone, or by heat transfer and mass transfer. The interaction coefficients are defined.

Section 1.1.3 focuses on the various ways in which the temperatures of the streams may vary with position in the equipment. The interfaces between the com-

municating streams may take many different geometric forms. The commonly occurring types are described in Sec. 1.1.4.

In Sec. 1.1.5 appear the names of types of heat exchanger equipment, classified according to function. No-phase-change exchangers, boilers, condensers, and other types of plant are encountered here. Finally, the various ways in which time-dependent operation can occur are distinguished in Sec. 1.1.6.

Other classifying principles could be employed, for example, by reference to the industry in which the equipment is used (aircraft, petroleum, etc.); but the function of Part 1 of this handbook is to provide **only** an introduction, not to anticipate the contents of later parts. This limitation of intention must explain and justify the brevity of the descriptions that are provided.

1.1.1

Types of flow configuration

D. Brian Spalding

A. Introduction

Heat exchangers transfer heat between two or more streams of fluid that flow through the apparatus. A major characteristic of heat exchanger design is the relative flow configuration, which is the set of geometric relationships between the streams. Section 1.1.1 describes the more common types of configuration.

It must be emphasized that the configurations described represent idealizations of what truly occurs; it is never possible, in practice, to make the flow patterns conform to the ideal.

B. Counter flow

In a counter-flow heat exchanger, the two fluids flow parallel to each other, but in opposite directions. Figure 1 represents such a configuration schematically, by showing a single smaller-diameter tube placed co-axially within a tube of larger diameter. The two fluids flow respectively within the inner tube and through the annular space that separates the two tubes. In practice, a large number of tubes can be inserted within a single surrounding tube, of much larger diameter, known as the shell.

Here the symbol T is used for temperature; subscript 1 denotes the first stream, and subscript 2 the second stream; the subscript in denotes the entry conditions, whereas out denotes the leaving condition.

Counter-flow exchangers are the most efficient, in that they make the best use of the available temperature difference, and can obtain the highest change of temperature of each fluid. This remark is expanded upon below.

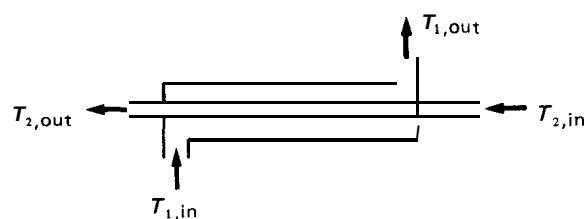


Figure 1[†] Schematic representation of a counter-flow heat exchanger.

C. Parallel flow

In a parallel-flow heat exchanger, the two fluids flow parallel to each other, and in the same direction. Figure 2 represents this configuration schematically. Parallel-flow exchangers make poor use of the available temperature difference when both fluids change appreciably in temperature, in which case they are not used if

[†]Within a three-digit section the equation, figure, and table numbers do not have the three-digit identifier; for cross-references between sections, the appropriate three-digit number will be given.

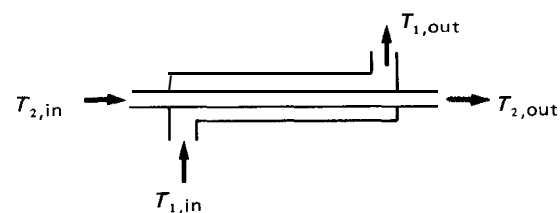


Figure 2 Schematic representation of a parallel-flow heat exchanger.

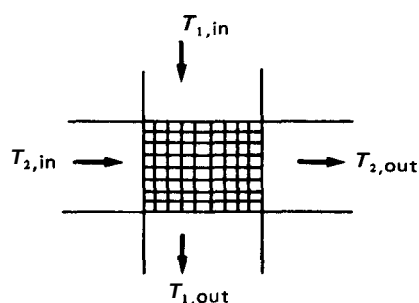


Figure 3 Schematic representation of a cross-flow heat exchanger.

efficiency is the most important factor in design. They do, however, tend to have more uniform wall temperatures than do counter-flow exchangers.

D. Cross flow

In a cross-flow exchanger, the two streams flow at right angles to each other. For example, stream 1 might flow through a set of tubes, arranged in a bank, whereas stream 2 might thread its way through the spaces between the tubes in a direction generally at right angles to the tube axes.

Schematically, cross-flow exchangers are usually represented as shown in Fig. 3. They are intermediate in efficiency between parallel-flow and counter-flow exchangers and, for practical reasons concerned with the ducting of the fluids toward the heat transfer surface, they are often easier to construct than either of the other two. Automobile radiators are of cross-flow design.

E. Cross counter flow

Sometimes, real heat exchanger flow configurations conform approximately to the idealizations shown in

Fig. 4. They are termed cross-counter-flow exchangers. Two-, three-, and four-pass types are represented; and, of course, the possible number of passes is unlimited.

Cross-counter-flow exchangers can be regarded as compromises between the desiderata of efficiency and ease of construction. The greater the number of passes, the closer is the approach to counter-flow economy.

F. Multipass shell and tube

Parallel-flow and counter-flow features may be combined within the same exchanger, as when tubes double back, once or more, within a single shell; and the same effect can be achieved, with straight tubes, by the provision of suitably subdivided headers. The U-tube, or hairpin, arrangement has the advantage of easy construction because only one end of the shell needs to be perforated, not two.

Examples of the idealized configurations are shown in Fig. 5. Also shown in Fig. 5 are flow configurations in which several shells are coupled together. It is, of course, impossible to represent here all the possibilities that may be encountered in practice or imagined, but the common ones are described in Sec. 4.2.

G. The general case

The idealized flow configurations described above are all particular examples of what may be called multiple flows in interpenetrating continua, of which the characteristics are as follows:

1. Several streams enter a common volume at a number of distinct entry points, and they leave at a number of distinct exit points.
2. Individual streams subdivide after entry, within the volume, and reunite at their exit points.
3. The individual streams come into thermal con-

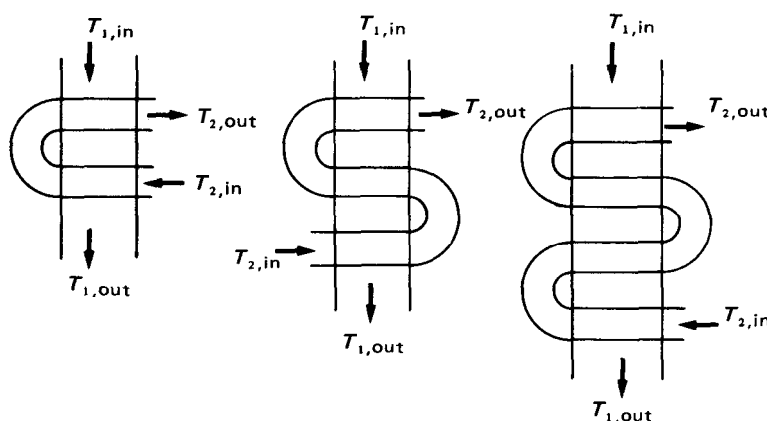


Figure 4 Schematic representations of cross-counter-flow heat exchangers.

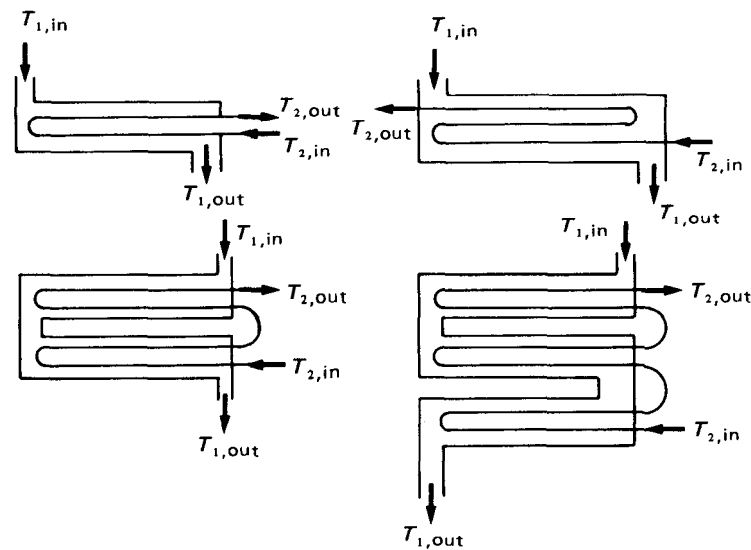


Figure 5 Schematic representations of flow configurations for multipass shell-and-tube heat exchangers.

tact with one another within the heat exchanger volume, and suffer consequential changes of temperature.

Figure 6 is a schematic representation of what is intended for the two streams. The fluids are distributed three-dimensionally within the space and there may be recirculation regions in which the streamlines are closed.

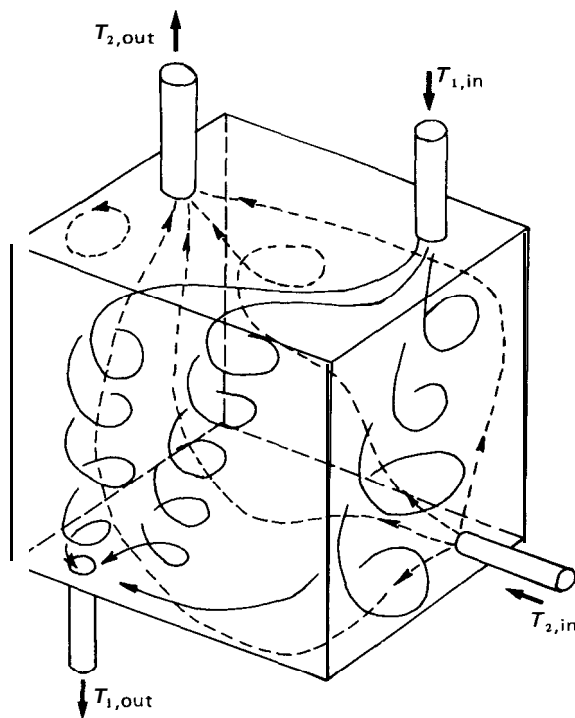


Figure 6 Schematic representations of the general case of interpenetrating continua. Only two streams are represented, and no indication is given of solid contents such as tubes, baffles, and so on.

H. Nonsimultaneous flow configurations: Regenerators

In all the examples given so far, it has been implied that the flows are steady and that both streams flow simultaneously. Now it must be disclosed that the implication is valid only for the class of heat exchangers known as recuperators. There is another class, called regenerators, for which the two streams flow alternately through the same space. In regenerators, heat transferred by one fluid to the solid walls of the ducts is stored there, and given up to the second fluid when it in turn flows past.

Regenerators can embody counter-flow, parallel-flow, and cross-flow configurations, just like recuperators. Thus, a simple counter-flow regenerator would be a straight horizontal tube (Fig. 7) through which one fluid flowed, when it flowed, from left to right; and through which the second fluid flowed, when the first fluid was

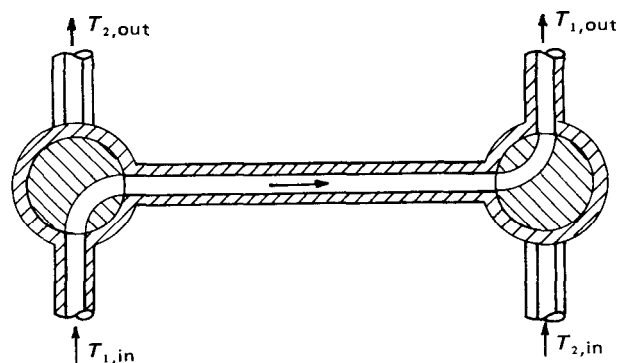


Figure 7 Schematic representation of a regenerative heat exchanger

not flowing, from right to left. Heat transfers to and from the tube walls would take place because of the different inlet temperatures of the two streams. As a consequence, the hotter stream would become cooler, and the colder would become warmer.

Regenerators are periodic-flow devices, for it is always arranged that the two streams alternate in accordance with a regular and predetermined rhythm. Figure 7 indicates how rotary valves at each end of the tube might control the flows appropriately.

I. Conclusion

In order to predict the performance of a heat exchanger, it is necessary to establish first, what is its flow configuration; second, what are the rates of flow along the prescribed paths; and third, what are the resistances to heat transfer from one stream to another at each point within the heat exchanger volume. The determina-

tion of the temperature distributions in the individual streams is then a matter of mathematics.

When the configurations are simple, as are those of **Secs. B through F**, and when the resistances to heat transfer are uniform with respect to volume, it is often possible to solve the relevant performance equations analytically. This possibility is extensively illustrated in sec. 1.3.

When, on the other hand, the flow configurations are complex, as in the general case illustrated in Fig. 6, or when the heat transfer resistances vary from place to place, the relevant equations can be solved only by numerical means. This matter is discussed in Sec. 1.4.

It is, of course, far from easy to know just what values must be ascribed to the resistances under the conditions that arise in practical heat exchangers; and indeed they often depend on the local temperatures of the heat exchanging fluids. Much of this handbook is devoted to establishing formulas for the resistances.

1.1.2

Types of interactions between streams

D. Brian Spalding

A. Heat transfer

Heat transfer is an interaction that occurs between materials by reason of the temperature difference between them. It is the most common type of interaction in heat exchange equipment, and it has pride of place in this handbook.

Heat transfer between the streams is usually effected indirectly: the streams are separated by a solid material, such as a metal tube wall or plate, or even a plastic membrane, and the heat passes from the first fluid through the solid material to the second fluid. The consequences of the heat transfer are often local increases in the temperature of the cooler fluid and decreases in that of the warmer fluid; they may also entail the change of phase of one or both fluids.

Heat transfer can also take place when the fluids are in direct contact, for example, when one fluid is warm water and the other is cool air. Direct-contact heat transfer is very common when the cooling water of a steam power station is to be cooled in its turn. The relevant heat exchange device is then usually called a **cooling tower**.

Sometimes a cloud of solid particles exchanges heat with a stream of fluid. In a **fluidized-bed heat exchanger**, a hot gas may pass upward through a dense cloud of solid particles which, although they are in violent motion of a semirandom character, are prevented by gravity from rising with the gas. These particles impinge upon, and transfer heat to, solid surfaces (for example, cooling water tubes) that pass through the bed.

There are other ways in which heat passes first from a fluid to a solid, and later from the solid to a second fluid; they are mentioned in Sec. 1.1.1 H.

B. Simultaneous heat and mass transfer

The presence of direct contact permits mass transfer to occur between the streams; this is often effected by way of the vaporization of a part of the liquid stream or the condensation of one component of the gaseous stream. The first occurs in the cooling tower situation just described; the latter occurs in some dehumidifying equipment.

The phase change is often accompanied by appreciable thermal effects associated with the latent heat of phase change, and these effects may be essential to the operation of the exchanger. This is true of the direct-contact (or wet) cooling tower, which needs only about one-fifth the amount of contact surface, for a given water-cooling effect, that is needed by an exchanger (a dry tower) in which mass transfer is prevented by the use of an indirect-contact design.

Mass transfer can occur, in a sense, in **indirect-contact** exchangers. A steam condenser is such a device, for the cooling water flows inside tubes, and the steam condenses on the outside. Strictly speaking, however, this is a three-fluid device; the fluids are the cooling water, the steam (mixed with a small quantity of air), and the condensate. The last two streams are in direct contact.

C. The interaction coefficient

If two streams interpenetrate within the volume of a heat exchanger, the magnitude of the interaction per unit volume is conveniently quantified by way of a coefficient. Thus, if heat transfer is in question, the

overall **volumetric heat transfer coefficient**, U_{vol} , may be defined by

$$\dot{q}_{\text{vol}} \equiv U_{\text{vol}}(T_1 - T_2) \quad (1)$$

where $\dot{q}_{\text{vol}}$ is the heat transfer rate per unit volume from fluid 1 to fluid 2, and T_1 and T_2 are the local fluid temperatures.

A corresponding expression for the overall **volumetric mass transfer coefficient**, β_{vol} , is

$$\dot{m}_{\text{vol}} = \beta_{\text{vol}} \rho (x_1 - x_2) \quad (2)$$

where $\dot{m}_{\text{vol}}$ is the mass transfer rate per unit volume from fluid 1 to fluid 2, x_1 and x_2 are the mass fractions in the two fluids of whatever substance is being transferred, and ρ is the reference density of the local fluid.

The volumetric interaction coefficients U_{vol} and β_{vol} are convenient measures of the ease with which fluid conditions are equalized within the exchanger, especially when the interface between the two fluids is extensive and perhaps irregular. Thus, splash-bar packings for cooling towers, and the packed beds of gas-absorption plants, are best characterized in these terms, and the same practice may be adopted for finned-tube matrices.

However, it is also possible to focus attention on the interface area itself, when this is known. Then the corresponding **superficial interaction coefficients can** be defined. This practice is more common among heat exchanger designers; it will therefore be used predominantly below.

The relevant coefficients are the heat transfer coefficient U , and the mass transfer coefficient β , without subscript. (The use of the subscript “vol” for volumetric coefficients could be matched by the use of the subscript *area* for superficial coefficients; but the

latter are so widely used that the practice would be tiresome. If “vol” is absent, *area* is meant.) Their definitions are

$$\dot{q} = U(T_1 - T_2) \quad (3)$$

and

$$\dot{m}_n = \beta \rho (x_1 - x_2) \quad (4)$$

where $\dot{q}$ and $\dot{m}_n$ are respectively the heat and mass fluxes, per unit time and unit area. Obviously, if A stands for interface area and V for exchanger volume, and if sufficiently small elements of each are taken for the fluxes and coefficients to be regarded as uniform, the various quantities can be connected by

$$\frac{\dot{q}U}{\dot{q}_{\text{vol}}U_{\text{vol}}} = \frac{\dot{m}_n\beta}{\dot{m}_{n,\text{vol}}\beta_{\text{vol}}} = \frac{V}{A} \quad (5)$$

This is just a matter of definition.

In practice, the interaction coefficients are rarely uniform throughout the heat exchanger space. This is true whether the volumetric or the superficial quantities are in question. Nevertheless, analysis is often simplified by the assumption that they are uniform; and this assumption will frequently be made below. The users of analyses based on this assumption should always treat their results with caution.

The interaction coefficients are discussed further in Secs. 1.2.2 and 1.2.3. There will be introduced the distinction between the overall heat transfer coefficient U , defined above, and the individual or “film” coefficient α , which pertains to the transfer between a fluid stream and its immediate boundary surface; and it will also appear that, to be most useful, the overall mass transfer coefficient is best defined rather differently from above [see Eq. 1.2.2(12)].

1.1.3

Types of temperature change pattern

D. Brian Spalding

A. Single-phase heat exchangers

In a large proportion of the heat exchangers encountered in practice, each fluid stream leaves in the same phase state as that in which it entered (a gas leaves as a gas, a liquid as a liquid). The result is that, as heat is transferred from the hotter fluid to the cooler one, the former temperature diminishes while the latter rises.

Often the temperature change is proportional to the heat transferred. This is the case most commonly analyzed (see, for example, Secs. 1.3.1 and 1.5).

Figure 1 illustrates the temperature changes that occur in the two streams along the length of heat exchangers of counter-flow and parallel-flow configuration. Evidently, it is possible in the former case for the outlet temperature of one of the streams to approach closely the inlet temperature of the other; the best that can occur in the case of parallel-flow exchangers is that the two outlet temperatures may be close together.

It is not possible to illustrate so easily the temperature distribution in heat exchangers having cross-flow configurations, for temperature is no longer a function of a single distance variable, even ideally.

B. Boilers and condensers

In other heat exchanger types, however, especially in boilers and condensers, the essential function is to change the phase of one of the streams. In such cases, the temperature change in that stream is often small enough to be neglected.

In the case of the boiler, the phase-changing fluid may enter as a subcooled liquid, and subsequently rise in temperature to its boiling point; then, throughout the two-phase region, the temperature will vary only because of the small difference of pressure that may prevail; thereafter, before leaving the heat exchanger, the fluid may become superheated. Temperature changes therefore do occur in the phase-changing fluid within a boiler, but are often disregarded by the designer who wants to employ an analytical formula that is valid only for the case of zero change.

Figure 2 illustrates the temperature distribution in a parallel-flow steam boiler. Stream 1 represents water and steam, which pass from the subcooled-water to the superheated-steam states; stream 2 represents the combustion gases. Practical boilers, it should be mentioned,

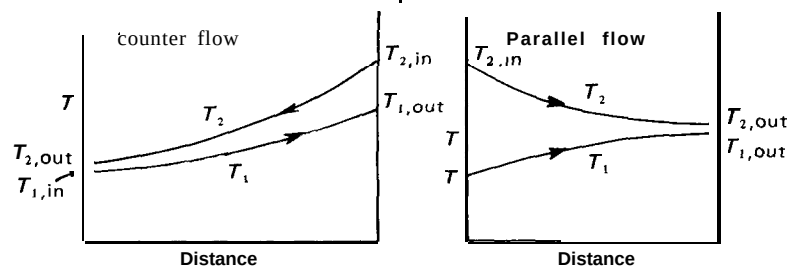


Figure 1 Temperature distribution diagrams for counter-flow and parallel-flow heat exchangers

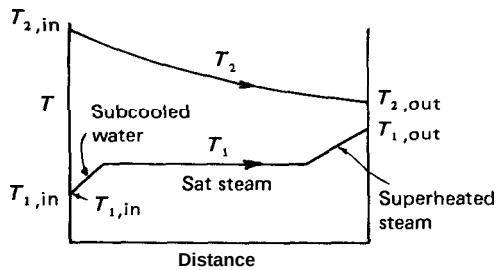


Figure 2 Temperature-distribution diagram for a parallel-flow boiler, producing superheated steam from subcooled water.

will have a mixture of parallel-flow, counter-flow, and cross-flow features.

It is much the same for steam condensers: the presence of air ensures that the temperature of the gas phase does **not** remain uniform throughout, for it must fall as the steam concentration (and therefore partial pressure also) diminishes. Often, however, designers will ignore such effects, or take account of them only approximately. Similar problems arise in condensers for multicomponent mixtures.

C. The general case

In general, the temperature of a fluid stream varies in a nonlinear fashion as heat is transferred to it. Such nonlinearities exist, as has just been explained, even in the steam boilers and condensers where they are often ignored. In other types of equipment, especially those involving chemical reaction (petroleum crackers) or phase-change (direct-contact cooling towers; drying plants for textiles, paper, or foodstuffs; condensers for multicomponent mixtures), the nonlinearities are too significant to be ignored. Even in no-phase-change heat exchangers the variation of specific heat with temperature (see Sec. 1.2.1) exhibited by many materials may need to be taken into account.

Such nonlinearities, like the variations of transfer coefficient with position in the exchanger, limit the applicability of the analytical approach to heat transfer design, represented by Secs. 1.3.3 and 1.5. However, they can be handled by the numerical solution procedures described in Sec. 1.4.

1.1.4

Types of interface between streams

D. Brian Spalding

A. Introduction

The purpose of this brief section is to draw attention to the fact that the fluid streams in a heat exchanger are brought into contact (direct or indirect) in a wide variety of ways. The purpose can best be met by describing fluid-interface types briefly, and indicating where more complete descriptions can be found.

B. Plain tubes

The most common arrangement is for one fluid to flow within straight or curved tubes of circular cross section, while a second fluid washes the outside of these tubes in a longitudinal, perpendicular, or oblique direction.

The interface is thus represented by a tube wall, and contact is indirect. The tubes may, of course, be noncircular in cross section; often they are bent into hairpin shapes and sometimes they take the form of helical coils.

C. Finned tubes

Sometimes, when heat transfer is effected more easily on the inside of the tubes than on the outside, the latter surface is extended by the provision of fins, as illustrated in Fig. 1. These excrescences may be integral with the tube wall, or they may be soldered, brazed, or welded to it; they may comprise annular disks, helical tapes, or plane sheets aligned with the tube axis.

The presence of the **fins** entails that, per unit volume of exchanger, there is more interface area

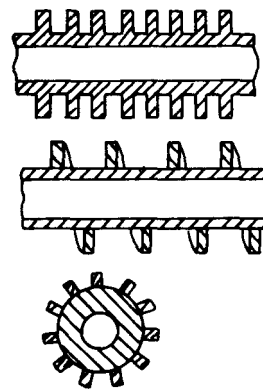


Figure 1 Illustration of various kinds of externally finned tubes.

between metal and fluid for one stream than for the other. Occasionally it is the inside of the tube that is provided with such fins; this is appropriate when it is the internal heat transfer coefficient that is the lower.

The fact that the metal-fluid interface area per unit volume is different for each fluid renders it necessary to specify precisely which area is in question when a heat flux per unit area, or a superficial interaction coefficient, is being employed. The problem is avoided, of course, if volumetric equivalents are used; in this connection, it is useful to distinguish the problem of specifying the surface area per unit volume from that of measuring it. In a sense, the former is both easier and more important; for it does not matter precisely what the interface area per unit volume actually is, provided that a definite value is ascribed, and used consistently, when it is the heat transfer rate per unit volume that is measured.

D. Matrix arrangements

The **ingenuity** of designers has been widely employed in devising economical ways in which the amount of interfacial area can be increased so that the volumetric interaction coefficients are high. Configurations devised with this intention, and departing significantly from the plain-tube or finned-tube types, are commonly called heat exchange matrices.

The integrated blocks of tubes and **fins** found in automobile radiators and in air conditioning devices are familiar examples.

E. Films

Two examples have already been mentioned (in **Secs. 1.1.2C** and **1.1.3B**, of direct contact between a stream of gas on the one hand and, on the other, a film of liquid that runs over the surface of a solid. There are more examples to be found in engineering practice, such as the gas-absorption tower **in** which the liquid solvent is caused to flow downward in an irregularly configured film, over the surfaces of some packing of Raschig rings, saddles, or even fragments of stone, while a gaseous mixture, from which one component has to be removed by dissolution in the liquid, flows upward through the interstices.

Another example is found in cooling towers for power stations, in which the water to be cooled may flow downward over the surface of corrugated asbestos sheets, similar in type to those used for roofing (Fig. 2). Air flowing upward in the spaces between these sheets becomes heated and absorbs water vapor at the same time.

F. Sprays

It is perhaps rare that, in film-type contact devices, the liquid adheres always to the solid surface: some droplets are inevitably formed when the gas velocity is large. There exist also heat and mass exchange devices in which

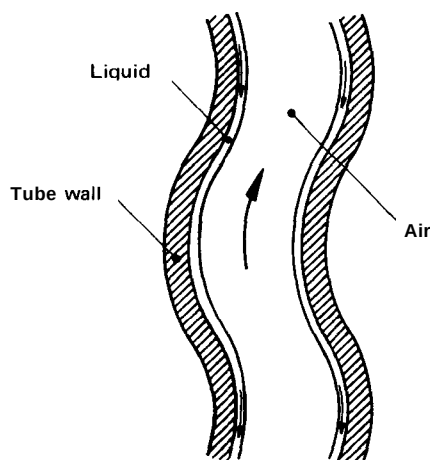


Figure 2 Illustration of the downward flow of a liquid film and the upward flow of air in the corrugated asbestos (roofing sheet) packing of a direct-contact cooling tower for a power station.

a major part of the interaction takes place while the droplets, formed by means of a sprayer, are falling freely through the gaseous phase.

Equipment of this kind includes desuperheaters in steam power plants, humidifiers in air conditioning practice, devices for the production of powdered milk, and numerous other types of apparatus. Even furnaces and engine combustion chambers fall into this category, for the fuel is often injected as a spray of droplets that must vaporize before it can burn.

G. Scraped surfaces

Only for fluids of low viscosity is it practicable to bring about **filmwise** flow, at an economically satisfactory rate, while relying only on gravitational and **fluid-friction** processes. If **filmwise** behavior is required of a high-viscosity fluid, it is necessary to use some special means for forming and maintaining a thin film.

Scraping devices are used for this purpose. These involve the introduction and operation of a powered mechanical system.

1.1.5

Types of heat exchange equipment

D. Brian Spalding

A. Introduction

The role of Sec. 1.1, as a whole, is to introduce heat exchange equipment by way of a series of classifications, each from a different point of view. In this section, this purpose is prosecuted further by enumerating heat exchanger types, mainly from the viewpoints of function and construction. This section gives the names of the main types and highlights their special features.

B. Shell-and-tube no-phase-change heat exchangers

Most unfired heat exchangers, operating with fluids that do not change phase, are of baffled shell-and-tube design, by which is meant that one of the fluids flows within straight or hairpin-bent tubes, whereas the other flow between and around those tubes, within an ah-confining shell, being guided in its path by baffles. These baffles serve to hold the tubes apart, so that the shell-side fluid can flow between them; they also control the path of that fluid to some extent.

Many different designs are used; the choice depends on the relative importance, for the fluids and the application in question, of such factors as the following:

- Capital (construction) cost
- Running (especially pumping) cost
- Cleanability
- Tendency to corrosion
- Pressure differences to be sustained
- Dangers associated with leakage

Temperature range, and liability to thermal stress
Tendency to tube vibration and subsequent fatigue

The designs differ in the following ways, among others:

The tube-side fluid may be constrained to pass through the shell once only, or two or more times.

When there are many passes, the tube-side fluid may be turned around by the hairpinlike nature of the tubes; or it may enter headers, at either end of the shell, that receive fluid from one set of tubes and deliver it to another set.

The tubes entering the headers may be **fixed** to the tube plates that separate their contents from the shell in various ways—for example, by welding.

The headers themselves may be welded to the shell, or they may be permitted to slide relative to it in response to thermal expansion.

The shell may be provided with removable openings to permit inspection and/or cleaning of either the inside or outside of the tubes.

The baffles may be of various shapes, and they may be few or many in number.

The shell-side fluid may enter or leave through one or more apertures.

How the design features connect with the performance features is described elsewhere in this handbook. For the purposes of the present discussion, the most important features are those that affect the flow configuration, namely, the number of passes of the tube side fluid and the extent to which the baffling of

the shell causes the shell-side fluid to be well mixed over the cross section.

Not all no-phase-change heat exchangers are of shell-and-tube type. An alternative construction involves the provision of more-or-less flat plates, by which the two fluids are separated and through which the heat is transferred. Such **plate heat exchangers** are used when the pressure difference between the two streams is not excessive, and when easy cleanability is desired. They are often employed in the food and pharmaceutical industries.

C. Evaporators, boilers, and reboilers

When the function of the heat exchanger is to cause one of the fluid streams to change from the liquid to the vapor phase, it is common for a modified shell-and-tube construction to be employed. Compared with the no-phase-change equipment discussed in Sec. B, **vapor-producing** heat exchangers are usually supplied with enlarged spaces within which separation of the vapor from the liquid can occur.

When the vapor is formed in the tube-side fluid, these phase-separation spaces are either enlarged **upper-level** headers, or else they are additional vessels connected with the headers. When it is the shell-side fluid that is evaporated, the shell may be provided with a tube-free region, possibly of enlarged diameter; but there **are** many variants, with the shell vertical or horizontal, and with or without provision of external "**down-comers**", permitting liquid to recirculate to the bottom of the shell. Tube-side evaporators also often allow for such a recirculation, which may be either natural (i.e., driven by gravity) or assisted (e.g., by a pump).

D. Condensers

Gravitational separation of liquid and vapor also plays an important part in the functioning of condensers, of which many designs exist, their nature determined largely by the ratio of the condensed to the uncondensed components in the fluid stream being cooled.

Even when the cooled stream is nominally wholly **condensible**, as is true of stream condensers for power stations, inward leakage of air may make it necessary to extract some uncondensed fluid, and the flow velocities must be such that the upward flow of air will not entrain any of the (nominally) downward-flowing water. This requirement is met by the provision of lanes between banks of tubes; and baffles are employed to ensure that there is not short-circuiting from the steam inlet to the air outlet.

In power station condensers, the steam is usually on

the shell side, and the cooling water runs through horizontal tubes. In the process industry, tube-side condensation is common, and, in those circumstances, the tubes are usually vertical.

E. Cooling towers

Cooling water passing through the condenser of a power station, or used in a chemical plant, often has to be cooled, in its turn, by contact with the atmosphere. Devices for effecting this are called cooling towers. They may be broadly classified by reference to:

Whether contact between the water and the air is direct or indirect; and

Whether the circulation of the air is procured by gravitational or mechanical effects.

The smallest running cost is provided by gravitationally actuated, that is, natural-draft, cooling towers. These consist of tall chimneys, open at the base to atmospheric air, which is drawn in past heat exchange surfaces situated near the opening. The taller the chimney, the larger the air flow; but capital cost increases as a consequence, so that an economic optimum height exists.

Mechanical-draft cooling towers are shorter and therefore cheaper to build, but the fans in them, which cause the air to circulate, absorb appreciable quantities of electrical power. Once again, an optimum must be sought.

When water is not in short supply, so that the inevitable vaporization loss can be tolerated, and when the resulting pollution of the atmosphere by moisture can be tolerated, the air is allowed to make direct contact with the water. The reasons are that the latent-heat-of-vaporization effect enlarges the cooling capacity of a given quantity of air; and that, for natural-draft towers, the low molecular weight of steam increases the buoyancy of the air in the chimney. Direct-contact heat transfer is common in both mechanical- and natural-draft equipment.

Indirect-contact cooling towers, as the name implies, provide that the cooling water is separated from the air by a solid barrier, usually the wall of a metal tube. The tube wall is often provided with fins on the air side to facilitate heat transfer. Such indirect-contact towers (often **called** dry, as opposed to wet **direct-contact** ones) are expensive to construct but they do prevent loss of water to the atmosphere.

The foregoing classification fails to cover all types; for example there now exist wet-dry towers and some designs employ both gravitational and mechanical effects.

F. Drying plant

In some heat exchangers, vaporization of water (or some other liquid) is desired. Evaporators, mentioned above, serve this purpose, as do dryers, in which a solid material enters with a high moisture content and should leave with a lower one as a consequence of interactions with a second stream of fluid.

Drying equipment varies very much in its configuration and mode of operation, depending on the material to be dried. Thus, milk powder may be manufactured by spraying milk into hot gases resulting from combustion of a fuel; paper is dried by being conveyed over the surface of a steam-heated rotating drum while air is blown over it; textiles are dried by being exposed to an array of steam or air jets while being drawn along a conveyor track, and so on.

G. Furnaces

The use of the combustion products of a fuel and air as one of the streams in a heat exchanger has been mentioned on more than one occasion. **When** the combustion of the fuel takes place inside the heat exchanger, rather than in an external combustion chamber (as in a gas turbine plant), the equipment may be called a furnace or ***fired heater***.

Heat exchangers of this type take many forms, depending on the nature of the fuel (gaseous, liquid, or solid); the material to be heated (which may be oil in tubes lining the furnace wall, or a pool of molten iron, or a stack of solid **clayware** articles); and the magnitude of the throughput that must be achieved. Apart from stressing the importance of radiative transport of heat, there is little of general character that may be said.

1.1.6

Unsteady operation

D. Brian Spalding

A. Transient behavior of steady-state heat exchangers

Although not emphasized, it has been implied above, and is true, that all the heat exchangers mentioned in Sec. 1.1.5 are designed for steady-state operation. However, every heat exchanger operation must have a beginning and an end, and the requirements of industrial use necessitate changes from one steady state to another. Each such change occupies a finite amount of time, and it may be that the sum of the transitional periods forms an appreciable fraction of the whole time of operation of the heat exchanger.

This is one reason why the quantitative study of the unsteady behavior of heat exchangers is often desirable, for performance predicted solely on the basis of a succession of steady states may be considerably at variance with reality.

There is another reason: sometimes the start-up and shut-down performance of a heat exchanger has implications for the safety of a plant, especially when the transient is unexpected, resulting perhaps from a power failure. Thus, thermal stresses may result from rapid changes of temperature and water hammer effects, associated with the sudden stopping of slugs of liquid, can cause fractures of pipes and fittings.

B. Periodically operating heat exchangers (regenerators)

Some heat exchangers are essentially unsteady in operation. They are of the kind known as **regenerators**, and

their characteristic feature is that the fluids that are exchanging heat occupy the same space, in repeated succession; this was mentioned in Sec. 1.1.1H.

A typical example of a regenerator is that of an industrial furnace in which a pool of molten glass is heated by a flame playing over its surface. The task of the regenerator is to preheat the combustion air by extraction of heat from the combustion products. This is effected by passing air and combustion products, in turn, through the same labyrinth of bricks. When the combustion gases pass through, the bricks are heated and the gases cooled; when the air passes through in its turn, the air is heated and the bricks are cooled. This sequence of events recurs periodically.

In order that the supply of air to the furnace can be continuous, there are normally at least two regenerators for each furnace; the air is switched from the one to the other by a periodically actuated valve, and the combustion gases flow into the one that is not being used to heat air.

Regenerators are of two main types. In one, such as the example just given, the solid heat transferring material is **fixed**; in the other, the solid material rotates continuously past ports from which hotter and colder fluids are alternately admitted.

1.2

Definitions and quantitative relationships for heat exchangers

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1.2.0

Structure of the section

D. Brian Spalding

■ Whereas Sec. 1.1 has been concerned with qualitative descriptions, the concern of Sec. 1.2 is with quantitative concepts needed for the design of heat exchanger equipment.

Section 1.2.1 reviews the relevant concepts from the subject of thermodynamics. The treatment of the various topics should be adequate for most purposes; however, readers who are concerned with, say, **nonideal** mixtures will have to turn to specialist works for further explanations.

Section 1.2.2 introduces and defines the important notions of interphase heat flux and mass flux, and of the interaction coefficients between the phases. Although the relationships between heat flux and temperature difference, and between mass flux and concentration difference, are rarely quite linear, it is still useful to divide the one by the other and to give the resulting coefficients prominence in the resulting relationships.

The designer needs to ascribe values to the coefficients that must be used; and, since their values depend on local fluid velocities, properties, and geometric factors, the designer needs to know the formulas that describe these relationships. Although such formulas are provided, for particular situations described in other parts of this handbook it is necessary to introduce here

the terms in which such relationships can be described. This involves, in particular, the definition of the widely used dimensionless groups, such as the Nusselt and Stanton numbers. Some of the more common formulas are also presented by way of example.

Section 1.2.4 applies the conservation equations of mass and energy to complete heat exchanger equipment; the relationships generated form part of the mathematical foundation of heat exchanger theory. Section 1.2.5 applies the equations to cross sections of the equipment, thereby generating the ordinary differential equations that must be solved when the performance of a heat exchanger is to be predicted by conventional means.

Section 1.2.6 goes further; it applies the conservation equations to infinitesimal volume elements of the equipment. The result is the set of partial differential equations that must be solved when a more thorough analysis of heat exchangers is made, that is, one that is free from assumptions about the distributions of fluid temperature and velocity across a section through the equipment. These differential equations are the foundation of the numerical approach to heat exchanger analysis discussed in Sec. 1.4.

1.2.1

Thermodynamics: Brief notes on important concepts

D. Brian Spalding

A. Temperature

For present purposes, temperature is that property of matter, differences of which are cause of heat transfer. It is an intensive property. Its symbol in this book is **T**, and it is measured in kelvins (K) or degrees Celsius (°C).

B. Specific internal energy

The specific internal energy **u** of a material is the extensive property which changes as a consequence of heat and work transfers in accordance with the linear relationship

$$M \Delta u = Q - W \quad (1)$$

where **M** stands for the mass of the material, **A** signifies increase of, **Q** is the symbol for the heat transferred to the material, and **W** is the external work done by it during the transaction. The units of **u** are joules per kilogram (J/kg).

C. Specific enthalpy

The specific enthalpy **h** of a material is the extensive property that is related to the specific internal energy **u**, to the pressure **p**, and the density **ρ** by the relationship:

$$h \equiv u + \frac{p}{\rho} \quad (2)$$

Like **u**, **h** is usually a function of two variables, for example, pressure and temperature; its units are joules per kilogram (J/kg).

Pressure is here understood as the force that the material exerts on its surroundings, normal to its surface, per unit area of that surface; its units are newtons per square meter (N/m²). Density is the mass of the material per unit volume; its units are kilograms per cubic meter (kg/m³).

Specific enthalpy is of particular importance in heat exchanger practice because it enters the steady-flow energy equation

$$\dot{M} \Delta \left(h + \frac{v^2}{2} + g_n z \right) = \dot{Q} - \dot{W}_s \quad (3)$$

where **Ṁ** stands for the mass rate of flow (kg/s), **A** again stands for increase of, **(h + v²/2 + g_nz)** is the sum of the enthalpy, the kinetic energy, and the gravitational potential energy, **Q̇** is the rate of heat transfer into the region under study, and **Ẇ_s** is the shaft work, that is, the work transmitted to the outside, as by the shaft of a turbine. In many heat exchanger circumstances, the only terms of importance in this equation are **Ṁ**, **Ah**, and **Q̇**.

D. Quality

In a two-phase mixture, for example, of steam and water, the quality **x** is the mass of fluid, per unit mass of mixture, that is in the vapor phase; its units are kilograms per kilogram (kg/kg), and its value must always lie in the range 0.0-1.0.

The specific enthalpy of a two-phase mixture is related to the specific enthalpies of the saturated liquid and vapor, **h_l** and **h_g**, respectively, by

$$h = h_l (1 - x) + h_g x \quad (4)$$

A similar relationship holds for the reciprocals of the densities,

$$\frac{1}{\rho} = \frac{1-x}{\rho_l} + \frac{x}{\rho_g} \quad (5)$$

where ρ_l and ρ_g are the densities of the saturated liquid and vapor, respectively. As a consequence, h and ρ may be connected by

$$x = \frac{h - h_l}{h_g - h_l} = \frac{1/\rho - 1/\rho_l}{1/\rho_g - 1/\rho_l} \quad (6)$$

This relationship is useful in the design of boilers. Note that x has just been defined in terms of the quantities of the two phases that are present in a defined volume of space, but another definition is sometimes used that relates to flow rate ratios. Consider a two-phase mixture flowing in a duct, and suppose that the mass flow rate of liquid is $\dot{M}_l$ whereas that of gas is $\dot{M}_g$; then the flow quality x is defined by

$$\dot{x} \equiv \frac{\dot{M}_g}{\dot{M}_l + \dot{M}_g} \quad (7)$$

where $\dot{x}$ and x will have the same value only when the two phases are flowing at the same velocity. This sometimes occurs in practice, for example, when the subdivision of one phase and its dispersion in the other are on a very fine scale; but it is exceptional.

In some parts of the two-phase-flow literature, the term quality is used where flow quality is meant. The reader is advised to check what definition is implied in every particular case.

E. Mass fraction

The composition of mixtures of materials having different chemical natures may be described in many ways. The most convenient for heat exchanger purposes is by reference to the mass fractions x_i of the various components. These are defined by the statement that x_i stands for the mass of species i per unit mass of mixtures; its units are kilograms per kilogram (kg/kg).

An implication of the definition is that the sum of all x_i 's equals unity. Thus,

$$\sum_{all\ i} x_i = 1 \quad (8)$$

The specific enthalpy of a mixture, it is useful to mention, is sometimes given accurately enough for purposes of heat exchanger design by the relationship

$$h = \sum_{all\ i} x_i h_i \quad (9)$$

where h_i is the specific enthalpy of species i at its

prevailing partial pressure. This is true of ideal mixtures, for example, mixtures of permanent gases at pressures well below the critical pressure. However, in the most general case, the relationship of h to x_i is nonlinear. The reader must turn to specialized works for advice.

F. Specific heat capacities

It is not the purpose of this handbook to supply all the knowledge of thermodynamics that a heat exchanger designer will need, but rather to refresh the designer's memory about the most commonly needed concepts. In relation to specific heat capacities at constant volume and constant pressure, therefore, it suffices to give their definitions and to indicate that they can often be regarded as constants for the material.

The definitions of the two specific heat capacities are, respectively,

$$c_v = \left(\frac{\partial u}{\partial T} \right)_v \quad (10)$$

and

$$c_p = \left(\frac{\partial h}{\partial T} \right)_p \quad (11)$$

where the right-hand sides denote differentiation with respect to temperature, with v or p being held constant when the differentiated quantity is u or h , respectively.

In general, c_v and c_p for any particular substance are functions of temperature and pressure. However, they are often slowly varying properties and, over the range of temperatures likely to be encountered in a heat exchanger, the variations can frequently be neglected. This is the practice that is followed throughout most of this handbook.

A consequence is that it is possible to represent the specific enthalpy of a single-phase material by the relationship

$$h = c_p(T - T_0) \quad (12)$$

where T_0 is an arbitrarily chosen base temperature.

For an ideal mixture, the specific enthalpy can be correspondingly written as

$$h = \sum_{all\ i} x_i c_{p,i}(T - T_{0,i}) \quad (13)$$

where $c_{p,i}$ and $T_{0,i}$ are the c_p and T_0 of species i .

This is a useful working approximation for nonideal mixtures also, but of course the problem of ascribing a value to the $c_{p,i}$'s has to be solved.

Sometimes it is useful, for a material of which the specific heat varies with temperature, to define the average specific heat, $\bar{c}_p$, by

$$\bar{c}_{p,1,2} \equiv \frac{\int_{T_1}^{T_2} c_p dT}{T_2 - T_1} \quad (14)$$

which has of course the same meaning as

$$c_{p,1,2} \equiv \frac{h_2 - h_1}{T_2 - T_1} \quad (15)$$

Because steady flows, to which c_p is more relevant than c_v , are so prevalent in heat exchanger practice, the symbol c is sometimes used without subscript, to stand for the constant-pressure specific heat capacity, c_p .

G. Interphase equilibrium

In all practical heat exchanger situations, the fluids immediately on either side of an interface between a liquid and a vapor, or between two liquids, can almost always be taken as being in thermodynamic equilibrium, even though fluxes of heat and matter across the interface are in progress. The reason is that although the finite fluxes must entail some departures from equilibrium, the actual magnitudes of these are often negligibly small. The only exceptions to this rule would occur at very low pressures (far below those prevailing, for example, in any steam condenser), in condensation of liquid metals, and in fast transients.

Consequences of the existence of equilibrium are that (1) the temperatures of the gas and liquid immediately on either side of the interface are equal, and (2) the mass fractions of a given species of the mixture on either side of the interface are related to each other, and to the temperature and pressure, in a known way. Thus,

$$x_{i,g} = x_{i,g}(x_{i,l}, p, T) \quad (16)$$

where $x_{i,g}$ and $x_{i,l}$ are respectively the gas- and liquid-side mass fractions of species i , and p and T are respectively the pressure and temperature.

An example of special importance is that of the mass fraction of steam in air, adjacent to a body of pure water, as illustrated in Fig. 1. Since, for the water, $x_{i,l}$ equals 1.0 by definition, the relationship is

$$x_{H_2O,g} = x_{H_2O,g}(p, T) \quad (17)$$

To be more specific, Eq. (17) may be rewritten, with the presumption that the molecular weights of steam and air are 18.0 and 29.0, respectively, as

$$x_{H_2O,g} = \left(\frac{29.0}{18.0} \cdot \frac{p}{p_{H_2O}} - \frac{11.0}{18.0} \right)^{-1} \quad (18)$$

where p_{H_2O} stands for the partial pressure of water vapor in contact with liquid water at the prevailing temperature, and is obtainable from steam tables.

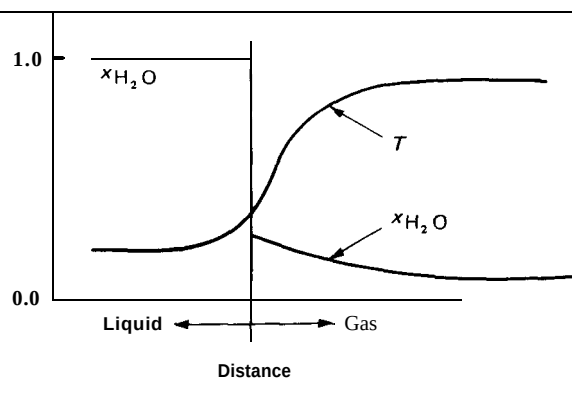


Figure 1 Distributions of T and x_{H_2O} along a normal to the interface between water (on the left) and air (on the right).

When p_{H_2O} equals p , $x_{H_2O,g}$ equals unity, for here the temperature equals the boiling point. For all lower values of p_{H_2O} , $x_{H_2O,g}$ is less than unity. Equation (18), together with the $p_{H_2O} \approx T$ relationship from steam tables, is needed for the design of direct-contact cooling towers, of drying plants, and of steam condensers.

H. Conservation laws

The steady-flow energy equation, in the form relevant to heat exchangers, has already been cited as Eq. (3). There is a corresponding equation for the conservation of a chemical species i :

$$\Delta(\dot{M}x_i) = \dot{M}_i \quad (19)$$

where $\dot{M}$ stands, as before, for the total mass flow rate in the stream, summed for all species, and $\dot{M}_i$ stands for the mass transfer rate of species i into the stream; it is presumed, in this relationship, that all components of the mixture travel at the same velocity, that is, that axial diffusion is absent. In contrast to Eq. (3), Eq. (19) has the $\dot{M}$ inside the bracket on which A operates; for $\dot{M}$ varies as a result of the mass transfer, as a rule. Indeed, there exists an equation for the conservation of total mass:

$$(\dot{M}) = \sum_{\text{all } i} \dot{M}_i \quad (20)$$

These conservation laws, coupled with equations for the heat and mass fluxes, are the foundation of the theory of heat exchangers.

It should be noted that h and x were first defined as local values and then used, in Eqs. (3) and (19), as values representative of a stream in a duct. Since h and x are often not uniform across such a stream, the h and x used in the two cited equations are, strictly speaking, flow average quantities.

Nomenclature for Section 1.2.1 appears at the end of Section 1.4.3.

1.2.2

Flux relationships

D. Brian Spalding

A. Individual and overall heat transfer coefficients

Equation 1.1.2(3) is repeated here, by way of a reminder:

$$\dot{q} = U(T_1 - T_2) \quad (1)$$

where $\dot{q}$ is the heat flux from stream 1 to stream 2, having cross-sectional average temperatures T_1 and T_2 , per unit area of interface, and U is the overall heat transfer coefficient.

It is now time to raise and answer the questions of what precisely is meant by the interface area, and what is the relation of U to the individual resistances to heat transfer represented by the solid wall (if it exists)

separating the two fluids, the deposits of dirt that may cling to the surfaces of the wall, and the low-velocity fluid films that adjoin those deposits or, if they are absent, the wall itself.

Figure 1 illustrates the temperature profile discontinuities of slope at locations *a* (inner surface of inner deposit), *b* (outer surface of inner deposit and inner surface of tube wall), *c* (outer surface of tube wall and inner surface of outer deposit), and *d* (outer surface of outer deposit). It can also be seen that the fluid temperatures lie both above and below T_1 and T_2 , for these are defined as flow averages, that is, averages for the duct cross sections in which each element of area is weighted by the specific heat c times the local mass flow rate ρu . Thus,

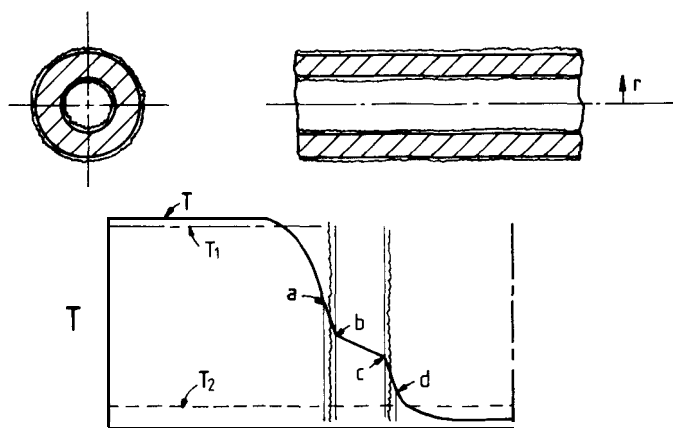


Figure 1 Illustration of the temperature distribution in the neighborhood of a tube wall, with dirt deposits on its inner and outer surfaces, across which heat is transferred from an inner fluid at bulk temperature T_1 to an outer fluid at bulk temperature T_2 .

$$T_1 \text{ or } 2 \equiv \left(\frac{\int c_p \rho v T dS}{\int \rho v T dS} \right)_{1 \text{ or } 2} \quad (2)$$

where the averages are taken over the whole of the cross section of the ducts in question.

Let α_{1a} and α_{d2} now represent the film heat transfer coefficients on the inside and outside of the tube, respectively; let α_{ab} and α_{cd} represent heat transfer coefficients for the two dirt deposits; and let α_{bc} represent the heat transfer coefficient for the tube wall itself. α_{1a} and α_{d2} will have to be obtained from one of the correlations discussed later; and α_{ab} , α_{bc} , and α_{cd} will all be related to the relevant material thicknesses δ and thermal conductivities λ by

$$\alpha_{ab} = \frac{\lambda_{ab}}{\delta_{ab}} \quad (3)$$

$$\alpha_{bc} = \frac{\lambda_{bc}}{\delta_{bc}} \quad (4)$$

$$\alpha_{cd} = \frac{\lambda_{cd}}{\delta_{cd}} \quad (5)$$

Each heat transfer coefficient, to be meaningful, must be associated with an area A . When this is done, the fact that the heat flow through all the elements in question is the same leads to

$$\begin{aligned} UA(T_1 - T_2) &= \alpha_{1a} A_{1a}(T_1 - T_a) \\ &= \alpha_{ab} A_{ab}(T_a - T_b) \\ &= \alpha_{bc} A_{bc}(T_b - T_c) \\ &= \alpha_{cd} A_{cd}(T_c - T_d) \\ &= \alpha_{d2} A_{d2}(T_d - T_2) \end{aligned} \quad (6)$$

Here A_{ab} , A_{bc} , A_{cd} can most appropriately be taken as the arithmetic means of the inner and outer surface areas of the materials in question; and A_{1a} and A_{d2} are best taken as the surface areas pertaining to locations a and d , respectively. The area A without subscript, associated with U , requires arbitrary definition; it can be set equal to any of the other areas, for example.

Elimination of T_a , T_b , T_c , and T_d now leads to the following important connection between the overall coefficient U and the individual coefficients α_a etc.:

$$U = \left(\frac{A}{\alpha_{1a} A_{1a}} + \frac{A}{\alpha_{ab} A_{ab}} + \frac{A}{\alpha_{bc} A_{bc}} + \frac{A}{\alpha_{cd} A_{cd}} + \frac{A}{\alpha_{d2} A_{d2}} \right)^{-1} \quad (7)$$

Evidently, the relationships is best understood as implying that the reciprocal of UA is the overall resistance to heat transfer, and that this is equal to the sum of the individual resistances $(\alpha_{1a} A_{1a})^{-1}$, $(\alpha_{ab} A_{ab})^{-1}$, and so on.

In the case of heat transfer across a plane wall, all the areas take identical values. Then Eq. (7) reduces to

$$U = \left(\frac{1}{\alpha_{1a}} + \frac{1}{\alpha_{ab}} + \frac{1}{\alpha_{bc}} + \frac{1}{\alpha_{cd}} + \frac{1}{\alpha_{d2}} \right)^{-1} \quad (8)$$

This equation is also often used for tubular heat transfer surfaces, by the device of arbitrarily selecting one of the areas, for example, A , as the reference area in all the transfer coefficient definitions.

B. Individual and overall mass transfer coefficients

Although the mass transfer coefficient β was introduced in Eq. 1.1.2(4) in association with U , the overall heat transfer coefficient, the reasonable presumption that β too is an overall coefficient was not there disclaimed; it is time now to disclaim it. The reason is that, although it is possible to treat β as an overall coefficient, it is rarely useful to do so because of the different roles of concentration in the two phases that the interface usually separates.⁺

As a rule, when mass transfer occurs across a phase interface, there are only two resistances to consider: that opposing diffusion between the bulk of the first phase and the interface, and that opposing further diffusion from the interface onward to the bulk of the second phase.

For the sake of concreteness, let the first phase be a gas and the second a liquid, and let attention be focused on the transfer rate per unit area of the i th component of the mixture, $\dot{m}_i$. This quantity can be related to gas-side and liquid-side coefficients, $\beta_{i,g}$ and $\beta_{i,l}$, respectively, thus:

$$\dot{m}_i = \beta_{i,g} \rho_g (x_{i,1} - x_{i,g}) \quad (9)$$

$$\dot{m}_i = \beta_{i,l} \rho_l (x_{i,l} - x_{i,2}) \quad (10)$$

Here $x_{i,g}$ and $x_{i,l}$ are, respectively, the mass fractions of component i in the gas and liquid phases immediately adjacent to the phase interface; $x_{i,1}$ and $x_{i,2}$ are of course the flow average mass fractions of the two streams; ρ_g and ρ_l are the densities of the two phases at interface conditions.

To proceed further, one must introduce a term, sometimes called the Henry number, He , for the ratio of gas-side to liquid-side mass fraction in thermodynamic equilibrium:

[†]If two gas-phase streams, for example, were separated by a porous metal wall, and some component of the gaseous mixture were to diffuse from one stream to the other, there would be an exact parallelism with heat transfer, and the notion of an overall coefficient as the reciprocal of the sum of individual coefficients would be useful. However, this situation rarely arises.

$$\text{He} \equiv \frac{x_{i,g}}{x_{i,l}} \quad (11)$$

With the aid of this quantity, both $x_{i,g}$ and $x_{i,l}$ can be eliminated from Eqs. (9) and (10). The result is

$$\frac{\dot{m}_i}{x_{i,1} - \text{He } x_{i,2}} = \left(\frac{1}{\beta_g} + \frac{\text{He}}{\beta_l} \right)^{-1} \quad (12)$$

The expression on the right can obviously be regarded as a kind of overall mass transfer coefficient; but it is worth noting that to yield the mass transfer rate, it must be multiplied not by $(x_{i,1} - x_{i,2})$ but by $(x_{i,1} - \text{He } x_{i,2})$; so it is not the overall coefficient that was in mind when Eq. 1.1.2(4) was introduced.

Fortunately, in many problems it is the gas-side resistance alone that is important. There is therefore no need to pursue the matter further here.

C. Simultaneous heat and mass transfer

In the remainder of this section, attention is confined to the air-steam-water system. The problem considered is how one can calculate the heat transfer rate when both heat transfer and mass transfer take place simultaneously at the air-water interface.

The general answer is that it is necessary to solve simultaneously the equations governing:

Liquid-side heat transfer

Gas-side heat transfer

Gas-side mass transfer

The equilibrium composition on the gas side of the interface

The energy equation accounting for the latent heat of vaporization

These five equations permit the following five unknowns to be determined:

$\dot{q}_l$, the liquid-side flux

$\dot{q}_g$, the gas-side heat flux

$\dot{m}_{\text{H}_2\text{O}}$, the vaporization (or condensation) rate

$x_{\text{H}_2\text{O},g}$, the gas-side mass fraction+ of steam at the interface

$T_{g,l}$, the temperature at the interface

[†]In the present work, the concentration variable $x_{\text{H}_2\text{O}}$ represents the mass of water vapor per unit mass of mixture. However, it is also common to represent the composition in terms of the ratio of the mass of H_2O to the mass of air (often called dry air). The advantage of so doing is that, for practical purposes (air being scarcely soluble in liquid water), the reference mass is often a constant, whereas the total mass of vapor-phase mixture, flowing in a cooling tower for example, varies with its moisture content. For the same reason, the specific enthalpy is also presented, in specialist publications, in terms of joules per kilogram of dry air. This practice is not adopted in Part 1 of this handbook because the advantage does

This matter is discussed at length in [1], where an account will also be found of the simplified treatment that now follows. The essential points to note are the following:

1. The water-side heat transfer coefficient is ordinarily much greater than the gas-side coefficient in cooling tower and drying plant situations, and so exerts little resistance to the transfer process.

2. It is the heat transfer rate to (or from) the liquid, $\dot{q}_l$, which is of major interest.

3. By a fortunate accident, if the specific enthalpies of both air and water are defined to be zero for 0°C , the equations reduce to

$$\dot{q}_l = \beta_g \rho_g (h_1 - h_2) \quad (13)$$

where h_1 is the specific enthalpy of the bulk of the air stream and h_2 is the specific enthalpy of air saturated with water vapor at the temperature of the bulk of the water stream, T_2 .

4. The expression for h_1 is

$$h_1 = [(1 - x_{\text{H}_2\text{O},1})c_{p,\text{air}} + x_{\text{H}_2\text{O},1}c_{p,\text{steam}}](T_1 - T_0) + x_{\text{H}_2\text{O},1}\Delta h_0 \quad (14)$$

where T_0 is 0°C and Δh_0 is the latent heat of vaporization of steam at that temperature.

The expression for h_2 , on the other hand, is correspondingly

$$h_2 = [(1 - x_{\text{H}_2\text{O},\text{sat},2})c_{p,\text{air}} + x_{\text{H}_2\text{O},\text{sat},2}c_{p,\text{steam}}] \times (T_2 - T_0) + x_{\text{H}_2\text{O},\text{sat},2}\Delta h_0 \quad (15)$$

where $x_{\text{H}_2\text{O},\text{sat},2}$ is the mass fraction of steam in saturated air at T_2 .

5. The value of $\beta_g \rho_g$ is equal to the value of $\alpha_g/c_{p,\text{air}}$.

Consideration of Eqs. (13)-(15) shows that the heat transfer rate to the water increases with both the temperature and the steam content of the air, and it is therefore not zero when the temperatures of the two phases are equal.

The first recognition of the approximate validity of the equations for air and water under near-atmospheric conditions appears to have been made by Merkel [2].

D. General remarks

The concepts of the heat transfer and mass transfer coefficients are so convenient, and so reminiscent of electrical circuit theory, that there is some danger that their validity may be presumed to be unlimited. A few remarks on their limitations are therefore in order.

not justify the departure from standard thermodynamic conventions. However, the reader of specialized works is warned to take careful note of the units employed.

First, it must be pointed out that the equations of Sec. C already indicate a kind of breakdown of the heat transfer coefficient concept, for the heat flux (to the liquid) is not proportional to the temperature difference alone. Other examples could be found, for example, those involving chemical reaction rather than phase change.

Second, the values of the coefficients are by no means independent of the values of the property differences that multiply them. In the case of heat transfer by natural convection, it is well known that the coefficient increases with the imposed temperature difference. In the case of mass transfer, there are further nonlinearities, which are discussed extensively in [1].

Third, there are circumstances in which nonuniformities of temperature or concentration along the interface are so steep as to affect the values of the transfer coefficients, and even to change their sign. In these circumstances, it is best to work with the fluxes and property differences directly, avoiding all reference to coefficients.

Fourth, and the last remark notwithstanding, it is useful to generalize the transfer coefficient concept by way of the definition of a conductance, g_ϕ , which enters the following relationships:

$$\dot{m}_n \equiv g_\phi B_\phi \quad (16)$$

$$B_\phi \equiv \frac{\phi_b - \phi_i}{\phi_i - \phi_t} \quad (17)$$

Here B_ϕ is a dimensionless driving force for mass transfer, and ϕ is a conserved property of the fluid, either concentration or specific enthalpy. The subscripts b, i , and t stand, respectively, for bulk of fluid stream; interface on the stream side; and transferred substance. Here $\dot{m}_n$ is defined as positive when the mass transfer is directed from the interface into the bulk of the fluid.

The full implications of Eqs. (16) and (17) are

explained in [1], but the following notes suffice for present purposes:

g_ϕ has the units $\text{kg}/\text{m}^2 \text{ s}$.

g_ϕ may be identified, in simple heat transfer circumstances, as α/c_p , with ϕ identified as h , and $\dot{q}$ as $\dot{m}_n/(h_i - h_t)$.

g_ϕ may be identified as $\beta_i \rho_i$, with ϕ standing for x_i , when the values of x_i are much smaller than unity, that is, the mixtures are dilute.

In many circumstances it is advantageous to think in terms of g_ϕ rather than of α or β , because g_ϕ can be readily given a physical significance as the Reynolds flux, and because it can often be given an approximate numerical value with little labor of computation.

The Reynolds flux is the mass of material from the bulk of the stream, per unit interface area and time, which, if it were brought into equilibrium with the interface, would effect the transfer of heat and mass that actually occurs. It is just such a bringing-into-equilibrium with respect to velocity, temperature, and concentration that is envisaged in the thinking that leads to the well-known Reynolds analogy.

Another significance of g_ϕ is that it is equal to the exchange coefficient of entity ϕ divided by the thickness of a stagnant film of fluid that exerts the major resistance to heat and mass transfer; the exchange coefficient is equal to the diffusion coefficient times the density when ϕ stands for concentration, and to the thermal conductivity divided by the specific heat when ϕ stands for enthalpy. The notion of the stagnant film is the conceptual foundation of many treatments of mass transfer in chemical engineering texts, and indeed it has great physical appeal. The phenomenon of diffusion through a stagnant film, in this context, is sometimes called a Stefan flow.

Nomenclature and References for Section 1.2.2 appear at the end of Section 1.4.3.

1.2.3

Transfer coefficient dependences

D. Brian Spalding

A. Introduction

Whether in terms of α , β , or g_ϕ , a major part of the analyst's concern is with ascribing numerical values to the interaction coefficients. These depend on the properties of the fluid, on the velocity of flow over the surface, on the shape and size of the interface, and on other factors.

What is derived is a formula, or set of formulas, of the kind

$$\alpha = \alpha(\lambda, L, c_p, \rho, \nu, \eta, AT, \dots) \quad (1)$$

where $\alpha(\dots)$ represents “ α as a function of $\dots$,” and the quantities in the bracket represent all those expected to influence the value of α , namely, the thermal conductivity, a linear dimension, the specific heat, the density, the velocity, the viscosity, the temperature difference, and so on.

Such formulas exist, although not yet in sufficient measure, and they are usually expressed, for compactness and for maximum generality, in terms of dimensionless quantities such as the Stanton or Reynolds numbers. Therefore, in order to facilitate the presentation of the formulas, the next few sections are devoted to introducing and discussing the more important of these quantities.

B. Stanton number?

The Stanton number is the ratio of the conductance to the mass velocity of the stream. Thus, in terms of the general conductance g_ϕ , the definition is

[†]In French literature, the Margoulis number is used for what is here called the Stanton number.

$$St_\phi \equiv \frac{g_\phi}{\dot{m}} \quad (2)$$

where $\dot{m}$ is the mass velocity of the stream flowing over the surface (i.e., the product of the density ρ and the velocity v), and St is the Stanton number related to property ϕ .

Usually the Stanton number is connected with heat transfer alone. Then the definition can be regarded as implying that

$$St = \frac{\alpha}{c_p \dot{m}} \quad (3)$$

which is the conventional relation.

In practice, Stanton numbers often lie in the range 0.001-0.01. They have the significance of the ratio of the Reynolds flux to the mass flux of the mainstream.

A remark that must be made about all the dimensionless quantities that will be introduced is that when the properties of the fluid vary within the field, it is not enough to refer to the specific heat c_p ; one must add, for example, evaluated at the bulk field condition.

Indeed, it is not sufficient to refer to the mass velocity of the stream without qualification: one could have in mind, for example, the total mass flowing in a duct divided by the cross-sectional area of the duct; or the maximum mass velocity in the cross section; or the mass velocity far upstream of the point in question; and so on. There exist no universally agreed prescriptions for such auxiliary aspects of the definitions.

C. Nusselt and Sherwood numbers

The heat transfer coefficient α appears also in another dimensionless quantity, the Nusselt number Nu , defined by

$$Nu \equiv \frac{\alpha L}{\lambda} \quad (4)$$

where L is a typical dimension of the apparatus, and λ is the thermal conductivity of the fluid. Obviously, further prescription is needed of precisely what dimension is in question (such as duct diameter), and just where the thermal conductivity is to be measured (such as at the interface, at the bulk condition, or in some reference state).

A simple example of a heat transfer formula embodying the Nusselt number is

$$\frac{\alpha D}{\lambda} = 2 \quad (5)$$

This is valid for heat transfer to the surface of a sphere of diameter D from an infinite stagnant medium of thermal conductivity λ in which the sphere is suspended. This formula is useful when, for example, it is necessary to compute the rate of temperature rise and the rate of subsequent vaporization when a spray of droplets is injected into a hot gas.

An analogous quantity for mass transfer is the Sherwood number Sh , defined by

$$Sh \equiv \frac{\beta L}{\delta} \quad (6)$$

where δ is the diffusion coefficient of the species in question.

The values of the Nusselt and Sherwood numbers vary more widely, in practice, than do those of the Stanton number. Usually their values exceed unity, for they can be given the significance of the ratio of L to the thickness of the laminar layer adhering to the interface (the so-called stagnant film) that would have the effect of resisting transfer to the same extent as the actual flow. This film is much thinner than the typical dimension as a rule.

Both the Nusselt and the Sherwood numbers, when expressed in terms of the relevant conductance, g_ϕ , are equivalent to the quantity $g_\phi L / \Gamma_\phi$, where Γ_ϕ is the **exchange coefficient** for the entity ϕ . This will be recognized, after perusal of the next section, as a Peclet number based on the Reynolds flux.

The exchange coefficient Γ_ϕ is equal to λ/c_p when heat transfer is in question, and to $\rho\delta$ when mass transfer is in question. Its units, in either case, are kg m/s; these, it is worth noting, are the same as those of viscosity.

D. The friction coefficient f

A dimensionless quantity frequently appearing in heat transfer studies is the friction coefficient f , defined by

$$f \equiv \frac{\tau}{\rho v^2} = \frac{\tau}{v \dot{m}} \quad (7)$$

where τ is the shear stress exerted by the fluid at its boundary, ρ is the fluid density, and v is the bulk velocity of the fluid. Other symbols for f may be found in the literature, for example, $c_f/2$. Further, the same symbols may be found having **significances** differing from those used here by factors of 2, 4, 8, etc.; the reader should be **wary**.[†]

The friction coefficient is in many respects akin to the Stanton number. Indeed, the famous Reynolds analogy between heat transfer and friction can be expressed by the equality

$$St \approx f \quad (8)$$

in which, as throughout below, f is defined by Eq. (7).

This approximate equality is closest to the truth when the flow is of a gas along a smooth wall. If the **wall** is made rough, St falls below f , as it does also for a nonmetallic liquid; for a liquid metal, on the other hand, St is usually much greater than f .

Equation (8), despite its approximate character, is still useful to designers in that it usually provides results of order-of-magnitude correctness, with very little computational labor.

E. Reynolds and Peclet numbers

If the mass velocity in the stream, $\dot{m}$, is multiplied by the typical length and divided by the viscosity of the fluid, η , the resulting dimensionless quantity is called the Reynolds number, Re :

$$Re \equiv \frac{\dot{m} L}{\eta} \quad (9)$$

The Reynolds number measures the relative importance of the inertial stresses to the viscous stresses, for, if v is a typical velocity, so that $\dot{m}$ equals ρv , it is clear that Eq. (9) can be written as

[†]The so-called Fanning friction factor is defined as $\tau/(\rho v^2/2)$, that is, as one-half of the above f . Schlichting [41], on the other hand, employs a dimensionless friction coefficient A , which equals one-quarter of the Fanning factor and therefore one eighth of the f of Eq. (7).

Probably the Fanning factor is the most widely used; and it appears in the well-known Moody chart (Fig. 1), representing the resistance to steady flow of a fluid in a pipe of circular cross section. The abscissa is the Reynolds number (see below), and the parameter ϵ/D is the ratio of the roughness dimension ϵ to the pipe diameter D .

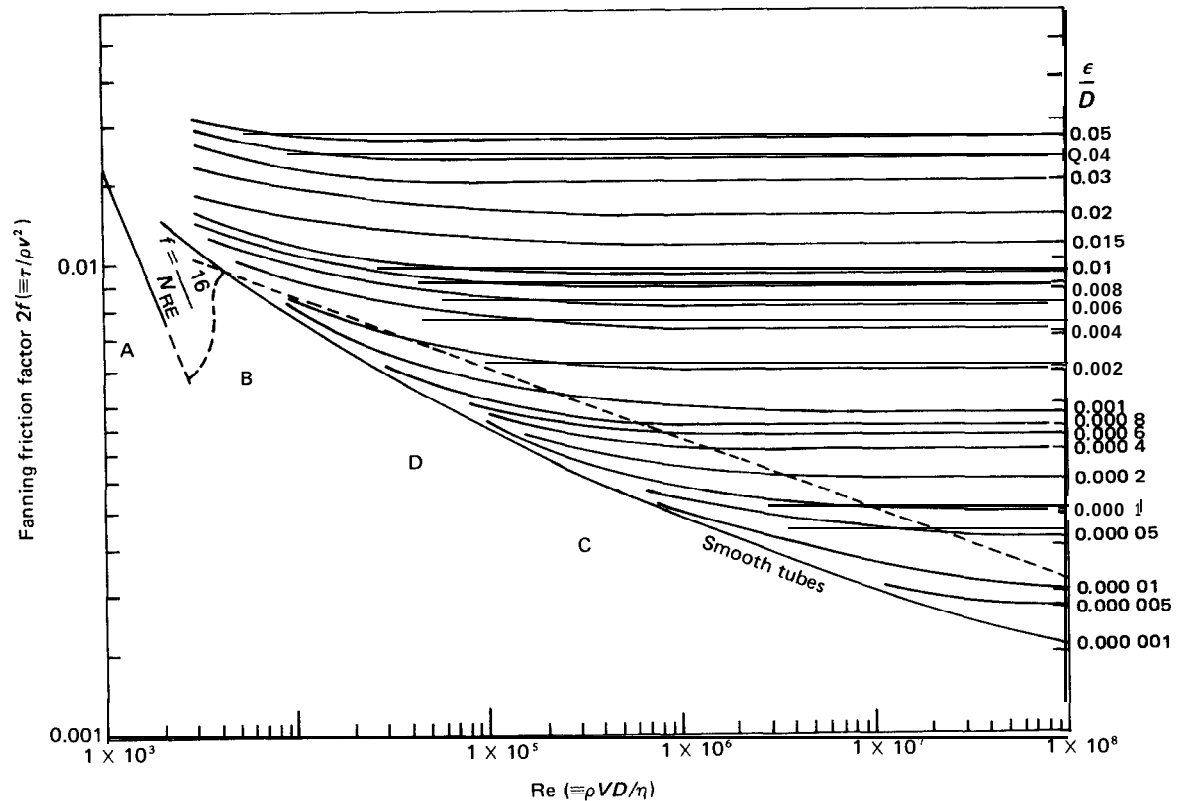


Figure 1 Fanning friction factors. Adapted from *Perry's Chemical Engineers' Handbook*, McGraw-Hill, 1963, p. S-20, Fig. S-25.

$$\text{Re} = \frac{\rho v^2}{\eta v/L} \quad (10)$$

where the numerator represents the inertial stress and the denominator is the viscous stress.

The Reynolds number plays an important part in all formulas representing the hydrodynamic behavior of real fluids. For example, the friction factor for fully developed flow in a smooth-walled pipe can be represented by the formula

$$f = \frac{16}{\text{Re}} \quad (11)$$

for laminar flow, and

$$f \approx 0.026 \text{Re}^{-0.2} \quad (12)$$

for turbulent flow, where the L appearing in Re is the pipe diameter.

If Γ_ϕ appeared in the denominator of Eq. (9) instead of η , the resulting dimensionless quantity would be called the Peclet number, Pe_ϕ , of property ϕ :

$$\text{Pe}_\phi \equiv \frac{GL}{\Gamma_\phi} \quad (13)$$

Usually, Peclet numbers are referred to only in connection with heat transfer, for which the definition becomes

$$\text{Pe} \equiv \frac{c_p \dot{m} L}{\lambda} \quad (14)$$

Both Reynolds and Peclet numbers vary over very wide limits in practical equipment, for example, in the range 10 – 10^5 . Of course, the precise definition of L , for example, influences the magnitude.

In a few situations, it is possible to relate the Nusselt or Stanton number to the Peclet number alone. Thus, when heat transfer occurs between a liquid metal stream in laminar motion and a flat plate, its rate is given by

$$\text{Nu} = 0.565 \text{Pe}^{1/2} \quad (15)$$

where the length appearing in the definition of Nu and Pe is that of the plate. More usually, **both** the Reynolds number and the Peclet number influence the heat transfer rate.

F. Prandtl, Schmidt, and Lewis numbers

The ratios of the transport properties of a fluid in respect of momentum, heat, and material transport are given names and symbols as follows:

Momentum/heat, the Prandtl number

$$\text{Pr} \equiv \frac{\eta}{\lambda/c_p} = \frac{c_p \eta}{\lambda} \quad (16)$$

Momentum/matter, the Schmidt number

$$\text{Sc} \equiv \frac{\eta}{\rho \delta} = \text{Pr} \text{ Le} \quad (17)$$

Heat/matter, the Lewis[†] number

$$\text{Le} \equiv \frac{\lambda/c_p}{\rho \delta} = \frac{\text{Pr}}{\text{Sc}} \quad (18)$$

For gases, all these quantities are of the order of unity, because heat, mass, and momentum transport take place by way of molecular motion. For nonmetallic liquids, the Prandtl numbers and Lewis numbers exceed unity, and the Schmidt numbers, being the product of the Prandtl and Lewis numbers, are still higher; the reason is that momentum transport is easy in liquids, that heat transport is less so, and that diffusion takes place only with difficulty.

In liquid metals, the motion of the electrons facilitates heat transport. Therefore Prandtl numbers are much smaller than unity, and Lewis numbers are much greater than unity.

It has been remarked above that both the Reynolds and Peclet numbers influence the Nusselt and Stanton numbers, as a rule. Usually, however, only the Reynolds number appears in the relevant formula; the place of the other is taken by the Prandtl or Schmidt number, this substitution being made possible because both the latter have the nature of the Peclet number divided by the Reynolds number. Some examples will now be given.

For the flat plate in laminar flow, with moderate Pr,

$$\text{Nu}_{\text{loc}} = 0.332 \text{ Re}^{1/2} \text{ Pr}^{1/3} \quad (19)$$

For the flat plate in turbulent flow, with moderate Pr,

$$\text{Nu}_{\text{loc}} \approx 0.0295 \text{ Re}^{0.8} \text{ Pr}^{1/3} \quad (20)$$

For fully developed turbulent flow in a smooth-walled tube,

$$\text{Nu}_{\text{loc}} \approx 0.023 \text{ Re}^{0.8} \text{ Pr}^{1/3} \quad (21)$$

For flow outside and normal to a circular-sectioned tube, deep within a tube bank,

[†]Sometimes the Lewis number is defined as the reciprocal of the expression given here.

$$\text{Nu}_{\text{arc}} \approx 0.3 \text{ Re}^{0.6} \text{ Pr}^{1/3} \quad (22)$$

The Chilton-Colburn analogy between friction and heat transfer is

$$\text{St} \approx f \text{ Pr}^{-2/3} \quad (23)$$

In all these relations, the quantity L should be taken as either plate length or tube diameter, according to the configuration in question.

G. Grashof and Rayleigh numbers

When body forces play a significant role, it is customary to represent them in terms of dimensionless groups in which their magnitudes are compared with those of viscous shear. Usually the body force is that of gravity, measured by the gravitational acceleration g_n . The two most commonly occurring dimensionless quantities are

The Grashof number

$$\text{Gr} \equiv L^3 g_n \rho \frac{\Delta \rho}{\eta^2} \quad (24)$$

The Rayleigh number

$$\text{Ra} \equiv L^3 g_n \rho c_p \frac{\Delta \rho}{\eta \lambda} \quad (25)$$

where $\Delta \rho$ is a typical density change associated with the changes of temperature or composition.

Examples of heat transfer formulas involving the Grashof number are as follows:

For a vertical flat plate in laminar flow,

$$\text{Nu} = 0.535 \text{ Gr}^{1/4} \text{ Pr}^{1/3} \quad (26)$$

For a vertical flat plate in turbulent flow,

$$\text{Nu} \approx 0.135 \text{ Gr}^{1/3} \text{ Pr}^{1/3} \quad (27)$$

Grashof and Rayleigh numbers can attain large values (3.6×10^9) in practice; usually, when there is an imposed (forced convection) flow, body-force effects are insignificant unless Gr exceeds, say, 10^3 .

From Eqs. (24), (25), and (16), it is evident that the following identity holds:

$$\text{Ra} \equiv \text{Gr} \text{ Pr} \quad (28)$$

H. Other dimensionless quantities

Whenever two commensurable physical quantities are brought into relationship, a dimensionless ratio can result; and usually the name of some scientific worker can be found to be appropriate to it. Some of these names are thought to be useful, and are remembered; others fall into decay. Among the more memorable in heat transfer contexts are the following:

The Fourier number

$$Fo = \frac{\lambda t}{c_p \rho L^2} \quad (29)$$

where t stands for time; it can be regarded as dividing elapsed time by the time that is typically sufficient to effect temperature equalization.

The Froude number

$$Fr \equiv \frac{v^2}{Lg_n} \quad (30)$$

This represents the relative importance of fluid inertia and gravitational potential, and is relevant to the flow of liquids with free surfaces.

The Graetz number

$$Gz \equiv Pe \frac{D}{\phi L} \quad (31)$$

where D is the cross-stream dimension of a duct, which also appears in the Pe definition, and L is the distance from its entrance; the Nusselt number in laminar pipe flow depends on it.

The Weber number

$$We = \frac{v^2 \rho L}{\sigma} \quad (32)$$

where σ stands for the surface tension of the fluid; this influences, for example, the tendency of liquid droplets to become fragmented while in flight.

There are many other dimensionless quantities that though they lack names, will be found to influence heat transfer behavior. These include the following:

Geometric ratios, for example, length/diameter; pitch/diameter; free volume/total volume

Velocity ratios, for example, swirl velocity/axial velocity; streamline angle

Temperature ratios, for example, stream temperature/surface temperature

Enthalpy ratios, for example,

$$c_p \frac{T_{in} - T_{out}}{Ah}$$

where Ah stands for the latent heat of vaporization

Other property ratios, for example, vapor density/liquid density; fluid conductivity/solid conductivity

These, and many more, will make their appearance in this handbook.

I. Correlations

The heat and mass transfer coefficients needed by the heat exchanger designer must be obtained, as has already been stated, from formulas connecting them with the

known (or guessed) conditions of the flow situations. These formulas may represent the results of a theoretical analysis or may derive from experimental data, but they will nearly always be expressed in terms of dimensionless groups. The typical form, of which examples have been encountered above, is

$$St = St-Re, Pr, Gr, \frac{L}{D}, \frac{\eta_1}{\eta_2}, \frac{\rho_1}{\rho_2}, \dots - \quad (33)$$

Frequently, the formulas have power law form; for example,

$$St = \text{constant } Re^{\text{exponent}} Pr^{\text{exponent}} \quad (34)$$

and, if so, they are usually valid over only a restricted range of Reynolds and Prandtl numbers. Sometimes the relations are represented by way of graphs or nomograms, although the widening use of the digital computer is rendering these obsolete.

Because relationships such as Eqs. (33) or (34) are so often derived by correlating experimental data and finding formulas to fit, they are usually called correlations. Many will be found later in this handbook. Here it is necessary to make only general remarks:

Correlations are rarely capable of predicting the transfer coefficients accurately within $\pm 10\%$, partly because of inaccuracies in the experiments from which they were derived, and partly because the conditions to which they must be applied are seldom precisely those for which they are valid. They must therefore be used with caution.

Despite the facts that heat transfer research has been in progress for many years, and that publications of that research clog library shelves, the designer frequently searches in vain for correlations that will fit the conditions even approximately. This is partly a consequence of the inaccessibility of most published material, a defect that this handbook seeks to remedy, but it is mainly a consequence of the fact that if it takes 10 experiments to determine the influence of a **single**-argument function, it would take 10^6 experiments to determine the behavior of a six-argument function like that of Eq. (33); and, in truth, more than six dimensionless quantities may influence the Stanton number. Although the experienced specialist can afford to dismiss some of the arguments as being without influences in the circumstances that concern him or her, it remains true that the experimental path to general correlations is a prohibitively expensive one.

The absence or inadequacy of correlations can be made good, when the need is great, either by the making of expensive ad hoc experimental studies, or by the conduct of detailed theoretical analyses, based on solution of the relevant differential equations. The former is often difficult, expensive, and time consuming;

and the latter may overstrain the available resources of physics, mathematics, or computing power. Discussion of these, in any case, lies beyond the scope of this handbook.

However, although the use of correlations to give accurate heat transfer or pressure drop predictions often leaves much to be desired, their utility is far from negligible; indeed, they are the basis of most practical heat exchanger designs. Very simple correlations can be usefully employed so as to yield order-of-magnitude results, or upper and lower bounds for the heat transfer rates, prior to the initiation of more precise experimental or theoretical studies; and sometimes these cheaply obtained prognostications show that the more precise and expensive studies will not be worth carrying out. This is especially true when effects of fouling, of uncertain physical properties, and of other **imponderables** are impairing the accuracy of prediction.

Sometimes, available experimental data on pressure drop can be interpreted in terms of skin friction, and some kind of analogy between friction and heat transfer can be invoked by means of which the corresponding heat transfer coefficient can be computed. How this may be done is described in the next section.

J. Relations between transfer coefficients for heat, mass, and momentum transfer

At the end of Sec. 1.2.1D, the Reynolds analogy was mentioned; its nature, and the further formulas to which it has given rise, will now be described.

Reynolds [3] in 1874 made a proposal for the relation between friction and heat transfer that has already been expressed in Eq. (8): the Stanton number is approximately equal to the friction factor. His recommendations were based primarily on observations relating to gases.

Widely used by chemical engineers, and founded on experiments covering a wide range of fluids, is the much-later-published **Chilton-Colburn** analogy [4], namely,

$$St = f \sigma^{-2/3} \quad (35)$$

where St stands for the Stanton number, whether for heat or mass transfer, f is the friction factor defined in accordance with Eq. (7), and σ stands for the Prandtl number or the Schmidt number according to whether it is heat or mass transfer that is in question. This relationship implies that the heat transfer coefficient α increases with the thermal conductivity of the fluid, but in accordance with only the two-thirds power; the mass transfer coefficient β is similarly dependent on the appropriate diffusion coefficient.

The Chilton-Colburn analogy implies that the ratios

of heat, mass, and momentum transfer coefficients are independent of the actual level of the friction factor. Starting, however, with Prandtl [5] and Taylor [6], numerous authors have proposed that the relationship between the Stanton number, friction factor, etc., should be of the form

$$(St)^{-1} = b(af^{-1/2} + f^{-1}) \quad (36)$$

where a and b are constants, at least one of which depends on σ . The underlying concept is that the overall resistance to heat and mass transfer usually consists of two parts: one is associated with the semilaminar layer of fluid immediately in the vicinity of the solid wall, and the other is associated with the turbulent fluid lying at a greater distance. The two terms on the right-hand side of Eq. (36) express these two influences quantitatively.

It is in Eq. (3) that depends more strongly on the value of σ (i.e., of Pr or Sc , as appropriate). Expressions for it have become more elaborate with increase in the number of experimental observations to be fitted. Among the best documented are those of Jayatilaka [7], for $\sigma \geq 0.7$, and of Elhadidy [8] for $\sigma < 0.7$. They are

$$\sigma \geq 0.7: \quad a = 8.32 \left[\left(\frac{\sigma}{0.9} \right)^{3/4} 1 \right] \times [1 + 0.29 \exp(-0.0078\sigma)] \quad (37)$$

$$\sigma < 0.7: \quad a = -f^{1/2} + 12.5 \frac{f^{1/2}}{Re} \frac{h}{\sigma} \left(-10.58 \left(\frac{\sigma}{b} \right)^{1.44} - \ln \left[1 + 4.6 \left(\frac{\sigma}{b} \right)^{0.72} + 4.6 \left(\frac{\sigma}{b} \right)^{0.72} \right] + \left(1 + 0.2 \frac{Re}{f^{1/2}} \frac{\sigma}{b} \right) \ln \left(1 + 0.2 \frac{Re}{f^{1/2}} \frac{\sigma}{b} \right) + 0.2 \frac{Re}{f^{1/2}} \frac{\sigma}{b} \left\{ 4.6 \left(\frac{\sigma}{b} \right)^{0.72} - \ln \left[1 + 4.6 \left(\frac{\sigma}{b} \right)^{0.72} \right] - 1 \right\} \right) \quad (38)$$

The value of b , which has the significance of a turbulent Prandtl-Schmidt number, is commonly taken as being around 0.9.

Equation (36), coupled with the recommendations just made as to how a and b are to be computed, permit heat and mass transfer data to be computed from friction data, or from each other for that matter [f being regarded as an unknown to be eliminated between two equations (36), one for each σ]. The procedure is expected to be valid when the transfer surface is smooth and when the Reynolds number of the flow is high enough for the major part of the flow to be turbulent.

When the surface is rough, or when the available values off contain a large component of form drag as distinct from skin friction drag, the values of Stanton number predicted by Eq. (36) are likely to exceed those that will prevail in practice; the equation is then an overoptimistic one from the point of view of the engineer for whom heat transfer is desirable and friction is undesirable.

Friction is undesirable in heat exchangers because it increases the pressure drop across the equipment, and hence the power that must be expended in pumping the fluids through it. Other things being equal, a high ratio of St to f is therefore desired.

Some further implications of the heat transfer-friction interrelationships for equipment design are described in Sec. 1.2.41.

Nomenclature and References for Section 1.2.3 appear at the end of Section 1.4.3.

1.2.4

Balance equations applied to complete equipment

D. Brian Spalding

A. Relation between enthalpy changes

Let a control volume be considered, enveloping a heat exchanger, through which there flow two streams, denoted by subscripts 1 and 2. Then the steady-flow version of the first law of thermodynamics leads to

$$\dot{M}_1(h_{1,\text{out}} - h_{1,\text{in}}) + \dot{M}_2(h_{2,\text{out}} - h_{2,\text{in}}) = 0 \quad (1)$$

where $\dot{M}$ stands for the mass flow rate and h for the specific enthalpy.

Of course, it is here implied that kinetic energy, heat transfer to the outside, and shaft work input are all negligible, but anyone with elementary knowledge of thermodynamics will know how to include the relevant terms when necessary, by reference, for example, to Eq. 1.2.1(3).

Equation (1) is of great use in connecting the inlet and outlet states of the streams that interact in a heat exchanger.

B. Relation between temperature changes

If the two streams passing through the heat exchanger experience no phase change, Eq. (1) can be expressed in terms of temperature changes as follows:

$$\dot{M}_1 c_{p,1}(T_{1,\text{out}} - T_{1,\text{in}}) + \dot{M}_2 c_{p,2}(T_{2,\text{out}} - T_{2,\text{in}}) = 0$$

that is,

$$\dot{C}_1 \Delta T_1 + \dot{C}_2 \Delta T_2 = 0 \quad (2)$$

where $c_{p,1}$ and $c_{p,2}$ are the average specific heats at constant pressure of the two streams, over the indicated

temperature ranges; these quantities multiplied by the mass flow rates are given the symbols $\dot{C}_1$ and $\dot{C}_2$, and the temperature ranges are given the symbols ΔT_1 and ΔT_2 .

It may happen that one of the materials experiences a phase change while the other does not. For example, in the steam generator of a nuclear power plant, the primary fluid (hot water) does not change phase, whereas the secondary one does. In such a case, the appropriate form of equation would be

$$\dot{C}_1 \Delta T_1 + \dot{C}_2 \left(\Delta T_2 - \frac{\Delta h_2}{c_{p,2}} \right) = 0 \quad (3)$$

Here, note has been taken of the phase change in stream 2 by the inclusion of the latent heat of vaporization of the latter, Δh_2 ; and the steam leaving the generator is supposed to be dry saturated, that is, of 100% quality ($x_2 = 1.0$).

C. Relation between concentration changes

When there is a mass transfer process within the apparatus, it may be necessary to carry out a mass balance for a particular chemical species denoted by subscript i . Provided that this species is neither created nor destroyed by chemical reaction within the apparatus, and provided that the flow is steady, the inlet and outlet mass fractions, x_i , of i in the two streams are connected by

$$\dot{M}_{1,\text{out}} x_{1,\text{out}} - \dot{M}_{1,\text{in}} x_{1,\text{in}} + \dot{M}_{2,\text{out}} x_{2,\text{out}} - \dot{M}_{2,\text{in}} x_{2,\text{in}} = 0 \quad (4)$$

where the subscript i to x has been omitted for ease of writing and reading.

Comparison of this equation with Eqs. (1)–(3) reveals that it allows the mass flow rate in a given stream to be different as between inflow and outflow. This is of course a necessity, for the transfer of mixture component i from one stream to another is hardly likely to be balanced by an exactly equal and opposite transfer of other components.

The stream mass fluxes must, of course, obey the overall mass balance relation

$$\dot{M}_{1,\text{out}} - \dot{M}_{1,\text{in}} + \dot{M}_{2,\text{out}} - \dot{M}_{2,\text{in}} = 0 \quad (5)$$

and the differences in the stream mass flow rates can be connected also with the sum of the mass transfers, as follows:

$$\begin{aligned} \dot{M}_{1,\text{out}} - \dot{M}_{1,\text{in}} &= \dot{M}_{2,\text{in}} - \dot{M}_{2,\text{out}} \\ &= \sum_{\text{all } i} \dot{M}_{i,1 \rightarrow 2} \end{aligned} \quad (6)$$

wherein $\dot{M}_{i,1 \rightarrow 2}$ is the mass species i transferred from stream 1 to stream 2.

D. Approximate relation between enthalpies for the water-steam-air system

In cooling tower design, it is useful to make use of certain simplifications in the equations that result, by a fortunate accident, when the enthalpy base is chosen so that h is zero for both air and water at 0° C (see Sec. 1.2.2C). Further, the increase of mass of the air stream, resulting from its increased moisture content, is small enough to be neglected.

In these circumstances, it remains permissible to employ Eq. (1) for the simultaneous heat and mass transfers that occur within the apparatus. To be more specific, one should write

$$\dot{M}_{a,\text{in}}(h_{a,\text{out}} - h_{a,\text{in}}) + \dot{M}_{w,\text{in}}(h_{w,\text{out}} - h_{w,\text{in}}) \simeq 0 \quad (7)$$

where the subscripts a and w , replacing 1 and 2, stand for the air and water, respectively, and the approximate nature of the balance is represented by the $\simeq$ sign.

For safety's sake, it will again be emphasized, Eq. (7) is valid only when the specific enthalpies are defined as in Eqs. 1.2.2(4) and 1.2.2(15). The matter is discussed at considerable length in [1].

E. Average interaction coefficients and driving forces

Although the stream-to-stream coefficients for heat and mass transfer vary from point to point within the equipment, and although the differences of temperature

and concentration vary still more widely as a rule, it is common practice to focus attention on formulas involving space average quantities. Thus, for heat transfer in the absence of mass transfer, the formula is

$$\dot{Q}_{1 \rightarrow 2} = A \overline{U(T_1 - T_2)} \quad (8)$$

where $\dot{Q}_{1 \rightarrow 2}$ denotes the heat transfer rate from stream 1 to stream 2, A stands for the total amount of interface area within the heat exchanger, U is the local overall heat transfer coefficient, and the bar over $U(T_1 - T_2)$ signifies that the space average of this quantity is in question. The latter can be defined formally by

$$\overline{U(T_1 - T_2)} \equiv \frac{\int_A U(T_1 - T_2) dA}{A} \quad (9)$$

where dA is the element of interface area and $\int_A$ signifies integration over the whole area.

A question that immediately arises is how $\overline{U(T_1 - T_2)}$ is to be evaluated. The answer will be given below (see Secs. 1.3 and 1.5). However, it can immediately be seen that, if U is taken as uniform (a common presumption), the question reduces to that about the space average value of the temperature difference, $\overline{T_1 - T_2}$, where T_1 and T_2 are the local temperatures. This difference is bound to be smaller than $T_{1,\text{in}} - T_{2,\text{in}}$, but how much smaller? The answer depends partly on the flow configuration (the topic of Sec. 1.1.1), partly on whether or not phase changes are present, and partly on whether the heat exchanger has a large or small surface area (the topic of Sec. F).

Because of their importance, and in anticipation of later developments, the following useful statements are now made concerning $\overline{T_1 - T_2}$.

$\overline{T_1 - T_2}$ is often called the effective or mean temperature difference and given the symbol ΔT_M .

ΔT_M is often conveniently calculated from the relationship

$$\Delta T_M = \Delta T_{LM} F \quad (10)$$

where ΔT_{LM} is called the logarithmic mean temperature difference, defined below, and F is a correction factor that is unity for counter-flow heat exchangers under ideal conditions and less than unity for all other configurations (see Sec. H for further discussion).

The definition of the logarithmic mean temperature difference (also referred to as LMTD) is

$$\Delta T_{LM} \equiv \frac{(T_{1,\text{in}} - T_{2,\text{out}}) - (T_{1,\text{out}} - T_{2,\text{in}})}{\ln [(T_{1,\text{in}} - T_{2,\text{out}})/(T_{1,\text{out}} - T_{2,\text{in}})]} \quad (11)$$

which implies that ΔT_{LM} is the logarithmic mean of the temperature differences $(T_{1,\text{in}} - T_{2,\text{out}})$ and $(T_{1,\text{out}} - T_{2,\text{in}})$. If the heat exchanger is of the counter-flow variety, these are the temperature differences prevailing

at the two ends, but the definition is more generally valid.

The employment of Eq. (10) replaces the problem of determining the effective temperature difference ΔT_M by that of determining the correction factor F ; the advantage is that F is dimensionless, equals unity for counter-flow exchangers and less than unity for all others, and can be deduced from graphs and formulas to be described later.

For future reference, Eq. (8) will be brought into relation with Eqs. (1) and (2) as follows:

1. The general heat exchanger case:

$$\begin{aligned}\dot{Q}_{1 \rightarrow 2} &= A \overline{U(T_1 - T_2)} = \dot{M}_{2, \text{out}} - h_{2, \text{in}}) \\ &= \dot{M}_1 (h_{1, \text{in}} - h_{1, \text{out}}) \quad (12)\end{aligned}$$

2. The case of no phase change:

$$\begin{aligned}\dot{Q}_{1 \rightarrow 2} &= A \overline{U(T_1 - T_1)} = \dot{C}_2 \text{ AT}, \\ &= \dot{C}_1 \text{ AT}, \quad (13)\end{aligned}$$

3. The water-steam case:

$$\begin{aligned}\dot{Q}_{\ell, 1 \rightarrow 2} &= A \rho_g \beta_g (h_a - h_v) \simeq \dot{M}_{w, \text{in}} (h_{w, \text{out}} - h_{w, \text{in}}) \\ &\simeq \dot{M}_{a, \text{in}} (h_{a, \text{in}} - h_{a, \text{out}}) \quad (14)\end{aligned}$$

Here, Eq. (13) has been generalized, the ℓ subscript to $\dot{Q}$ signifies that it is the heat transfer on the liquid side of the interface that is in question, and a and w replace 1 and 2 as subscripts to conform with the notation of Eq. (7). The approximation in the latter contributes the $\simeq$ signs.

The determination of the appropriate average value of $(h, -h_w)$ is no less a problem than that of determining an average temperature difference.

F. The number of transfer units, NTU

The size of a heat exchanger can be measured meaningfully only in relation to the amount of material that it has to process. A measure sometimes chosen in practice rests directly on Eq. (3); it is called the number of transfer units of the exchanger, is sometimes abbreviated to NTU, and is here given the symbol NTU_1 or NTU_2 with the definitions:

$$\begin{aligned}\text{NTU}_1 &\equiv \frac{A \bar{U}}{\dot{C}_1} \\ \text{NTU}_2 &\equiv \frac{A \bar{U}}{\dot{C}_2} \quad (15)\end{aligned}$$

Evidently it is necessary, when settling the NTU of a heat exchanger, to state which stream, 1 or 2, is to be referred to, for, as a rule, $\dot{C}_1$ will not equal $\dot{C}_2$.

When a cooling tower is in question, it is conven-

tional (because convenient) to use the water stream as the reference, for only this has its enthalpy change proportional to the temperature difference. Then the definition is

$$\text{NTU} = \frac{A \rho \beta}{\dot{M}_{w, \text{in}} c_{p, w}} \quad (16)$$

G. Heat exchanger effectiveness in the absence of phase change

The NTU of a heat exchanger or cooling tower is important for determining its performance. This performance is often defined in terms of the effectiveness E , which stands for the actual heat transfer rate divided by the maximum possible heat transfer rate.

Now the maximum possible heat transfer rate, $\dot{Q}_{\text{max}}$, is that which would cause the one of the streams with the smaller $\dot{M}c_p$ product, that is, $\dot{C}_{\text{min}}$, to leave the exchanger at the inlet temperature of the other. Thus,

$$\dot{Q}_{\text{max}} = \dot{C}_{\text{min}} \Delta T_{\text{max}} \quad (17)$$

where $\text{AT}_{,,}$ stands for $(T_{1, \text{in}} - T_{2, \text{in}})$. Hence the effectiveness E is given by

$$E = \frac{\dot{Q}}{\dot{Q}_{\text{max}}} \quad (18)$$

$$= \frac{A \bar{U} \Delta T_M}{\dot{C}_{\text{min}} \Delta T_{\text{max}}} = \text{NTU}_{\text{min}} \frac{\Delta T_M}{\Delta T_{\text{max}}} \quad (19)$$

Here it is noticed that:

NTU_{min} stands for the number of transfer units based on the smaller value of $\dot{C}$.

Equation (19) permits the effectiveness E of a heat exchanger to be computed, in principle, as the product of (1) a measure of the size of the heat exchanger, viz., NTU_{min} , and (2) the ratio of the mean temperature difference to the maximum possible temperature difference.

The latter quantity, viz., $\Delta T_M / \Delta T_{\text{max}}$, is always less than unity; and its value depends both on the flow configuration of the exchanger (e.g., parallel-, counter-, cross-flow) and on the values of both NTU_{min} and of $\dot{C}_1 / \dot{C}_2$.

Because of the connections between $\Delta T_M / \Delta T_{\text{max}}$, NTU_{min} and $\dot{C}_1 / \dot{C}_2$ (to be discussed, among other things, in Sec. 1.3), it is convenient to express the effectiveness E as a function of NTU_{min} and $\dot{C}_1 / \dot{C}_2$ directly. One may thus write

$$E = E - \text{NTU}_{\text{min}}, \frac{\dot{C}_1}{\dot{C}_2}, \text{ flow configuration} - \quad (20)$$

Examples of the E function are to be found in Figs. 1.3.1(1) and 1.3.1(2).

The purpose of heat exchanger analysis (also called rating) is to predict the value of E , for given values of A , $\dot{C}$ ratio and configuration. Heat exchanger design has a somewhat different emphasis, namely, to select a configuration, and a corresponding value of A , that will result in a desired change of fluid stream temperature. These differing tasks have given rise to a number of different methods for expressing the relations that describe heat exchanger performance, as the following section will reveal.

H. Some conventional representations of no-phase-change heat exchanger performance

A. C. Mueller, in the *Handbook of Heat Transfer* [9], distinguishes three methods of presenting heat exchanger performance:

- The F-correction-factor method
- The P-NTU method
- The θ method

The concepts employed in each of these methods are now introduced, and their interrelations made plain. The actual use of the concepts for the representation of performance are deferred to Secs. 1.3 and 1.5.

(a) The F correction factor method

F has already been defined in Eq. (10), here rewritten as

$$F \equiv \frac{\Delta T_M}{\Delta T_{LM}} \quad (21)$$

In use, F can be employed to yield the required heat transfer area from the equation

$$A = \frac{Q}{UF \Delta T_{LM}} \quad (22)$$

where its position in the denominator shows how its fall in value below unity tends to increase the necessary area.

F can also be employed so as to yield the effectiveness E of the heat exchanger, by combination of Eqs. (19) and (22):

$$E = NTU_{\min} F \frac{\Delta T_{LM}}{\Delta T_{\max}} \quad (23)$$

Before F can be used, its value must be known, and it is common to express it as a function of the $\dot{M}c_p$ ratio and of the proportional temperature change of one of the streams. The symbols commonly employed are R and P , respectively, with definitions

$$R \equiv \frac{C_2}{C_1} \quad (24)$$

and

$$P \equiv \frac{T_{2,\text{out}} - T_{2,\text{in}}}{T_{1,\text{in}} - T_{2,\text{in}}} = \frac{\Delta T_2}{\Delta T_{\max}} \quad (25)$$

Thus

$$F = F(R, P, \text{configuration}) \quad (26)$$

Algebraic expressions, graphs, and tables have been determined for the F function, under certain simplifying assumptions. An example of a graphical representation of the function is shown in Fig. 1, which is valid for 1-2 multipass exchangers.

The quantity P is a kind of effectiveness; its definition differs from that of ϵ in that it refers to whatever stream is given the subscript 2, whereas E refers to the stream of minimum $\dot{C}$. It therefore follows that

$$\text{for } R \equiv \frac{C_2}{C_1} < 1, P \equiv E \quad (27)$$

$$\text{and for } R \equiv \frac{C_2}{C_1} > 1, P \equiv ER \quad (28)$$

Many heat exchanger specialists prefer to work with P rather than E , because its definition, Eq. (25), refers always to a single stream (number 2) without the necessity to determine which $\dot{C}$ is the smaller.

Inspection of Fig. 1 shows that F diminishes as either P or R increases; the first influence expresses the fact that it is hard to attain a high effectiveness, and the second influence expresses the difficulty of raising the temperature of a fluid stream of large mass flow rate.

(b) The P-NTU method

Since P is the effectiveness of the heat exchanger in transferring heat to stream 2, it may itself conveniently be expressed in terms of the number of transfer units, for a given ratio of $\dot{C}$ quantities. We may thus write

$$P = P(NTU_2, R, \text{flow configuration}) \quad (29)$$

whereby the NTU_2 definition of Eq. (15) has been

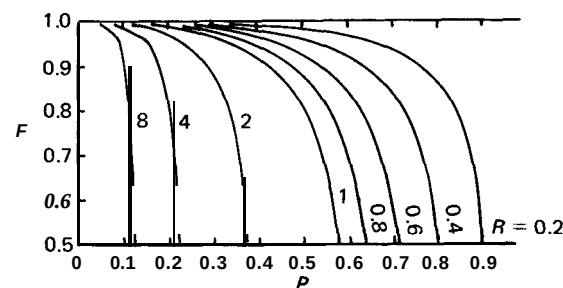


Figure 1 Mean temperature correction factor F versus performance parameter P for various flow ratios R , for 1-2 multipass heat exchangers with uniform U . From Hartnett and Rohsenow [9].

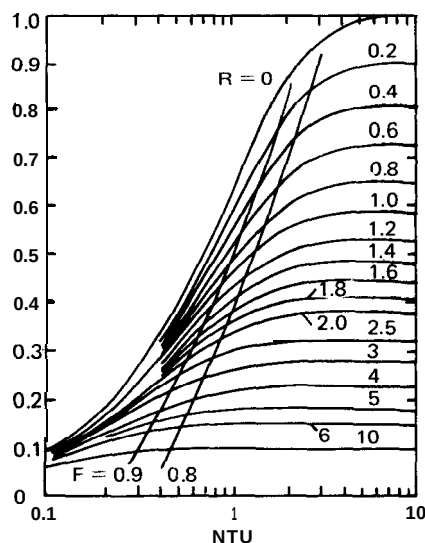


Figure 2 Performance parameter P ($\equiv$ stream 2 effectiveness) versus number of transfer units NTU_2 , for various flow ratios R , for 1-2 multipass heat exchangers with uniform U . From Hartnett and Rohsenow [9].

chosen, to accord with the definition of P ; both refer to stream 2.

This function can be employed for the determination of the performance of a heat exchanger, without any necessity to calculate ΔT_{LM} or F . It is convenient for rating, but less so for design.

Figure 2 represents the same information contained in Fig. 1, but in P - (NTU, R) form. Inspection shows that the performance of the heat exchanger improves with increase in NTU and diminution of R . The former increase represents the value of increasing the heat transfer area, and the latter expresses the fact that it is easier to heat a smaller flow than a larger one.

It will be observed that P can reach unity (i.e., the effectiveness attains 100%), even for indefinitely large NTU , only for R ($\equiv \dot{C}_2/\dot{C}_1$) equal to zero. There is not otherwise any way in which this class of heat exchanger, that is that of 1-2 multipass configuration, can cause stream 2 to leave at the entering temperature of stream 1.

Only the portion of Fig. 2 lying to the left of the $F = 0.8$ line is of much practical significance, for heat exchangers operating to the right of the line would seldom be uneconomical. Because of the concentration of curves on the left, the diagram is not very convenient for use in design.

(c) The θ method

The definition of θ is

$$\theta \equiv \frac{\Delta T_M}{\Delta T_{\max}} \quad (30)$$

θ is useful because when its value is known the effectiveness of the heat exchanger can be computed directly, as Eq. (19) makes plain, from

$$E = NTU_{\min} \theta \quad (31)$$

Alternatively, if reference is to be made to stream 2 rather than to the one of minimum $\dot{C}$, the performance parameter P can be computed directly from the relation

$$P = NTU_2 \theta \quad (32)$$

θ can of course be expressed, like P itself, as a function of NTU_2 and R , and the result can be given algebraic, tabular, or graphic form. An example of the last of these is supplied in Fig. 3, which has been taken from the VDI *Wärmeatlas* [10]. Here, θ is the parameter, E_1 is the ordinate, and E_2 (i.e., P) is the abscissa; values of R can be read from the scales on the right-hand and upper margins. NTU_2 does not appear directly, but its value can be determined for any point on the diagram by way of Eq. (32). θ is also employed, plotted against P , with R and NTU as parameters, in Sec. 1.5. The reasons for using this representation in combination with the F-factor method are given in Sec. 1.5.1.

I. Pressure drop and pumping power

Both heat transfer and momentum transfer depend, as Reynolds recognized in 1874, on similar processes of

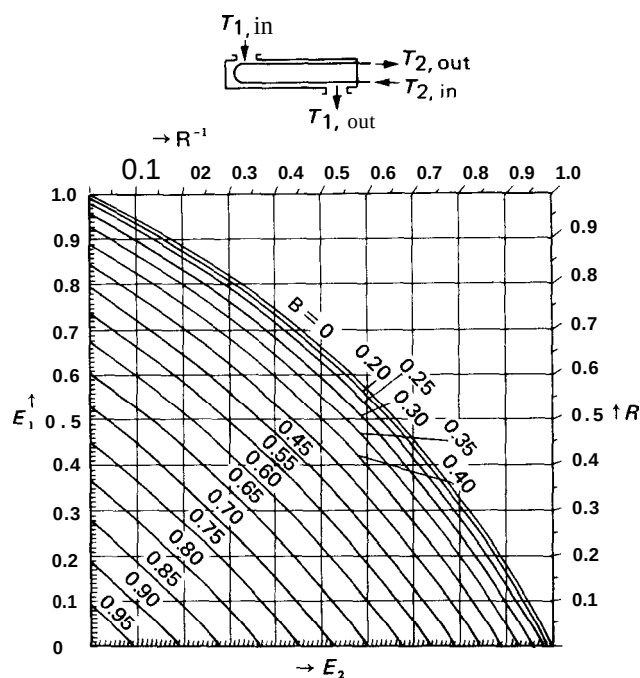


Figure 3 Variation of E_1 with E_2 , for various θ , for a multipass heat exchanger with one outer pass and two, or a multiple of two, inner passes [10].

fluid motion. Since the number of transfer units has now been introduced as a measure of the heat transfer capability of a heat exchanger, it is useful to introduce a parallel measure of its friction properties, namely, NVH, the number of velocity heads[†] of pressure drop. NVH is defined by

$$\text{NVH} \equiv \frac{p_{\text{in}} - p_{\text{out}}}{\rho_{\text{ref}} v_{\text{ref}}^2} \quad (33)$$

where p_{in} and p_{out} refer to the inlet and outlet pressures of the stream in question, ρ_{ref} refers to the density of the fluid in a reference state, and v_{ref} refers to its velocity at a reference cross section.

If, as is usual, two fluid streams flow through the equipment, there are two values of NVH to be calculated. For the shell-side fluid, the reference velocity is best computed by dividing the mass flow rate $\dot{M}$ by ρ_{ref} times the cross-sectional area of the shell; and for the tube-side fluid the total cross-sectional area of a single pass of tubes is appropriate. Thus,

$$v_{\text{ref}} = \frac{\dot{M}}{\rho_{\text{ref}} S_{\text{ref}}} \quad (34)$$

where S_{ref} is the maximum area available to the fluid stream. The reference density is conveniently taken as that for the fluid in its inlet state.

Combination of Eqs. (33) and (34) leads to

$$\text{NVH} \equiv (p_{\text{in}} - p_{\text{out}}) \frac{\rho_{\text{ref}} S_{\text{ref}}^2}{\dot{M}^2} \quad (35)$$

Since NTU has been related by its definition, Eq. (15), to a heat transfer coefficient, U , it is useful to do the same to NVH. For this purpose the effective friction factor, f_{eff} is introduced by way of the relationship

$$f_{\text{eff}} A \rho_{\text{ref}} v_{\text{ref}}^2 = (p_{\text{in}} - p_{\text{out}}) S_{\text{ref}} \quad (36)$$

in which A is the heat transfer area employed in the definition of NTU. Elimination of $(p_{\text{in}} - p_{\text{out}})$ then leads to

$$\text{NVH} = f_{\text{eff}} \frac{A}{S_{\text{ref}}} \quad (37)$$

Comparison of heat transfer and friction performance can be effected by dividing NTU by NVH. The result is

$$\frac{\text{NTU}}{\text{NVH}} = \frac{\bar{U} S_{\text{ref}}}{\dot{M} c_p f_{\text{eff}}} \quad (38)$$

and, since $\bar{U}/c_p$ is an overall conductance and $\dot{M}/S_{\text{ref}}$

[†]As in the case of the friction factor (Sec. 1.2.3), definitions of velocity head may differ. Thus, some authors would insist that the term stands for $\frac{1}{2} \rho_{\text{ref}} v_{\text{ref}}^2$. In Part I of this handbook, however, the view is taken that such numerical factors are best omitted from such definitions.

reference value of G , the mass velocity appearing in the definition of the Stanton number, Eq. 1.2.3(2), the equation can be meaningfully expressed in terms of an effective Stanton number St_{eff} , as follows:

$$\frac{\text{NTU}}{\text{NVH}} = \frac{U}{f_{\text{eff}}} St_{\text{eff}} \quad (39)$$

In Sec. 1.2.31, methods were provided of computing the ratio of Stanton numbers to friction factors; for example, the easy-to-use Chilton-Colburn formula, Eq. 1.2.3(35), indicates that the ratio may be of the order of $(Pr)^{-2/3}$, so that, if gases are in question, St/f is of the order of unity.

Whatever value of St/f is given by the Chilton-Colburn (or other) formula, there are at least two reasons why the value of $St_{\text{eff}}/f_{\text{eff}}$ (and so of NTU/NVH) will be appreciably lower. The first is that it is the overall heat transfer coefficient that has entered into the definition of St_{eff} ; but it is the usually much-larger film coefficient that appears in St . The second reason is that f_{eff} has been deduced from the overall pressure drop in the stream in question, and this pressure drop, which contains contributions from baffles and other flow restrictions as well as from friction of heat transfer surfaces, is probably several times as great as that which would equal $f A/S_{\text{ref}}$, that is, which would balance the skin friction alone.

However, a separate point needs now to be made: although the ratio of NTU/NVH may have to be accepted by the designer as having some value distressingly below unit, he or she still has some control over the pumping power that is expended for a fixed NTU. Pumping power, P , is expressible (if there is no very large density change in the heat exchanger) as

$$\frac{P}{\dot{M}} = \frac{p_{\text{in}} - p_{\text{out}}}{\rho_{\text{ref}}} \quad (40)$$

that is, as

$$\frac{P}{\dot{M}} = \frac{\text{NTU } V_{\text{ref}}^2}{\text{NTU}/\text{NVH}} \quad (41)$$

Therefore the pumping power, per unit mass of material pumped, can be reduced by lowering V_{ref} , that is, by increasing S_{ref} , the cross-sectional area of the heat exchanger.

Of course, increasing S_{ref} ordinarily increases other dimensions of the heat exchanger, and in most circumstances the capital cost of the equipment will also be increased thereby. The designer must then conduct an economic analysis, in which running costs are balanced against capital costs, before he or she can determine the optimum values of V_{ref} , S_{ref} , and so on.

J. The choice of formulation

Heat exchanger analysts and designers are not united about which formulation of heat exchanger performance parameters is to be preferred. It would therefore be unwise to do more than explain that choice is possible, and that it can be based on reason.

For the analyst who knows what the heat transfer area and flow configuration are but whose task it is to determine the performance, it is convenient to use formulations in which the numbers of transfer units are inputs; then, either by way of the effectiveness E , or the

dimensionless mean temperature difference θ , the performance can be determined.

The designer, on the other hand, knows what performance is desired and so can evaluate the logarithmic mean temperature difference, and also the quantity P [Eq. (25)]. What the designer needs to find out is the effective mean temperature difference, which can conveniently be done by way of the temperature difference correction factor F . If this is known, which is often the case provided that U is uniform, the designer can proceed to evaluate the necessary heat transfer area without considering the number of transfer units at all.

Nomenclature and References for Section 1.2.4 appear at the end of Section 1.4.3.

1.2.5

The differential equations governing streams

D. Brian Spalding

A. The differential equations for enthalpy

Figure 1 represents an elementary portion of stream 1 in a heat exchanger. The length of path in the flow direction is dz_1 , and the associated interface area is dA . If the flow is steady, and effects of kinetic heating, external work, and gravitational potential energy are negligible, the application of the steady-flow energy equation, 1.2.1(3), yields

$$\frac{\dot{M}_1 dh_1}{dz_1} = -\dot{q}_{1 \rightarrow 2} \frac{dA}{dz_1} \quad (1)$$

where $\dot{q}_{1 \rightarrow 2}$ is the heat flux per unit area from stream 1 to stream 2. This heat flux can of course be calculated from

$$\dot{q}_{1 \rightarrow 2} = U(T_1 - T_2) \quad (2)$$

where U , T_1 , and T_2 are all local values.

Figure 1 also shows, displaced to one side, the

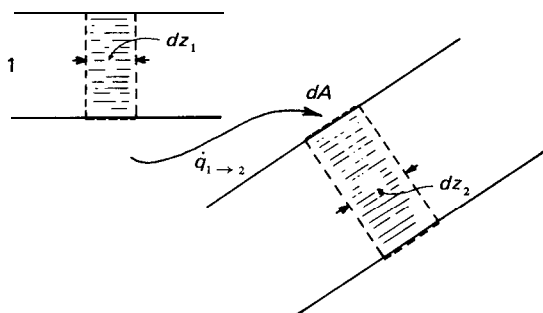


Figure 1 Control volumes for derivation of differential balance equations.

corresponding control volume for stream 2. For this control volume, the differential form of the steady-flow energy equation is

$$\dot{M}_2 \frac{dh_2}{dz_2} = \dot{q}_{1 \rightarrow 2} \frac{dA}{dz_2} \quad (3)$$

The similarities and differences between Eqs. (1) and (3) are obvious, and they can be usefully exploited by combination as follows:

$$\dot{M}_1 \frac{dh_1}{dz_1} + \dot{M}_2 \frac{dh_2}{dz_2} = 0 \quad (4)$$

Here it should be noted that in the case of a parallel-flow heat exchanger, dz_1 and dz_2 are equal in both magnitude and sign, whereas, for a counter-flow heat exchanger, dz_1 and dz_2 are equal in magnitude but opposite in sign. For cross-flow configurations, the equations are not useful, for the condition of each stream depends on at least two position coordinates, not one.

B. The differential equation for temperature

If there is no phase change in stream 1 at the point in question, Eq. (1) can be expressed as

$$\dot{C}_1 \frac{dT_1}{dz_1} = -\dot{q}_{1 \rightarrow 2} \frac{dA}{dz_1} \quad (5)$$

and, if stream 2 is without phase change, its differential equation becomes

$$\dot{C}_2 \frac{dT_2}{dz_2} = \dot{q}_{1 \rightarrow 2} \frac{dA}{dz_2} \quad (6)$$

Direct combination, and elimination of $\dot{q}_{1 \rightarrow 2}$, then yields:

$$\dot{C}_1 \frac{dT_1}{dz_1} + \dot{C}_2 \frac{dT_2}{dz_2} = 0 \quad (7)$$

However, there is also another interesting method of combination, which comes from multiplying Eq. (6) by $(C_1/C_2)(dz_2/dz_1)$ and subtracting it from Eq. (5). The result is

$$\dot{C}_1 \frac{d}{dz_1} (T_1 - T_2) = -\dot{q}_{1 \rightarrow 2} \frac{dA}{dz} \left(1 + \frac{\dot{C}_1 dz_2}{\dot{C}_2 dz_1} \right) \quad (8)$$

Here the subscript has been dropped from the z in dA/dz to make plain that dA/dz_1 and dA/dz_2 have the same essentially positive value whether parallel flow or counter flow is in question.

The final step in this development is to eliminate the heat flux from Eq. (2); a differential equation results in which the temperature difference $(T_1 - T_2)$ is the dependent variable. It is

$$\frac{1}{T_1 - T_2} \frac{d(T_1 - T_2)}{dz} = -U \frac{dA}{dz} \left(\frac{1}{\dot{C}_1} + \frac{1}{\dot{C}_2} \frac{dz_2}{dz_1} \right) \quad (9)$$

It is this equation that, upon solution, yields the logarithm of which mention has already been made in the logarithmic temperature difference. This solution is presented in Sec. 1.3.1B.

C. The differential equation for concentration

Although little use is made of them in this handbook, the analogous differential equations for concentration, being easy to derive and present, are given here. They are

$$\frac{d}{dz_1} (\dot{M}_1 x_{i,1}) = -\dot{m}_{i,1 \rightarrow 2} \frac{dA}{dx_1} \quad (10)$$

and

$$\frac{d}{dz_2} (\dot{M}_2 x_{i,2}) = \dot{m}_{i,1 \rightarrow 2} \frac{dA}{dz_2} \quad (11)$$

Combination by simple addition leads to

$$\frac{d}{dz_1} (\dot{M}_1 x_{i,1}) + \frac{d}{dz_2} (\dot{M}_2 x_{i,2}) = 0 \quad (12)$$

A further combination is possible, by introduction of Eq. 1.2.2(12), which leads to an equation having $(x_{i,1} - \text{He } x_{i,2})$ as its dependent variable. This is analogous to Eq. (9), but not so neat. Details can be found in [1], but they are not needed there.

D. The approximate differential equation for enthalpy in the steam-air-water system

It will be remembered from Sec. 1.2.2C that, by what was called a fortunate accident, the energy and mass

exchanges between water and air can be expressed in an especially simple form. Renewed attention is now be given to that system, which is especially relevant to cooling towers. The differential forms of the balance equations, given in Sec. 1.2.4B, are as follows:

$$\dot{M}_a \frac{dh_a}{dz_a} \simeq -\dot{q}_{l,a \rightarrow w} \frac{dA}{dz_a} \quad (13)$$

and

$$\dot{M}_w \frac{dh_w}{dz_w} \simeq \dot{q}_{l,a \rightarrow w} \frac{dA}{dz_w} \quad (14)$$

Addition yields

$$\dot{M}_a \frac{dh_a}{dz_a} + \dot{M}_w \frac{dh_w}{dz_w} \simeq 0 \quad (15)$$

The heat flux can be conveniently introduced (after obvious minor notation changes) from Eq. 1.2.2(13). A useful result is then

$$dh(w)/dz_w = \text{RoBg} / (\text{Mw}(\text{h}_a - \text{h}_s)) \quad (16)$$

Now h_a can be expressed as a linear function of h_w , by integration of Eq. (15); however, h_s is a nonlinear function of h_w , dictated by the partial pressure-temperature curve of steam in air. Integration of Eq. (16) must therefore proceed numerically. The logarithmic mean enthalpy difference can therefore be used in cooling tower calculations only under infrequent circumstances.

E. Some remarks about the solution of the equations

Nonlinearity has just been mentioned in connection with the $h_s(h_w)$ function. There are other nonlinearities that may arise, even when the concern is with heat transfer uncomplicated by mass transfer. For example, the overall heat transfer coefficient U in Eq. (9) may depend on the local temperature levels, as when the viscosity of the fluid diminishes with rising temperature. Further, the specific heats may vary significantly with temperature, as when one of the fluids is near its critical point.

Analytical solution may still be possible when the equations are nonlinear. For example, if U were a simple function of temperature difference $(T_1 - T_2)$, for example, a constant times $(T_1 - T_2)$ raised to some power, solution would be easy. However, such cases are rare.

In practice, solutions of the equations given above are obtained in two ways: for the case of uniform U , $c_{p,1}$, and $c_{p,2}$ the analytical solutions are employed that will be described in Sec. 1.3.1; but when any nonlinearities are present, solutions are obtained numerically, that is, by **stepwise** integration.

F. Closure

In the foregoing section, the fluid enthalpies, temperatures, and compositions have been presumed to depend on a single space variable, namely, z ; the differential equations are therefore what are called ordinary ones, to distinguish them from the partial differential equations that are needed when two or more independent variables (space or time) play a part.

In real heat exchangers, fluid properties do vary in two or three space dimensions, and every departure from the steady state entails attention to the time dimension

also; so partial differential equations are needed for the realistic analysis of heat exchanger phenomena. These are the subject of Sec. 1.2.6.

Nevertheless, it is mainly on the one-dimensional equations that conventional analysis and design procedures are based. Some readers may therefore prefer to skip Sec. 1.2.6, proceeding directly to Sec. 1.3, which is devoted to analytical solutions. Section 1.2.6 can be returned to later, as a prelude to the understanding of Sec. 1.4, which is concerned with numerical solution procedures.

Nomenclature for Section 2.5 appears at the end of Section 1.4.3.

1.2.6

Partial differential equations for interpenetrating continua

D. Brian Spalding

A. Introduction

The differential equations given in Sec. 1.2.5 are useful if the two fluids flow in parallel or counter flow, or in some combination of these. They cannot be used, however, in cross-flow situations, for these give rise to two- or three-dimensional temperature variations.

The task of the present section is to develop the differential equations that govern the temperature distribution in cross-flow and more complex situations. Mass transfer and phase-change effects will be neglected for simplicity; a more advanced treatment, allowing for these effects, is to be found in [11].

It has been common practice for heat exchanger theorists to suppose that they know quantitatively the pattern of flow adopted by the fluids in the equipment, and that their task is to predict the resulting temperature distributions. This is the first level at which the differential equations can be approached: the velocities appearing in them can be regarded as known functions of position.

However, knowledge of actual flow patterns in, say, baffled shells, is in fact very unreliable; little more is known than that the common assumptions are considerably at fault. It therefore becomes desirable to be able to calculate the velocity fields on the basis of the best general information that is available concerning flows in media containing distributed resistances (such as baffles per banks of tubes).

To do this involves solving the differential equations for the velocity fields as well as those for the temperature fields. These equations will therefore also be developed in the present section. Section 1.4 describes

methods for numerically solving all these equations and leading to detailed predictions of heat exchanger performance.

Because it is as easy to formulate and solve the equations for the transient case as those for steady flow, it is the former that will be presented. They can therefore serve for the description of regenerator flows as well as for those appearing in recuperators.

In order to mark clearly the two levels of approach, Sec. B is concerned with the temperature equations only; the equations for velocity are dealt with in Sec. C.

B. Differential equations for temperature

(a) Independent variables

Because most heat exchangers are cylindrical in form, it is convenient to employ the θ, r, z coordinate system, in which θ stands for angular position, measured from an arbitrarily chosen plane enclosing the axis; r stands for distance from a symmetry axis (which is usually either vertical or horizontal, but may be inclined); and z stands for distance measured parallel to the axis from a plane lying normal to that axis.

Figure 1 pictures the coordinate system, with the axis placed vertically. It may be useful to imagine that the cylinder depicted there represents the shell of a heat exchanger in which tubes run mainly along vertical lines of constant θ and r , whereas baffles are horizontal surfaces or constants; however, the equations to be given in no way depend on such a presumption.

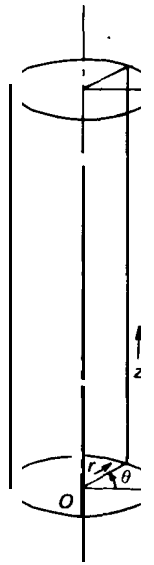


Figure 1 Illustration of the $\theta \sim r \sim z$ polar coordinate system used for describing temperature and velocity fields.

(b) Dependent variables

The dependent variables considered will be T_1 and T_2 , the temperatures of the two fluids that are supposed to flow through the heat exchanger, and T_m , the temperature of the metal of the intervening tube walls.

Analysis proceeds as though these three variables, and others for that matter, vary continuously in space. Of course, this is not so, for the first fluid, second fluid, and metal are coarsely interspersed, rather than finely mixed. However, the idea of the three interpenetrating continua is convenient and leads to no disadvantage if properly recognized as a fiction.

(c) Porosities

The fact that the whole of space is not accessible to each material is introduced into the analysis by way of volume fractions R_1 , R_2 , and R_m . These represent the local fractions of the volume of space that are occupied by fluid 1, fluid 2, and metal, respectively. As a matter of definition, all three quantities lie between zero and unity, and they obey the additive relationship

$$R_1 + R_2 + R_m = 1 \quad (1)$$

The manner of calculating the R 's is straightforward. Let it be supposed that fluid 1 is the tube-side fluid, that the tubes have external diameter D_e and internal diameter D_i , and that they are spaced in a rectangular array with distances between their centers denoted by X and Y .

Then the number of tubes per unit cross-sectional

area is $1/XY$, and the internal volume of unit length of a single tube is $\pi D_i^2/4$. Therefore the expression for R_1 is

$$R_1 = \frac{\pi D_i^2}{4XY} \quad (2)$$

Correspondingly, the expression for R_2 , the proportion of volume accessible to the shell-side fluid, is

$$R_2 = 1 - \frac{\pi D_e^2}{4XY} \quad (3)$$

Finally, the proportion of volume occupied by the metal is

$$R_m = \frac{\pi(D_e^2 - D_i^2)}{4XY} \quad (4)$$

The R 's, which were defined in terms of volumes, are also conveniently used as dictating free area proportions, and so for defining velocities. The velocities of the two fluid phases in the coordinate directions θ , r , and z will be given the symbols $u_1, u_2, v_1, v_2, w_1, w_2$, respectively, and these are so defined that, for example, the mass flow rate of fluid 1 per unit area of space in the θ direction is given by $\rho_1 u_1 R_1$, that of fluid 2 is given by $\rho_2 u_2 R_2$, and so on.

(d) The energy equations

The interpenetrating continua idea allows the thermal behavior of the contents of the heat exchanger to be described by partial differential equations, as follows:

For fluid 1, the temperature field is governed by the equation

$$\begin{aligned} \frac{\partial}{\partial t} (R_1 \rho_1 c_1 T_1) + \frac{1}{r^2} \frac{\partial}{\partial \theta} (R_1 \rho_1 u_1 r c_1 T_1) \\ + \frac{1}{r} \frac{\partial}{\partial r} (R_1 \rho_1 v_1 r c_1 T_1) = \frac{1}{r} \frac{\partial}{\partial \theta} \left(R_1 \Lambda_1 \frac{\partial T_1}{\partial \theta} \right) \\ + \frac{1}{r} \frac{\partial}{\partial r} \left(R_1 \Lambda_1 r \frac{\partial T_1}{\partial r} \right) + \frac{\partial}{\partial z} \left(R_1 \Lambda_1 \frac{\partial T_1}{\partial z} \right) + \dot{q}_{m \rightarrow 1} \end{aligned} \quad (5)$$

where c_1 stands for the constant-pressure specific heat of fluid 1, Λ , stands for its effective thermal conductivity, account being taken of turbulence and other factors, and $\dot{q}_{m \rightarrow 1}$ stands for the rate of heat transfer, per unit volume of exchanger space, from the metal to fluid 1.

The following terms have been omitted because they are of little importance in heat exchangers:

The term $R_1 \partial p / \partial t$, representing the effect of changes of pressure p on the temperature

The terms representing the kinetic energy of motion, and the effect of viscous dissipation of energy

Terms representing chemical changes within the fluid

Terms representing phase changes

The last of these, of course, cannot always be neglected in practice, but the method of analysis can be straightforwardly extended to phase-change situations by the replacement of temperature by enthalpy.

It should also be noted that constancy of specific heat is not being presumed, so a simple method of extending to phase-change situations can be easily envisaged: for the one-degree temperature range enclosing the boiling point, the specific heat can be artificially raised to equal the latent heat of vaporization. Of course, this device applies only to the change of phase within a single stream; transfer of mass from one stream to another requires the more extended treatment described in [11].

The energy equation for fluid 2, that is, in the shell, can be written in a similar manner as

$$\begin{aligned} & \frac{\partial}{\partial t} (R_2 \rho_2 c_2 T_2) + \frac{1}{r^2} \frac{\partial}{\partial \theta} (R_2 \rho_2 u_2 r c_2 T_2) \\ & + \frac{1}{r} \frac{\partial}{\partial r} (R_2 \rho_2 v_2 r c_2 T_2) + \frac{\partial}{\partial z} (R_2 \rho_2 w_2 c_2 T_2) \\ & = \frac{1}{r} \frac{\partial}{\partial \theta} \left(R_2 \Lambda_2 \frac{\partial T_2}{\partial \theta} \right) + \frac{1}{r} \frac{\partial}{\partial r} \left(R_2 \Lambda_2 r \frac{\partial T_2}{\partial r} \right) \\ & + \frac{\partial}{\partial z} \left(R_2 \Lambda_2 \frac{\partial T_2}{\partial z} \right) + \dot{q}_{m \rightarrow 2} \end{aligned} \quad (6)$$

where, of course, Λ_2 is the effective thermal conductivity of the second fluid, and $\dot{q}_{m \rightarrow 2}$ is the heat flux to it from the metal per unit exchanger volume.

It may be remarked that the r 's inside the brackets in the second terms of both Eqs. (5) and (6) could be cancelled. This is not done, however, because $u_1 r$ and $u_2 r$, representing the angular momenta per unit mass, are the quantities that are obtained from the hydrodynamic analysis, discussed in Sec. C.

The energy equation for the metal is, of course, simpler, for the metal does not move. The equation is

$$\begin{aligned} \frac{\partial}{\partial t} (R_m \rho_m c_m T_m) &= \frac{1}{r} \frac{\partial}{\partial \theta} \left(R_m \Lambda_m \frac{\partial T_m}{\partial \theta} \right) \\ &+ \frac{1}{r} \frac{\partial}{\partial r} \left(R_m \Lambda_m r \frac{\partial T_m}{\partial r} \right) \\ &+ \frac{\partial}{\partial z} \left(R_m \Lambda_m \frac{\partial T_m}{\partial z} \right) - \dot{q}_{m \rightarrow 1} - \dot{q}_{m \rightarrow 2} \end{aligned} \quad (7)$$

Here Λ stands for the effective thermal conductivity of the metal, and account has been taken of the fact that the heat transfers to fluids 1 and 2 are taken from the metal.

(e) Common (but unnecessary) simplifications

It is common to neglect the thermal conductivity effects in the metal entirely, on the grounds that they are negligible. Then Eq. (7) reduces to

$$\frac{\partial}{\partial t} (R_m \rho_m c_m T_m) = -\dot{q}_{m \rightarrow 1} - \dot{q}_{m \rightarrow 2} \quad (8)$$

It is also common, with one exception, to neglect the thermal conductivity effects in the fluids, the equation of which then reduces to

$$\begin{aligned} & \frac{\partial}{\partial t} (R_1 \rho_1 c_1 T_1) + \frac{1}{r^2} \frac{\partial}{\partial \theta} (R_1 \rho_1 u_1 r c_1 T_1) \\ & + \frac{1}{r} \frac{\partial}{\partial r} (R_1 \rho_1 v_1 r c_1 T_1) + \frac{\partial}{\partial z} (R_1 \rho_1 w_1 T_1) = \dot{q}_{m \rightarrow 1} \end{aligned} \quad (9)$$

$$\begin{aligned} & \frac{\partial}{\partial t} (R_2 \rho_2 c_2 T_2) + \frac{1}{r^2} \frac{\partial}{\partial \theta} (R_2 \rho_2 u_2 r c_2 T_2) \\ & + \frac{1}{r} \frac{\partial}{\partial r} (R_2 \rho_2 v_2 r c_2 T_2) + \frac{\partial}{\partial z} (R_2 \rho_2 w_2 T_2) = \dot{q}_{m \rightarrow 2} \end{aligned} \quad (10)$$

The one exception is when it is presumed that intense mixing takes place in one or two directions. Thus, if it were to be supposed that the shell-side fluid is fully mixed over constant z planes, then Λ_2 would be set to a large number in the $\partial/\partial\theta$ and $\partial/\partial r$ terms, and to zero in the $\partial/\partial z$ term.

This practice, incidentally, is a good illustration of how the Λ 's can, and normally should, exhibit directional properties. Obviously, it is easier for heat to be conducted in tube-side fluid in the tube-length directions, in which the fluid is continuous, than it is in either of the other two directions, in which it is interrupted; this can be selected by giving Λ_1 a different value according to whether it is the θ , r , or z component of the gradient of T_1 that is in question.

The word unnecessary has been inserted parenthetically in the above heading. This means unnecessary from the point of view of practicability of solving the equations; the simplifications have been made, in the past, mainly because efficient equation-solving devices were not available. The coming of the digital computer has changed this.

(f) The heat flux terms

The heat flux terms $\dot{q}_{m \rightarrow 1}$ and $\dot{q}_{m \rightarrow 2}$ can be written as

$$\dot{q}_{m \rightarrow 1} = U_1 (T_m - T_1) \quad (11)$$

$$\dot{q}_{m \rightarrow 2} = U_2 (T_m - T_2) \quad (12)$$

Here the U_1 and U_2 are partial volumetric heat transfer coefficients, representing the ease with which heat can

flow through one half of the metal thickness, and through the adjacent dirt and fluid films, to the core of the relevant fluid stream.

U_1 and U_2 can be, and usually will be, functions of the local fluid temperature and velocities, as well as of the purely geometric quantities. As a rule, U_1 and U_2 will both increase with R_m , which represents the amount of metal that is present per unit volume.

(g) The steady state

Often it is required to analyze only the steady-state temperature distribution in a heat exchanger. Then solution of the differential equation for T_m can be dispensed with, for this reduces to

$$\dot{q}_{m \rightarrow 1} + \dot{q}_{m \rightarrow 2} = 0 \quad (13)$$

from which can be deduced, by way of Eqs. (11) and (12),

$$T_m = \frac{U_1 T_1 + U_2 T_2}{U_1 + U_2} \quad (14)$$

and

$$q_{m \rightarrow 1} = \dot{q}_{m \rightarrow 2} = \frac{T_2 - T_1}{1/U_1 + 1/U_2} \quad (15)$$

The expression in Eq. (15) can then be substituted into the differential equations for T_1 and T_2 , solution of which is what is needed to predict the heat exchanger performance. If it should be desired subsequently to know what is the metal temperature at any point, all that is needed is to deduce it from T_1 and T_2 via Eq. (14).

(h) The prediction of heat exchanger performance

The solution of the differential equations, which can be obtained by the numerical methods described in Sec. 1.4, yields three-dimensional distributions of temperatures at all times: $T_1(\theta, r, z, t)$, $T_2(\theta, r, z, t)$, and $T_m(\theta, r, z, t)$.

If the steady-state solution is provided, the distribution functions are simply $T_1(\theta, r, z)$, $T_2(\theta, r, z)$, and $T_m(\theta, r, z)$.

These distribution functions are provided by the computer either as tables of numbers, representing temperatures at particular locations and instants, or as graphs.

In order to convert these results into terms familiar to the heat exchanger specialist, very minor adaptations are needed; these can, of course, be done by the computer. Thus, it will usually be desired to know what are the outlet temperatures of the streams 1 and 2. This

information can be deduced from the temperature fields by simple integration, weighted appropriately by the mass flux (density X velocity) distributions.

The heat transfer rate can then be deduced either from these quantities, together with an overall heat balance, or the quantities $\dot{q}_{m \rightarrow 1}$ and $\dot{q}_{m \rightarrow 2}$ can be integrated over the whole volume of the heat exchanger.

(i) Application to a cross-flow heat exchanger

Although it is not intended to follow the analysis very far, a useful link between this section and the material of Secs. 1.3 and 1.4 may be provided by indicating how the equations derived so far could be applied to the prediction of the steady-state behavior of an idealized cross-flow heat exchanger.

Such a heat exchanger can be represented in a two-dimensional manner, so variations in the θ direction will be neglected. It will further be presumed that fluid 1 flows only in the r direction and fluid 2 only in the z direction, and that mixing occurs, if at all, only in directions at right angles to the direction of flow. The differential equations for T_1 and T_2 then become

$$\frac{1}{r} \frac{\partial}{\partial r} (R_1 \rho_1 v_1 r c_1 T_1) = \frac{\partial}{\partial z} \left(R_1 \Lambda_1 \frac{\partial T_1}{\partial z} \right) + \frac{T_2 - T_1}{1/U_1 + 1/U_2} \quad (16)$$

$$\frac{\partial}{\partial z} (R_2 \rho_2 w_2 c_2 T_2) = \frac{1}{r} \frac{\partial}{\partial r} \left(R_2 \Lambda_2 r \frac{\partial T_2}{\partial r} \right) + \frac{T_1 - T_2}{1/U_1 + 1/U_2} \quad (17)$$

As indicated above, in order to conform with the usual idealized assumptions, the A 's will be set either to zero (for no mixing) or a large number (for perfect mixing).

In the classical cross-flow heat exchanger problem, the terms $R_1 \rho_1 v_1 c_1$, $R_2 \rho_2 w_2 c_2$, and $(1/U_1 + 1/U_2)$ are regarded as constants, and the variation in r as small enough to be neglected. Thus the equations can be condensed to

$$C_1 \frac{\partial T_1}{\partial r} = C_2 \frac{\partial T_1}{\partial z} + C_3 (T_2 - T_1) \quad (18)$$

and

$$C_4 \frac{\partial T_2}{\partial z} = C_5 \frac{\partial T_2}{\partial r} + C_3 (T_1 - T_2) \quad (19)$$

where C_1 through C_5 are constants.

It is at this point that the analysis will be broken off. Solution of these equations by numerical means is straightforward, and the results of some such solutions appear in Sec. 1.3.1.

C. Differential equations for velocity and pressure

(a) The physical principles and phenomena involved

When guessing or idealizing the distributions of velocity in the shell of the heat exchanger ceases to be satisfactory, these distributions can be computed by reference to the principles of conservation of mass and energy. These principles can be expressed in terms of partial differential equations; and, if information can be provided about certain auxiliary functions appearing in these, the equations can be solved. Solution simultaneously with the solving of the temperature equations leads directly to the prediction of heat exchanger performance.

The auxiliary functions are algebraic expressions of the laws of momentum transfer; and they correspond to the A s and U s that appeared in the temperature equations. Specifically, they concern the mixing of nearby streams of the same fluid with each other (the counterpart of the A s), and the losses of momentum that occur when fluid is forced through tube bundles, along baffle surfaces, and so on (the counterpart of the U s).

In shell-and-tube heat exchangers, the flow pattern of the tube-side fluid truly is known from the start; it is therefore necessary to predict the velocity distribution only in the shell-side fluid. There will be four differential equations to solve, one for each velocity component, and one for the pressure. The first three equations are those for momentum conservation; the pressure is obtained from the mass conservation equation.

The momentum equations exhibit a source term for which there is no counterpart in the energy equations. This is the buoyancy term, representing a source of momentum in the vertical direction proportional to the difference between the density of the fluid at the point in question and that which is the average for that horizontal level. In the following equations, the convention that the z direction is vertical will be retained; if the shell were inclined to the vertical, buoyancy terms would appear in the equations for u and v , as well as in that for w .

(b) The mass conservation equation

The conservation-of-mass principle dictates the following relationship between the density and the velocity components:

$$\frac{\partial}{\partial t}(R\rho) + \frac{1}{r^2} \frac{\partial}{\partial \theta} (R\rho ur) + \frac{1}{r} \frac{\partial}{\partial r} (R\rho ur) + \frac{\partial}{\partial z} (R\rho w) = 0 \quad (20)$$

Here, no subscript has been supplied, for ease of writing. If fluid 2 is that in the shell, as implied earlier, it is of course subscript 2 that would be appropriate.

If the flow is steady, or if the fluid can be regarded as incompressible, a simpler equation applies, namely,

$$\frac{1}{r^2} \frac{\partial}{\partial \theta} (R\rho ur) + \frac{1}{r} \frac{\partial}{\partial r} (R\rho ur) + \frac{\partial}{\partial z} (R\rho w) = 0 \quad (21)$$

(c) The momentum equations

The differential equation for the conservation of angular momentum, ur , is

$$\frac{\partial}{\partial t} (R\rho ur) + \frac{1}{r^2} \frac{\partial}{\partial \theta} (R\rho ur ur) + \frac{1}{r} \frac{\partial}{\partial r} (R\rho ur ur) + \frac{\partial}{\partial z} (R\rho w ur) = -R \frac{\partial p}{\partial \theta} - f_{\theta} r \quad (22)$$

where p stands for the pressure of the fluid, and f_{θ} is a force per unit exchanger volume, acting in the negative θ direction, and expressing all frictional effects, whether between one part of fluid 1 and another or between fluid 1 and some solid structure.

The differential equation for momentum conservation in the r direction is

$$\frac{\partial}{\partial t} (R\rho v) + \frac{1}{r^2} \frac{\partial}{\partial \theta} (R\rho ur v) + \frac{1}{r} \frac{\partial}{\partial r} (R\rho ur v) + \frac{\partial}{\partial z} (R\rho w v) = -R \frac{\partial p}{\partial r} + R\rho \frac{(ur)^2}{r^3} - f_r \quad (23)$$

where f_r represents the force per unit exchanger volume in the negative r direction, again expressing all frictional effects. The term containing $(ur)^2$ represents the centrifugal force.

The differential equation for momentum conservation in the z (i.e., vertical) direction is

$$\frac{\partial}{\partial t} (R\rho w) + \frac{1}{r^2} \frac{\partial}{\partial \theta} (R\rho ur w) + \frac{1}{r} \frac{\partial}{\partial r} (R\rho ur w) + \frac{\partial}{\partial z} (R\rho w w) = -R \frac{\partial p}{\partial z} - f_z + g_n(\bar{\rho} - \rho) \quad (24)$$

where f_z represents all the frictional effects acting in the negative z direction and the term $g_n(\bar{\rho} - \rho)$ represents the effect of the gravitational acceleration, g_n , in causing the fluid to tend to rise when its density ρ falls below the average density $\bar{\rho}$. The latter quantity may be defined as a constant for the whole exchanger, or it may be a function of z .

(d) The friction terms

As has already been explained, the terms f_{θ} , f_r , and f_z account for frictional effects of two kinds: what might be called internal friction, and momentum losses

caused by the presence of the solid structure of tubes, baffles, and so on.

The first kind of effect can be expressed in terms of effective viscosity, and of gradients of velocity, in the manner that is familiar from texts on hydrodynamics. However, in order to save space, and because the terms are frequently unimportant in heat exchangers as compared with the others, no attempt will be made here to write out the full expressions. Instead, the viscous components will be simply given the symbols $f_{\eta\theta}$, $f_{\eta r}$, and $f_{\eta z}$, to indicate their connection with viscosity, and left in the equations to be evaluated or ignored as appropriate.

The other parts of the f terms will be written as products of quantities f'_{θ} , f'_{r} , f'_{z} , with the relevant velocity. This practice indicates that the magnitudes of the terms increase with the velocity; but the linearity is only superficial, because it is to be understood that the f 's can all depend on the velocities in complex ways.

These practices can now be represented by the equations

$$f_{\theta} = f_{\eta\theta} + \frac{1}{r} f'_{\theta} u r \quad (25)$$

$$f_r = f_{\eta r} + f'_{r} v \quad (26)$$

$$f_z = f_{\eta z} + f'_{z} w \quad (27)$$

and the appropriate substitutions can be made in the three momentum equations.

(e) Initial and boundary conditions

In order to complete the specification of the mathematical problem represented by the differential equations and the auxiliary functions, it is necessary to apply boundary conditions, and, if the flow is not of the steady-state variety, initial conditions are required as well.

In the heat exchanger context, the boundary conditions usually take the form of:

Specifications of finite-fluid velocity, temperature, and phase at the inflow ports

Setting all velocity components to zero at the solid boundaries of the flow domain (shell walls)

Setting the temperature gradients normal to, and at, these boundaries equal either to zero (for a well-insulated shell) or to a constant times the difference of temperature from that of the surroundings (for a shell exhibiting surface heat loss)

If the problem is one of transient flow, all the above quantities may vary with time.

The initial conditions required for transient-flow problems entail, ideally, a specification of the distribution of all the fluid variables throughout the flow

domain at the instant at which the process starts. Such information is hardly ever available directly; however, it often suffices to suppose that the conditions within the heat exchanger at the start are those that are appropriate to steady-state operation with the initially prevailing boundary conditions; the transient analysis can then proceed to determine what transpires as a consequence of the specified changes in the boundary conditions.

(f) Discussion of the mathematical problem

Comparison of the momentum equations with the energy equations shows that solution of the former is likely to be appreciably harder than solution of the latter. Not only are the equations more numerous, but they are more obviously nonlinear and interlinked.

Nonlinearity could be present in the energy equations, even though the mass flow rates were prescribed, because of dependencies of the U 's on the T 's; and interlinkages are effected through the temperature differences appearing in the heat flux terms. However, nonlinearity in the momentum equations appears even in the convection terms, where one velocity multiplies another; and, since all pairs of velocities are multiplied together, the interlinkages are also more extensive.

If there were any lingering hope that analytical methods could be employed to solve the equations, the foregoing discussion proves the hope vain, for analytical methods for interlinked equations are rare, and for nonlinear ones they are almost nonexistent. It follows that numerical methods must be used. Once this decision has been made, the problem of solving the momentum equations differs from that of solving the energy equations more in magnitude than in quality; as Sec. 1.4 explains, both problems are, nowadays, readily soluble.

(g) The nature of the solution

What will result from solving the equations is a set of three-dimensional velocity fields, for successive instants of time, together with the concomitant pressure fields:

$$u\{\theta, r, z, t\}$$

$$v\{\theta, r, z, t\}$$

$$w\{\theta, r, z, t\}$$

$$p\{\theta, r, z, t\}$$

The pressure field, or rather that part of it which yields the pressure difference between inlet and outlet, will tell the designer how much power must be used to force the fluid through the shell. It will also tell much more, should it be necessary to know it, about forces on baffles, headers, and so on, but this information can be held in the computer until it is needed. The practically

oriented designer does not have to be inundated with numbers merely because the computer has calculated them.

The velocity fields, of course, are needed to put the solution of the temperature equation on a sound basis, for they replace the guesses that, at the level of discussion in Sec. B, had to be made for the terms ρu , ρv , and ρw .

The velocity fields also permit the best available knowledge concerning the connections between the convective heat transfer coefficients and the local relative velocities to be employed when the quantity U_2 is computed. It is true that this best available knowledge is still not very good, especially when the flow direction is neither normal nor parallel to the tubes in the bank, and when indeed the mean streamlines are strongly curved. Nevertheless it is useful to be able to discover where the relative velocities are high and where they are

low, so that at least the worst nonuniformities of heat transfer coefficient can be brought to light.

In summary, the valuable results that can result from a complete solution of the momentum and energy equations are as follows:

More accurate predictions of performance

Insight into the ways in which the actual flow pattern departs from the ideal one

Quantitative knowledge of transient behavior

A stimulus to acquire further general knowledge of convective heat transfer formulas (e.g., concerning oblique flow in tube banks) so as to increase the realism of predictions

This theme is taken up again in Sec. 1.4. Next, however, attention is given to those (rather few) situations for which analytical solutions can be procured.

Nomenclature and References for Section 1.2.6 appear at the end of Section 1.4.3.

1.3

Analytic solutions to heat exchanger equations

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1.3.1

Uniform-transfer-coefficient solutions for the no-phase-change heat exchanger

D. Brian Spalding

A. Introduction

In this section, solutions are assembled of the ordinary differential equations for temperature appearing in Sec. 1.2.5B; these are valid for heat exchangers of idealized (one-dimensional) multipass parallel- and counter-flow types, such as are described in Sec. 1.1.1.

Solutions are also provided of the partial differential equations for temperature appearing in Sec. 1.2.6B; these are valid for heat exchangers of multipass cross-flow types, which are also illustrated in Sec. 1.1.1.

The solutions are valid for heat exchangers without change of phase. However, if any stream should both enter and leave at the same temperature, having changed phase meanwhile, the solutions can still be used; this is effected by the device of imagining that there is a small, arbitrarily chosen temperature rise between the saturated liquid and saturated vapor states, δT_{lg} , say; this could be 10^{-5}°C , for example. Then the specific heat equivalent to the latent heat of vaporization, $c_{p,lg}$, is given by

$$c_{p,lg} \equiv \frac{\Delta h}{\delta T_{lg}} \quad (1)$$

If this specific heat is inserted into the heat exchanger equations, these become capable of predicting the with-phase-change behavior. The heat transfer coefficient U is presumed uniform.

The solutions are presented in both algebraic and graphic form, various formats of the latter being illustrated. Additional solutions and alternative representations are provided in Sec. 1.5.

B. Single-pass counter flow

For a counter-flow heat exchanger, dz_2/dz_1 in Eq. 1.2.5(9) equals -1. If the inlet of stream 1 is at the lower value of z_1 , integration of the equation yields

$$(T_{1,\text{out}} - T_{2,\text{in}}) = (T_{1,\text{in}} - T_{2,\text{out}}) \exp (NTU_2 - NTU_1) \quad (2)$$

Now the effective temperature difference can be related to the temperature changes in the two streams by equations derived from the definitions of the numbers of transfer units, NTU_1 and NTU_2 [Eq. 1.2.4(15)], and that of the effective temperature difference itself [Eq. 1.2.4(13) with U taken as constant]. Thus,

$$\begin{aligned} \Delta T_M \equiv T_1 - T_2 &= \frac{T_{1,\text{in}} - T_{1,\text{out}}}{NTU_1} \\ &= \frac{T_{2,\text{out}} - T_{2,\text{in}}}{NTU_2} \end{aligned} \quad (3)$$

from which it is easy to derive

$$\begin{aligned} \Delta T_M &= \frac{(T_{1,\text{in}} - T_{1,\text{out}}) - (T_{2,\text{out}} - T_{2,\text{in}})}{NTU_1 - NTU_2} \\ &= \frac{(T_{1,\text{in}} - T_{2,\text{out}}) - (T_{1,\text{out}} - T_{2,\text{in}})}{NTU_1 - NTU_2} \end{aligned} \quad (4)$$

Now Eq. (2) can be rearranged to yield:

$$NTU_1 - NTU_2 = \ln \left(\frac{T_{1,\text{in}} - T_{2,\text{out}}}{T_{1,\text{out}} - T_{2,\text{in}}} \right)$$

so that combination of Eqs. (4) and (5) yields a useful expression for the mean temperature difference in terms of the inlet and outlet temperatures of the two streams. It is

$$T_M = \frac{(T_{1,in} - T_{2,out}) - (T_{1,out} - T_{2,in})}{\ln [(T_{1,in} - T_{2,out}) / (T_{1,out} - T_{2,in})]} \quad (6)$$

Expressed in words, this equation states that the mean temperature difference causing heat transfer from one stream to the other is equal to the logarithmic mean of the temperature differences that prevail at the two ends of the exchanger.

Comparison of Eq. (6) and Eq. 1.2.4(11) shows that the symbol ΔT_{LM} has already been provided for this particular expression: For a single-pass counter-flow heat exchanger, ΔT_M equals ΔT_{LM} .

It is important to be aware of how the expression in Eq. (6) behaves when the temperature changes in the two streams become equal, as occurs when C_1 equals C_2 (and $R = 1$, therefore).

For algebraic convenience, let Eq. (6) be rewritten as

$$\Delta T_M = \frac{\Delta T_x - \Delta T_y}{\ln (\Delta T_x / \Delta T_y)} \quad (7)$$

Expansion of the logarithm as a series gives

$$\begin{aligned} \Delta T_M &\rightarrow \frac{\Delta T_x - \Delta T_y}{(\Delta T_x - \Delta T_y) / \Delta T_y + \frac{1}{2} [(\Delta T_x - \Delta T_y) / \Delta T_y]^2 + \dots} \\ &\rightarrow \frac{\Delta T_y}{[1 + \frac{1}{2} (\Delta T_x - \Delta T_y) / \Delta T_y]} \quad (8) \end{aligned}$$

Since the denominator of Eq. (8) can be expanded by the binomial theorem to give

$$\Delta T_M \simeq \Delta T_y \left[1 - \frac{1}{2} \frac{(\Delta T_x - \Delta T_y)}{\Delta T_y} \right] \quad (9)$$

it can be concluded that

$$\Delta T_M \simeq \frac{1}{2} (\Delta T_x + \Delta T_y) \quad (10)$$

Thus, as is easily understood, the logarithmic mean temperature difference reduces to the arithmetic mean of the temperature differences prevailing at the two ends of the heat exchanger.

The results of the analysis can now be expressed in terms of the three methods described in Sec. 1.2.4H.

(a) The *F*-correction-factor method

The fact that ΔT_M and ΔT_{LM} are identical for the counter-flow heat exchanger with uniform U implies, from Eq. 1.2.4(21), that F equals unity for the single-pass counter-flow exchanger.

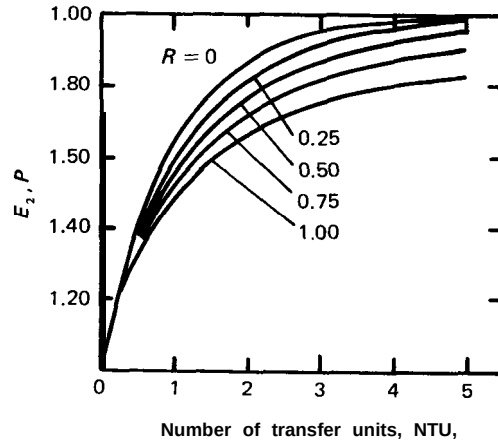


Figure 1 Effectiveness versus number of transfer units for a counter-flow heat exchanger with uniform U , for various values of flow ratio R .

(b) The *P-NTU₂* method

In order to determine the expression for the performance parameter P in terms of the number of transfer units NTU_2 , and the flow ratio R , it is necessary only to eliminate the temperatures from Eq. 1.2.4(25) and Eqs. (2) and (3). The result is

$$P = \frac{1 - \exp [-NTU_2 (1 - R)]}{1 - R \exp [-NTU_2 (1 - R)]} \quad (11)$$

This equation implies that P increases with the increase of NTU_2 and with decrease of R . Figure 1 represents the behavior graphically, for values of R between 0 and 1 ($C_2 < C_1$), for which P and E are identical.

When R tends to unity, Eq. (11) becomes indeterminate. However, conventional limit analysis reveals the following behavior:

$$R \rightarrow 1: \quad P \rightarrow \frac{NTU_2}{1 + NTU_2} \quad (12)$$

(c) The θ method

Since θ is connected to P and NTU_2 by Eq. 1.2.4(32), it is easy to deduce the following relation between θ and the two NTUs:

$$\theta = \frac{1}{NTU_2} \frac{1 - \exp [-NTU_2 (1 - R)]}{1 - R \exp [-NTU_2 (1 - R)]} \quad (13)$$

which may be expressed more symmetrically as

$$\theta = \frac{\exp (NTU_2) - \exp (NTU_1)}{NTU_2 \exp (NTU_2) - NTU_1 \exp (NTU_1)} \quad (14)$$

As the flow ratio tends to unity, and so NTU_1 and NTU_2 become equal, these expressions reduce to

$$R \rightarrow 1: \theta \rightarrow \frac{1}{1 + NTU} \quad (15)$$

C. Single-pass parallel flow

For a parallel-flow heat exchanger, dz_2/dz_1 in Eq. 1.2.5(9) equals +1. Integration of that equation therefore yields

$$T_{1,out} - T_{2,out} = (T_{1,in} - T_{2,in}) \exp [-(NTU_1 + NTU_2)] \quad (16)$$

Now Eq. (3), being no more than a matter of definition combined with a heat balance, is valid for any configuration of heat exchanger. With its aid, the effective mean temperature difference can be expressed as

$$\Delta T_M = \frac{(T_{1,in} - T_{2,in}) - (T_{1,out} - T_{2,out})}{NTU_1 + NTU_2} \quad (17)$$

and this can be combined with Eq. (1) with the result

$$\Delta T_M = \frac{(T_{1,in} - T_{1,out}) - (T_{2,out} - T_{2,in})}{\ln [(T_{1,in} - T_{2,in}) / (T_{1,out} - T_{2,out})]} \quad (18)$$

This equation differs from Eq. (6), which was the equivalent result for the counter-flow heat exchanger, only in the in and out subscripts T_2 , and these changes correspond precisely to the fact that stream 2 now enters at the same end of the exchanger as does stream 1, whereas previously it entered at the opposite end. It follows that the expression for the mean temperature difference of both counter-flow and parallel-flow heat exchangers is still the logarithmic mean, T_{LM} ; this can be defined generally, in the manner of Eq. (7), as

$$\Delta T_{LM} = \frac{AT_c - AT_h}{\ln (\Delta T_x / \Delta T_y)} \quad (19)$$

where AT_c and AT_h stand for the temperature differences between the streams prevailing at the two ends of the heat exchanger.

The results of this analysis can now be represented in terms of the three methods described in Sec. 1.2.4H.

(a) The F -correction-factor method

The ratio of the effective mean temperature difference for a parallel-flow exchanger to that for a counter-flow exchanger with the same number of transfer units and the same flow ratio can be expressed, with the aid of the θ 's given by Eq. (26), below, and Eq. (14), above. This however is not what is required in the F -correction-factor method, for F is defined as the ratio of the mean temperature differences for the same values of P and R .

The formula for F defined in this way is

$$F = \frac{\eta/\xi}{\ln \{ [2 - P(1 + R - \eta)] / [2 - P(1 + R + \eta)] \}} \quad (20)$$

where

$$\eta \equiv \sqrt{R^2 + 1} \quad (21)$$

and

$$R \neq 1: \quad \xi \equiv \frac{R + 1}{\ln |(1 - P)/(1 - PR)|} \quad (22)$$

$$R=1: \quad \xi = \frac{1 - P}{P} \quad (23)$$

The general shape of the $F(P, R)$ function corresponding to these expressions is similar to that appearing in Fig. 1.2.4(1), but there are quantitative differences.

(b) The P - NTU_2 method

The performance parameter P , which is also the effectiveness based on stream 2, E_2 , can be determined by elimination of temperatures from Eqs. (3) and (17), and Eq. 1.2.4(25). The result is

$$P = \frac{1 - \exp [-(R + 1) NTU_2]}{R + 1} \quad (24)$$

This equation shows that P increases as NTU_2 increases and R diminishes; Fig. 2 represents its implications graphically.

Comparison of Figs. 1 and 2 shows that the counter-flow heat exchanger is always superior to (i.e., has a higher P value than) the parallel-flow heat exchanger, for a given number of transfer units and flow ratio.

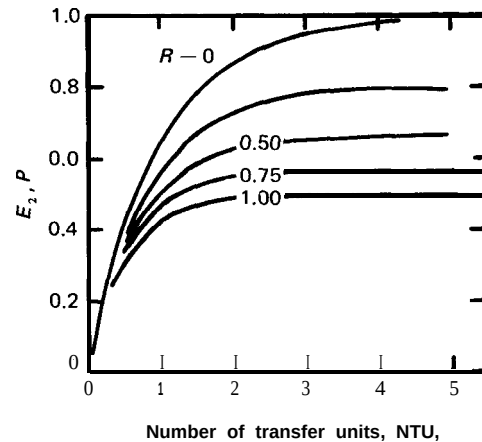


Figure 2 Effectiveness versus number of transfer units for a parallel-flow heat exchanger with uniform U , for various values of R .

(c) The θ method

The ratio of the effective mean temperature difference to the maximum temperature difference, that is, θ , is simply P/NTU_2 . Thus,

$$\theta = \frac{1 - \exp [-(R + 1)NTU_2]}{(R + 1)NTU_2} \quad (25)$$

or, more symmetrically,

$$\theta = \frac{1 - \exp [-(NTU_1 + NTU_2)]}{NTU_1 + NTU_2} \quad (26)$$

D. Cross-flow exchangers

There are three cross-flow configurations that require attention:

1. Neither stream is laterally mixed.
2. Both streams are laterally mixed.
3. One stream is laterally mixed and the other is not.

For case 1, there exists an exact solution of the relevant equations first published by Nusselt [39, 40] and discussed by Stevens et al. [35], also, Eckert and Drake [12] present an approximate formula, namely,

$$P = 1 - \exp \left[\frac{\exp (-NTU_2 R \eta) - 1}{R \eta} \right] \quad (27)$$

where η is defined by

$$\eta \equiv NTU_2^{0.22} \quad (28)$$

For case 2, [5] gives the formula

$$P = \{ [1 - \exp (-NTU_2)]^{-1} + R [1 - \exp (-R NTU_2)] - 1 \}^{-1} \quad (29)$$

For case 3, when it is stream 1 that is unmixed, [5] (and the more accessible [35] also) gives

$$P = 1 - \exp \frac{-[1 - \exp (-R NTU_2)]}{P} \quad (30)$$

and, when it is stream 2 that is unmixed, the same reference gives

$$P = R^{-1} (1 - \exp \{-R[1 - \exp (-N_2)]\}) \quad (31)$$

Of course, other quantities, such as θ and F , can be deduced from the above expressions for P , by way of the appropriate definitions,

E. Other configurations

The more complex the flow configuration, the more complex the corresponding analytical expression, if it exists, for the performance parameters of the heat exchanger. Thus, for exchangers in which the shell fluid flows in a single pass and the tube fluid in two, or a multiple of two, passes, the P quantity can be expressed in terms of NTU_2 and R as

$$P = 2 \left[1 + R + \frac{1 + \exp (-NTU_2 \sqrt{1 + R^2})}{1 - \exp (-NTU_2 \sqrt{1 + R^2})} \cdot \sqrt{1 + R^2} \right]^{-1} \quad (32)$$

Reference [13], from which this formula is taken, contains numerous further expressions and their graphic counterparts for a wide variety of flow configuration. Section 1.5 also contains a comprehensive collection of equations, graphs, and tables, so there is no need to provide further material here.

Nomenclature and References for Section 1.3.1 appear at the end of Section 1.4.3.

1.3.2

Other analytic solutions

D. Brian Spalding

■ Prior to the advent of the digital computer, the ingenuity of research workers in heat exchanger theory was exercised in devising analytical solutions for idealizations of heat exchanger phenomena that were somewhat closer to reality than the uniform- U , steady-state idealization of Sec. 1.3.1 [36,37].

Thus, it is possible to allow U to vary with either position or temperature difference without thereby rendering the equations of parallel- or counter-flow exchangers totally intractable, and transient effects have also been treated analytically in simple cases.

However, this whole stream of work has been rendered obsolete by the ease with which numerical solutions can be produced, nowadays, by recourse to a digital computer and the appropriate computer program. Numerical methods can allow much greater departures from ideality, and therefore closer representations of

reality, than were ever envisaged by the workers with analytical methods, and they do it without significantly greater penalty. It therefore appears that there is no need in this handbook to review the analytical work that has just been referred to.

Even the formulas and charts of Sec. 1.3.1 are more of educative than practical value, for when it is easy to do so, one might as well take nonuniformities of heat transfer coefficient, and of fluid properties, into account right from the start. Graphs have been provided in this handbook because of their central place in the thinking of heat exchanger specialists. However, there is little doubt that the attention given to the graphic representation of idealized configurations will diminish in the future. The reasons should be fully apparent by the time that Sec. 1.4 has been read.

1.4

Numerical solution procedures for heat exchanger equations

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1.4.4

Cases with prescribed flow patterns

D. Brian Spalding

A. Introduction

This section discusses how the behavior of a heat exchanger can be computed when the idealizations of Sec. 1.3 are too remote from reality, because, for example,

The flow is transient; or

The heat transfer coefficient is nonuniform; or

The fluid properties are nonuniform; or

The geometry of the inlet flow, and the leakiness of the baffles, are expected to cause large departures from the ideal flow pattern.

The last point, involving calculation of the flow pattern, is dealt with in Sec. 1.4.2; in the present section, attention is confined to problems in which the flow pattern can be presumed.

In almost all real heat exchangers, temperatures are distributed in a three-dimensional manner, for even when the major part of the configuration is **axisymmetrical** in character, the nozzles and headers through which fluid enters and leaves the exchanger almost inevitably involve departures from that symmetry.

A natural-draught cooling tower of hyperboloidal form is an exchanger that conforms as closely as any to the axisymmetrical ideal, yet a small amount of (necessarily horizontal) wind makes conditions within and around the tower three-dimensional. This section, therefore, **focusses** on the prediction of three-dimensional flow phenomena.

The application of numerical methods to heat exchanger design has a long history, which has led to many publications. There is an even vaster literature

concerned with the application of numerical methods to problems of fluid flow, and of heat and mass transfer in general. Anyone wishing to trace origins and compare alternatives will **find** it easy to begin (although hard to make an end) by turning to the appropriate specialized textbooks and journals.

B. Discretization

(a) Subdivision of space

Discussion is conducted, as is done in Sec. 1.2.6, by reference to the $\theta \sim r \sim z$ (i.e., cylindrical polar) coordinate system [Fig. 1.2.6(1)].

Numerical methods work with a limited set of numbers; it is therefore advisable to divide the space to be considered into a finite number of continuous nonoverlapping subdomains, usually called cells, and to associate one value of each dependent variable with each cell.

Specifically, let the cylindrical space representing the heat exchanger be subdivided by three sets of surfaces: planes of **fixed** θ , cylinders of **fixed** r , and planes of **fixed** z . Then the resulting spatial subdivision may be as illustrated in Fig. 1. There is no need for the intervals of θ , r , and z to be uniform, and as a rule there are advantages in making them nonuniform, so as to concentrate particular attention on parts of the heat exchanger that are of special concern.

Let it now be supposed that the dimensions of the cells are large compared with those of the heat transfer surfaces, for example, the tube diameters, as Fig. 2

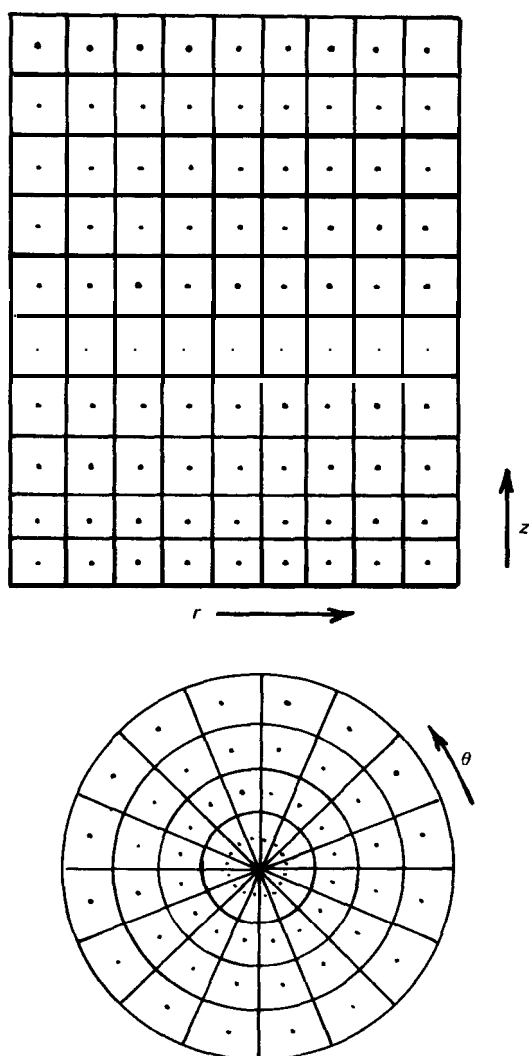


Figure 1 Discretization of the cylindrical heat exchanger space by planes or cylinders of constant θ , r , z .

illustrates. Since it is necessary to ascribe values T_1 , T_2 , and T_m to the values of the tube-side fluid, shell-side fluid, and metal, respectively, it is clear that some averages must be taken. These averages are defined as those that, when substituted into the algebraic equations

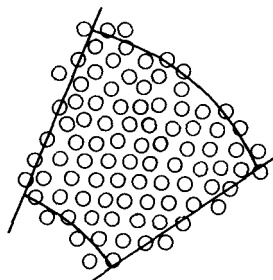


Figure 2 Illustration of the fact that many tubes or other heat transfer elements may fit into the cross section of a single cell.

to be developed, give the correct fluxes of energy, etc., for the control volume surfaces as a whole.

It may aid the imagination to conceive of these average temperatures as existing at some typical points within the cells. It is for this reason that the dots, representing these typical points, have been placed within the cells illustrated in Fig. 1.

Figure 3 illustrates a single cell to a larger scale, and introduces some notation that will be useful. The central point is denoted by P , and the central points of neighboring cells on the north, south, east, and west, high, and low sides are denoted appropriately by N, S, E, W, H , and L . A memory point, $P-7$, is also shown; this is supposed to represent the point P , and so, of course, the cell surrounding it, at the immediately previous instant of time. The six faces of the cells are given the symbols n, s, e, w, h, l in the appropriate order. The main use of all these letters will be as subscripts to symbols for temperature, density, velocity, etc., indicating to what cell volume or cell wall the latter apply.

The subdivision of space is performed, for a heat exchanger study, at the beginning of the computation. The cell wall locations are neither required nor allowed to vary as the computation proceeds.

(b) Subdivision of time

Just as space is subdivided into finite intervals, so is time, as the above mention of the immediately previous instant has revealed. The calculation proceeds more like a digital clock than one with imperceptibly moving hands, but the size of the time interval is adjustable. Thus, in the analysis of the response of a heat exchanger to a sudden change in inlet temperature, it might be appropriate to employ small time intervals at first, so that rapid changes could be perceived by the computation; the subsequent more gradual reversion to the

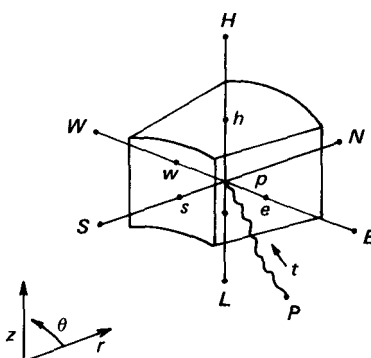


Figure 3 Representation of a typical cell, with central point P ; neighbor points N, S, E, W, H , and L ; and face points n, s, e, w, h, l . P represents point P at the immediately prior instant of time.

steady state might well be computed by way of a series of larger time intervals.

C. Finite-difference equations

(a) Nature of the finite-difference equations

If there are n_θ subdivisions of space in the θ direction, n_r is the r direction, and n_z in the z direction, there will be $n_\theta n_r n_z$ values of each of the three dependent variables T_1, T_2 , and T_n to compute at each instant of time. The n 's are typically of the order of 10; so around 3 000 values require computation at each time. The question now arises: how are they to be computed?

The answer is this: a set of algebraic equations for the variables is contrived, in which each variable, T_{1P} , say, is connected with the values pertaining to six neighbors ($T_{1N}, T_{1S}, T_{1E}, T_{1W}, T_{1H}, T_{1L}$) and to the relevant earlier time value, T_{1P-} . The equations are called finite-difference equations for historical reasons; but the name is now more misleading than helpful, because of the presence in the literature of invalid generalizations about equations with that name.

The equations are normally given a linear appearance, of which the following is typical:

$$\begin{aligned} \phi_P = & a_N \phi_N + a_S \phi_S + a_E \phi_E + a_W \phi_W + a_H \phi_H + a_L \phi_L \\ & + a_P \phi_{P-} + S_P / a_N + a_S + a_E + a_W + a_H \\ & + a_L + a_{P-} - S'_P \end{aligned} \quad (1)$$

Here ϕ stands for any one of the dependent variables T_1, T_2, T_m , and the coefficients a_N , etc., are characterized by the following statements:

They are all greater than or equal to zero.

They represent the effects of convection and diffusion (a_N increases with the magnitude of the wind from the north).

They may be functions of the ϕ 's (as when the effective conductivity is temperature-dependent), which explains why only an appearance of linearity can be claimed for the equations.

The quantities S_P and S'_P are representative of the linearized source of the entity of which ϕ measures the amount, in accordance with the equation

$$\text{Source of } \phi \equiv S_P + S'_P \phi_P \quad (2)$$

(b) Origin of the equations

Finite-difference equations such as Eq. (1), of which there will be $3 \times n_\theta n_r n_z$, are best obtained from the relevant partial differential equations for the three variables, T_1, T_2, T_m (see Sec. 1.2.6), by integration of these equations over the relevant cell volumes.

When this integration has been formally completed, resulting in an equation with treble and quadruple integrals for the faces and the volume, respectively, further progress can be made only by way of the prescription of interpolation procedures. This human input is inevitable and just, because it is humans who have decided to remove some of physical reality from attention by replacing the continuum for which the partial differential equations held by the finite set of subdomains.

The interpolation procedures recommended here are as follows:

Evaluate all fluxes through cell walls in terms of values prevailing at the end of the time instant.

Evaluate convective fluxes through cell walls by imagining that the average ϕ value at a wall is that prevailing on the upwind side of that wall.

Evaluate diffusive fluxes at walls by reference to the ϕ values at the center points of the cells adjoining that wall.

Treat the movement through time as a convective flux in which (density $\times$ area $\times$ velocity $\times$ time interval) is replaced by (density $\times$ volume).

Evaluate sources in terms of the ϕ values prevailing at the end of the time interval.

These are not the only practices that could be advocated with plausible justification, but they have served well and will continue to do so.

(c) Some details

As a consequence of these recommendations, the coefficient a_N when ϕ stands for a temperature, for example, can be computed from the expressions

$$a_N = \frac{1}{2} (R_P + R_N) A_n (D_n + C_n) \quad (3)$$

$$D_n = \frac{1}{2} \frac{\Lambda_P + \Lambda_N}{r_N - r_P} \quad (4)$$

$$\text{If } v_n \geq 0: \quad C_n = 0 \quad (5)$$

$$\text{If } v_n < 0: \quad C_n = -\rho_N v_N c_{PN} \quad (6)$$

Here R_P and R_N are the volume fractions accessible to the fluid in question, A stands for thermal conductivity, A_n is the area of the northern face of the cell, D_n is the diffusional (i.e., conductive) contribution to the heat flux, C_n is the convective contribution to the heat flux, and c_{PN} is the specific heat appropriate to the temperature range from T_P to T_N .

The other a 's are computed in the same way, except for a_{P-} , which is given by

$$a_{P-} = R_P V_P \rho_{P-} \quad (7)$$

where V_P is the volume of the cell and ρ_{P-} is the density of the fluid at the start of the time instant in question.

D. A straightforward solution procedure

If it is supposed that all the flow velocities are known, that values can be ascribed to all the A 's, and that the volume fractions are all deducible from the prescribed heat exchanger geometry, all the a 's in the finite-difference equations can be evaluated for the time interval in question.

If it is further supposed that the volumetric heat transfer coefficients U_1 and U_2 , appearing in the energy equations of Sec. 1.2.6B, are also known for every location, it is also possible to ascribe values to the S_p 's and S_p' 's, for, typically :

$$S_p = U \times \text{cell volume} \times T_m \quad (8)$$

$$S_p' = -U \times \text{cell volume} \quad (9)$$

Solution can now proceed, for a particular time step, by the simple method known as the Gauss-Seidel relaxation procedure. In this method, the values of all variables in store are brought successively into accordance with the finite-difference equations for which they occupy the left-hand position, and for which the in-store values of the neighboring values are substituted into the right-hand side. Since, if points are being visited in a south-to-north order, a northward move adjusts the temperature by reference to which the just-vacated southerly point was adjusted, it is obvious that the process must be repeated many times. What is not obvious, but is true, is that if it is repeated many times, the magnitudes of the adjustments that are necessary become smaller and smaller, so that, finally, they are negligible. The solution is then said to have converged, whereafter attention can be given to the next time step.

Because repeated adjustments, that is, iterations, are needed, the opportunity repeatedly presents itself to change the values of the coefficients if there is any reason for so doing. There is a reason if they exhibit nonlinearity, for example, if the heat transfer coefficients depend (as they always do to some extent) on the temperatures or temperature differences; and there would be no point, in that case, in not updating the values. This is how it can be asserted that, when numerical methods are employed, nonlinear problems present no special difficulty.

E. More economical solution procedures

In practice, the Gauss-Seidel procedure is rarely employed in computer codes designed for the economical solution of large-scale problems. Instead, methods of successive adjustment of variables are employed that allow the solution to be approached more rapidly. Alternating direction implicitly, strongly

implicit, and successive overrelaxation are names for procedures, or parts of procedures, that are commonly used.

These matters are for the numerical methods specialist. From the point of view of the readers of this handbook, the only statement necessary is that many methods of solution do exist and that, if the appropriate ones are employed, the solution of a problem in which the heat exchanger volume is represented by a few thousand cells is not an expensive matter. More time and money are likely to be consumed in specifying the problem adequately, so that all the relevant information is supplied to the computer, than in actually solving it.

F. The utility of calculations based on prescribed flow distributions

It may be useful, now that an insight has been provided into how numerical predictions of heat exchanger performance can be made, to examine a list of uses to which the procedure can be put. In doing so, it must be remembered that, in what has been presented so far, ability to prescribe the flow pattern has been presumed.

(a) The sensitivity to the flow pattern prescription

It may be that a heat exchanger has been designed on the basis of conventional presumptions about where cross flow exists, and where parallel flow or counter-flow; and it may be suspected that the true flow pattern deviates from the presumed one appreciably. Does it matter? How much does it matter, in a variety of circumstances?

Such questions are easy to answer. One makes a series of systematically differing presumptions about the flow pattern, spanning the range that can be expected in practice; then, by examining the extent to which these changes alter the thermal performance of the heat exchanger, one determines what features of the flow pattern, if any, are crucial.

What if the flow distribution in the tubes is not uniform, either because some tubes had to be removed, or because of poor header design? The probable effect on thermal performance can be estimated in the same way.

(b) Transient performance

The way in which the outlet temperatures from a heat exchanger will vary with time, as a consequence of arbitrary changes to the rates or temperatures of the inflowing streams, is easy to compute with the aid of a numerical method of the kind described, but difficult or impossible without.

(c) Influences of nonuniform heat transfer coefficients

Viscous liquids, for example, exhibit strongly varying heat transfer coefficients if significant temperature changes occur as they flow through the heat exchanger. This nonlinear interaction necessitates a numerical procedure for its correct modeling.

This part is often recognized, in practice, by the performance of one-dimensional numerical computations, but much of the reality is missed thereby. Unevennesses of temperature that exist in the cross section of a heat exchanger lead to unevennesses in mass flow distribution also; a kind of instability can even result, according to which the dependence of resistance coefficients on temperature acts so as to increase the resistance to flow in regions that are already in disproportionately slow motion. Such effects can be detected only by two- and three-dimensional analyses.

(d) Parametric studies

The paucity of existing performance charts for practically occurring situations is commented upon in Sec. 1.3. Possession of a general numerical solution procedure would enable as large a number as necessary of these charts to be generated. Then the designer would more often be able to get the needed answer from existing manuals, and would less frequently need to resort to the computer. There is undoubtedly still some scope for such parametric studies, and for the publication of their results.

G. The influence of the fineness of discretization

The user of numerical procedures for simulating continuum-flow processes has always to effect a compromise between the requirements of accuracy and of economy; and the fineness of the discretization of space is that which determines whether the choice is optimal.

If the discretization is coarse, which is to say that the exchanger volume is represented by a rather small number of cells, the accuracy of prediction is low, but so, also, is the computational expense. A fine discretization, on the other hand, that is, subdivision into a very large number of cells, produces high accuracy, but the cost of the computation is often unacceptably large.

Let it be supposed that a single heat exchanger problem is computed several times, and that the number of cells into which the exchanger is divided is doubled for each computation. It will be found that the effectiveness of the exchanger, when plotted against the number of cells, will tend to an asymptotic value (Fig. 4). When the effectiveness is sufficiently close to the

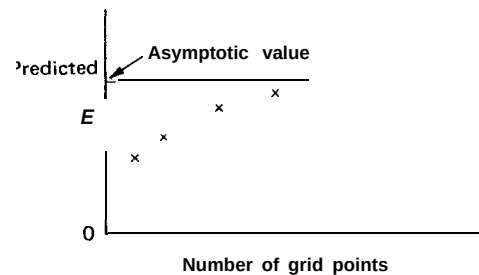


Figure 4 Illustration of the influence of fineness of discretization on the predicted heat exchanger effectiveness.

asymptotic value for practical purposes, the discretization can be regarded as *fine* enough; the results are then termed grid-independent, and regarded as acceptable. Since the cost of computation increases in proportion to the number of grid points (or even more steeply with some methods), there is no point in using a finer grid.

No general rule can be given as to how many cells are needed to ensure grid independence, for the necessary number depends on the flow configuration, the number of transfer units, the way in which the heat transfer coefficient varies with (say) temperature, and indeed on the precision of prediction that is required; 10 cells may suffice for one problem, whereas 1 000 may be needed for another. Usually, therefore, it is necessary to perform a computation with differing discretization finenesses, in order to establish the satisfactoriness of the results.

It is worth noting that the necessary fineness can be reduced by taking care to employ the most suitable effective heat transfer coefficient U , appearing in Eqs. (8) and (9), for these equations give the heat sources as proportional to the differences between the representative temperatures of fluid and metal in the cell; and, if

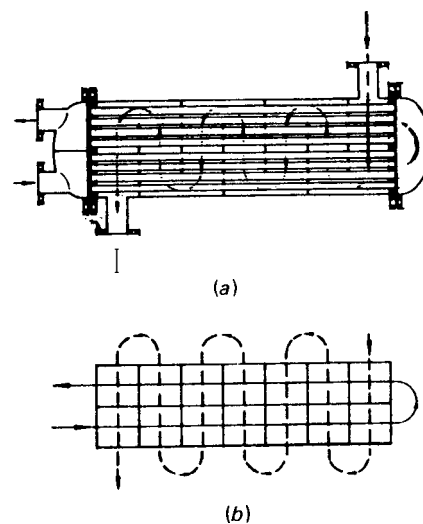


Figure 5 (a) Multipass shell-and-tube heat exchanger with baffles. (b) Idealized heat exchanger.

the cell is large, the variations of temperature within the cell should be allowed for. The best way to understand this idea is by regarding the individual cell as a microexchanger, having a small but finite number of transfer units, for which the **value** of θ [see Eq. 1.3.1(25)] is less than unity. What value depends, of course, on NTU and on the local flow configuration; and the equations of Sec. 1.3.1, and the graphs of Sec. App. 1.5 provide guidance as to its choice. The U of Eqs. (8) and (9) should be θ times the U .

Approaching the subject from the microexchanger point of view, the heat exchanger analyst may scarcely recognize that he or she has begun to use **finite-difference** procedures; and the specialist in the latter may not recognize the merit of introducing θ into the formulas. A recent reference that is representative of the former approach is the paper by Gaddis and Schlünder [38], from which Fig. 5 has been reproduced.

Nomenclature and References for Section 1.4.1 appear at the end of Section 1.4.3.

1.4.2

Cases in which the flow patterns must be calculated

D. Brian Spalding

A. Introduction

It has been seen that the energy equations can be solved by numerical analysis, and that the solution is in no way hindered by the existence of the nonlinearities and interlinkages between equations. Why, then, should not the momentum and mass continuity equations also be solved numerically? Their nonlinearity and interlinkedness may exceed that of the energy equation, but is there reason to suppose that this quantitative change entails a qualitative one also?

As will be shown, the momentum and continuity equations can indeed be solved numerically, and although the interlinkedness does present a special problem, it is one that is amenable to a simple solution.

Of course, suitable input information must be provided. It is no longer necessary to specify the velocity components at every point, but it is necessary to supply formulas for the resistances to flow, in the various directions, presented by the tubes, baffles, and other features.

It is the flow in the shell only, as a rule, that requires this treatment, for the tube-side fluid is so well controlled by its confinement within the tubes that conventional procedures suffice for determining its distribution. However, there do exist heat exchangers—for example, those of plate type—in which numerical solution of the flow distribution equations is needed for both fluids.

The direct-contact cooling tower is another device for which it may be desirable to solve the equations of motion for the water as well as for the air, for the

friction between the two fluids may cause the water to depart appreciably from a purely vertical flow direction.

In general, therefore, it may be necessary to solve six momentum equations. However, present purposes are adequately served by concentrating on the three equations appropriate to a single fluid.

B. Discretization

The subdivision of space into the $n_\theta n_r n_z$ cells used for temperature remains useful when velocities are to be computed; and the very same cells will be employed when the pressure is being adjusted so as to satisfy the continuity equation. However, additional cells are required to allow the velocity components to be computed.

Figure 1 illustrates the locations of these additional cells. They can be imagined as formed by taking the cells used for temperature and pressure and displacing them by half a θ interval to make the u cells, by half an r interval to make the v cells, and by half a z interval to make the w cells. This arrangement of cells, which results from the fact that velocities are being computed for locations that lie on the walls of the temperature and pressure cells, is sometimes known as the staggered grid.

C. Finite-difference equations

Equation 1.4.1(1) can serve to describe the finite-difference equations for the variables u, v , and w , as

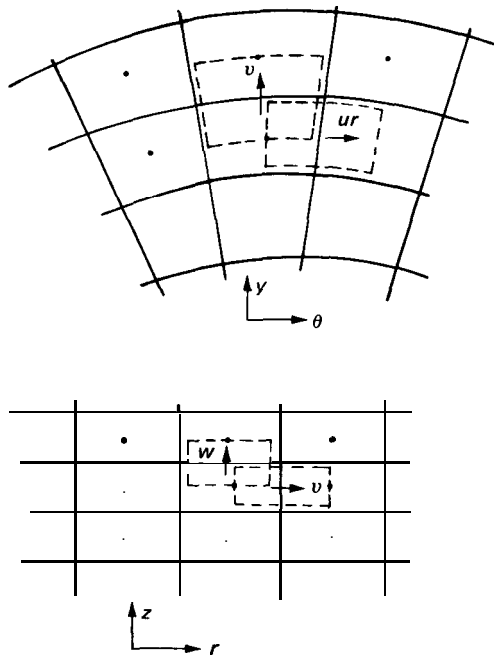


Figure 1 Illustrations of the cells employed for the setting up of finite-difference equations for u_r , v , and w .

well as those for the temperatures, provided only that it is recognized that the locations of all the points must be regarded as having shifted half an interval in the appropriate direction, viz., the θ direction when ϕ stands for u_r , the r direction when ϕ stands for v , and the z direction when ϕ stands for w .

The convection contributions to the finite-difference coefficients are not now quite so easily calculated, for the velocities at the new control volume boundaries have to be determined by interpolation; however, this presents no difficulty.

The source terms now contain several components, namely :

Forces exerted on the control volume by pressure differences

Centrifugal forces

Buoyancy forces

Forces associated with the flow resistance of tube bundles and baffles

Such of the internal friction forces as are not easily included within the a s

Each of these can be represented in terms of the variables provided in a straightforward way.

The mass conservation equation, however, requires special attention. It can be represented as

$$b_n + b_s + b_e + b_w + b_h + b_l + RV(\rho_P - \rho_{P-}) = 0 \quad (1)$$

where the b 's represent the amounts of mass outflow during the time interval; for example,

$$\text{If } v_n \geq 0: \quad b_n = \frac{1}{2}(R_N + R_P)A_n \rho_P v_n \quad (2)$$

$$\text{If } v_n < 0: \quad b_n = \frac{1}{2}(R_N + R_P)A_n \rho_N v_n \quad (3)$$

Here upwind densities multiply velocities, as was already tacitly shown in Eq. 1.4.1(6).

From Eq. (1) there is derived a finite-difference equation for the pressure, or rather for a correction to the pressure. The procedure is as follows. Before the converged solution has been achieved, the velocities in store will not, in general, satisfy Eq. (1). If the in-store values of b 's are indicated by an asterisk, and the necessary corrections to them by a prime, Eq. (1) can then be rewritten as

$$b'_n + b'_s + b'_e + b'_w + b'_h + b'_l = -RV(\rho_P - \rho_{P-}) - b_n^* - b_s^* - b_e^* - b_w^* - b_h^* - b_l^* \equiv E_P \quad (4)$$

Next, the primed b 's are supposed to be linearly related to pressure adjustments, denoted by p , which are to be made at nearby points. Thus,

$$b'_n = b_n''(p'_P - p'_N) \quad (5)$$

$$b'_s = b_s''(p'_P - p'_S) \quad (6)$$

$$b'_e = b_e''(p'_P - p'_E) \quad (7)$$

$$b'_w = b_w''(p'_P - p'_W) \quad (8)$$

$$b'_h = b_h''(p'_P - p'_H) \quad (9)$$

$$b'_l = b_l''(p'_P - p'_L) \quad (10)$$

The coefficients $b_n'', b_s'',$ etc., are actually obtained by differentiation of the finite-difference equations for the velocities (to which the b 's are proportional) with respect to the S_p 's (which contain the pressures).

Substitution of Eqs. (5) through (10) into Eq. (4) leads to

$$p'_P = b_n''p'_N + b_s''p'_S + b_e''p'_E + b_w''p'_W + b_h''p'_H + b_l''p'_L + E_P/b_n'' + b_s'' + b_e'' + b_w'' + b_h'' + b_l'' \quad (11)$$

This equation, the finite-difference equation for the pressure correction, bears a great formal similarity to the general finite-difference Eq. 1.4.1(1). It is necessary only to identify p' with ϕ , b'' with a , and E_P with S_p , and to put S_P equal to zero.

Equation (11) completes the set of finite-difference equations that must be solved.

D. A straightforward solution procedure

Section 1.4.1D, indicates how the Gauss-Seidel point-by-point adjustment procedure, applied iteratively to all

the grid nodes, can lead finally to a converged solution to the temperature equations. The counterpart of this procedure, which solves the associated velocity equations, can now be described.

There are six velocities associated with each temperature cell. Let it therefore be supposed that a Gauss-Seidel adjustment is made to each of these six velocities by reference to the appropriate finite-difference equation immediately after the temperatures have been adjusted.

Now let these velocities be substituted into the expressions for the b 's. It will be found, in general, that the mass conservation principle is not satisfied by them, that is, that the quantity Ep , defined by Eq. 1.4.1(4), is not zero. This is a sign that the pressure at point P needs adjustment, and the velocities also.

The increment to p_P that is needed, namely, p'_P can now be obtained from Eq. (11), which reduces, because the neighboring p 's are zero (neighbor values not being adjusted in a single-point adjustment procedure), to

$$p'_P = \frac{Ep}{b''_n + b''_s + b''_e + b''_w + b''_h + b''_l} \quad (12)$$

The final step is to evaluate the associated corrections to the b 's, viz., b'_n, b'_s, b'_e , etc., and to apply the corresponding corrections to the stored values of the six velocities. Then attention can pass to the next point.

This procedure, if conducted in an iterative manner over the whole field, will lead finally to a set of values of temperatures, pressures, and velocities that satisfies all the finite-difference equations as closely as is desired. Of course, it takes longer than the procedure for the temperatures alone, for more variables have to be computed; nevertheless, the increased expense has brought with it freedom from the necessity to prescribe (often without real knowledge) the velocity distributions.

E. More economical procedures

Just as the energy equations can be solved by procedures that are more economical than the Gauss-Seidel one, so can the complete set of equations be solved by procedures that, because they handle more than one point at a time, can greatly reduce the convergence time as compared with what would result from direct application of the procedure described in Sec. D.

There is no need to describe these here, for the reader is more likely to wish to become a user of an established computer program than one who develops the methodology further. It should therefore be sufficient to state that the available methods provide solutions at an expense, in terms of computer time, that

is acceptable in many practical circumstances, and that the continued improvements in computer software and hardware are steadily reducing the costs.

F. Some uses to which numerical procedures for solving all the equations can be put

In addition to the uses for numerical procedures described in Sec. 1.4.1 F, there are further ones that may be envisaged once the velocity fields are capable of being computed. Consideration of the following list may excite interest and stimulate the finding of further uses.

(a) Detection of the role of buoyancy forces

Heat exchangers are large; the fluids within them are often in fairly slow motion, so as to reduce pumping costs; and temperature variations are present. These are circumstances liable to give rise to buoyancy-induced flows that, though they may sometimes be advantageous, may on other occasions have deleterious effects on performance.

In boilers, of course, the buoyancy forces are often employed as the main or only means of promoting circulation of the water past the heating surfaces, but in other types of heat exchanger the effects of buoyancy may have been disregarded during the design phase.

A numerical procedure allows such effects to be quantified. Indeed, it is easier to conduct the relevant numerical experiment than it would be to perform a physical one even on the most elaborately instrumented apparatus, for, in the computation, the gravitational acceleration can be easily switched off by setting the value of g_n to zero, but a visit to outer space is needed before the equivalent effect can be achieved in practice!

(b) Influences of baffle leakage

A baffle is a resistance to flow that is highly directional: it impedes flow normal to it, but augments only slightly the resistance to flow along its surface.

Such directional resistances are easy to introduce into the numerical procedure, mainly by way of the S'_p terms; and S'_p for u and v can be zero while that for w is very large.

All baffles are leaky to some extent, and the effects of this leakiness can be predicted only by a complete calculation, of the kind indicated above, that takes account of the resistances to flow in all directions and so computes how the various velocity components adjust themselves to comply with the loss of momentum and mass conservation.

(c) Fouling, corrosion, vibration, thermal stress

Heat exchanger performance deteriorates in the course of time as a consequence of deposition of material that builds insulating layers on heat transfer surfaces.

Sometimes the deposited material interacts deleteriously with the material of which the heat exchanger is composed, which is thereby corroded, or suffers other damage.

Effects of this kind are strongly influenced by details of the flow pattern, for deposition occurs most readily in regions of low velocity, and even in regions of the equipment where the average velocity is sufficient to prevent deposition, there may be small subregions where this condition is not satisfied. These small departures from the average may suffice to put the heat exchanger out of commission, with serious economic consequences.

Numerical prediction procedures that disclose details of velocity distributions are especially useful for the light they can cast on processes of this kind, for they indicate where deposition is likely to occur, and they can also be useful in determining what design changes, such as provision of flow redistribution plates, are likely to be advantageous.

Flow-pattern predictions can also reveal regions in

which tubes are likely to vibrate as a consequence of the high velocity of the fluid flowing between them, and detailed knowledge of the temperature distribution in the tubes, tube supports, and shell permits thermal stresses, both steady and transient, to be computed.

It therefore follows that, even when the thermal performance of a heat exchanger is so insensitive to the flow pattern (under design conditions) that the complete numerical study is regarded as unnecessary by the thermal specialist, the structural designer may need to commission such a study for his or her own purposes.

G. References

So far, few publications have appeared in which calculations of flow patterns in heat exchangers, and of their influence on performance, have been presented. However, two papers by Patankar and Spalding [14,15], provide preliminary exemplifications of what can be achieved; and [16] gives a more extensive discussion of possible applications. Further information about the numerical procedures outlined above can be found in [17],[18],[19], and [20].

Nomenclature and References for Section 1.4.2 appear at the end of Section 1.4.3.

1.4.3

Special applications of numerical solution procedures

D. Brian Spalding

A. The prediction of two-phase flows

When a liquid stream begins to boil, or a vapor phase to condense, two new factors have to be taken account of by the numerical analyst. The first is that rather larger changes take place in the density and in other properties of the **fluid**, while the temperature remains practically constant. Thus, the variable property problem, which will have been dealt with to some extent in connection with the single-phase fluid, here assumes a greater quantitative significance.

The second factor is that the two phases, although present in the same region of the heat exchanger, and fairly intimately interspersed, may be proceeding in different directions and at different velocities. Typically, the vapor is likely to travel upward at a greater velocity than the liquid, because the pressure gradient, which is shared by the two phases (apart from surface tension effects, not affecting the argument) exerts the same upward force on each per unit volume, whereas the gravitational field exerts a much greater downward force per unit volume on the liquid than on the vapor.

This second factor substantially increased the numerical analyst's task, for he or she must now calculate two sets of velocity components rather than one, and the relevant equations are interlinked not only by their common pressure but also by interphase momentum transfer terms, which depend on the relative velocities of the two phases.

This is a branch of numerical fluid mechanics that is

undergoing rapid and intense development. The problem of obtaining solutions to the equations has been solved, and the need is now for an increase in knowledge concerning the interphase transfer laws.

It must be recognized that to regard the problem as that of obtaining just two sets of velocity components is itself an idealization that will not always be acceptable, for at any particular point, not all particles of the liquid phase will be proceeding at the same velocity. A familiar example in which wide velocity differences occur is the fountain formed when water flows from a nozzle vertically upward into the atmosphere; all the water falls down again, and so, in the absence of a significant lateral wind, there is a large volume near the axis of the fountain in which both upward-moving and downward-moving water can be found.

Likewise, when the vapor is dispersed within the liquid in the form of bubbles, it is possible to observe situations in which the larger bubbles are moving upward while the smaller ones are moving downward. It follows that to ascribe a single velocity to the vapor phase is a distortion of reality.

In general, therefore, a two-phase flow may require a more elaborate treatment than the **six-velocity-component** one outlined above. However, there is much work to be done with the latter before further advances are seriously envisaged.

Some account of problems and methods of the numerical treatment of two-phase flows will be found in [11], [42], and [43].

B. Turbulent flow in empty spaces

In the foregoing account of numerical prediction procedures, the emphasis has been on heat exchanger shells that are extensively filled by tubes and baffles, the presence of which dominates the flow. However, there are often some regions that are free from such inclusions, and sometimes, as when the dominant mode of heat transfer is by radiation, the shell may be entirely empty.

Because of the large sizes of practical equipment, Reynolds numbers are high; the flow is therefore turbulent. The question therefore arises: Are the equations that have been presented in Sec. 1.2.6 and in the present section capable of describing and predicting turbulent flow in empty spaces?

The answer is that the equations, although still valid, must be augmented by further differential equations that will predict important features about the turbulence, and how its properties are distributed in space.

The topic touched upon here is that of turbulence modeling, introductory accounts to which can be found in [21] and [22]. A few remarks will be made here, so that the essence of the matter can be appreciated.

What the energy and momentum equations need to obtain from a turbulence model is a set of values of the effective thermal conductivity and the effective viscosity, for all relevant locations of the finite-difference grid. The concept implicit in most turbulence models is that these transport properties are functions primarily of the local fluctuation energy of the turbulence and of the local size of the eddies (more elaborate presumptions are sometimes also made).

What a turbulence model therefore often supplies is a pair of differential equations, to be solved simultaneously with the equations of energy and momentum, the dependent variables of which are either the fluctuation energy and the length scale themselves, or some quantities from which they can be derived.

The problem of turbulent flow in an empty space is therefore somewhat (but only somewhat) more complex than that of flow in a space filled with flow obstructions: the number of equations rises by two, because of the turbulence model, and these equations are strongly linked with those for velocity; but the equations for the temperatures of the tube-side fluid, and of the metal, have disappeared by way of compensation.

It is therefore quite easy to make predictions of turbulent flow in empty spaces, and many successful computations (in the sense of quantitatively in accord with reality) have been made and reported in the literature. This is not to say, however, that agreement

with experiment is always good, or that all problems concerned with the development of turbulence models have been solved. Particular uncertainty still attaches to those regions of flow in which buoyancy effects are influential, and to those in which the effective transport properties are not much greater than the laminar ones. Research can be expected to have to continue to this topic for many years before even the fairly modest (in this respect) needs of heat exchanger designers are fully met.

C. Flows involving chemical reaction

Although the emphasis in this handbook is on unfired heat exchanger equipment, it may be useful in the present context to examine briefly the special problems that arise when numerical prediction procedures are applied to flow and heat transfer phenomena in which an exothermic chemical reaction, for example, the combustion of a fuel, plays a significant role.

Usually the combustion takes place in empty spaces, and the flow is turbulent. Therefore the remarks of Sec. B are relevant. However, there are further complications: specifically, if the behavior of the flame is to be predicted (and the heat transfer rate can scarcely otherwise be computed), it is necessary to solve at least some partial differential equations having concentration as their dependent variables.

There is one situation that permits fairly satisfactory predictions to be based on the solution of a single additional equation; this may be that for the concentration of any chemically inert constituent, such as nitrogen; or it may be the equation for the mass fraction of a chemical element, regardless of the chemical species between which it is distributed; or yet again the equation for the mass fraction of material derived from the fuel system, irrespective of whether or not the fuel has reacted chemically.

This special situation is characterized as follows:

The fuel enters in a separate stream from the oxygen.

The chemical reactivity of the fuel is such that the fuel burns just as soon as it encounters oxygen (the so-called physically controlled or mixed-is-burned phenomenon).

The flow is fully turbulent.

It is a situation that is quite commonly approached in furnaces and other heating equipment, and it is easy to handle numerically because there is only one additional differential equation, and that is one, as it happens, without any complicated source terms.

Of the more general situation, in which the chemical reactions proceed too slowly for the mixing rate to be

limiting, it is appropriate to record the following major facts:

Most chemical reactions proceed by way of many intermediate stages, so that a complete description may involve quantitative concern with some tens of concentration variables.

Although knowledge of the rate constants of the reactions into which these species enter is never as complete or as precise as would be desirable, such progress has been made by chemical kineticists in recent years that shortage of rate constant information is seldom the major cause of inadequacy of predictions.

When the flow is laminar, numerical solution procedures are available for solving the flame prediction problem, albeit often at rather great expense (because there are many equations to solve, and because fine grids are needed for thin-reaction zone regions).

When the flow is turbulent, uniting knowledge of chemical kinetics with knowledge of turbulence presents special difficulties that have not yet yielded to entirely general and valid solution procedures.

Consequently, the prediction of kinetically influenced turbulent combustion phenomena must still be regarded as a matter for the research worker rather than for the practicing engineer. However, there are reasons for hope that this situation may soon change.

This brief account will close by drawing the reader's attention to a few sources of further enlightenment, namely, [23], [24], [25], and [26].

D. Flows involving radiation

Often a major mode of heat transfer from a flame is by way of radiation. It is therefore appropriate to touch upon the question of how radiative transfer can be incorporated into numerical solution procedures.

In principle, of course, the Hottel zone method [27] can be applied. In practice, this becomes enormously expensive if it is to take account of the strong nonuniformities of concentration and temperature that exist in practical combustion chambers, and that are brought to light by the prediction procedures of the kind under discussion here. In practice, therefore, it is seldom used in conjunction with these procedures.

What is used, as a rule, is some variant of the flux method, usually ascribed to Schuster [28] and Hamaker [29]. Examples of its use in furnace calculations are represented by [30], [31], and [32]. It involves the solution of additional finite-difference equations, rather simpler in form than those given earlier (because radiation is neither convected nor stored), but soluble by similar techniques.

Whereas the finite-difference equations for tempera-

ture, concentration, and velocity are in full conformity (when the grid is fine enough) with the basic differential equations, it is not true, unfortunately, that the finite-difference equations of the radiation flux method exhibit that conformity, no matter how fine (or coarse) the grid. The reason is that all versions of the flux method that have existed until now overidealize the angular distribution of radiation, treating it as being collected into bundles having direction normal to the main cell surfaces. This idealization, essential if the problem is to be numerically tractable, ignores an important feature of real radiation phenomena, namely, that radiation travels obliquely as well.

No solution to this difficulty has yet been found that is simple enough for practical use. Consequently, radiative transfer problems can still be only approximately solved.

It ought to be mentioned, however, that this mathematical difficulty is not the only hindrance to the reliable prediction of radiation from flames, for knowledge of the local radiative properties of the gases is required, and those are wavelength-dependent and often not known. As a last source of difficulty, it should be further remarked that often the most active radiating species in a flame is the soot that is suspended in it, and the kinetics of soot formation and oxidation constitutes one of the least well-documented chapters of chemical kinetics.

Radiation from flames must therefore also be regarded as the research worker's preserve at the present time.

E. Calculation of heat transfer coefficients

The last topic in this section is the role of numerical methods in predicting heat transfer coefficients, that is, as a replacement for the experimentally based correlations that are discussed in Sec. 1.2.3.

This application requires a radical change of scale in the object of study. Instead of being concerned with the heat exchanger as a whole, attention is confined, for example, to a short length of a single passage between three heat exchanger tubes in a bank. Then, of course, the problem becomes one of turbulent flow in an empty space, of the kind discussed in Sec. B; on the other hand, the space in question is now much smaller, and the characteristic Reynolds number will also be smaller.

This last feature is important for if now the solution of the finite-difference equations is to lead, as it well may, to predictions of Nusselt number-Reynolds number relationships, these equations must hold for regions close to the walls in which the turbulent contributions to the transport properties may not be much greater than the laminar contributions. This is just that domain in which, as has been stated above, turbulence models are least

well developed; therefore the extent to which heat transfer formulas can currently be obtained in this manner is severely limited.

Nevertheless, prospects for future progress along these lines are good. References [33] and [34] expatiate upon this theme and contain further relevant references.

NOMENCLATURE FOR SECTIONS 1.1.0-1.4.3

A	interface area across which heat transfer occurs, m^2	$\dot{m}_{vol}$	mass transfer rate across an interface per unit volume of the space within which the interface area is distributed, $kg/m^3 s$
a	coefficient in finite-difference equation, $kg/m^2 s$	NTU	number of transfer units
a	a constant	NVH	number of velocity heads (pressure drop)
B	dimensionless driving force for mass transfer	Nu	Nusselt number
b	mass flow rates across cell boundaries, $kg/m^2 s$	n_θ, n_r, n_z	number of subdivisions of the θ , r , and z dimensions
b	a constant	P	heat exchanger performance parameter
b'	corrections to b , $kg/m^2 s$	P	power needed to pump a fluid through a heat exchanger, W
b''	differential coefficient of b' with respect to p , $kg/N s$	Pe	Peclet number
b*	approximate value of b , $kg/m^2 s$	Pr	Prandtl number
$\dot{C}$	product of mass flow rate and constant-pressure specific heat capacity, W/K	p	pressure, N/m^2
c	upwind specific heat times density times velocity, $W/m^2 K$	p'	pressure correction, N/m^2
c, c_p	constant-pressure specific heat capacity, $J/kg K$	Q	quantity of heat, J
c_v	constant-volume specific heat capacity, $J/kg K$	$\dot{Q}$	rate of heat supply or exchange, W
c_f	friction factor	$\dot{q}$	heat transfer rate across an interface per unit interface area, W/m^2
D	diameter, m	q_{vol}	heat transfer rate across an interface per unit volume of the space within which the interface area is distributed, W/m^3
D	thermal conductivity divided by internode distance, $W/m^2 K$	R	ratio of mass flow $\times$ specific heat capacity products for a heat exchanger
E	effectiveness of a heat exchanger	R	porosity, i.e., fraction of volume accessible to a phase
E	error in continuity equation, kg/s	Ra	Rayleigh number
F	ratio of effective temperature difference to logarithmic mean temperature difference	Re	Reynolds number
Fo	Fourier number	r	radius, m
Fr	Froude number	S	cross-sectional area of a fluid stream, m^2
f	friction coefficient	S_p, S'_p	contributions to the source of a transported property at grid point P , various/s
f_θ, f_r, f_z	friction terms in momentum equations, N/m^3	SC	Schmidt number
$f_{\eta\theta}, f_{\eta r}, f_{\eta z}$	components of f_θ, f_r, f_z dependent on intraphase viscous action, N/m^3	Sh	Sherwood number
f'_θ, f'_r, f'_z	remaining components of f_θ, f_r, f_z , divided by the relevant velocity component, $N s/m^4$	St	Stanton number
$\dot{m}$	product of density and velocity, $kg/m^2 s$	T	temperature, $^\circ C$
Gr	Grashof number	t	time, s
GZ	Graetz number	U	heat transfer coefficient, $W/m^2 K$
g	conductance, $kg/m^2 s$	U_{vol}	volumetric heat transfer coefficient, $W/m^3 K$
g_n	gravitational acceleration, m/s^2	u	specific internal energy, J/kg
He	Henry number	u	velocity (component in θ direction), m/s
h	specific enthalpy, J/kg	V	volume, m^3
Ah	latent heat of vaporization, J/kg	v	velocity (component in r direction), m/s
L	length dimension, m	W	quantity of work, J
Le	Lewis number	$\dot{W}_s$	rate of performance of shaft work, W
M	mass of material, kg	We	Weber number
A	mass flow rate of material in a stream, kg/s	w	velocity (component in z direction), m/s
$m_n, \dot{m}_i$, etc.	mass transfer rate across an interface per unit interface area, $kg/m^2 s$	X	spacing of tubes in one direction, m
		x	mass fraction

x	quality	min, max	streams of minimum or maximum mass
$\dot{x}$	flow quality		flow rate times specific heat capacity, respectively
Y	spacing of tubes in a second direction, m	max	largest possible
z	distance in axial (or flow or vertical) direction, m	ref	reference
α	individual heat transfer coefficient, $W/m^2 K$	N, S, E, W, H, L	north, south, east, west, high, low neighbors of a grid point, respectively
β	individual mass transfer coefficient, m/s	n, s, e, w, h, l	control volume faces separating a grid point from its N, S, E, W, H, L neighbors, respectively
β_{vol}	volumetric mass transfer coefficient, s^{-1}	P	grid point under consideration
Γ	exchange coefficient, $kg/m s$	P^-	the same grid point at the previous time instant
ΔT_{LM}	logarithmic mean temperature difference, K	a	air
ΔT_M	mean temperature difference causing heat transfer, K	b	bulk
Δ, δ	increase in	i	component i
δ	diffusion coefficient, m^2/s	i	interface
η	dynamic viscosity, $kg/m s$	m	metal
η	parameter in expressions for F and P	s	saturation condition of gas phase adjacent to liquid
θ	ratio of effective to maximum possible temperature difference	t	transferred substance
θ	angular coordinate	w, H_2O	water
ξ	parameter in expression for F	x, Y	ends of a heat exchanger
Λ	effective thermal conductivity, $W/m K$	1, 2	streams in a heat exchanger
λ	thermal conductivity, $W/m K$	1, 2, 3	coordinate direction
ρ	density, kg/m^3	$\dots \rightarrow \dots$	from $\dots$ to $\dots$
σ	surface tension constant, N/m	loc	
σ	either Prandtl number or Schmidt number, as appropriate	arc	
τ	shear stress, N/m^2		
ϕ	a transported fluid property, various		

Subscripts

eff effective
in, out entry and exit points, respectively, of fluid streams

REFERENCES FOR SECTIONS 1.1.0-1.4.3

1. Spalding, D. B., **Convective Mass Transfer**, Edward Arnold, London, 1963.
2. Merkel, F., Verdunstungskühlung, **Forschungsarb. Wes.**, no. 275, 1925.
3. Reynolds, O., On the Extent and Action of the Heating for Steam Boilers, **Proc. Manchester Lit. Phil. Soc.**, vol. 8, 1874.
4. Chilton, T. H., and Colburn, A. P., Mass-Transfer (Absorption) Coefficients. Prediction from Data on Heat-Transfer and Fluid Friction, **Ind. Eng. Chem.**, vol. 26, pp. 1183-1187, 1934.
5. Prandtl, L., Eine Beziehung Zwischen Wärmeaustausch und Strömungswiderstand der Flüssigkeiten, **Phys. Z.**, vol. 11, p. 1072, 1910.
6. Taylor, G. I., Conditions at the Surface of a Hot Body Exposed to the Wind, **Br. Aero. Res. Commun.**, R&M no. 272, p. 423, 1916.
7. Jayatilaka, C. V. L., The Influence of Prandtl Number and Surface Roughness on the Resistance of the Laminar Sub-Layer to Momentum and Heat Transfer, **in Progress in Heat and Mass Transfer**, ed. U. Grigull and E. Hahne, vol. 1, Pergamon Press, London, 1969.
8. Elhadidy, M. A., Applications of a Low-Reynolds-Number Turbulence Model and Wall Functions for Steady and Unsteady Heat-Transfer Computations, Ph.D. thesis, London University, 1980.
9. Hartnett, J. P., and Rohsenow, W. (eds.), **Handbook of Heat Transfer**, McGraw-Hill, New York, 1972.
10. VDI, **Wärme-Atlas**, VDI Verlag, Düsseldorf, 1974.
11. Spalding, D. B., Numerical Computation of Multi-Phase Fluid Flow and Heat Transfer, **in Recent Advances in Numerical Mechanics**, ed. C. Taylor, Pineridge Press, London, 1980.
12. Eckert, E. R. G., and Drake, R. M., **Heat and Mass Transfer**, McGraw-Hill, New York, 1959.
13. Kays, W., and London, A. L., **Compact Heat Exchangers**, 2d ed., McGraw-Hill, New York, 1964.
14. Patankar, S. V., and Spalding, D. B., A Calculation Procedure for the Transient and Steady-State Behaviour of Shell-and-Tube Heat Exchangers, in **Heat Exchangers Design and Theory Sourcebook**, eds. N. M. Afgan and E. U. Schlünder, Hemisphere, Washington, D.C., 1976.
15. Patankar, S. V., and Spalding, D. B., Computer Analysis of the Three-Dimensional Flow and Heat Transfer in a Steam Generator, **Forschung Ing.-Wes.**, vol. 44, pp. 47-52, 1978.

16. Spalding, D. B., The Calculation of Convective Heat Transfer **in** Complex Flow Systems, *Proc. 5th Int. Heat Transfer Conf.*, pub. by the Science Council of Japan, vol. VI, pp. 44-60, Sept. 74.
17. **Caretto**, L. S., Gosman, A. D., Patankar, S. V., and Spalding, D. B., Two Calculation Procedures for Steady and Three-Dimensional Flows with **Re-Circulation**, *Proc. 3d Int. Conf. on Numerical Methods in Fluid Mechanics*, Springer-Verlag, Berlin, pp. 60-68, 1973.
18. Patankar, S. V., and Spalding, D. B., A Calculation Procedure for Heat, Mass and Momentum Transfer in **Three-Dimensional** Parabolic Flows, *Int. J. Heat Mass Transfer*, vol. 15, pp. 1787-1806, 1972.
19. **Singhal**, A. K., and Spalding, D. B., Numerical Modelling of Two-Phase Flow in Steam Generators, *2d Multi-Phase Flow and Heat Transfer Symp., Miami Beach*, April 1979.
20. Spalding, D. B., Numerical **Modelling** of Fluid-Mechanics, Heat-Transfer and Mass-Transfer Processes; A Lecture Course, Imperial College, **Mech. Eng. Dept.**, Heat Transfer Section Report ref **HTS/80/1**, 1980.
21. Launder, B. E., and Spalding, D. B., Mathematical Models of Turbulence, Academic Press, London, New York, 1972.
22. Spalding, D. B., Turbulence Models; A Lecture Course, Imperial College, **Mech. Eng. Dept.**, Heat Transfer Section Report ref **HTS/80/2**, 1980.
23. Afgan, N., and Beer, J. M. (eds.), **Heat Transfer in Flames**, Hemisphere, Washington, D.C., 1976.
24. Kennedy, L. A. (ed.), **Turbulent Combustion**, AIAA, New York, 1978.
25. Spalding, D. B., Combustion Theory Applied to Engineering, Imperial College, London, **Mech. Eng. Dept.**, Heat Transfer Section Report ref **HTS/77/1**, 1977.
26. Spalding, D. B., A General Theory of Turbulent Combustion, *J. Energy*, vol. 2, no. 1, pp. 16-23, 1978.
27. Hottel, H. C., and Sarofim, A. F., **Radiative Transfer**, McGraw-Hill, New York, 1967.
28. Schuster, A., *Astrophys. J.*, vol. 21, pp. 1-22, 1905.
29. **Hamaker**, H. C., Radiation and Heat Conduction in Light-Scattering Material, Philips Res. Rep. 2, pp. 55-67, 103, 112, 420, 1947.
30. Patankar, S. V., and Spalding, D. B., A Computer Model for Three-Dimensional Flow in Furnaces, *Proc. 14th Symp. Znt. Comb.*, pub by Combustion Institute, pp. 605-614, 1973.
31. Patankar, S. V., and Spalding, D. B., Simultaneous Predictions of Flow Pattern and Radiation for Three-Dimensional Flames, in **Heat Transfer in Flames**, ed. N. M. Afgan and J. M. Beer, pp. 73-94, Hemisphere, Washington, D.C., 1974.
32. **Pai**, B. R., **Michelfelder**, S., and Spalding, D. B., Prediction of Furnace Heat Transfer with a Three-Dimensional Mathematical Model, *Int. J. Heat Mass Transfer*, vol. 21, pp. 571-580, 1978.
33. Spalding, D. B., Convective Heat Transfer, *Fortschr. Verfahrenstech.*, vol. 15, pp. 55-66, 1978.
34. Spalding, D. B., Turbulence Models for Heat Transfer, 6th Int. Heat Transfer **Conf. Proc.**, vol. 6, paper KS4, pp. 33-43, Hemisphere, Washington, D.C., 1978.
35. Stevens, P. A., Fernandez, J., and Woolf, J. R., Mean Temperature Difference in One, Two-, and **Three-Pass** Cross Flow Heat Exchangers. *J. Heat Transfer*, vol. 79, pp. 287-297, 1957.
36. Roetzel, W., and Nicole, F. J. L., Mean Temperature Difference for Heat-Exchanger Design-A General Approximate Explicit Equation. *J. Heat Transfer*, vol. 97, pp. 5-8, 1975.
37. Gardner, K., and Taborek, J., Mean Temperature Difference-A **Reappraisal**. *AIChE J.*, vol. 23, no. 6, pp. 1-9, 1977.
38. Gaddis, E. S., and Schlinder, E. U., Exchange Temperature Distribution and Heat in Multi-Pass Shell-and-Tube Exchangers with Baffles, *Heat Transfer Eng.*, vol. 1, no. 1, pp. 43-52, 1979.
39. Nusselt, W., Der **Wärmeübergang** im Kreuzstrom, *Z. Ver. Dt. Zng.*, vol. 55, pp. 2021-2024, 1911.
40. Nusselt, W., Eine Neue **Formel** für den **Wärmedurchgang** im Kreuzstrom, *Tech. Mech. Thermodynam.*, vol. 1, pp. 417-422, 1930.
41. Schlichting, H., **Boundary-Layer Theory**, 4th ed., McGraw-Hill, New York, 1960.
42. Spalding, D. B., Mathematical Methods in Nuclear-Reactor Society Meeting on Nuclear-Reactor Thermal Hydraulics, Saratoga, N.Y., Oct. 6, 1980.
43. Spalding, D. B., Numerical Computation of Multi-Phase Flow; A Lecture Course, Heat Transfer Section, **Mech. Eng. Dept.**, Imperial College, Rept. 1980.

1.5

Charts for mean temperature difference in industrial heat exchanger configurations

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1.5.1

Introduction

J. Taborek

A. Background

This section presents charts (together with their associated equations) for use in the performance evaluation or design of specific industrial heat exchanger configurations. The methods selected for these working charts are

1. F-correction-factor method (see Secs. 1.2.4 and 1.3.1)
2. θ -NTU method (see Secs. 1.2.4 and 1.3.1)

Since the two methods are derived from the same basic equations, they must give the same answer. However, one or the other of the two methods may be advantageous for specific calculations. In particular:

1. The F-correction-factor method has the unique advantage that it directly represents the departure of the effective temperature driving force AT , for any flow system from the best possible value for counter flow ΔT_{LM} . The method is particularly suitable for process heat exchanger calculations where usually the temperature profiles and heat balances are specified from the process conditions but the heat transfer rate U and often the heat exchanger surface A are not known. As a matter of fact, the overall heat transfer rate U will usually be a function of the design configuration considered. The problem to be solved is then to select an exchanger design type that would operate with the highest possible effective temperature driving force or, for that matter, eliminate quickly such design types as may be not operable at all.

2. The θ -NTU method is particularly suitable for preliminary evaluation of systems where the coefficient

U can be considered known and constant (as in compact heat exchangers) and/or the surface A can be controlled by addition of incremental moduli without affecting the rate U , such as plate exchangers and compact plate-fin units. In many of these cases, the overall plant economics are strongly dependent on the optimum utilization of the available AT versus capital investment in heat transfer surfaces. The heat transfer area and usually one of the terminal temperatures are not fixed a priori but are chosen as a result of the economic optimization or design process.

B. Conditions for validity

In Sec. 1.2.4, three of the primary assumptions for validity of the F-correction-factor and θ methods are stated, namely, that the overall heat transfer coefficient U , the flow rate, and specific heat of the two streams are constant through the heat exchanger. The assumptions of constant U is never exact but is often accepted for practical reasons. In addition, a number of other assumptions are implicit to the definition of the logarithmic mean temperature difference, and thus the F and θ charts. These include, in approximate order of importance:

1. For pure counter flow or **cocurrent** flow, the temperature of either fluid is uniform over any cross section of its path. In other words, the thermal history of any particle in either stream is identical to any other particle in that stream, that is, complete mixing, no stratification or bypassing.

2. For baffled shell-and-tube exchangers, the heat

transferred in each baffle compartment is small compared to the overall; that is, the number of baffles is large (usually ≥ 5).

3. There is no change of phase of either fluid in only a part of the exchanger. If condensation or boiling occurs, it must do so uniformly over the entire surface and in such a way that equal quantities of heat are exchanged for equal changes of the fluid temperature. This produces a linear plot of heat exchanged versus temperature. Isothermal boiling or condensation complies with these conditions.

4. There is equal heat transfer in each tube or shell pass.

5. Heat losses to surroundings are negligible.

C. Definitions of quantities

An arbitrary heat exchanger is illustrated in Fig. 1. An amount of heat Q is transferred between streams 1 and 2. The F -correction-factor and θ methods use the following definitions (see Sec. 1.2.4 for details). First, the general defining equation is

$$Q = UA AT_{m} \quad (1)$$

where U is the overall heat transfer coefficient, A is the exchanger surface area, and AT_{m} is a suitably defined effective mean temperature difference. For a pure counter current flow, with the assumptions given in Sec. B, AT_{m} becomes ΔT_{LM} , the logarithmic mean temperature difference, defined as

$$\Delta T_{LM} = \frac{[(T_1)_i - (T_2)_o] - [(T_1)_o - (T_2)_i]}{\ln \{[(T_1)_i - (T_2)_o] / [(T_1)_o - (T_2)_i]\}} \quad (2)$$

Equally, Q can be defined by the heat balances of streams 1 and 2,

$$Q = C_1 [(T_1)_i - (T_1)_o] = C_2 [(T_2)_o - (T_2)_i] \quad (3)$$

where C_1 and C_2 are the heat capacities of the two streams ($\dot{M}C_p$).

For configurations other than counter flow, a correction factor F is defined as

$$F = \frac{\Delta T_m}{\Delta T_{LM}} \quad (4)$$

The relationships for F are conveniently formulated in terms of the parameter R (ratio of stream heat capacity or stream temperature differences) and P (thermal effectiveness).

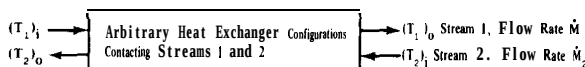


Figure 1 Flow streams to arbitrary heat exchanger.

$$R = \frac{C_2}{C_1} = \frac{(T_1)_i - (T_1)_o}{(T_2)_o - (T_2)_i} \quad (5)$$

$$P = \frac{(T_2)_o - (T_2)_i}{(T_1)_i - (T_2)_i} \quad (6)$$

The thermal effectiveness P is the ratio of the change of temperature of stream 2 and the maximum possible difference between the inlet temperatures of the two streams. The term ΔT_{LM} can also be expressed in terms of the R and P parameters as follows:

$$\Delta T_{LM} = \frac{(T_1)_i - (T_1)_o}{R} \left\{ \frac{R - 1}{\ln [(1 - P)/(1 - PR)]} \right\}_{R \neq 1} \quad (7)$$

The term in braces will be frequently repeated in the derivations and it is therefore assigned a symbol δ :

$$\delta = \frac{R - 1}{\ln [(1 - P)/(1 - PR)]} \quad R \neq 1 = \frac{1 - P}{P} \quad R \rightarrow 1 \quad (8)$$

The number of transfer units for stream 2 (NTU_2) is defined as

$$NTU_2 = \frac{AU}{C_2} = \frac{(T_2)_o - (T_2)_i}{\Delta T_m} \quad (9)$$

The NTU value itself has only a vague physical significance as roughly a heat exchanger size factor, being proportional to the product of U and A . The original reason for introducing the NTU method was that the unknown heat balance case, where both outlet temperatures are unknown, can be solved without reiterative calculations—a chore impossible with the F method. Further interpretation of the NTU methods is given in detail in Sec. 1.2.4. Of the various methods representing NTU , the θ method was selected because it is graphically compatible with the F -correction-factor method.

The parameter θ is defined as

$$\theta = \frac{\Delta T_m}{(T_1)_i - (T_2)_i} \quad (10)$$

The charts are such that knowing two of the three parameters, P , R , and NTU_2 , θ , and thus AT , can be directly determined. Only the θ method permits this convenience. Note that the above variables are inter-related as follows:

$$NTU_2 = \frac{1}{F\delta} \quad (11)$$

$$\theta = \frac{P}{FP\delta} = \frac{P}{NTU_2} \quad (12)$$

D. AT_{m} , $-\theta$ - NTU_2 - F charts and how to use them

Based on the definitions in Sec. C, a series of graphic charts have been calculated, based on equations docu-

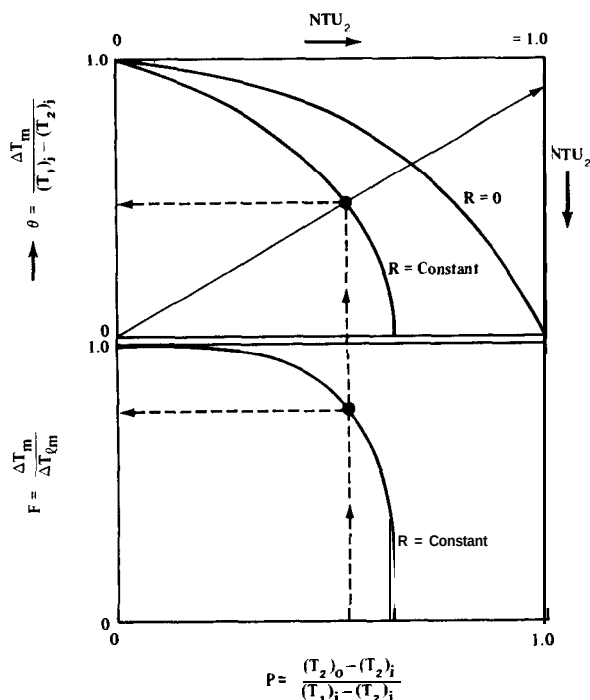


Figure 2 Schematic relation of the F - P and θ - P - NTU_2 charts.

mented in subsequent sections. The charts are arranged in a dual configuration, as schematically represented in Fig. 2:

1. θ versus P versus NTU_2 , with $R = \text{const}$ as parameter
2. F versus P , with $R = \text{const}$ as parameter

Notice that the interrelation of all the parameters can be easily established by visual inspection.

Graphs are presented as figures in Sec. 1.5.2 for the most common shell-and-tube exchangers and in Sec. 1.5.3 for a number of typical cross-flow configurations. In the following examples the use of the charts is demonstrated.

(a) Case 1: Calculation of effective mean temperature difference ΔT , knowing inlet and outlet temperature of both streams

This case can be solved using the F -factor charts as follows:

1. Calculate R and P from Eqs. (5) and (6).
2. For the particular configuration, read the F value corresponding to the calculated R and P from the F chart (or calculate F using the equations given).
3. Calculate ΔT_{LM} from Eq. (2).
4. Calculate $\Delta T = F \Delta T_{LM}$ [Eq. (4)].

Alternatively, the θ charts may be used:

1. Calculate R and P from Eqs. (5) and (6).
2. Read θ from upper charts for particular configuration (or calculate θ from equations given).
3. Calculate ΔT , from Eq. (10).

Note that in the latter case, NTU_2 can be read directly from the charts.

(b) Case 2: Calculation of outlet temperature of both streams where C_1 , C_2 , U , A , and the stream inlet temperatures are known

This case can be dealt with via the θ charts as follows:

1. Calculate R from Eq. (5) and NTU_2 from Eq. (9).
2. From the θ chart, read off the value of P corresponding to the calculated value of R and NTU_2 . (Alternatively, P can be calculated from the equations presented.)
3. By rearranging Eq. (6), determine $(T_2)_o$ from P and the inlet temperature values.

$$(T_2)_o = P[(T_1)_i - (T_2)_i] + (T_2)_i \quad (13)$$

4. Calculate $(T_1)_o$ from Eq. (5):

$$(T_1)_o = (T_1)_i - R[(T_2)_o - (T_2)_i] \quad (14)$$

The F -correction-factor method would require a reiterative procedure for the same calculation.

(c) Case 3: Calculation of required area A where inlet and outlet temperatures of both streams are known and the overall coefficient U is also assumed to be known

Here, the θ charts may be employed as follows:

1. Calculate R and P from Eqs. (5) and (6).
2. Read the value of NTU_2 corresponding to the calculated R and P from the θ chart.
3. Calculate A from

$$A = \frac{NTU_2 C_2}{U} \quad (15)$$

The same calculation can be achieved using the F -correction-factor charts, but this involves the additional calculation of ΔT_{LM} . Furthermore, the values of NTU_2 obtained via the θ method can be useful for comparative purposes.

(d) Case 4: Relationship between NTU_2 , F and F_{min} using visual inspection of charts

The relationship between NTU_2 and F can be read from the dual graphs. This is useful since good practice requires operation above a certain minimum F factor

($F_{\min}$). If the system is operated at too low a value of F , it falls into the region where large decreases in F occur for small differences in P . Thus, the performance is very sensitive to deviation from the given temperatures.

There is no straightforward criterion for definition of the value of $F_{\min}$. In principle it is a judgment of how much the exchanger designer is willing to sacrifice on loss of AT_{LM} for potential gain in other effects or cost of the unit. Two typical examples can be shown to illustrate this problem.

1. If outlet temperatures form a cross, $(T_1)_o > (T_2)_o$, a low value of F factor may result. A simple remedy is to use two (or more) exchangers in series, which will result in higher F values. This, however, causes a considerable cost increase, and the designer may decide to take the F-factor penalty and larger-size single unit.

2. A single tube-pass TEMA E shell-and-tube exchanger is considered to be in counter flow and therefore to have no limitation on temperature cross

with $F = 1$. However, tube-side flow velocity for such a tube-pass arrangement may be inherently low, resulting in low tube-side α and increased fouling. Switching to two (or more) tube-passes will improve these problems at a cost of having an F-factor penalty to AT_{LM} . Again, an engineering judgment will be necessary to arrive at the best possible solution, which in most cases is not quite obvious.

The suggestion has been made to limit $F_{\min}$ so that no outlet temperature cross will result. This criterion provides reasonable values of $F_{\min} \approx 0.8$ for TEMA E and J shells, but is too restrictive for units in series and multitube-pass cross flow ($F_{\min} \approx 0.95$), where larger F penalties can be tolerated and economically justified.

In summary, the best generalized recommendation that can be given is to use $F_{\min} \approx 0.8$ but to assure that this value on the F curves does not fall into the steeply decreasing region characteristic of low and high values of R . Otherwise, sound engineering judgment is the ultimate criterion.

Nomenclature for Section 1.5. I appears at the end of Section 1.5.3.

1.5.2

F and θ charts for shell-and-tube exchangers

J. Taborek

■ Charts for the following types of common shell-and-tube exchangers are presented in the following figures. Descriptive material, comments on proper usage, and documentation of equations from which the graphs have been constructed are included with each specific configuration.

- | | |
|---|---------|
| 1. Counter flow | Fig. 1 |
| 2. Cocurrent flow | Fig. 2 |
| 3. TEMA E shell, 1-2N | Fig. 3 |
| 4. TEMA E shell, 2 in series | Fig. 4 |
| 5. TEMA E shell, 3 in series | Fig. 5 |
| 6. TEMA E shell, 4 in series | Fig. 6 |
| 7. TEMA E shell, 5 in series | Fig. 7 |
| 8. TEMA E shell, 6 in series | Fig. 8 |
| 9. TEMA J shell, one tube pass | Fig. 9 |
| 10. TEMA J shell, 2N tube passes | Fig. 10 |
| 11. TEMA G shell, 2N tube passes | Fig. 11 |
| 12. TEMA F shell, partition thermal leakage | Fig. 12 |

A. Counter flow

Counter flow is included here only as being the best possible temperature-driving force utilization. For this case, by definition, AT_{LM} = ΔT_{LM} and $F = 1.0$. The θ method representation is given in Fig. 1.

$$\theta = P\delta = \frac{P}{NTU_2} \quad (1)$$

Credit and thanks is given to Mr. John Ward of AERE, Harwell, U.K., for computer calculation of the graphs presented in this section and Sec. 1.5.3.

The intercept of the $R = \text{const}$ curves at $\theta = 0$ occurs at

$$P_{\text{max}} = \frac{1}{R} \Big|_{R \geq 1.0} = 1.0 \Big|_{R \leq 1.0} \quad (2)$$

The line of $R = 0$ is the same for all the flow systems, as it corresponds to fluid 2 being isothermal. In such a case,

$$\theta = \frac{-P}{\ln(1-P)} \Big|_{R=0} \quad (3)$$

B. Cocurrent flow

Cocurrent flow AT_{LM} can be calculated directly from the ΔT_{LM} , but is presented here in the F-factor form, as it is the least effective flow configuration. The principal limitation is, of course, that the cold fluid outlet temperature must be below the hot fluid exit temperature—an unacceptable limitation in many cases, particularly where heat recovery is involved.

All other flow systems will be somewhere between the pure counter current and cocurrent flow. However, before dismissing cocurrent flow as undesirable, let us make the following observations.

1. Cocurrent flow produces the most uniform tube wall temperature between hot end inlet and cold end outlet, thus minimizing the thermal stress.

2. The highest wall temperature is lower than for counter flow, thus eliminating, in appropriate cases, problems associated with high wall temperature, such as fouling, material selection, etc.

3. In the case of flow reboilers, cocurrent flow

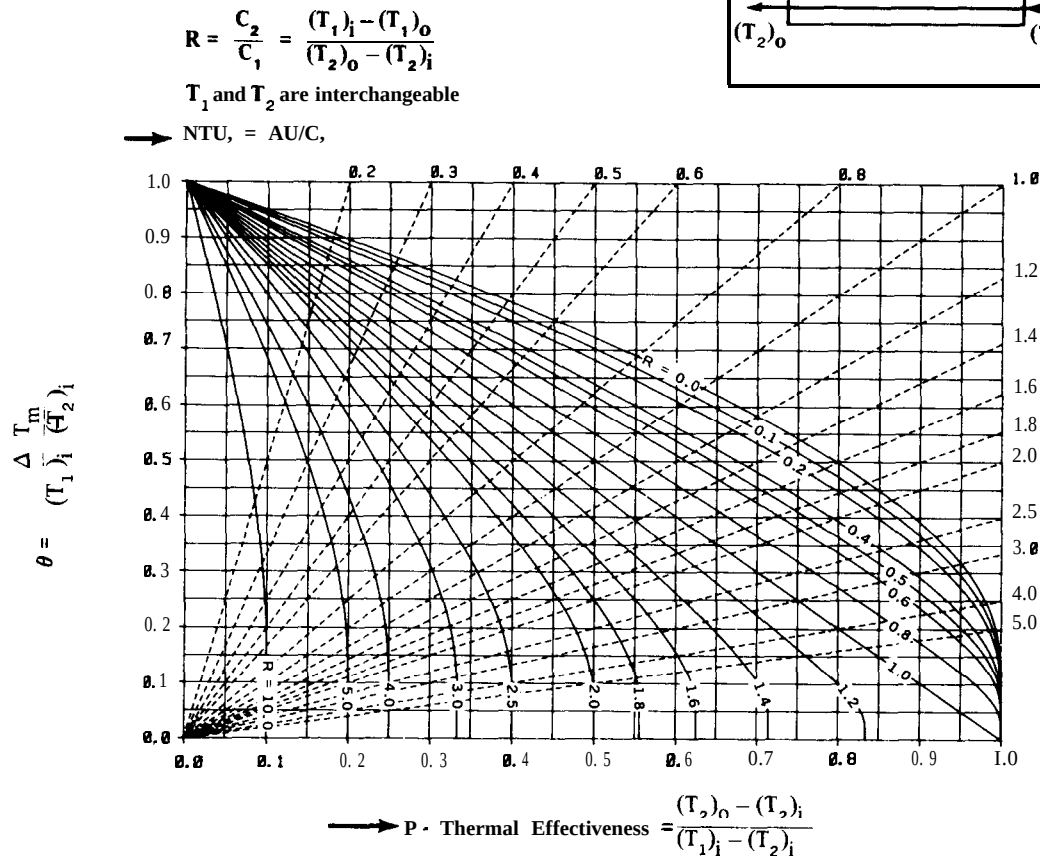


Figure 1 Mean temperature difference relationships: counter flow.

produces the highest ΔT at the entrance of streams, thus potentially contributing (in appropriate cases) to early initiation of nucleate boiling, which would not occur with counter flow.

The following equations are readily deduced from the basic definitions.

$$F = \frac{-(R + 1)}{\delta \ln [1 - P(R + 1)]} \quad (4)$$

The $P_{\max}$ at $F = 0$ is defined as

$$P_{\max} = \frac{1}{R + 1} \quad (5)$$

The results are shown in Fig. 2.

C. TEMA E shell-and-tube flow arrangements

This section will cover the most common flow arrangements in shell-and-tube exchangers, that is, a variety of shell flow geometries and tube pass arrangements.

(a) TEMA E shell with even number of tube passes (1-2N)

In commonly accepted nomenclature, the single shell pass (1) and the even number of tube passes (2N) are designated as a 1-2N configuration. This practice will be used in the following sections.

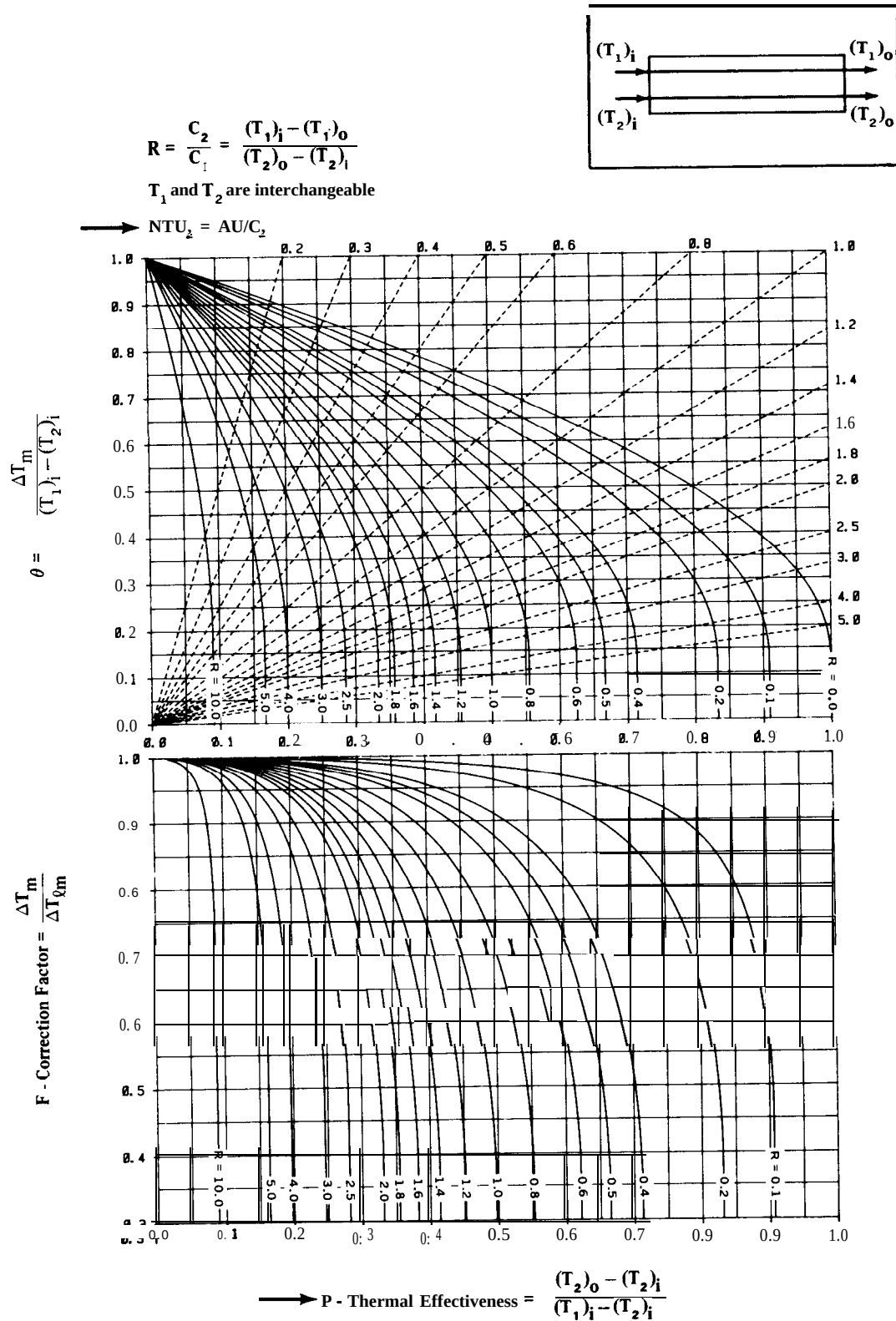
Under the assumption that the shell fluid is "mixed," that is, not stratified along the flow path, Underwood [1] performed the original integration for a 1-2 flow system, which after some rearrangements appears as

$$F = \frac{\eta}{(\delta) \ln \{ [2 - P(1 + R - \eta)] / [2 - P(1 + R + \eta)] \}} \quad (6)$$

where

$$\eta = \sqrt{R^2 + 1} \quad (6a)$$

Underwood also proved that it is immaterial whether tube-side flow direction in the first pass in counter flow

Figure 2 Mean temperature difference relationships: **cocurrent** flow.

(shown in Fig. 3) or **cocurrent** flow with respect to shell flow. Subsequently [2], four tube-pass arrangements were analyzed, but the differences were found to be very small compared to the validity of basic assumptions. Therefore, Eq. [6] is considered valid for any even number of tube passes. For an infinite number of tube passes, the correction factor F charts will transit to that of cross flow with both fluids mixed (see Sec. 1.5.3). Even for this extreme case, the F values above 0.7 are within 2% of the $1-2N$ solutions.

The asymptotic value of P is reached at

$$P_{\max} = \frac{2}{1 + R + \eta} \quad (7)$$

(b) Multiple E shells in series, each with two or any even number of tube passes

If the temperature range is very large, for example, $(T_2)_o \gg (T_1)_o$, a single unit with $2N$ tube passes would not be acceptable ($P > P_{\max}$) (see Figs. 4-8). In such cases (if counterflow is excluded for other reasons), resort is taken to employing several units where both the shell-side and tube-side flow are in series. Using M identical units of the TEMA E shell type in series, the F factor (or NTU_2) can be calculated from Eq. (6) if, according to Bowman [2], the P parameter is expressed as follows:

$$P = \frac{1 - (X)^{1/M}}{R - (X)^{1/M}} \bigg|_{R \neq 1} = \frac{P_o}{M - P_o(M - 1)} \bigg|_{R=1} \quad (8)$$

$$X = \frac{P_o R - 1}{P_o - 1} \quad (9)$$

and where R and P_o are based on the overall terminal temperatures of the system.

The asymptotic $P_{\max}$ values can be determined from

$$P_{\max} = \frac{Z^M - 1}{Z^M - R} \bigg|_{R \neq 1} = \frac{2M}{2M + \sqrt{2}} \bigg|_{R=1} \quad (10)$$

where

$$Z = \frac{\eta - R + 1}{\eta + R - 1}$$

and η is defined in Eq. (6a).

Although it has never been proven mathematically, according to Gardner [3], the same procedure will apply to any series of identical exchangers, with identical inlet-outlet configurations of each unit.

D. TEMA J shell divided-flow exchanger

The divided-flow J shell is used mainly to accommodate very low pressure drops. As only one-half the stream

traverses one-half the length, the shell-side pressure drop is only about one-eighth that of a TEMA E shell. A schematic sketch of the flow is shown in Fig. 9. Notice that a single nozzle inlet is shown (usual arrangement) and that the shell outlet temperatures from the two outlet nozzles will not be the same. The temperature profile refers to the mixed outlet temperature. Consequently, if two J shells in series should be calculated by Eq. (8), the shells will have to be connected so that the second shell again has a single nozzle entry-an inconvenience in piping arrangement. If the second shell is connected with two inlet nozzles to the two outlet nozzles of the first shell, the AT , relations do not strictly apply and may be on the unsafe side. The AT , analysis for J shells is well summarized by Jaw [4].

(a) J shell with single tube pass

The solution for this case (see Fig. 9) is identical to that for the TEMA E shell ($1-2N$) with the fluid unmixed and is due to Gardner [5].

$$P = 1 - \left(\frac{2R - 1}{2R + 1} \right) \frac{2R + \phi^{-(R+0.5)}}{2R - \phi^{-(R+0.5)}} \bigg|_{R \neq 0.5} \quad (11a)$$

$$= 1 - \frac{1 + 1/\phi}{2 + \ln \phi} \bigg|_{R=0.5} \quad (11b)$$

where ϕ is defined as

$$\phi = \exp \frac{1}{0F\delta} = \exp (NTU_2) \quad (12a)$$

$$\ln \phi = \frac{1}{F\delta} = NTU_2 = \frac{P}{\theta} \quad (12b)$$

The $P_{\max}$ values can be calculated as

$$P_{\max} = 1 - \frac{2R - 1}{2R + 1} \bigg|_{R > 0.5} = 1.0 \big|_{R \leq 0.5} \quad (13)$$

In definition of R and P , T_1 represents the shell-side fluid and T_1 and T_2 cannot be interchanged.

(b) J shell with two or any even number of tube passes

Jaw [4] analyzed cases with two and four tube passes and found that the results are within very small differences. Extending the number of tube passes to infinity, the case becomes identical to that of mixed-mixed cross flow, as derived by Gardner [6]. Again, the differences were found to be negligible compared to the two-tube-pass solution in the region of interest, that is, $F \geq 0.5$. Therefore, Gardner's solution is recommended for any even number of tube passes.

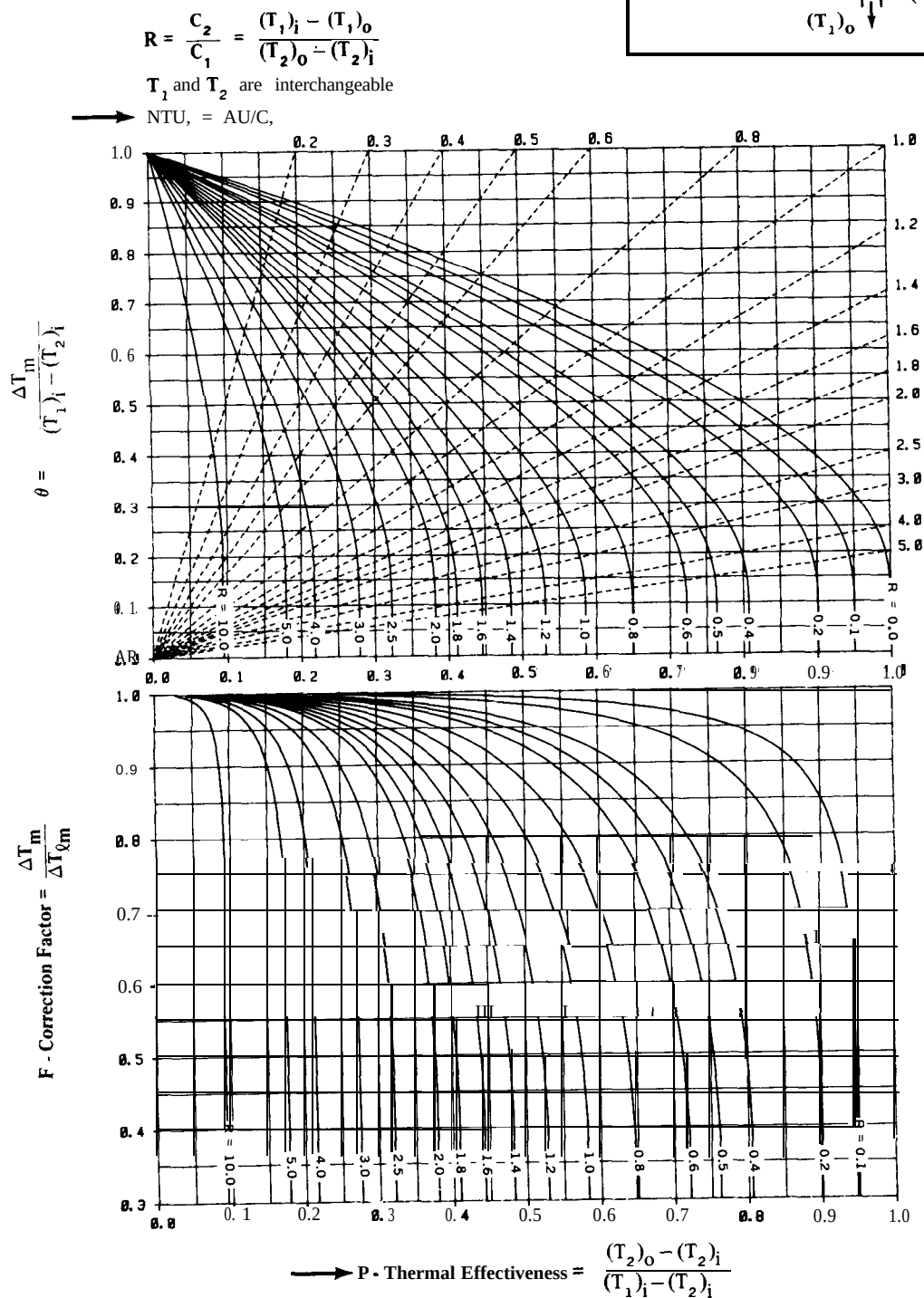


Figure 3 Mean temperature difference relationships: one E shell with any even number of tube passes (shell fluid mixed).

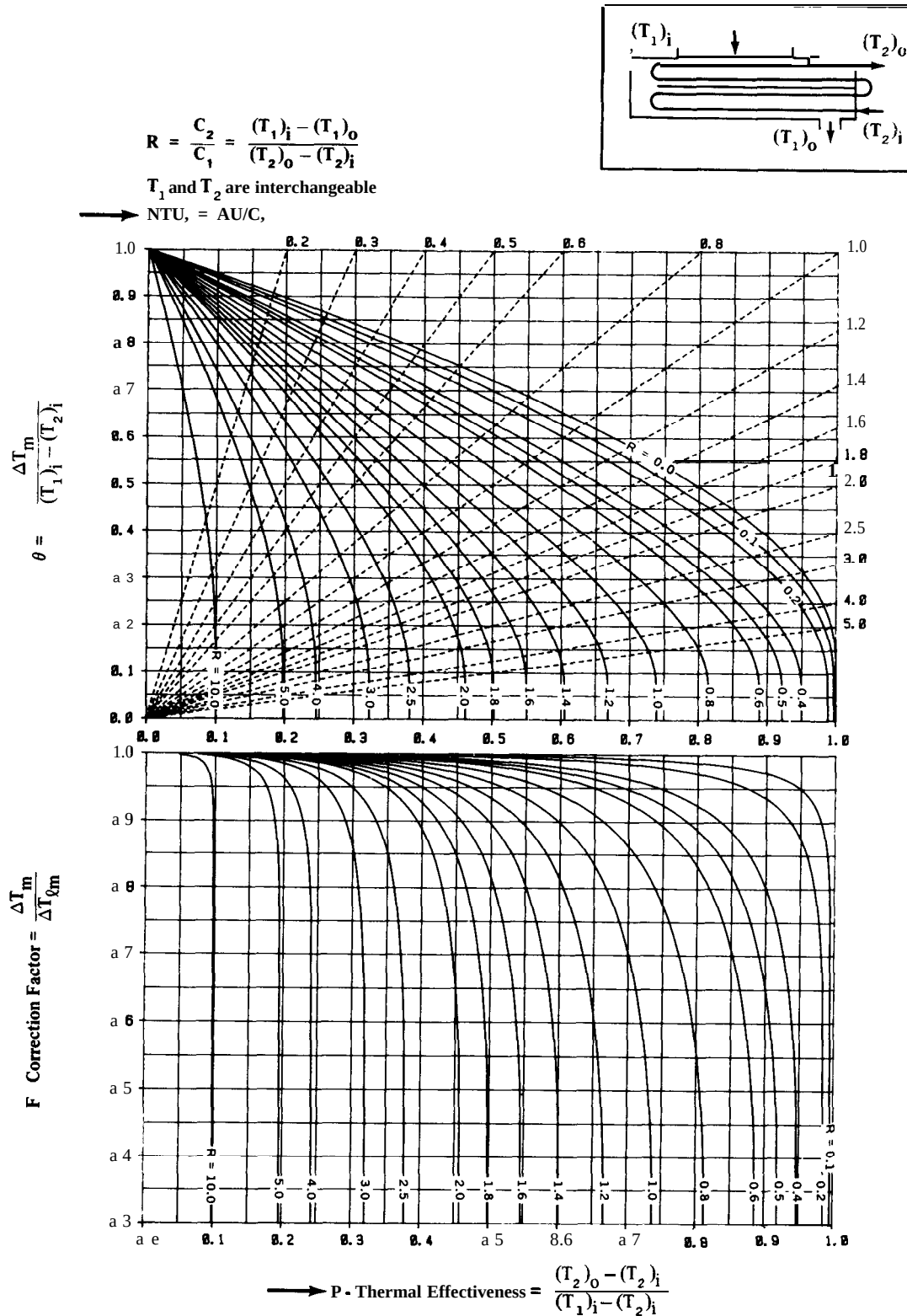


Figure 4 Mean temperature difference relationships: two E shells in series, each with two or any even number of tube passes.

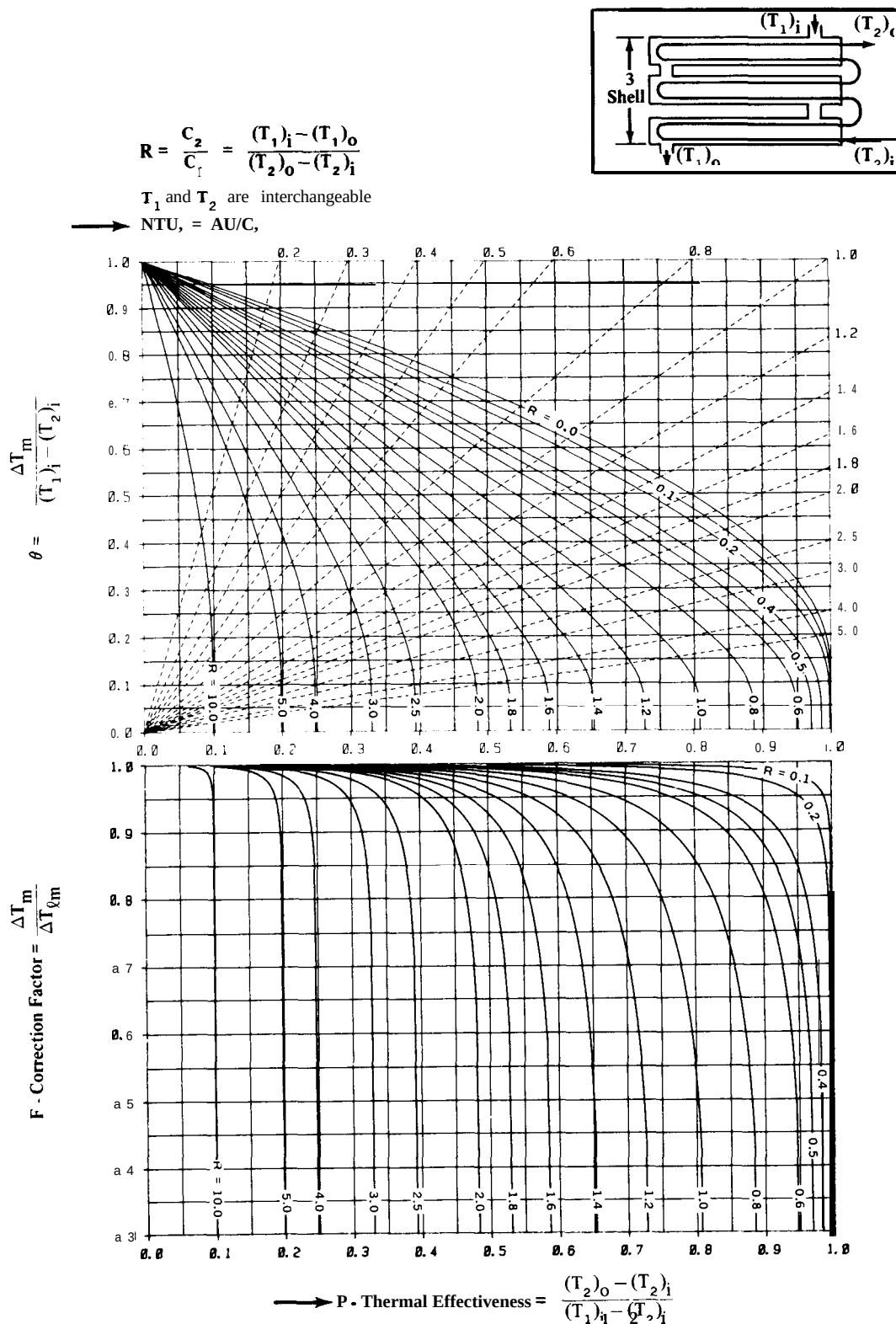


Figure 5 Mean temperature difference relationships: three E shells in series, each with two or any even number of tube passes.

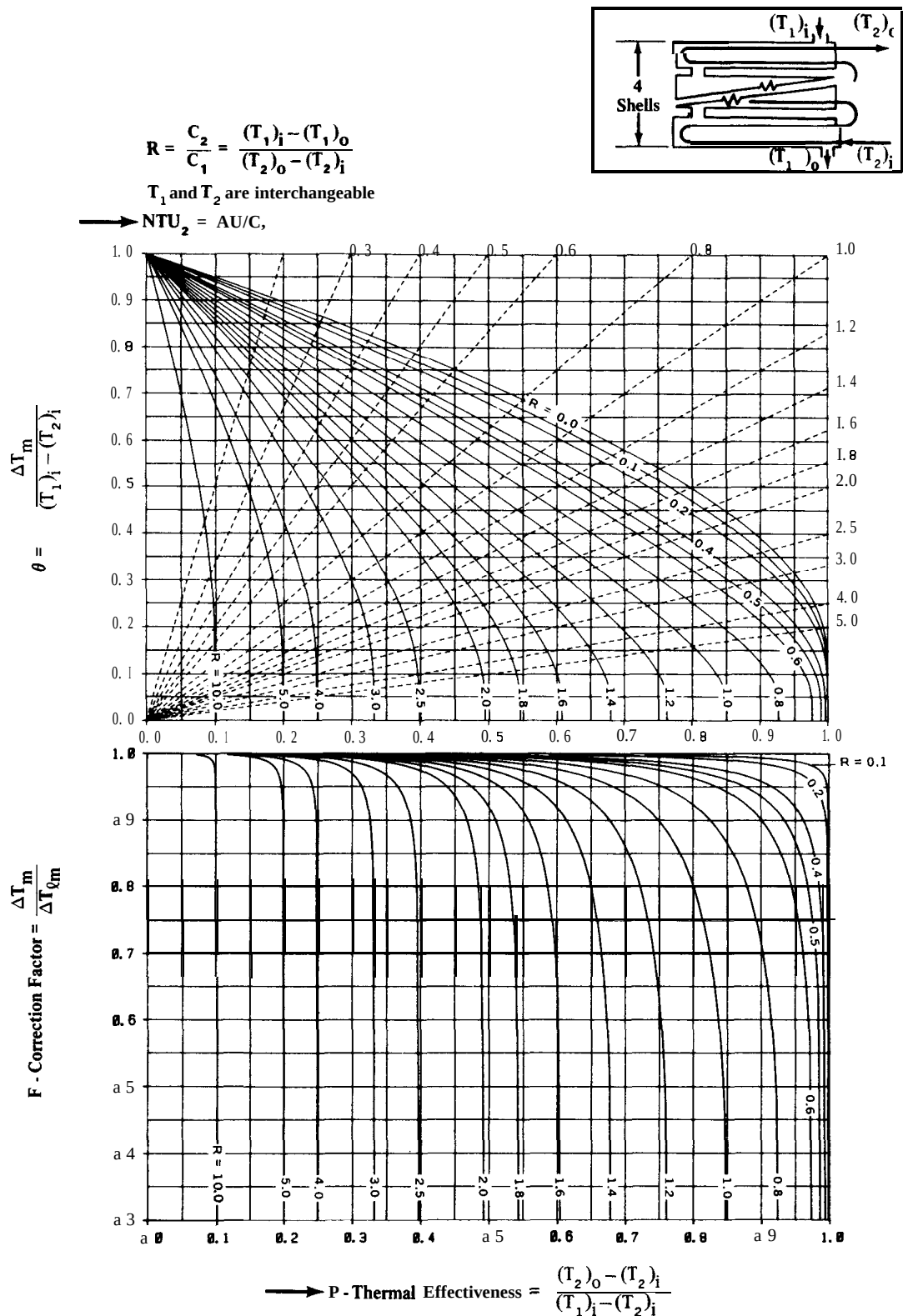


Figure 6 Mean temperature difference relationships: four E shells in series, each with two or any even number of tube passes.

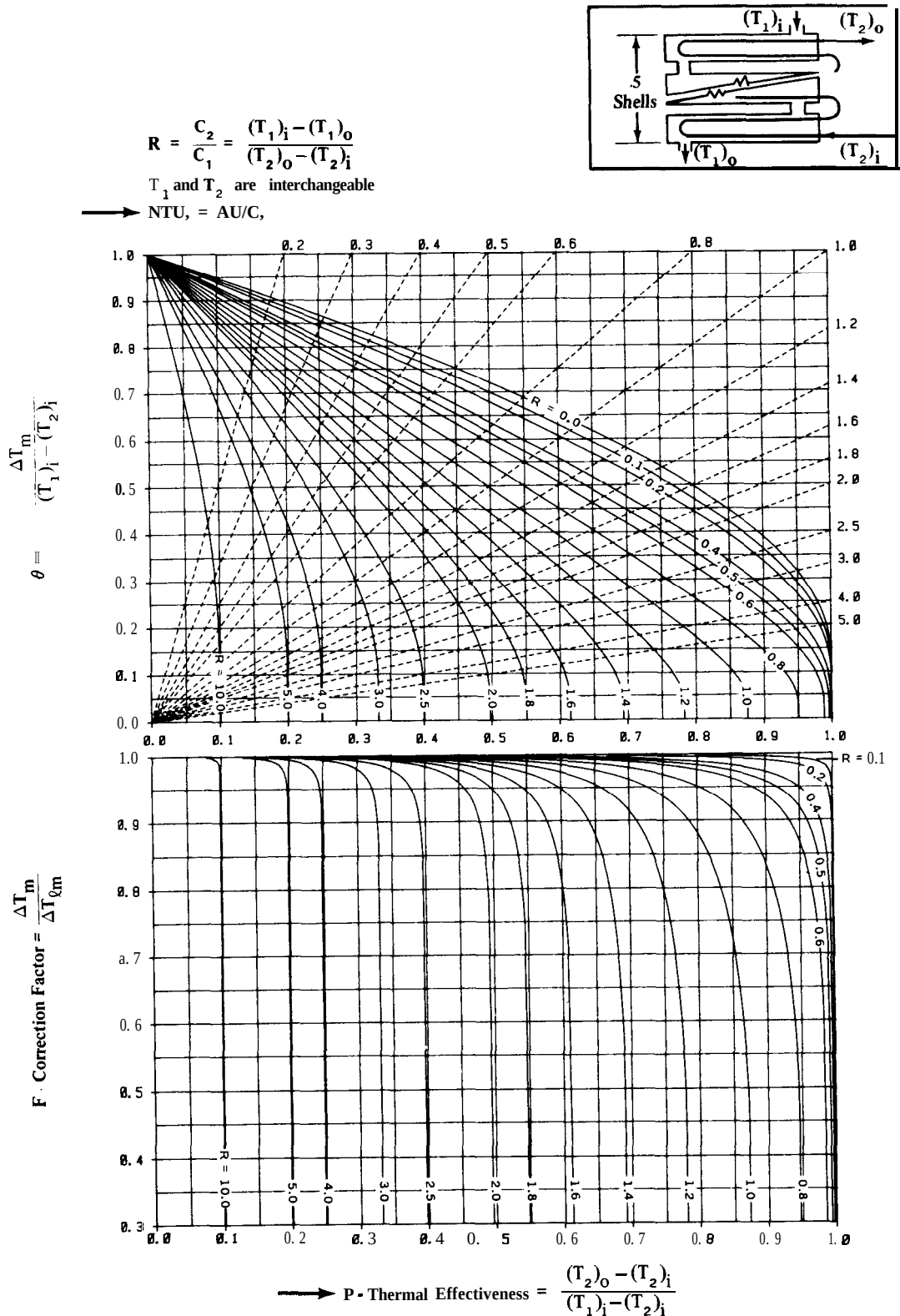


Figure 7 Mean temperature difference relationships: five E shells in series, each with two or any even number of tube passes.

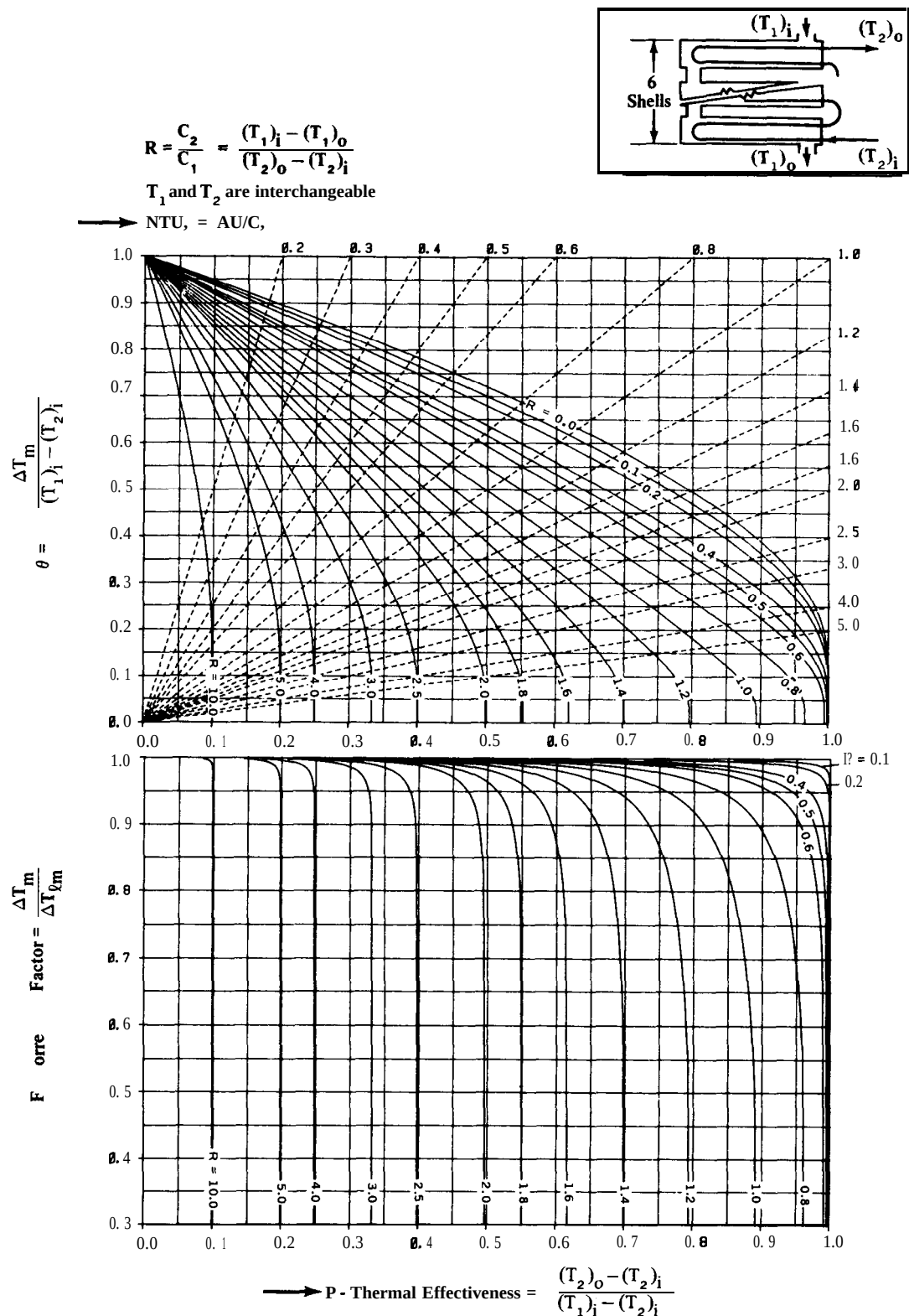


Figure 8 Mean temperature difference relationships: six E shells in series, each with two or any even number of tube passes.

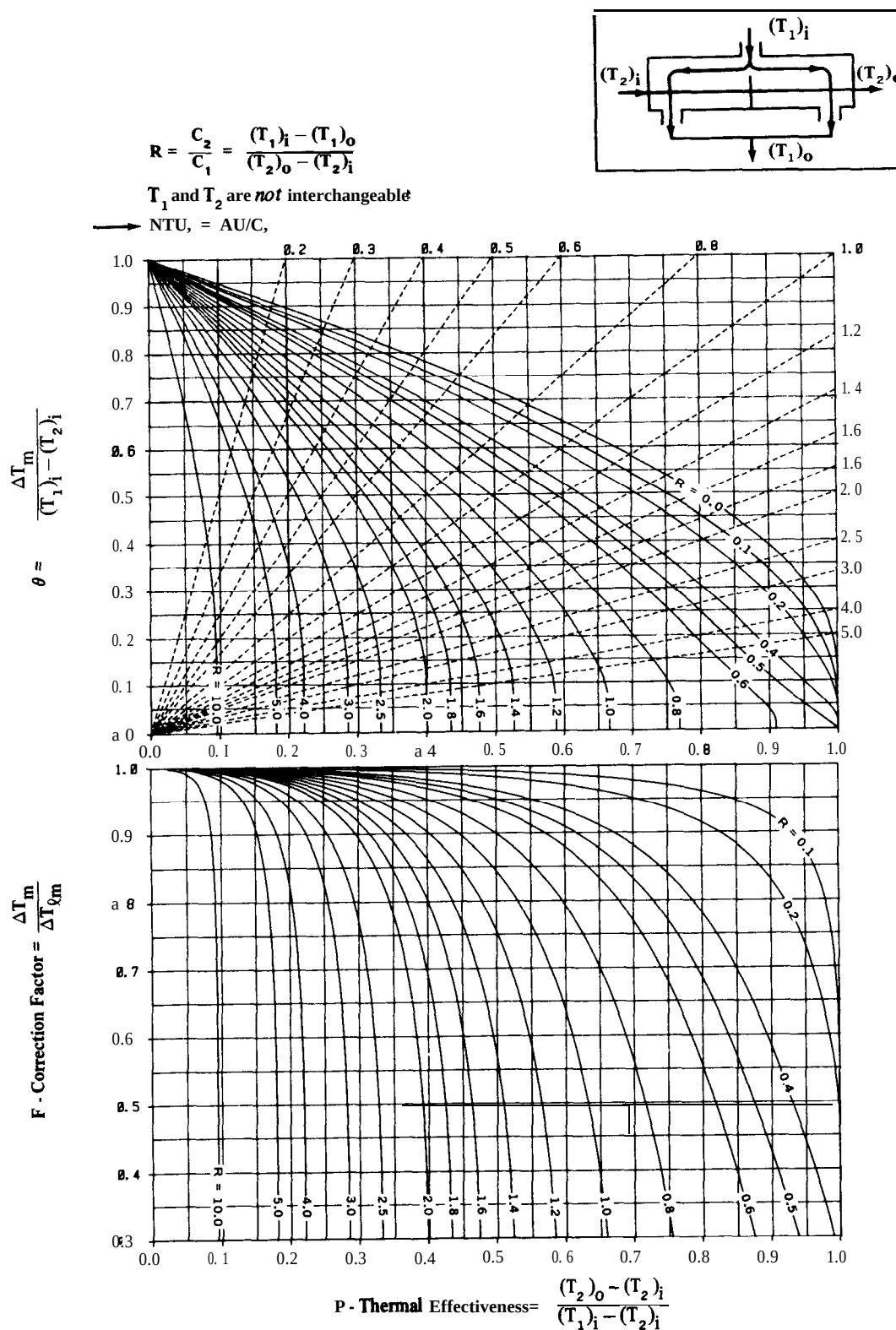


Figure 9 Mean temperature difference relationships: TEMA J shell, one tube pass.

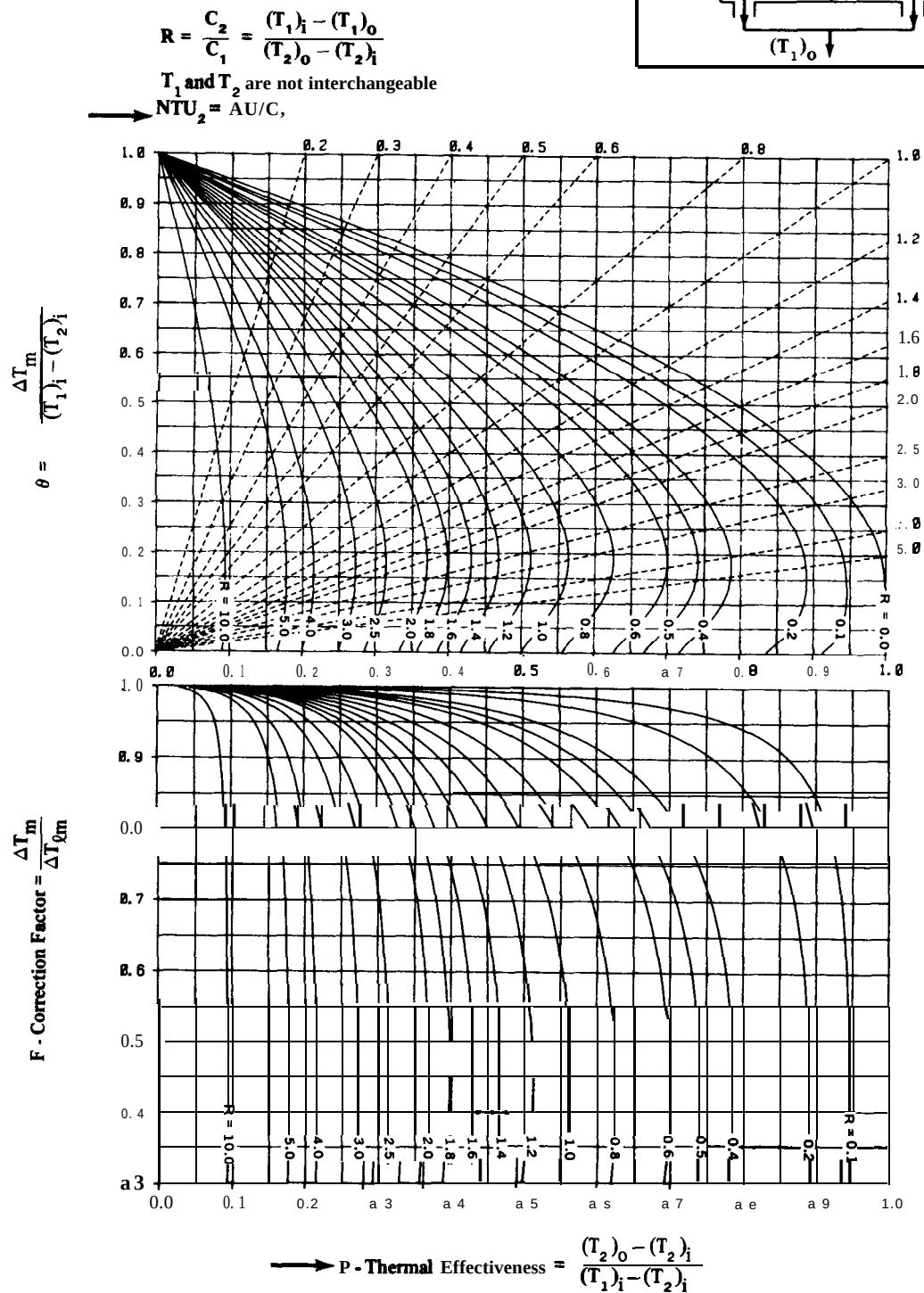


Figure 10 Mean temperature difference relationships: TEMA J shell, two or any even number of tube passes, and mixed-mixed cross flow.

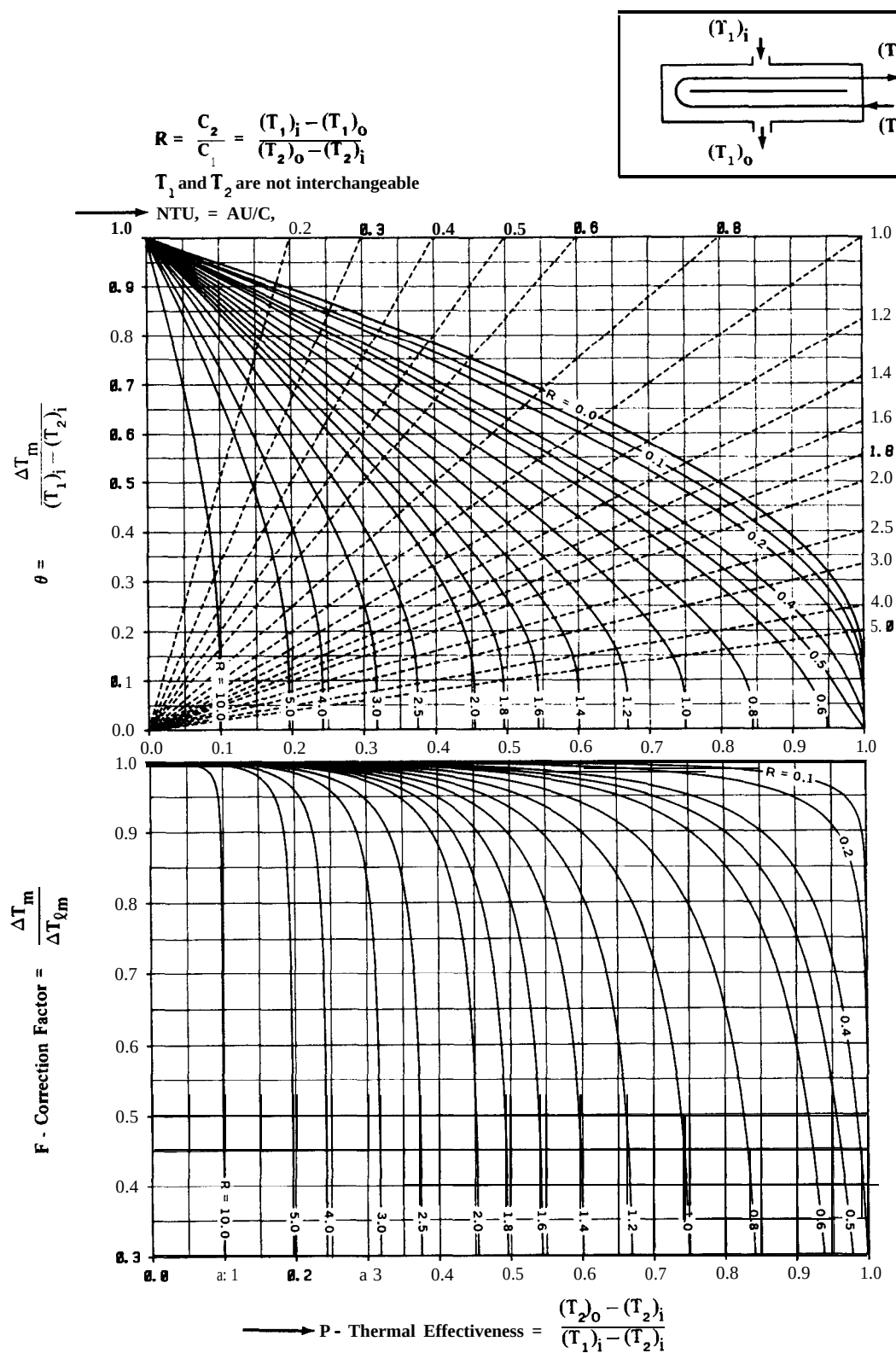


Figure 11 Mean temperature difference relationships: TEMA G shell, two or any even number of tube passes.

$$P = \left(\frac{R\phi^R}{\phi^R - 1} + \frac{\phi}{\phi - 1} - \frac{1}{\ln \phi} \right)^{-1} \quad (14)$$

The derivation assumes that the tube-side fluid enters at the side of shell outlet nozzles. Notice from Fig. 10 that the curves form a $P_{\max}$ at values approximately corresponding to $F = 0.5$ and bend backward to lower P values at the $\theta = 0$ intercept. Operation in the region below $P_{\max}$ is not recommended.

Comparison of the F values for one tube pass to 2N tube passes indicates that one-tube-pass solutions result in higher thermal effectiveness.

E. TEMA G shell split-flow exchanger with 2N tube passes

In TEMA G shells, the flow is split into two streams separated by a longitudinal baffle (see Fig. 11). The shell-side pressure drop is approximately the same as for an E-type exchanger. Under the assumption that the tube-side fluid enters at the shell outlet nozzle side, the thermal effectiveness is substantially higher than for an E (1-2N) exchanger.

The solution of this case is due to Schindler and Bates [7] and appears after some rearrangements as follows :

$$P = \frac{J - 1}{J + 2R e^a} \quad (15)$$

$$J = \frac{1}{1 - D} [(1 + G + 2RG)e^a + 2RD] \quad (16)$$

$$G = \frac{1 - e^{-b}}{2R - 1} \bigg|_{R \neq 0.5} \frac{1}{2F\delta} \bigg|_{R = 0.5} \quad (17)$$

$$D = \frac{1 - e^{-a}}{2R + 1} \quad (18)$$

$$a = \frac{2R + 1}{4F\delta} \quad (19)$$

$$b = \frac{2R - 1}{2F\delta} \quad (20)$$

Stream 1 in definitions of R and R is on the shell side and cannot be interchanged with stream 2. Definition of δ is in Eq. (8), Sec. 1.5.1.

F. TEMA F shell

The F shell has a longitudinal baffle dividing the shell into two compartments. If used, as is usual, with two tube passes, the arrangement results in pure counter flow with $F = 1$. For four tube passes, the configuration would be treated as two E (1-2N) exchangers in series.

There are two inherent problems in the F-shell application:

1. As the longitudinal baffle is exposed on one side to the hot end of the stream and on the other side to the cold end of the stream, there is a thermal conduction

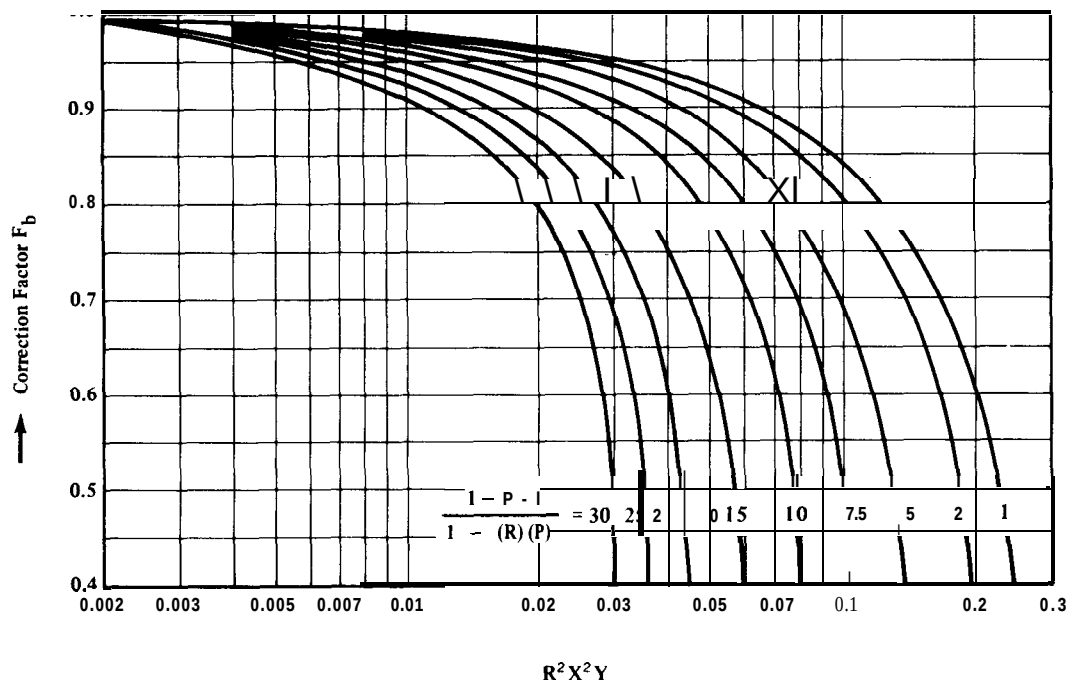


Figure 12 Correction factor F_b for thermal leakage through longitudinal baffle in TEMA F shell exchangers.

heat exchange that decreases the effectiveness of the thermal profile. A method to account for this inefficiency was first developed by Whistler [8] and later rearranged into a dimensionless form by Rozenman and Taborek [9]. It can be expressed as an additional correction factor F_b as follows:

$$AT = F(F_b) \Delta T_{LM} \quad (21)$$

For a two-tube-pass arrangement, $F = 1.0$. The baffle correction factor F_b is expressed as

$$F_b = \frac{Z\delta}{\ln[(1+ZX)/(1-ZX)]} \quad (22)$$

$$Z = \sqrt{4R^2 Y + (R-1)^2} \quad (23)$$

$$X = \frac{P}{2 - PR - P} \quad (24)$$

$$Y = \frac{A_b U_b}{A U_o} \quad (25)$$

where, in consistent units,

A = total outside surface of the tubes

A_b = surface of the longitudinal baffle

U_o = overall heat transfer coefficient

U_b = heat transfer coefficient through the partition, defined as

$$U_b = \left(\frac{2}{\alpha_s} + \frac{L_b}{\lambda_b} \right)^{-1} \quad (26)$$

α_s = convective coefficient at the baffle, usually assumed equal to the shell-side coefficient

L_b = baffle thickness

δ = defined in Eq. 1.5.1(8)

λ_b = thermal conductivity of the baffle material

Graphic representation is shown in Fig. 12.

2. The baffle can be welded to shell wall, in which case there is no leakage between the two shell halves. However, in most cases the tube bundle is removable for cleaning, and in such case a flexible sealing strip will effectively seal leakage of the fluid at the higher pressure in the shell inlet compartment to lower pressure at the outlet compartment. However, the seals are often damaged at times of tube bundle removal and the consequent leakage will further-and sometimes substantially-deteriorate the effective temperature profile.

This case has been analyzed in [9]. The difficulty is in assigning the proper value of the seal gap, which can only be estimated.

Nomenclature for Section 1.52 appears at the end of Section 1.5.3.

REFERENCES FOR SECTION 1.5.2

1. Underwood, A. J. V., *J. Inst. Pet. Technol.*, vol. 20, p. 145, 1934.
2. Bowman, R. A., *Ind. Eng. Chem.*, vol. 28, no. 5, 1936.
3. Gardner, K. A., private communication, 1964.
4. Jaw, L., *J. Heat Transfer, ser. C, vol. 86*, 408, 1964.
5. Gardner, K. A., *Ind. Eng. Chem.*, vol. 33, no. 12, p. 1495, 1941.
6. Gardner, K. A., *2nd. Eng. Chem.*, vol. 33, p. 1215, 1941.
7. Schindler, D. L., and Bates, H. T., *CEP Symp. Ser.* No. 30, vol. 56, Heat Transfer-Storrs, 1960.
8. Whistler, A. M., *J. Heat Transfer*, 683, 1947.
9. Rozenman, T., and Taborek, J., *CEP Symp. Ser.* No. 118, vol. 68, Heat Transfer-Tulsa, 1971.

1.5.3

Fand θ charts for cross-flow arrangements

J. Taborek

■ In this section we present solutions to the most common cases of cross-flow heat exchanger configurations. A very complete reference on this general subject is the work of Stevens et al. [1], where analytical solution of numerous configurations are given—many of them of rather academic interest only. However, there are no numerical or graphic solutions included. Another very useful reference with a good bibliography is by Nicole [2], which is devoted specifically to air-cooled units.

A. Definition of terms

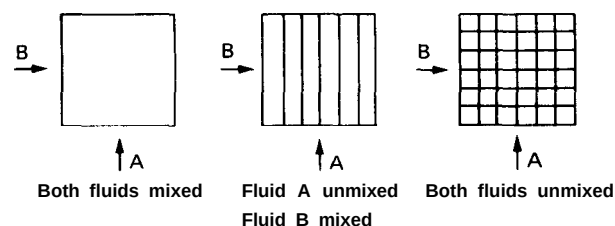
(a) Mixed and unmixed streams

The analytical developments distinguish two extremes of fluid: mixed or unmixed. The following definitions are based partially on [2].

Complete mixing implies that all the fluid in any given plane normal to the flow has the same temperature, although this temperature does change in the direction of flow. Unmixed flow implies, on the other hand, that temperature differences within the fluid in at least one direction normal to the flow can exist but that no heat flux due to these differences occurs.

When the fluid is unmixed, the terminal temperature of either stream is defined as that which would result after complete mixing.

The three possible basic flow combinations for a single-pass cross-flow exchanger, if it is assumed that either fluid is completely mixed or unmixed, are shown here schematically. It is assumed that there is no variation of temperature in the third dimension.



Bowman et al. [3] noted in 1940 that a method of calculating the AT , when either or both fluids are only partially mixed does not exist; this is still the case. They also noted that AT is lower when either fluid is mixed than when both are unmixed.

In many real-life exchangers, the actual flow will be partially mixed, although for high-fin tubes the fins under close spacing will act as flow straighteners and the flow will closely approach the unmixed case.

(b) Pass arrangement

In multipass arrangements in cross flow, the overall direction of flow of the one fluid relative to the other is either counter current or cocurrent. True counter-current flow yields the maximum attainable temperature difference, whereas cocurrent flow yields the smallest temperature difference for the same terminal temperatures. Hence, multipass air coolers are almost invariably of the counter-cross-flow type except in unusual cases, such as where low pour points or a high viscosity index of the process fluid may dictate the use of cocurrent cross flow. Primarily the counter-cross-flow type where the process fluid flows in alternate directions in alternate passes will be considered in this section. Some cases of

cocurrent cross flow are discussed later. Stevens et al. [4] noted that the **multipass** arrangement where the flow is in the same direction in each pass yields a slightly higher AT_{lm} for the same terminal temperatures. For obvious practical reasons, such arrangements are seldom used in the process industry. Nicole [2] lists a few cases for cocurrent cross flow with two and three passes, but the results obtained by numerical methods are not shown, the reader being referred to the extremely hard-to-obtain [4].

(c) Temperature profile considerations

For all nonisothermal conditions, all the restrictive limitations underlying the LMTD validity (see Sec. 1.5.1) must be respected with the addition of the following condition: **The flow of both fluids is evenly distributed over the total flow area** In air-cooled exchangers this condition is assumed to be reasonably satisfied, although some air maldistribution, occasionally quite severe, will exist. The effects on the validity of AT, have never been investigated, and it is a common industrial practice to assume that compensating errors will restore a reasonable balance. Some caution, especially in cases of close temperature approaches, is indicated. Similar comments of caution apply to the tube-side fluid distribution in the inlet headers, especially if viscous fluids or two-phase mixtures are involved.

Some additional observations regarding cross-flow AT, are of interest. In any cross-flow arrangement (even when both streams are completely mixed), the temperatures of the two streams may cross. The greater the number of passes in a counter-cross-flow arrangement, the larger will be the attainable temperature cross, and for the extreme case of very large number of passes, AT, will approach that of counter flow.

For single-tube-pass, multiple-tube-row configurations, the effectiveness increases with the number of rows to approach the values of unmixed-unmixed cross flow. This is practically true for $F > 0.8$ for a number of tube rows larger than four.

Graphic solutions in the form of θ and F charts are included for the most common cross-flow configurations as figures in this section, with equations and comments to proper usage.

1. Unmixed-unmixed cross flow (Fig. 1)
2. One tube pass, one tube row (Fig. 2)
3. One tube pass, two tube rows (Fig. 3)
4. One tube pass, three tube rows (Fig. 4)
5. One tube pass, four tube rows (Fig. 5)
6. Two tube passes, two tube rows, unmixed between passes (Fig. 6)
7. Three tube passes, three tube rows, unmixed between passes (Fig. 7)

8. Four tube passes, four tube rows, unmixed between passes (Fig. 8)

9. Two tube passes, four tube rows, mixed between passes (Fig. 9)

B. Solutions of T_m for most common cross-flow configurations

(a) Unmixed-unmixed cross flow

This is an industrially important flow arrangement representing the case of a large number of unmixed channels in both streams. The original solution is due to Nusselt [5], later reformulated into a more manageable equation by Mason [6].

$$P = \frac{1}{RN} \sum_{n=0}^p \left\{ \left(1 - e^{-N} \sum_{m=0}^n \frac{N^m}{m!} \right) \times \left[1 - e^{-RN} \sum_{m=0}^n \frac{(RN)^m}{m!} \right] \right\} \quad (1)$$

where δ is defined in Eq. 1.5.1(8) and

$$N = NTU = \frac{1}{F\delta} \quad (2)$$

Figure 1 was calculated from this equation with $p = 10$. Even with high-speed computers, the trial-and-error calculations are rather lengthy. An alternative method using numerical integration is faster, as described in Ref. [1].

(b) Mixed-mixed cross flow

This case has no industrial application known to the author and is shown here only as an extreme of the cross-flow arrangements. The solution quoted here is shown in Fig. 1.5.2(10) and is identical to TEMA J shell with 2N tube passes.

(c) One tube pass and one, two, three, or four tube rows

These flow configurations, shown schematically in Figs. 2-5, represent important industrial design arrangements. Essentially a number of tube rows n is connected to a common header in parallel.

The general solution for these cases is shown by Nicole in [2]. In our nomenclature, noting that Nicole's $p = PR$, $q = P$, and $r = \theta = FP\delta = P/NTU_2$, the equation becomes

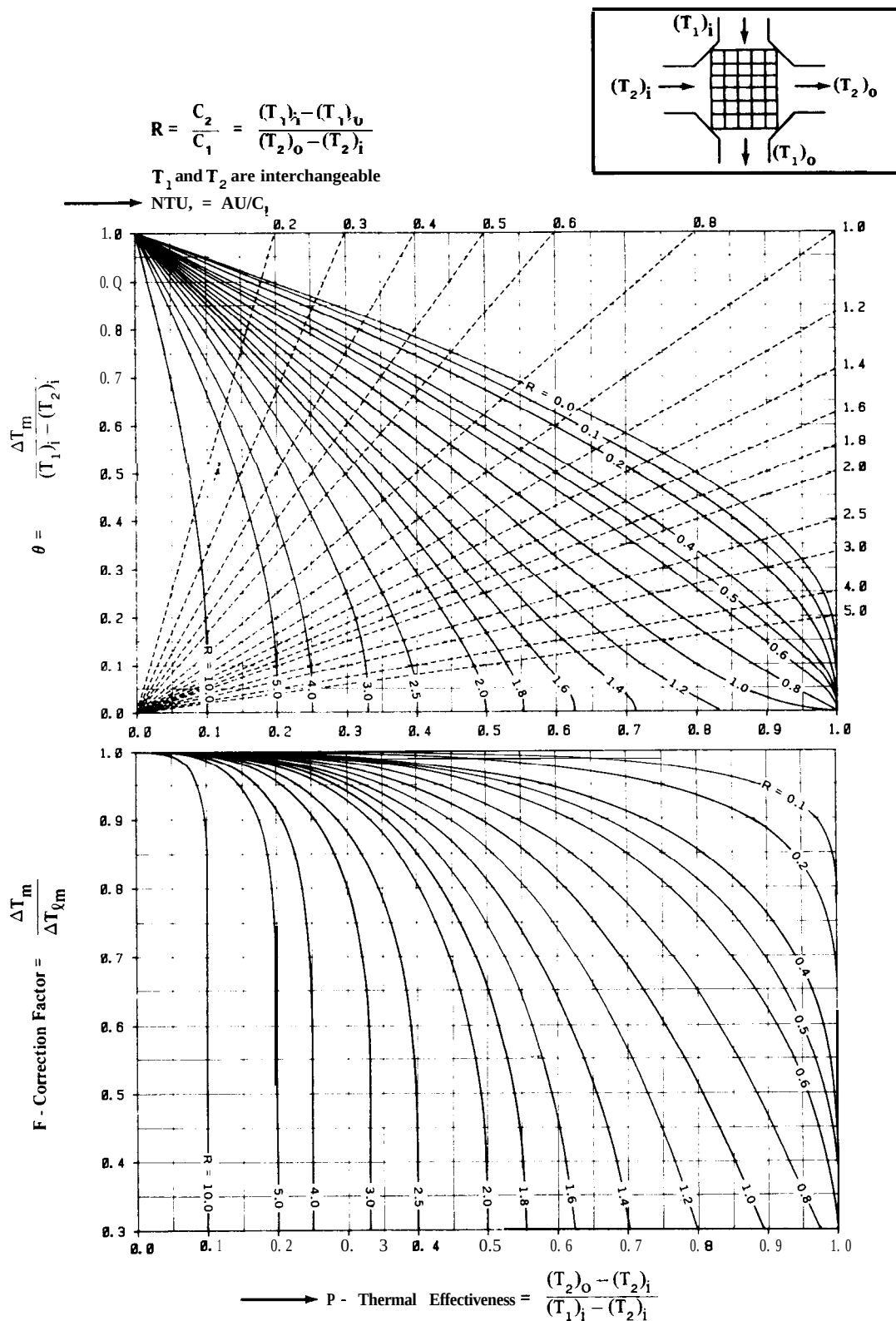


Figure 1 Mean temperature difference relationships: cross flow, both fluids unmixed.

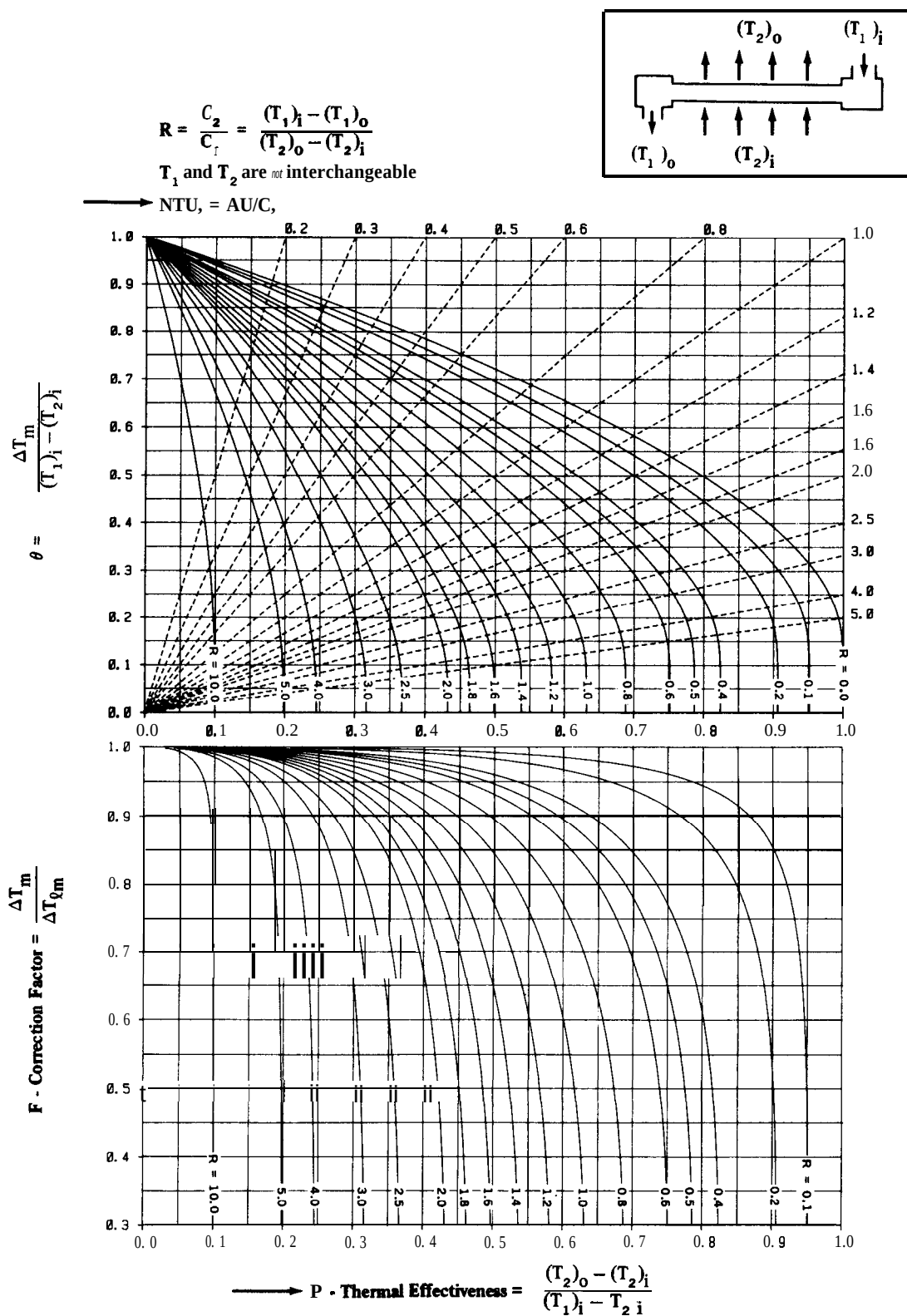


Figure 2 Mean temperature difference relationships: cross-flow unit, one tube row, unmixed.

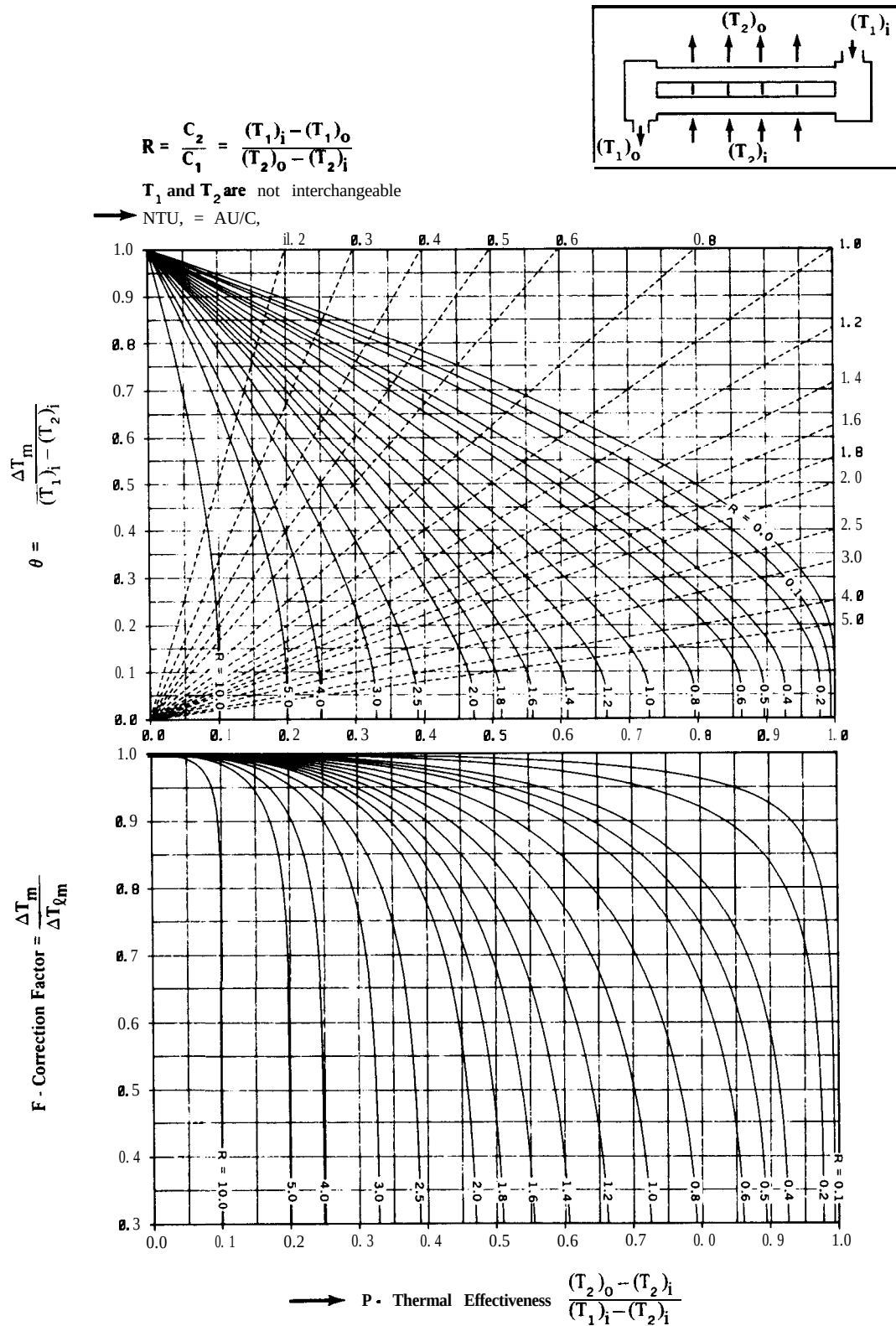


Figure 3 Mean temperature difference relationships: cross-flow unit, two tube rows, one pass, unmixed.

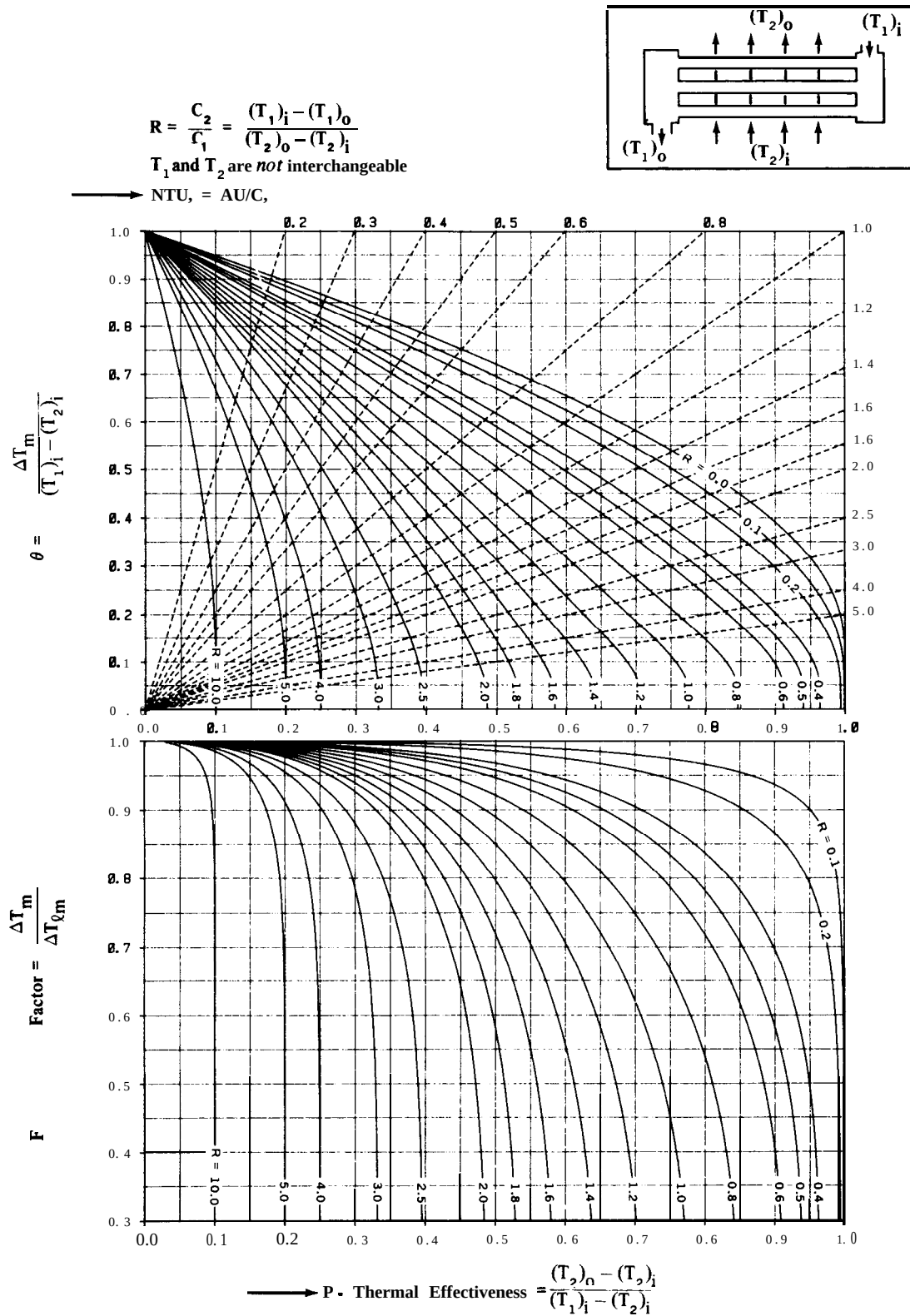


Figure 4 Mean temperature difference relationships: cross-flow unit, three tube rows, one pass, unmixed.

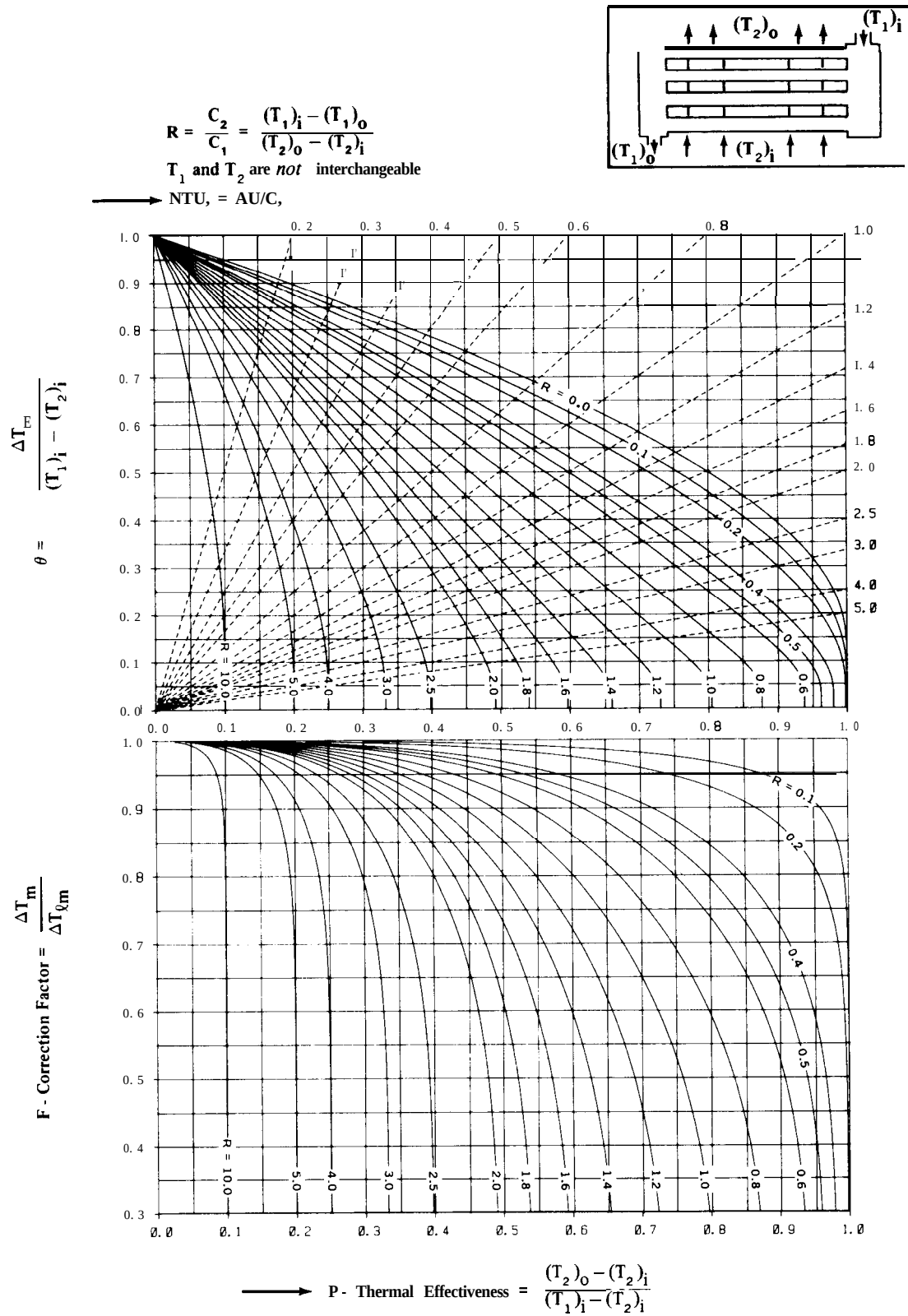


Figure 5 Mean temperature difference relationships: cross-flow unit, four tube rows, one pass, unmixed.

$$P = (1/R) \{1 - [(n \exp(nKR)) / (1 + \sum_{i=1}^{n-1} \sum_{j=0}^i (i_j) X K^j \exp\{-[(i-j)(P/nr)] \sum_{k=0}^j (nKR)^k / k!\})]\}^{-1} \quad (3)$$

where

$$\frac{i_j}{j} = \frac{i!}{(i-j)!j!}$$

that is, the number of combinations of i taken j at a time, and

$$K = 1 - \exp\left(-\frac{1}{nF\delta}\right) = 1 - \exp\left(-\frac{P}{n\theta}\right) \\ = 1 - \exp\left(-\frac{NTU_2}{n}\right) \quad (3a)$$

The expansions of Eq. (3) are shown in [2]. It should be mentioned that identical solutions were obtained by Gardner in 1946 [7] but never published. In our nomenclature, the solutions for the various tube rows are as follows:

1. One tube row, Fig. 2:

$$P = \frac{1}{R} \left(1 - e^{-KR}\right) \quad (4)$$

2. Two tube rows, Fig. 3:

$$P = \frac{1}{R} [1 - e^{-2KR} (1 + RK^2)] \quad (5)$$

3. Three tube rows, Fig. 4:

$$P = \frac{1}{R} \left\{1 - \left[\frac{e^{3KR}}{1 + RK^2(3-K) + (3/2)R^2K^4} \right]^{-1} \right\} \quad (6)$$

4. Four tube rows, Fig. 5:

$$P = (1/R) \{1 - [e^{4KR} / [1 + RK^2(6 - 4K + K^2) + 4R^2K^4(2 - K) + (8/3)R^3K^6]]\}^{-1} \quad (7)$$

For a larger number of tube rows, the solution approaches that of unmixed-unmixed flow (Fig. 1).

(d) Multitube-pass arrangements with equal numbers of multiple tube rows (U-bend sections)

A schematic representation of these flow arrangements is shown in Figs. 6-8. They are essentially U-bend sections with flow in tube passes alternating the flow direction in each subsequent tube row. The solutions are based on [2]. Again, the same solutions have been derived in the unpublished [7].

Using the general formula

$$P = \frac{1}{R} \left(1 - \frac{1}{A}\right) \quad (8)$$

the expressions for A are as follows:

1. Two tube rows, two passes (Fig. 6):

$$A = \frac{K}{2} + \left(1 - \frac{K}{2}\right) e^{2KR} \quad (9)$$

2. Three tube rows, three passes (Fig. 7):

$$A = K \left[1 - \frac{K}{4} - RK \left(1 - \frac{K}{2}\right)\right] e^{KR} + \left(1 - \frac{K}{2}\right)^2 e^{3KR} \quad (10)$$

3. Four tube rows, four passes (Fig. 8):

$$A = \frac{K}{2} \left(1 - \frac{K}{1} + \frac{K^2}{4}\right) + K \left(1 - \frac{K}{2}\right) \left[1 - \frac{R}{8} K \left(1 - \frac{K}{2}\right) e^{2KR}\right] + \left(1 - \frac{K}{2}\right)^3 e^{4KR} \quad (11)$$

4. Five tube rows, five passes (no figure):

$$A = \left\{ K \left(1 - \frac{3}{4}K + \frac{K^2}{2} - \frac{K^3}{8}\right) - RK^2 \left[1 - K + \frac{3}{4}K^2 - \frac{1}{4}K^3 - \frac{R}{2}K^2 \left(1 - \frac{K}{2}\right)\right]^2 \right\} e^{KR} \\ + \left[K \left(1 - \frac{3}{4}K + \frac{1}{16}K^3\right) - 3RK^2 \left(1 - \frac{K}{2}\right)^3 \right] e^{3KR} \\ + \left(1 - \frac{K}{2}\right)^4 e^{5KR} \quad (12)$$

5. Six tube rows, six passes (no figure):

$$A = \frac{K}{2} \left(1 - K + K^2 - \frac{1}{2}K^3 + \frac{1}{8}K^4\right) + K \left(1 - K + \frac{3}{4}K^2 - \frac{5}{16}K^3 + \frac{1}{32}K^4\right) e^{2KR} - RK^2 \\ \times \left[2 - 3K + 3K^2 - \frac{7}{4}K^3 + \frac{3}{8}K^4 - RK^2\right] \\ \times \left(2 - 3K + \frac{3}{2}K^2 - \frac{1}{4}K^3\right) e^{2KR} \\ + \left[\frac{K}{2} \left(2 - 2K + \frac{1}{2}K^3 - \frac{1}{8}K^4\right)\right]$$

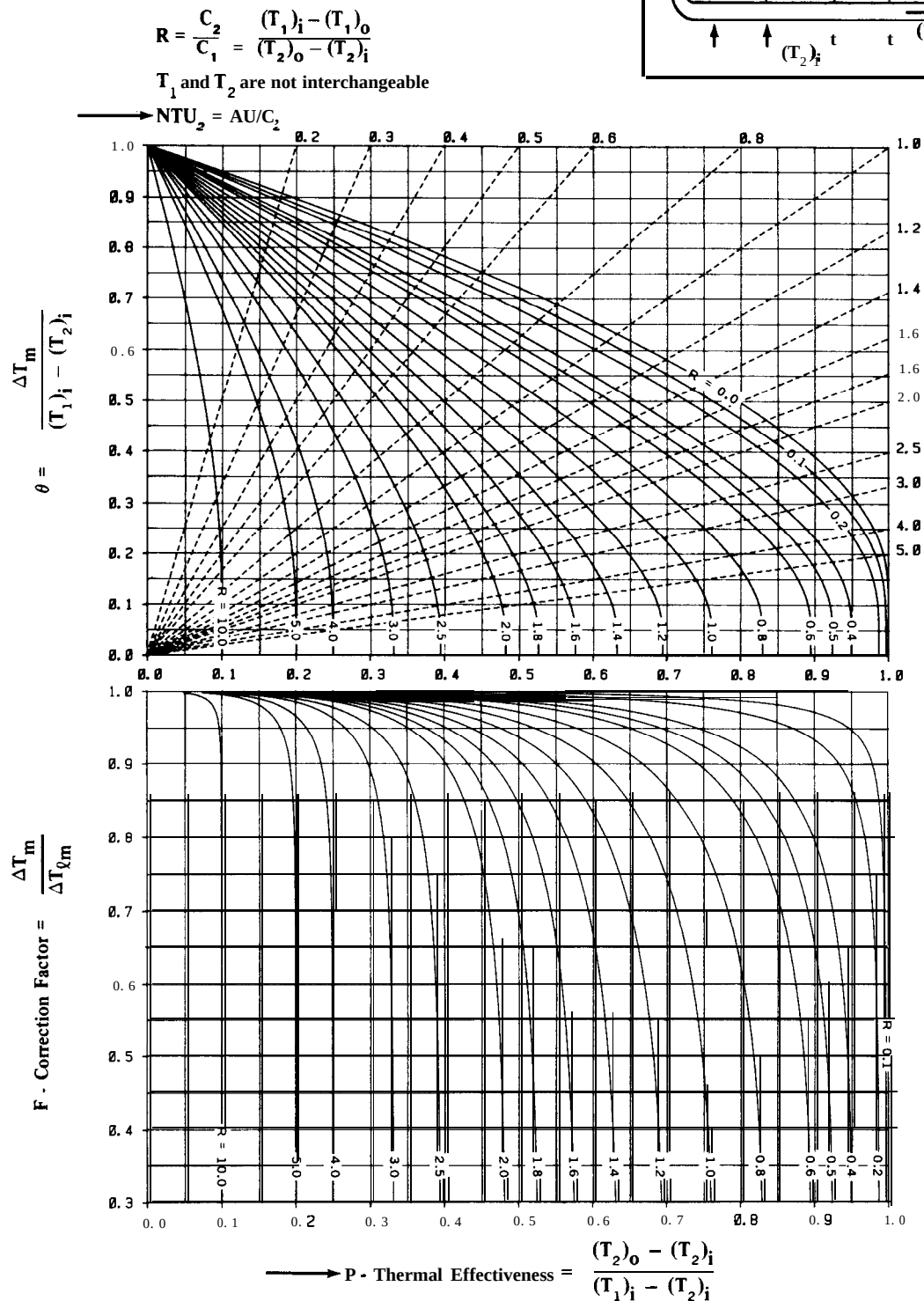


Figure 6 Mean temperature difference relationships: cross-flow unit, two tube rows, two passes, unmixed between passes.

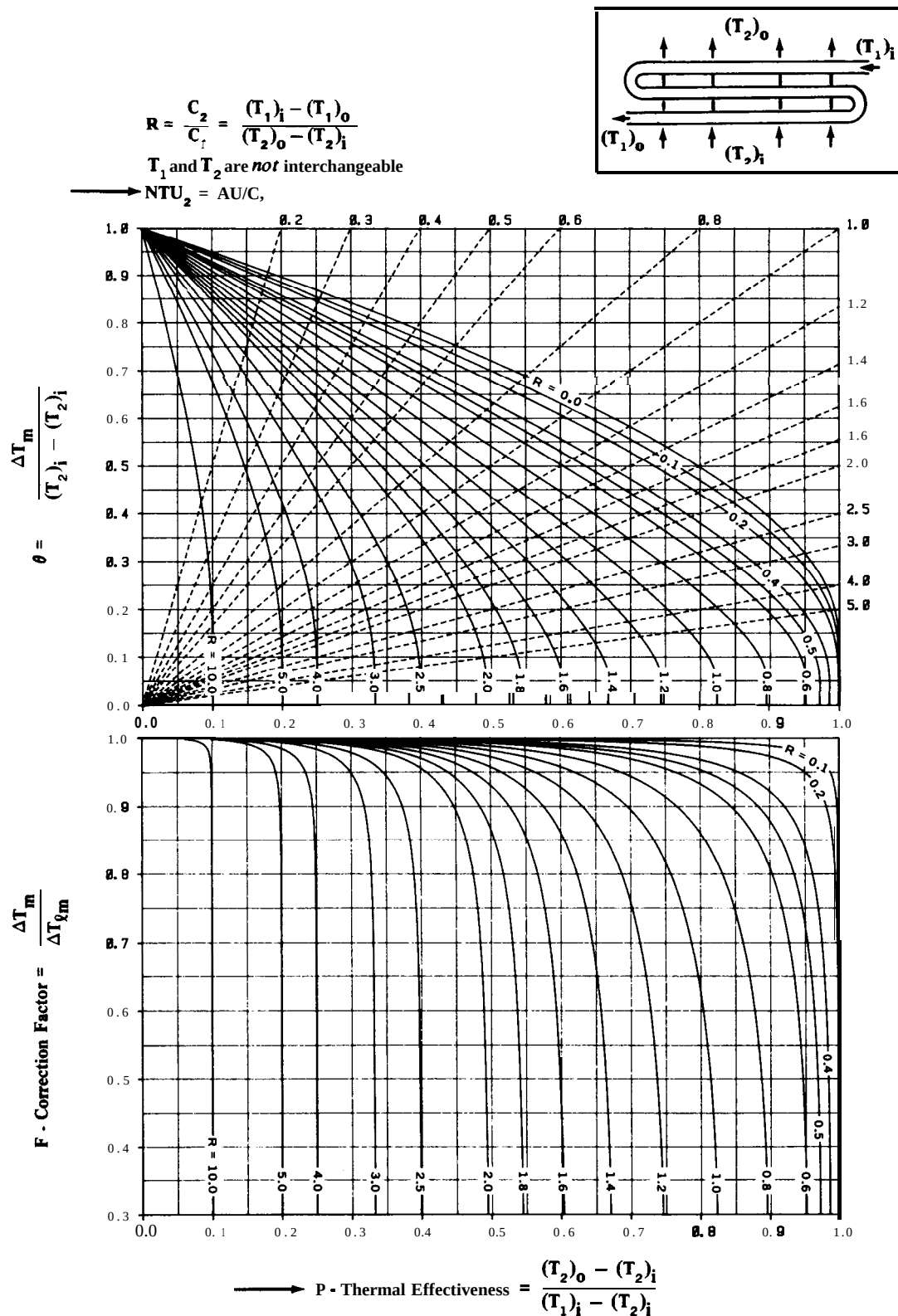


Figure 7 Mean temperature difference relationships: cross-flow unit, three tube rows, three passes, unmixed between passes.

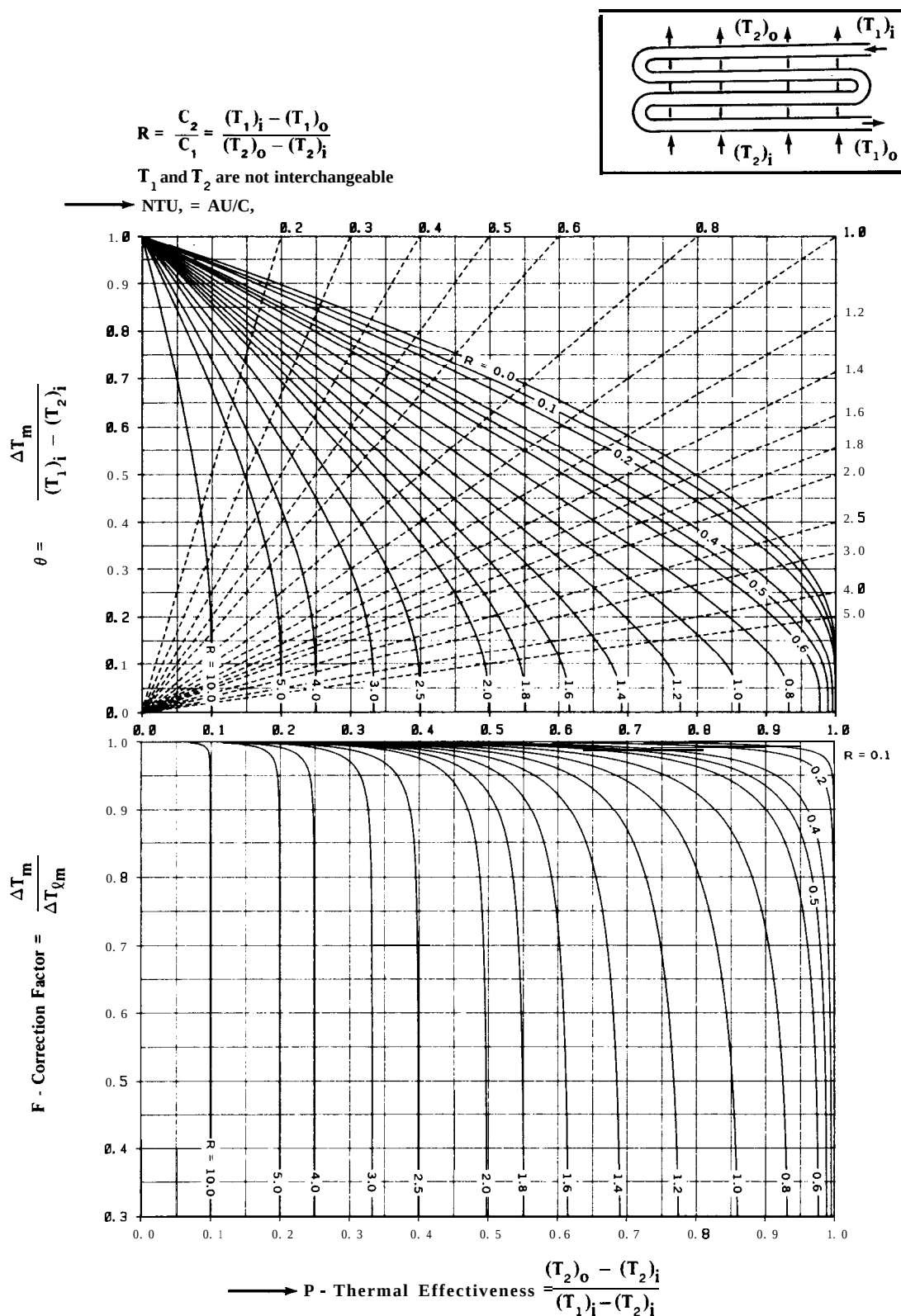


Figure 8 Mean temperature difference relationships: cross-flow unit, four tube rows, four passes, unmixed between passes.

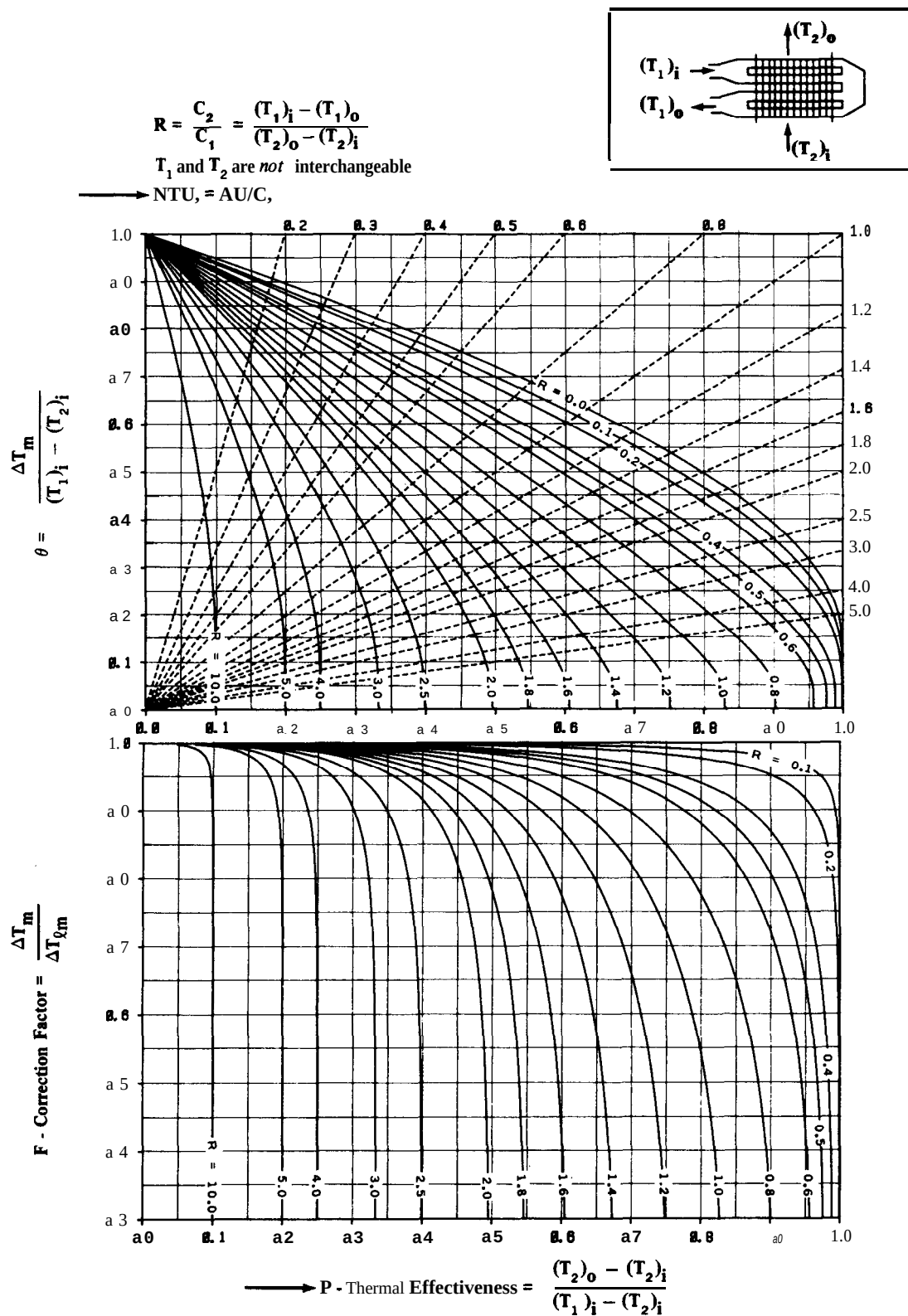


Figure 9 Mean temperature difference relationships: cross flow, four tube rows in two passes, two tube rows per pass, mixed at the header.

$$-4RK^2 \left(1 - \frac{K}{2}\right)^4 \Big] e^{4KR} + \left(1 - \frac{K}{2}\right)^5 e^{6KR} \quad (13)$$

Beyond six tube rows and six passes, the solutions for all practical purposes approach that of pure counter flow, at least in the region of industrial interest, that is, $F > 0.8$. Even the 44 solution is already so close to the counter-flow solution that figures for the 5-5 and 6-6 arrangements are not shown and the equations are documented for archival purposes only.

(e) Multitube-pass arrangements with any number of tube rows, mixed or unmixed between tube passes

The variety of the possible flow arrangements between tube passes and tube rows, mixed or unmixed, either in the flow through the exchanger itself or between passes, is extremely numerous and beyond the objectives of this handbook. Many solutions to such cases are given in [1] and [8], but only a limited number are of industrial

interest. Furthermore, as noted in the very pertinent Discussion to [1] by Landis, many of the solutions are within 2% of the base cases. Realizing the number of assumptions underlying the validity of the solutions (many of which are at least somewhat violated), it appears that some rational simplification could be established—a challenge to future researchers. From the many possible solutions, explicit or obtainable only by numerical integration, a relatively simple case but common in practice is presented here for two passes with four tube rows, that is, two tube rows per pass, Fig. 9. The analytical solution in [2], in terms of Eq. (8), appears as

$$A = \left[\frac{R}{2} K^3 (4 - K + 2RK^2) + e^{4KR} + K \left(1 - \frac{K}{2} + \frac{K^2}{8} \right) (1 - e^{4KR}) \right] (1 + RK^2)^{-2} \quad (14)$$

NOMENCLATURE FOR SECTIONS 1.5.1-1.5.3

A	heat exchanger total surface area, m ²	R	correlational parameter [Eq. 1.5.1(5)]
C_p	specific heat capacity, J/kg K	NTU₂	number of transfer units defined in Eq. 1.5.1(9)
C₁	heat capacities of streams 1 and 2, respectively	U	overall heat transfer coefficient, W/m ² K
C₂	[Eq. 1.5.1(3)], J/kg	δ	auxiliary term [Eq. 1.5.1(8)]
F	LMTD correction factor	ΔT_m	effective temperature difference, K
F_{min}	minimum recommended value of <i>F</i>	ΔT_{LM}	log mean temperature difference, K
F_b	longitudinal baffle correction [Eq. 1.5.2(21)]	θ	temperature effectiveness parameter, defined in Eq. 1.5.1(10)
$\dot{M}$	flow rate, kg/s		
P	thermal effectiveness [Eq. 1.5.1(6)]		

REFERENCES FOR SECTION 1.5.3

1. Stevens, R. A., Fernandez, J., and Woolf, J. R., *J. Heat Transfer*, vol. 79, p. 287, 1957.
2. Nicole, F. J. L., Council for Scientific and Industrial Research (CSIR), Rept. Chem. 223, Pretoria So. Africa, 1972.
3. Bowman, R. A., Mueller, A. C., and Nagle, W. M., Mean Temperature Difference in Design, *J. Heat Transfer*, p. 283, 1940.
4. Stevenes, R. A., Fernandez, J., and Woolfe, J. R., Convair Rept. FZA-095, 1955.
5. Nusselt, W., *Techn. Mech. Thermodynam.*, vol. 1, pp. 417-422, 1930.
6. Mason, J. L., *Proc. 2nd Nat. Congr. Applied Mechanics (ASME)*, p. 801, 1955.
7. Gardner, K. A., private communication, 1962.
8. Worsøe-Schmidt, P., and Høggard Knudsen, H. J., Thermal Modeling of Heat Exchangers for Simulation Purposes, *Proc. 1976 Heat Transfer and Fluid Mechanics Institute*, eds. A. A. McKillop, J. W. Baughn, and H. A. Dwyer, pp. 495-511, Stanford University Press, Stanford, Calif., 1976.

1.6

Effectiveness of multipass shell-and-tube heat exchangers with segmental baffles (cell method)

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1.6.1

Introduction

E. S. Gaddis

■ Single-phase multipass shell-and-tube heat exchangers provided with segmental baffles have a more or less cross flow through the tube bundle. **Baffle**-induced shell-side flow might influence the effective mean temperature difference and hence the effectiveness of the heat exchanger. The influence of the baffles on the exchanger effectiveness and on the mean temperature difference is ignored in most present thermal design calculations. This can be justified in the following cases:

1. Large number of baffles.
2. Heat capacity rates $\dot{C}_1$ and $\dot{C}_2$ ($\dot{C} = \dot{M}c$) of the two streams differ greatly from one another.
3. Number of transfer units (NTU) is small.

If neither of the above conditions is fulfilled, the error in computing the effectiveness of the heat exchanger due to ignoring the influence of the **baffle**-induced shell-side flow may not be small. In such cases it is possible to use numerical methods, as described in Sec. 1.4, to evaluate the thermal performance of the apparatus. The heat exchanger can be divided into a number of cells (subexchangers) as shown in Fig. 1.4.1(5). The purpose of this section is to illustrate with numerical examples the calculation procedure and to present rules that enable the engineer to choose the flow configuration with the highest heat exchanger effectiveness.

Nomenclature and References for Section 1.6 appear at the end of Section 1.6.9.

1.6.2

Calculation procedure

E. S. Gaddis

■ In this section the cross flow of the shell-side fluid across the tube bank between two adjacent baffles or between a baffle and the adjacent tubesheet is regarded as **one pass**. This usage should be distinguished from the more general **one shell-side pass**, which means that the fluid flows from one end of the shell to the other, with baffles treated as only a minor interruption of the basically one-dimensional flow. According to the present designation, the heat exchanger in Fig. 1.4.1(5a) has seven shell passes. Figure 1 in this section shows one of the heat exchanger cells. Leakage and bypass currents are ignored in this analysis. It is assumed in further calculations that stream 1 is the stream with the lower heat capacity rate. The following equations may be written in the steady state:

$$\dot{Q}_c = \dot{C}_1(T_1'' - T_1') \quad (1)$$

$$\dot{Q}_c = -\dot{C}_2(T_2'' - T_2') \quad (2)$$

$$\dot{Q}_c = E_c \dot{C}_1(T_2' - T_1') \quad (3)$$

where E_c is the effectiveness of the cell, defined by

$$E_c = \frac{T_1'' - T_1'}{T_2' - T_1'} \quad (4)$$

It is convenient to carry out the calculations with dimensionless temperatures defined by

$$\Theta = \frac{T - T_{1i}}{T_{2i} - T_{1i}} \quad (5)$$

where T_{1i} and T_{2i} are the heat exchanger inlet temperatures of stream 1 and stream 2, respectively. It is clear that for all cells $0 \leq \Theta \leq 1$ and the dimensionless heat exchanger inlet temperatures are $\Theta_{1i} = 0$ and $\Theta_{2i} = 1$.

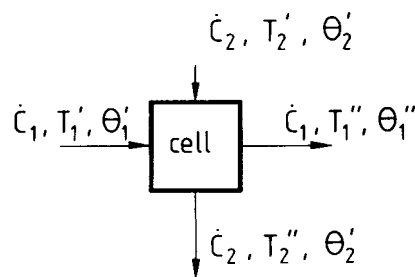


Figure 1 Heat exchanger cell.

Equations (1)-(3) with Eq. (5) give the following dimensionless equations:

$$\Theta_1'' = a\Theta_1' + b\Theta_2' \quad (6)$$

$$\Theta_2'' = e\Theta_2' + f\Theta_1' \quad (7)$$

where

$$a = 1 - E_c \quad (8)$$

$$b = E_c \quad (9)$$

$$e = 1 - RE_c \quad (10)$$

$$f = RE_c \quad (11)$$

$$R = \frac{C_1}{C_2} \quad (0 \leq R \leq 1) \quad (12)$$

The numerical value of the cell effectiveness E_c may or may not be the same in all cells (see Sec. 1.6.9). It is assumed at this stage **that** the cell effectiveness is known; this is discussed in Sec. 1.6.7. The constants **a**, **b**, **e**, and **f** in Eqs. (6) and (7) can be evaluated (**R** is known). Equations (6) and (7) may then be written for

each cell. For a heat exchanger of N_c cells there are $2N_c$ equations in $2N_c$ unknowns; these are the dimensionless outlet temperatures of **both** streams in each cell. The inlet cell temperatures for each stream are identical with the exit temperatures of the preceding cells, and **the** dimensionless inlet temperatures of the heat exchanger are known. The solution of the algebraic linear **equa-**tions gives the required temperature distribution.

The effectiveness E of the heat exchanger is by definition

$$E = \frac{T_{1o} - T_{1i}}{T_{2i} - T_{1i}} = \Theta_{1o} \quad (13)$$

where T_{1o} is the heat exchanger outlet temperature of stream 1. The effectiveness of the heat exchanger is thus identical with the dimensionless heat exchanger outlet temperature of the stream with the lower heat capacity rate.

Nomenclature and References for Section 1.6 appear at the end of Section I. 6.9.

1.6.3

Numerical examples

E. S. Gaddis

■ A shell-and-tube heat exchanger has two tube passes and two segmental baffles. The flow configuration of the idealized heat exchanger is shown in Fig. 1; both heat exchanger inlet nozzles are in the same cell. It is required to calculate the effectiveness of the heat exchanger and to calculate the outlet temperatures of both fluids. The following data are given:

Mass flow rate $\dot{M}_1 = \dot{M}_2 = 0.0834 \text{ kg/s}$
 Specific heat capacity $c_1 = c_2 = 10^3 \text{ J/kg K}$
 Inlet temperature of cold fluid $T_{1i} = 10^\circ \text{C}$
 Inlet temperature of hot fluid $T_{2i} = 60^\circ \text{C}$
 Cell effectiveness $E_c = 0.4$

It is assumed that E_c is constant in all cells; a calculation of the value of E_c is carried out in the example in Sec. 1.6.7.

Calculations:

$$\dot{C}_1 = \dot{M}_1 c_1 = 0.0834 \times 10^3 = 83.4 \text{ W/K}$$

$$\dot{C}_2 = \dot{M}_2 c_2 = 0.0834 \times 10^3 = 83.4 \text{ W/K}$$

Both streams have the same heat capacity rate; it is assumed that stream 1 is the tube-side stream

$$\text{Eq. 1.6.2(8): } a = 0.6$$

$$\text{Eq. 1.6.2(9): } b = 0.4$$

$$\text{Eq. 1.6.2(12): } R = 1.0$$

$$\text{Eq. 1.6.2(10): } e = 0.6$$

$$\text{Eq. 1.6.2(11): } f = 0.4$$

The 12 equations for the 6 cells are given in Table 1. The solution shown in Table 1 was obtained by iteration.

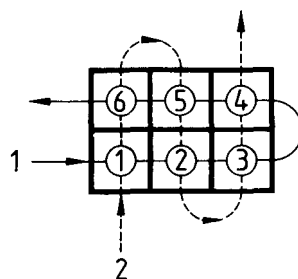


Figure 1 Flow configuration for the heat exchanger in the example.

tion methods. A constant temperature profile was assumed for each stream at the first iteration step ($\Theta_1'' = 0$ and $\Theta_2'' = 1.0$); better numerical values were obtained in the following iteration steps. The computations were carried out in the flow direction of each stream, substituting for the temperatures on the right-hand side of the equations the last calculated numerical values. Table 1 shows that the solution has converged after about eight iterations. More sophisticated methods, e.g., successive over-relaxation (see Sec. 1.4), may be used to accelerate convergence.

$$\text{Eq. 1.6.2(13): } E = 0.54$$

$$\begin{aligned} \text{Eq. 1.6.2(5): } T_{1o} &= (T_{2i} - T_{1i})\Theta_{1o} + T_{1i} \\ &= (60 - 10) \times 0.54 + 10 \\ &= 37^\circ \text{C} \end{aligned}$$

$$\begin{aligned} T_{2o} &= (T_{2i} - T_{1i})\Theta_{2o} + T_{1i} \\ &= (60 - 10) \times 0.46 + 10 \\ &= 33^\circ \text{C} \end{aligned}$$

Table 1 Equations for the example in Sec. 1.6.3 and their solution by iteration methods

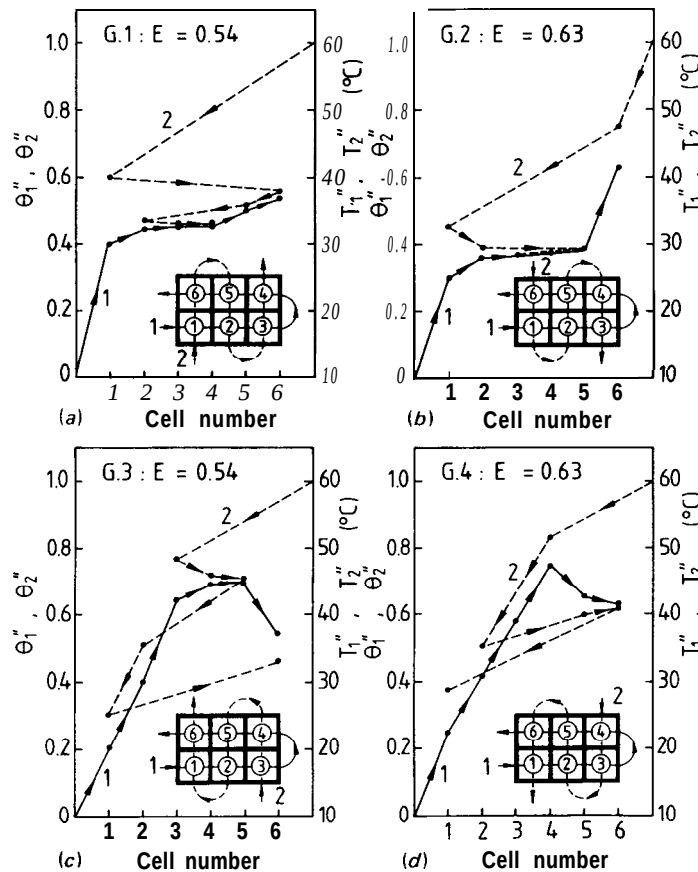
Equation	Iteration step									
	1	2	3	4	5	6	7	8	9	10
$\theta_{1(1)}'' = 0.4$	0	0.400	0.400	0.400	0.400	0.400	0.400	0.400	0.400	0.400
$\theta_{1(2)}'' = 0.6\theta_{1(1)}'' + 0.4\theta_{2(5)}''$	0	0.640	0.554	0.486	0.463	0.454	0.450	0.449	0.448	0.448
$\theta_{1(3)}'' = 0.6\theta_{1(2)}'' + 0.4\theta_{2(2)}''$	0	0.784	0.585	0.503	0.475	0.465	0.460	0.458	0.458	0.458
$\theta_{1(4)}'' = 0.6\theta_{1(3)}'' + 0.4\theta_{2(3)}''$	0	0.870	0.605	0.518	0.482	0.468	0.463	0.461	0.460	0.459
$\theta_{1(5)}'' = 0.6\theta_{1(4)}'' + 0.4\theta_{2(6)}''$	0	0.922	0.655	0.559	0.523	0.509	0.503	0.501	0.500	0.500
$\theta_{1(6)}'' = 0.6\theta_{1(5)}'' + 0.4\theta_{2(1)}''$	0	0.953	0.633	0.576	0.554	0.545	0.542	0.540	0.540	0.540
$\theta_{2(1)}'' = 0.6$	1.0	0.600	0.600	0.600	0.600	0.600	0.600	0.600	0.600	0.600
$\theta_{2(6)}'' = 0.6\theta_{2(1)}'' + 0.4\theta_{1(5)}''$	1.0	0.729	0.622	0.584	0.569	0.564	0.561	0.560	0.560	0.560
$\theta_{2(5)}'' = 0.6\theta_{2(6)}'' + 0.4\theta_{1(4)}''$	1.0	0.785	0.615	0.557	0.534	0.525	0.522	0.521	0.520	0.520
$\theta_{2(2)}'' = 0.6\theta_{2(5)}'' + 0.4\theta_{1(1)}''$	1.0	0.631	0.529	0.494	0.481	0.475	0.473	0.472	0.472	0.472
$\theta_{2(3)}'' = 0.6\theta_{2(2)}'' + 0.4\theta_{1(2)}''$	1.0	0.635	0.539	0.491	0.474	0.467	0.464	0.463	0.462	0.462
$\theta_{2(4)}'' = 0.6\theta_{2(3)}'' + 0.4\theta_{1(3)}''$	1.0	0.694	0.557	0.496	0.474	0.466	0.462	0.461	0.461	0.460

Numbers between brackets indicate cell number in Fig.1.6.3(1).

The temperature distribution (outlet cell temperatures) is shown in Fig. 2a.

If the flow direction of the shell-side fluid (stream 2) is reversed, the flow configuration in Fig. 2d is obtained. On the other hand, reversing the flow direction

of the tube-side fluid (stream 1) leads, depending on the flow direction of stream 2, to the flow configurations shown in Figs. 2b and c. The numbering of the cells in Fig. 2 is carried out always in the flow direction of stream 1, and cell 1 is located always at the same posi-

**Figure 2** Temperature distribution inside the heat exchanger for different flow configurations.

tion. The temperature distribution in the other three cases is shown also in Fig. 2; the computed heat exchanger effectivenesses are as follows.

For the flow configuration in Fig. **2b**: $E = 0.63$

For the flow configuration in Fig. **2c**: $E = 0.54$

For the flow configuration in Fig. **2d**: $E = 0.63$

Thus, two of the four flow configurations (Figs. **2a** and **c**) have the same lower effectiveness; the other two flow configurations (Figs. **2b** and **d**) have the same higher effectiveness. It is seen in Fig. 2 that the temperature distributions for the two streams, even for the

flow configurations with the same effectiveness, are different. For example, the outlet temperature of stream 1 for the two flow configurations with the same higher effectiveness (Figs. **2b** and **d**) is 41.5 °C. Nevertheless, stream 1 has a monotonic temperature distribution for the flow configuration in Fig. **2b** and has a maximum of 47.5 °C at the outlet of cell 4 for the flow configuration in Fig. **2d**. Knowledge of the highest or the lowest temperature of each fluid might be important in certain cases, for example, when phase change, polymerization, crystallization, etc. must be avoided inside the heat exchanger.

Nomenclature and References for Section 1.6 appear at the end of Section I. 6.9.

1.6.4

Rules for highest heat exchanger effectiveness

E. S. Gaddis

■ The effectiveness of a shell-and-tube heat exchanger with segmental baffles depends on the relative position of the inlet and outlet nozzles for both fluids, among other parameters. As demonstrated in Sec. 1.6.3, some flow configurations lead to the same higher and others to the same lower heat exchanger effectiveness. The inlet nozzle of stream 2 may be located relative to the inlet nozzle of stream 1 at four different positions. The four possible geometries (G) are shown in Fig. 1 and hereafter are called G. 1, G.2, G.3, and G.4. The relative position of the outlet nozzles for each geometry depends on the number of passes for each stream, whether this number is even or odd. In general:

$$E = f(E_{c,1}, E_{c,2}, \dots, E_{c,N_c}, R, G, M, N) \quad (1)$$

For the case where the cell effectiveness E_c is constant

$$E = f(E_c, R, G, M, N) \quad (2)$$

Here M and N are the numbers of passes for stream 1 and stream 2, respectively.

The rules illustrated in Fig. 2 were obtained from computations carried out for a large number of flow configurations [1, 2], and were verified in a variable-geometry heat exchanger [2]. It is seen in Fig. 2 that each flow configuration pair with the same lower or the same higher heat exchanger effectiveness has a common property: one of the two flow configurations can be obtained from the other by reversing the flow direction for both streams. Reversing the flow direction for only one of the two streams leads to a flow configuration of one of the other pair with either higher or lower heat exchanger effectiveness. It is possible to distinguish between the flow configurations with lower and with

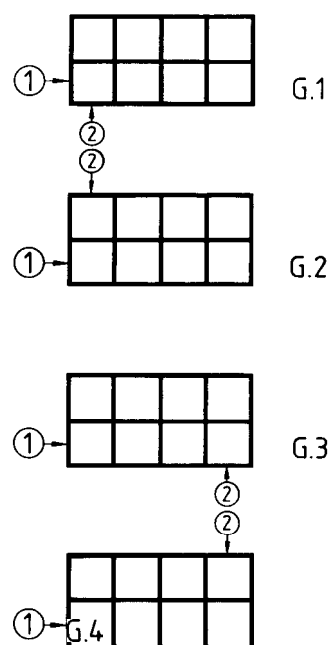


Figure 1 Relative position of the inlet nozzles for the two streams.

higher heat exchanger effectiveness by the following rule: each flow configuration with the lower heat exchanger effectiveness has either both inlet or both outlet nozzles in the same cell, and each flow configuration with the higher heat exchanger effectiveness has an inlet and an outlet nozzle in the same cell.

There are different rules for flow configurations with both odd numbers M and N . In that case, G. 1 always has the lowest and G.4 always has the highest heat exchanger effectiveness; the other two geometries, G.2

N	M	Equal lower heat exchanger Effectiveness		Equal higher heat exchanger Effectiveness	
even	even				
odd	even				
even	odd				

Figure 2 Heat exchangers with different flow configurations and equal heat exchanger effectiveness values.

Table 1 Rules for heat exchangers with odd numbers of passes M and N

$\frac{N}{M}$	Higher heat exchanger effectiveness in direction of arrow $\rightarrow$			
> 1	G.1	G.2	G.3	G.4
$= 1$	G.1	G.2 ^a	G.3 ^a	G.4
< 1	G.1	G.3	G.2	G.4

G.2 and G.3 are identical when $R = 1$.

and G.3, have heat exchanger effectivenesses that always lie between those of geometries G. 1 and G.4 (see Table 1). Reversing the flow direction of both fluids, when both M and N are odd, does not change the geometry G of the heat exchanger.

Nomenclature and References for Section 1.6 appear at the end of Section I. 6.9.

1.6.5

Special case of two tube passes

E. S. Gaddis

■ The special case of two tube passes has a simple flow configuration, which enables a quick calculation of the heat exchanger effectiveness [3]. The computation procedure is described by an example. Consider the heat exchanger treated in Sec. 1.6.3 with geometry G. 1. This heat exchanger is illustrated once more in Fig. 1a, while Fig. 1b is intended to clarify the type of connection between the cells. It is seen in Fig. 1b that parallel flow (P) exists between cells 2, 3, and 4 and counter flow (C) between cells 5 and 6.

The effectiveness of a heat exchanger composed of two units coupled in parallel flow or in counter flow may be evaluated from the following equations (4).

For parallel flow:

$$E = E_{u,1} + E_{u,2} - (1 + R)E_{u,1}E_{u,2} \quad (1)$$

and for counter flow:

$$E = \frac{E_{u,1} + E_{u,2} - (1 + R)E_{u,1}E_{u,2}}{1 - RE_{u,1}E_{u,2}} \quad (2)$$

where $E_{u,1}$ and $E_{u,2}$ are the effectivenesses of the two coupled units. The effectivenesses E , $E_{u,1}$, and $E_{u,2}$ are defined by Eq. 1.6.2(13) with the appropriate temperatures. For the special case where $E_{u,1} = E_{u,2} = E_u$ and $R = 1$, the heat exchanger effectiveness is given by:

For parallel flow:

$$E = 2E_u(1 - E_u) \quad (3)$$

and for counter flow:

$$E = \frac{2E_u}{1 + E_u} \quad (4)$$

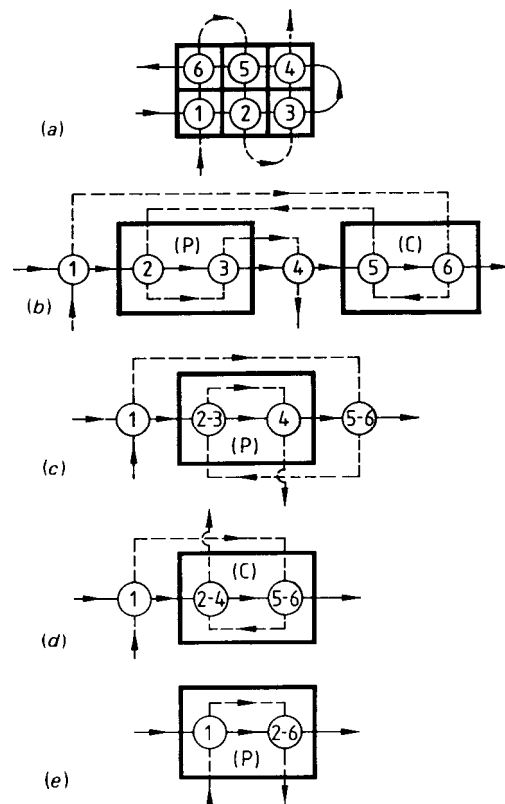


Figure 1 Steps for evaluating the effectiveness of a heat exchanger with two tube passes.

Combining cells 2 and 3 and cells 5 and 6 in Fig. 1 in part heat exchangers yields the following relationships:

$$E_{p,2,3} = E_{c,2} + E_{c,3} - (1 + R)E_{c,2}E_{c,3} \quad (5)$$

and

$$E_{p,5,6} = \frac{E_{c,5} + E_{c,6} - (1 + R)E_{c,5}E_{c,6}}{1 - RE_{c,5}E_{c,6}} \quad (6)$$

where $E_{c,i}$ is the effectiveness of cell i and $E_{p,i,j}$ is the effectiveness of a part heat exchanger from cell i to cell j .

Cell 4 and the part heat exchanger from cell 2 and cell 3 are connected in Fig. 1c in parallel. Thus

$$E_{p,2,4} = E_{p,2,3} + E_{c,4} - (1 + R)E_{p,2,3}E_{c,4} \quad (7)$$

In a similar way in Fig. 1d

$$E_{p,2,6} = \frac{E_{p,2,4} + E_{p,5,6} - (1 + R)E_{p,2,4}E_{p,5,6}}{1 - RE_{p,2,4}E_{p,5,6}} \quad (8)$$

and in Fig. 1e

$$E = E_{p,2,6} + E_{c,1} - (1 + R)E_{c,1}E_{p,2,6} \quad (9)$$

Assuming, as in the example in Sec. 1.6.3, that $E_{c,1} = \dots = E_{c,6} = E_c = 0.4$ and $R = 1$ yields

$$E_{p,2,3} = 2E_c(1 - E_c) = 0.480$$

$$E_{p,5,6} = \frac{2E_c}{1 + E_c} = 0.571$$

$$E_{p,2,4} = 0.496$$

$$E_{p,2,6} = 0.699$$

$$E = 0.54$$

It is worth mentioning that the part heat exchanger from cell 2 to cell 6 has an effectiveness $E_{p,2,6} = 0.699$. Thus, the heat exchanger effectiveness E increases by about 30% if for that geometry the heat transfer surface in cell 1 is insulated.

It is seen in Fig. 1 that reversing the flow direction for both streams does not change the type of interconnection (parallel or counter) between two adjacent cells, between a cell and the adjacent part heat exchanger, or between two adjacent part heat exchangers. This explains why the effectiveness of a heat exchanger does not change when the flow direction for both streams is reversed. On the other hand, reversing the flow direction for one of the two streams changes the type of interconnection between the heat exchanger components; the effectiveness of the heat exchanger, depending on the available interconnections with parallel and with counter flow, either decreases or increases.

Nomenclature and References for Section I. 6 appear at the end of Section I. 6.9.

1.6.6

Heat exchangers with longitudinal baffles

E. S. Gaddis

■ A shell-and-tube heat exchanger with a longitudinal baffle, as illustrated schematically in Fig. 1a, may be treated as two units coupled in parallel flow or in counter flow.

Example: The heat exchanger in Fig. 1a is pro-

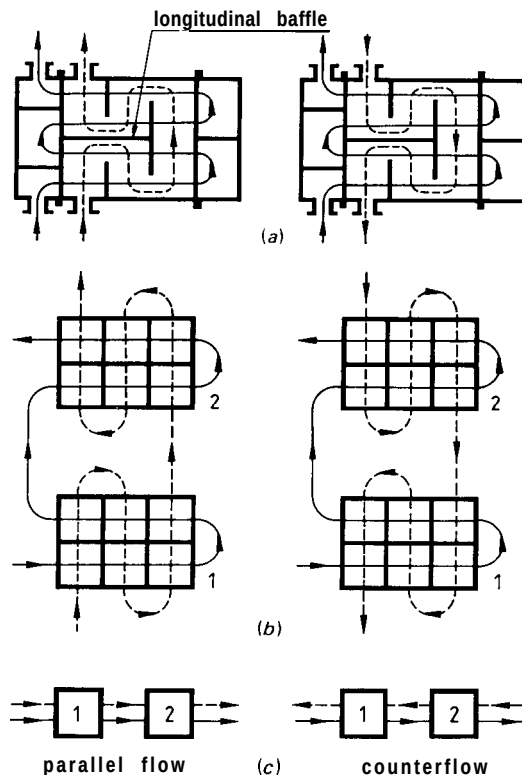


Figure 1 A shell-and-tube heat exchanger with a longitudinal baffle; shell-side nozzles are located near the tube-side nozzles.

vided from each side of the longitudinal baffle with two tube passes and two segmental baffles. It is required to calculate the heat exchanger effectiveness given the following data:

Cell effectiveness $E_c = 0.4$

Ratio of heat capacity rates $R = 1$

The calculations are as follows.

Case of parallel flow (Fig. 1a, left): The effectiveness $E_{u,1}$ of unit 1 with geometry G. 1 was calculated in the example in Sec. 1.6.3. It is clear from the rules given in Fig. 1.6.4(2) that the two coupled units shown in Fig. 1b, left, have the same effectiveness (geometries G.1 and G.3). Thus,

$$E_{u,1} = E_{u,2} = E_u = 0.54$$

From Eq. 1.6.5(3)

$$E = 2 \times 0.54(1 - 0.54) = 0.5$$

For parallel flow the heat exchanger has an effectiveness that is lower than the effectiveness of each unit.

Case of counter flow (Fig. 1a, right): The effectiveness of unit 1 with geometry G.4 is 0.63 (see the example in Sec. 1.6.3). Also in this case, both units (G.2 and G.4) have the same effectiveness. From Eq. 1.6.5(4)

$$E = \frac{2 \times 0.63}{1 + 0.63} = 0.77$$

Counter flow is thus superior not only because of the higher heat exchanger effectiveness attainable by virtue of the counter-flow coupling between the two units, but

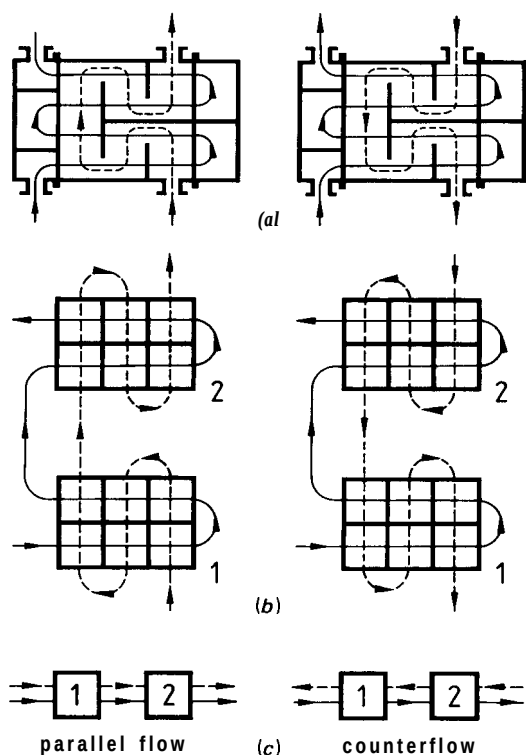


Figure 2 A shell-and-tube heat exchanger with a longitudinal baffle; shell-side nozzles are located apart from the tube-side nozzles.

also because of the higher effectiveness of each unit.

Shell-and-tube heat exchangers provided with segmental baffles and a longitudinal baffle are customarily illustrated as in Fig. 1a, with the shell-side nozzles located adjacent to the tube-side nozzles. Presumably it is considered that this layout alone with counter flow (Fig. 1a, right) yields the highest heat exchanger effectiveness. This is certainly the case when the heat exchanger is provided with an odd number of tube passes from each side of the longitudinal baffle. The number of shell passes from each side of the longitudinal baffle, induced by the segmental baffles, is bound to be odd by the construction. Let us consider the layout in Fig. 2a with an even number of tube passes from each side of the longitudinal baffle and with the shell-side nozzles located apart from the tube-side nozzles. It can be shown by the rules in Fig. 1.6.4(2) that the layout in Fig. 2a yields the same lower and the same higher heat exchanger effectiveness for parallel and for counter flow, respectively, as the layout in Fig. 1a [3].

Nomenclature and References for Section 1.6 appear at the end of Section I. 6.9.

1.6.7

Cell effectiveness

E. S. Gaddis

■ The cell effectiveness E_c , required for evaluating the heat exchanger effectiveness, may be expressed by (see Sec. 1.2.4)

$$E_c = f(\text{NTU}_c, R, \text{cell flow configuration}) \quad (1)$$

where

$$\text{NTU}_c = \frac{UA_c}{\dot{C}_1} \quad (2)$$

Apart from a few tube rows in the baffle cut near the shell, the shell-side flow is more or less across the tube bundle. The numerical value of the cell effectiveness lies between those for cross flow with both fluids unmixed and for cross flow with the shell-side fluid laterally mixed. Comparing the numerical values of cell effectiveness for the two limiting cases shows that both values in the range $\text{NTU}_c \leq 1$ are practically equal [2]. Since the number of transfer units of a cell is seldom higher than unity, either of the two relationships may be used. A number of the available equations are reviewed in Secs. 1.3.1 and 15.3.

Example: It is required to evaluate the cell effectiveness for the heat exchanger treated in Sec. 1.6.3 given the following additional data:

Total heat transfer area $A = 2.5 \text{ m}^2$

Overall heat transfer coefficient $U = 143 \text{ W/m}^2 \text{ K}$

It is assumed that the heat transfer area is the same in each cell and that the overall heat transfer coefficient is constant in the heat exchanger.

Calculations:

Number of cells in the heat exchanger $N_c = 6$

Heat transfer area in each cell $A_c = (2.5/6) = 0.417 \text{ m}^2$

Equation (2): number of transfer units of a cell

$$\text{NTU}_c = \frac{143 \times 0.417}{83.4} = 0.715$$

Since $\text{NTU}_c < 1$, the cell effectiveness may be evaluated from an equation for one of the two limiting cases. The equation for cross flow with one fluid laterally mixed is simpler than that for cross flow with both fluids unmixed, and is thus used hereafter for calculating E_c .

For the case where the fluid with the lower heat capacity rate is laterally mixed (shell-side flow), the cell effectiveness is given by [5]

$$E_c = 1 - \exp \left[- \frac{1 - \exp(-R \text{NTU}_c)}{R} \right] \quad (3)$$

and for the case where the fluid with the higher heat capacity rate is laterally mixed, the cell effectiveness is given by [5]

$$E_c = \frac{1 - \exp\{-R[1 - \exp(-\text{NTU}_c)]\}}{R} \quad (4)$$

When $R = 1$, Eqs. (3) and (4) become identical, or

$$E_c = 1 - \exp\{-[1 - \exp(-\text{NTU}_c)]\} \quad (5)$$

Even in the case $R \neq 1$, Eqs. (3) and (4) yield, in the range $\text{NTU}_c \leq 1$, almost the same value (maximum deviation $< 1\%$). Substituting in Eq. (5) gives cell effectiveness $E_c = 0.4$.

Nomenclature and References for Section 1.6 appear at the end of Section 1.6.9.

1.6.8

Comparison of the conventional method and the cell method

E. S. Gaddis

■ **AS** mentioned in Sec. 1.6.7, the cell effectiveness is not very sensitive to the flow configuration inside the cell ($NTU_c \leq 1$). Considering the case of constant cell effectiveness in all cells, substituting from Eq. 1.6.7(1) in Eq. 1.6.4(2) and rearranging gives

$$E = f(NTU, R, G, M, N) \quad (1)$$

The number of transfer units of the heat exchanger NTU is related to the number of transfer units of the cell NTU_c , by

$$NTU = \frac{UA}{\dot{C}_l} = MN NTU_c \quad (NTU_c = \text{constant}) \quad (2)$$

The dependence of the heat exchanger effectiveness on the number of transfer units for the heat exchanger geometries treated in Sec. 1.6.3 is shown in Fig. 1. For comparison, the heat exchanger effectiveness evaluated by the conventional method (ignoring baffle-induced flow) is also shown in Fig. 1 (dashed line). The effective mean temperature difference in the latter case was evaluated from the equations presented in Sec. 1.5, and Eq. 1.2.4(19) was used to relate the heat exchanger effectiveness to the effective mean temperature difference.

The following remarks can be made:

1. The conventional method fails to show the dependence of the effectiveness on the relative position of the inlet and outlet nozzles.
2. The heat exchanger effectiveness obtained by the conventional method lies between the higher and lower values obtained by the cell method.

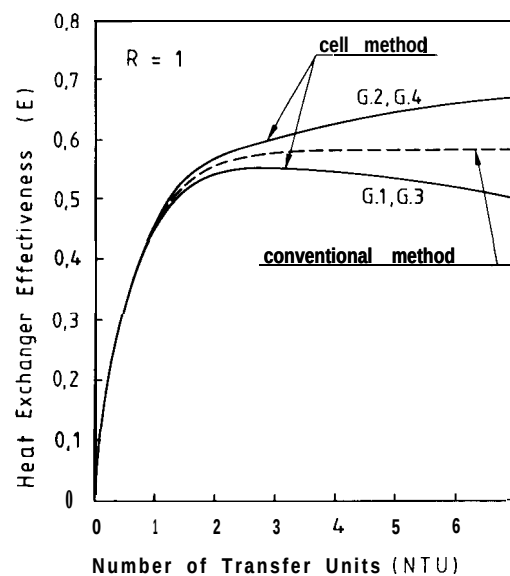


Figure 1 Comparison of the conventional method and the cell method for the heat exchanger treated in Sec. 1.6.3.

3. The conventional method shows a monotonic dependence of the heat exchanger effectiveness on the number of transfer units. The cell method shows a similar behavior for flow configurations G.2 and G.4; the other two flow configurations (G.1 and G.3) have a maximum at about $NTU = 3$.

4. At low NTU values both methods predict practically the same heat exchanger effectiveness. The differences between the higher and the lower values obtained by the cell method diminish.

Nomenclature and References for Section 1.6 appear at the end of Section 1.6.9.

1.6.9

General remarks

E. S. Gaddis

■ The numerical examples presented in Secs. 1.6.3 and 1.6.6 are for the case where the cell effectiveness is constant. This implies that the number of transfer units in a cell NTU, is constant. To accommodate the shell nozzles, the designer is obliged in many cases to use a spacing between the baffles and the tubesheets which is different from the spacing between the neighboring baffles. In such cases, and in many other cases, the overall heat transfer coefficient and the cell heat transfer area are different in the different cells. This leads to different numbers of transfer units NTU, and different cell effectiveness values E_c . However, the calculation procedure does not change.

The cell method was applied in Sec. 1.6.3 for one of the most common shell configurations (TEMA: E shell) and extended in Sec. 1.6.6 for the case where the shell is provided with a longitudinal baffle (TEMA: F shell). The cell method can be applied equally well for other shell flow configurations with segmental baffles, e.g., shells with divided flow (TEMA: J shell). The heat exchanger may be divided into a number of cells; the inlet and outlet temperatures in the different cells are then related to one another, as in the cases discussed, by using the information obtained from the flow configuration in the layout of the heat exchanger considered.

Nomenclature and References for Section I. 6 appear at the end of Section I. 6.9.

NOMENCLATURE FOR SECTIONS 1.6.1-1.6.9

a	dimensionless constant [Eq. 1.6.2(8)]		position of the inlet nozzles for the two streams—G.1, G.2, G.3, and G.4 [Fig. 1.6.4(1)]
A	total heat transfer area in the heat exchanger, m^2	M	number of passes for stream 1
A_c	heat transfer area in a cell, m^2	$\dot{M}$	mass flow rate, kg/s
b	dimensionless constant [Eq. 1.6.2(9)]	N	number of passes for stream 2
c	constant-pressure specific heat capacity, J/kg K	N_c	number of cells in a heat exchanger
$\dot{C}$	heat capacity rate ($=\dot{M}c$), W/K	NTU	number of transfer units of a heat exchanger, dimensionless [Eq. 1.6.8(2)]
e	dimensionless constant [Eq. 1.6.2(10)]	NTU,	number of transfer units of a cell, dimensionless [Eq. 1.6.7(2)]
E	heat exchanger effectiveness, dimensionless	$\dot{Q}_c$	rate of heat exchange in a cell, W
E_c	cell effectiveness, dimensionless	R	ratio of heat capacity rates of two streams, dimensionless [Eq. 1.6.2(12)]
E_u	unit effectiveness, dimensionless	T	temperature, $^{\circ}C$
E_p	effectiveness of a part heat exchanger composed of a number of cells, dimensionless		
f	dimensionless constant [Eq. 1.6.2(11)]		
G	geometry of a heat exchanger defining the relative		

U	overall heat transfer coefficient, $\text{W/m}^2 \text{K}$	$1i$	for stream 1 at heat exchanger inlet
Θ	dimensionless temperature [Eq. 1.6.2(5)]	$1o$	for stream 1 at heat exchanger outlet
Superscripts		$2i$	for stream 2 at heat exchanger inlet
		$2o$	for stream 2 at heat exchanger outlet
'	at cell inlet	i	for cell i or unit i , where i stands for a number
"	at cell outlet	$i-j$	for a part heat exchanger from cell i to cell j , where i and j stand for numbers and $j > i$
Subscripts			
1	for stream 1 (stream with lower $\dot{C}$)		
2	for stream 2 (stream with higher $\dot{C}$)		

REFERENCES FOR SECTIONS 1.6.1-1.6.9

1. Gaddis, E. S., and Schliinder, E. U., Temperaturverlauf und übertragbare Wärmemenge in Röhrenkesselapparaten mit Umlenkblechen, *Verfahrenstechnik*, vol. 9, no. 12, pp. 617-621, 1975.
2. Gaddis, E. S. Über die Berechnung des Austauschereffizienzes und der mittleren Temperaturdifferenz in mehrgängigen Rohrbündelwärmeaustauschern mit Umlenkblechen, *Verfahrenstechnik*, vol. 12, no. 3, pp. 144-149, 1978.
3. Gaddis, E. S., and Vogelpohl, A. Über den Wirkungsgrad und die mittlere Temperaturdifferenz von Rohrbündelwärmeübertragern mit Umlenkblechen und zwei rohreseitigen Gängen, *Chem. Eng. Process.*, vol. 18, pp. 269-273, 1984.
4. *VDI Wärmeatlas*, Sec. N, 3d ed., VDI-Verlag, Dusseldorf, 1977.
5. Kays, W. M., and London, A. L., Compact *Heat Exchangers*, 3d ed., McGraw-Hill, New York, 1984.



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