# 1.1. Basic Mechanisms of Heat Transfer

The basic mechanisms of heat transfer are generally considered to be conduction, convection, boiling, condensation, and radiation. Of these, radiation is usually significant only at temperatures higher than those ordinarily encountered in tubular process heat transfer equipment; therefore, radiation will not be considered in any great detail in this Manual. All of the others play a vital role in equipment design and will frequently appear in the discussion. In this section, the emphasis will be upon a qualitative description of the processes and a few very basic equations.

## *1.1.1. Conduction*

### Mechanism

Conduction in a metallic solid is largely due to the random movement of electrons through the metal. The electrons in the hot part of the solid have a higher kinetic energy than those in the cold part and give up some of this kinetic energy to the cold atoms, thus resulting in a transfer of heat from the hot surface to the cold. Since the free electrons are also responsible for the conduction of an electrical current through a metal, there is a qualitative similarity between the ability of a metal to conduct heat and to conduct electricity. In addition, some heat is transferred by interatomic vibrations.

### Fourier Equation

The details of conduction are quite complicated but for engineering purposes may be handled by a simple equation, usually called Fourier's equation. For the steady flow of heat across a plane wall (Fig. 1.1) with the surfaces at temperatures of $T_1$, and $T_2$ where $T_1$ is greater than $T_2$ the heat flow Q per unit area of surface A (the heat flux) is:

$$\frac{Q}{A} = q = k\left(\frac{T_1 - T_2}{X_1 - X_2}\right) = k\frac{\Delta T}{\Delta X} \tag{1.1}$$

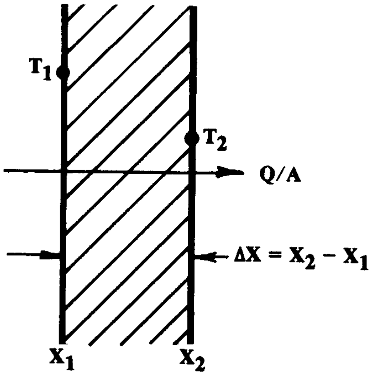

*Fig. 1.1 Diagram of Conduction Through a Plane Wall.*

The quantity k is called the thermal conductivity and is an experimentally measured value for any material. Eq. (1.1) can be written in a more general form if the temperature gradient term is written as a differential:

$$\frac{Q}{A} = -k\frac{dT}{dx} \tag{1.2}$$

The negative sign in the equation is introduced to account for the fact that heat is conducted from a high temperature to a low temperature, so that $\left(dT/dx\right)$inherently negative; therefore the double negative indicates a positive flow of heat in the direction of decreasing temperature.

### Conduction Through a Tube Wall

The main advantage of Eq. (1.2) is that it can be integrated for those cases in which the cross-sectional area for heat transfer changes along the conduction path. A section of tube wall is shown in Fig. 1.2. Q is the total heat conducted through the tube wall per unit time. At the radial position r in the tube wall ($r_i \le r \le r_o$), the area for heat transfer for a tube of length L is $A = 2\pi rL$. Putting these into Eq. (1.2) gives

$$\frac{Q}{2\pi rL} = -k\frac{dT}{dx} \qquad (1.3)$$

which may be integrated to

$$Q = \frac{2\pi L\ k(T_i - T_o)}{\ln(r_o/r_i)} \qquad (1.4)$$

If $T_i < T_o$, Q comes out negative; this just means that the heat flow is inward, reversed from the sense in which we took it. For thin-walled tubes, the ratio of the outer to the inner radius is close to unity, and we can use the simpler equation,

$$Q = \frac{2\pi r_o L\ k(T_i - T_o)}{r_o - r_i} \qquad (1.5)$$

with very small error.

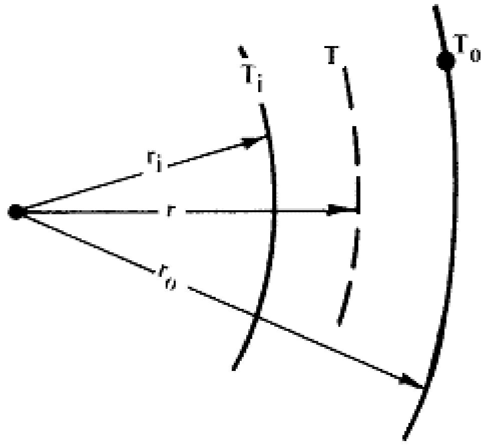

*Fig. 1.2 Diagram of Conduction Through a Cylindrical Wall.*

## Conduction Through a Bimetallic Wall

Sometimes, for reasons of corrosion, strength and/or economy, a tube is actually constructed out of two tightly fitting concentric cylinders of different metals as shown in Fig. 1.3. Note that r' is the outside radius of the inner tube **and** the inside radius of the outer tube, and T' is the corresponding temperature. From Eq. 1.4 we may write directly for the inner tube:

$$Q = \frac{2\pi L\ k_i(T_i - T')}{\ln(r'/r_i)} \qquad (1.6)$$

and for the outer tube:

$$Q = \frac{2\pi L\ k_o(T' - T_o)}{\ln(r_o/r')} \qquad (1.7)$$

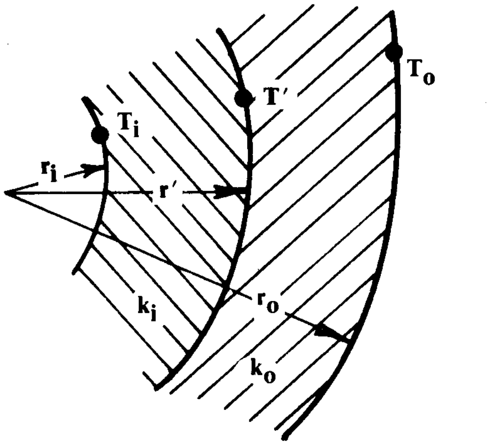

*Fig. 1.3 Diagram of Conduction Through Two Cylindrical Walls in Contact.*

Since the same amount of heat must flow through both tubes, the Q's are equal. Then the equations can be combined to eliminate the unknown temperature T', and Q is given by:

$$Q = \frac{T_i - T_o}{\dfrac{\ln(r'/r_i)}{2\pi L\ k_i} + \dfrac{\ln(r_o/r')}{2\pi L\ k_o}} \qquad (1.8)$$

**Contact Resistance**

In the previous section, the assumption was made that the outer surface of the inner cylinder and the inner surface of the outer cylinder were at the same temperature, implying that there was no resistance to heat transfer between the two. This assumption is essentially correct if the two surfaces are metallurgically bonded to one another, as can be achieved when one material is fused to the other or when they are bonded by a detonation wave. The assumption can be seriously in error if the two surfaces are merely in close physical contact, even at the very high pressures that can be exerted by shrink-fitting.

Practical metal surfaces generally have roughnesses ranging from 10 to 180 microinches, the degree and the form of the roughness depending upon the metal and the method of forming the surface. When two such surfaces are placed in contact, the "hills" are touching while the "valleys" are filled with the ambient atmosphere, usually air. Because of the low thermal conductivity of gases, practically all of the heat is conducted through the points in metal-to-metal contact. At low pressures, this will be only a small portion of the surface perhaps less than one percent and the resulting constriction of the heat flow lines can lead to an interface resistance several times greater than in the metal slabs themselves.

At higher contact pressures between the surfaces, the hills are flattened to give a greater surface area in contact in order to sustain the load, and the interface resistance decreases. Various methods are used to ensure good thermal contact including co-extrusions and shrink-fitting, but repeated thermal cycling in the normal operation of process equipment, together with creep, can cause long-term serious loss of efficiency.

Unfortunately it is almost impossible to predict contact resistance in process equipment applications. Here, as in all process equipment design, the engineer must assess the consequences of being wrong and the possible alternatives.

## *1.1.2. Single Phase Convection*

**Fluid Motion Near a Surface**

Convective heat transfer is closely connected to the mechanism of fluid flow near a surface, so the first matter of importance is to describe this flow.

Single phase flow must be characterized by both the geometry of the duct through which the flow occurs and by the flow regime of the fluid as it goes through the duct. There are two basically different types of duct geometry: constant cross-section, in which the area available for flow to the fluid has both the same shape and the same area at each point along the duct, and varying cross-section, in which the shape and/or the area of the duct vary with length, usually in a regular and repeated way.

The most common constant cross-section duct geometry that one deals with in process heat transfer applications is the cylindrical tube. In a cylindrical geometry, it is assumed that all parameters of the flow are a function only of the radial distance from the axis of the cylinder (or equivalently from the wall) and of the distance from the entrance (entrance effects).

For flow in ducts of varying cross section (flow across tube banks is the case of interest here), another phenomenon occurs that is of the utmost importance in calculating the pressure drop and which adversely affects the efficiency of conversion of pressure drop to heat transfer. Whereas in a duct of constant cross section only the pressure effect due to the friction of the fluid moving relative to the surface (skin friction) needs to be taken into account, in a bank of tubes **form drag** is a major contributor to the pressure drop. Form drag arises when the pressure on the front surface of a tube is greater than the pressure on the back surface. This pressure difference appears as part of the total pressure drop across the tube bank; unlike skin friction, form drag is not very efficient in promoting heat transfer. Form drag also gives rise to boundary layer separation and wake formation, which may cause destructive vibration of the tubes in a tube bank.

The type of flow in a duct can also be characterized by the flow regime; that is, laminar flow, turbulent flow, or some transition state having characteristics of both of the limiting regimes. All of the exact definitions of laminar flow are very complex, and an illustration (like Fig. 1.4) is much more useful.

If we have a round tube with a liquid flowing in it at a steady rate, and if we inject a dye trace with a needle parallel to the axis of the tube, one of two things can happen:

1) The dye trace may flow smoothly down the tube as a well-defined line, only very slowly becoming thicker, or
2) The dye trace may flow irregularly down the tube moving back and forth across the diameter of the tube and eventually becoming completely dispersed.

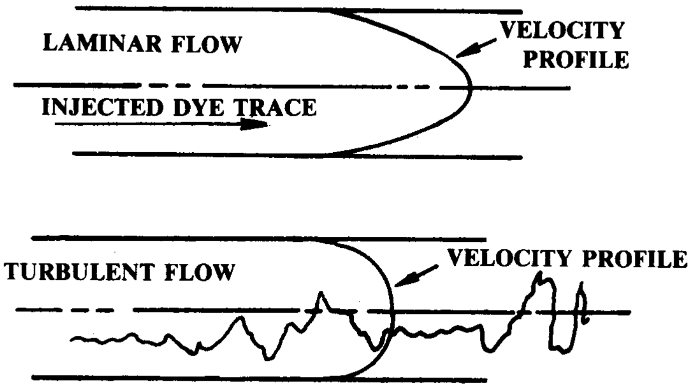

*Fig 1.4 Illustration of the Difference Between Laminar and Turbulent Flow.*

The first case is laminar flow and the second is turbulent flow. Laminar flow corresponds to the smooth movement of layers of fluid past one another without mixing; turbulent flow is characterized by a rapid exchange of packets or elements of fluid in a radial direction from one part of the flow field to another through turbulent eddies.

There are differences in the velocity pattern also. In laminar flow, the velocity at a given point is steady, whereas in turbulent flow the velocity fluctuates rapidly about an average value. If one measures the local velocity at various positions across the tube, one finds that laminar flow gives a parabolic velocity distribution whereas turbulent flow gives a blunter velocity profile, as shown in Fig. 1.4. In both flows, the fluid velocity is zero at the wall and a maximum at the centerline.

The flow regime that exists in a given case is ordinarily characterized by the Reynolds number. The Reynolds number has different definitions for flow in different geometries, but it is defined as in Eq. 1.9 for flow inside tubes:

$$\mathrm{Re} = \frac{d_i \rho V}{\mu} = \frac{d_i G}{\mu} \tag{1.9}$$

where $d_i$ is the inside diameter of the tube, V is the average velocity in the tube, $\rho$ the density of the fluid, and $\mu$ the viscosity of the fluid. Laminar flow is characterized by low Reynolds numbers, turbulent flow by high Reynolds Numbers.

For flow inside tubes, Reynolds numbers below about 2,000 result in laminar flow being the stable flow regime. Reynolds numbers above about 2100 give turbulent flow for pressure drop calculations, while Reynolds numbers above 10,000 give turbulent flow for heat transfer. The range from 2100 to 10,000 is generally referred to as the transition flow regime for heat transfer. Other geometries have different Reynolds number ranges to characterize flow regimes.

## Heat Transfer to a Flowing Fluid (Convection)

Convection heat transfer can be defined as transport of heat from one point to another in a flowing fluid as a result of macroscopic motions of the fluid, the heat being carried as internal energy. The convection process has received a great deal of both experimental and analytical attention and, although we are mainly concerned with using the results of these studies, a cursory look should be taken at the physical process of convection, both to define terms and to establish some intuitive sense of what really the correlations we use are trying to represent.

In laminar flow past a cold wall the heat is transferred to the tube surface from the fluid next to the wall. Within the fluid, heat is transferred from "layer" to "layer" of the fluid by conduction. There are no fluid motions perpendicular to the direction of flow to transport the heat by any other mechanism. Since the different "layers" of fluid are moving at different velocities, however, the conduction process is much more complex to analyze than for the solid wall previously discussed.

If we look at a fluid in turbulent flow past a cold surface and mark a few representative elements of fluid in order to trace their paths, we would obtain a picture something like Fig. 1.5.

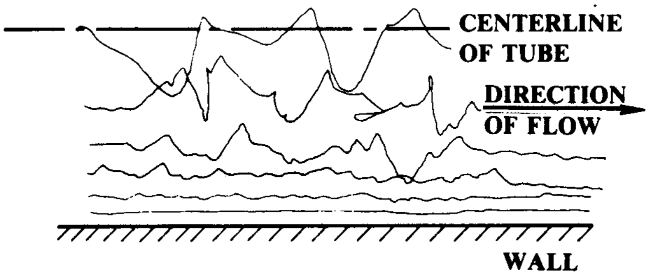

*Fig. 1.5 Schematic Diagram of Turbulent Fluid Flow.*

The corresponding time-averaged velocity and temperature profiles might look like Figs. 1.6 and 1.7.

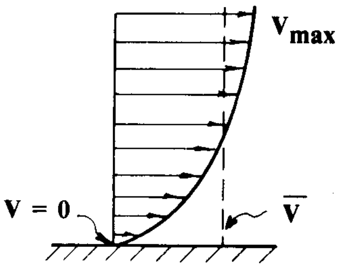

*Fig. 1.6 Time-Averaged Velocity Profile in Turbulent Flow.*

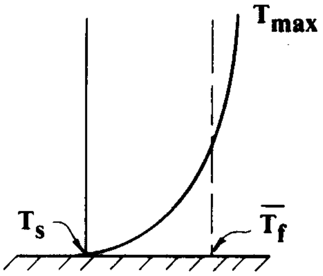

*Fig. 1.7 Time-Averaged Temperature Profile in Turbulent Flow.*

The flow near the wall has only a few small eddies, so that the predominant mechanism for heat transfer is conduction. At the wall, the fluid velocity is zero and the fluid temperature is the same as the wall. The velocity and temperature gradients near the wall are much steeper than those in the bulk flow where eddy transport becomes dominant. It is important to note that when we refer without further qualification to the velocity or temperature of a stream, we mean the volume-mean or **mixing-cup** values shown on the figures as $\overline{V}$ and $\overline{T}_f$. (We will drop the bars henceforth.) However, it is important to remember that some portions of the fluid are at possibly significantly higher or lower temperatures, where thermal degradation or phase change might occur. For the case shown, if the fluid had a freezing temperature between $T_s$ and $T_f$, a layer of solid would form on the wall, resulting in a major change in the heat transfer and fluid flow mechanisms.

## Film Heat Transfer Coefficients

For many convective heat transfer processes, it is found that the local heat flux is approximately proportional to the temperature difference between the wall and the bulk of the fluid, i.e.,

$$\frac{Q}{A} \propto (T_f - T_s) \qquad (1.10)$$

which causes us to define a constant of proportionality, called the "film coefficient of heat transfer," usually denoted by h:

$$\frac{Q}{A} = h(T_f - T_s) \qquad (1.11)$$

The value of h depends upon the geometry of the system, the physical properties and flow velocity of the fluid.

The concept of a heat transfer coefficient is useful to the designer only if there exists a quantitative relationship between these variables and the heat transfer coefficient. It is important also that this relationship be reasonably valid for the conditions existing in the particular application. These relationships, or correlations, may come from either theoretical or experimental studies, or from a combination of both. The correlation may be expressed as an equation, a graph, a table of values, or a computational procedure. These forms are more or less readily convertible from one into another according to the needs and convenience of the user. In using the correlation, the designer needs to know, at least roughly, how accurate the results are likely to be in his application.

We sometimes employ a film coefficient in cases where the flux is not even approximately proportional to the temperature difference (e.g., nucleate boiling), contradictory to the implication of Eq. (1.10). This practice offers a useful basis of comparison of the relative resistances of the several heat transfer processes in a given problem, but has no fundamental significance.

### 1.1.3. Two Phase (Liquid-Gas/Vapor) Flow

**Regimes of Two-Phase Flow**

In the present context, two-phase flow will usually refer to the simultaneous flow of a liquid and a gas or vapor through a duct. Such a flow occurs when a vapor is being condensed or a liquid is being vaporized; less commonly, a two-phase flow may involve a gas-liquid mixture (such as air and water) flowing together and being heated or cooled without any appreciable change of phase. The actual two-phase flow configuration, or regime, existing in a conduit in a given case depends upon the relative and absolute quantities and the physical properties of the fluids flowing, the geometric configuration of the conduit, and the kind of heat transfer process involved, if any. We will first consider the flow regimes observed by Alves (1) in his study of air-water flows in horizontal tubes. These regimes are diagrammed in rather idealized form in Fig. 1.8. The chief difference between flow regimes studied in a non-heat transfer situation and those existing during condensation is that a liquid film exists on the entire surface of the conduit during condensation. However, we may presume that this thin film of draining condensation does not cause any vital difference in the interaction between the vapor and the main inventory of liquid.

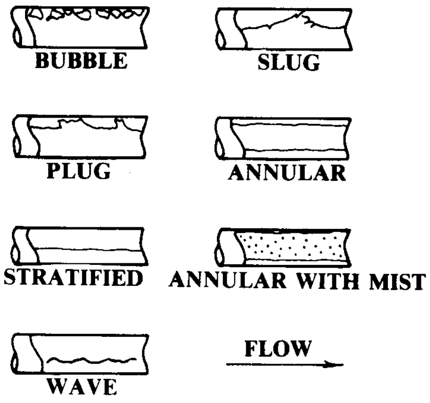

*Fig. 1.8 Flow Pattern Sketches for Two-Phase Adiabatic Flow in a Horizontal Pipe. (Slightly modified from Ref. 1)*

For boiling flows, there are at least two additional flow regimes required. In one, bubbles are formed on the walls which disrupt the flow pattern in the immediate vicinity of the wall. In order for this to occur, the wall must be above the boiling point (though this does not guarantee that bubbles will form); if the bulk fluid temperature is saturated, the bubbles will be carried off downstream very much like the bubble flow regime shown in the Fig. 1.8, but if the bulk flow is subcooled, the bubbles will quickly collapse. In the latter case, there will be only a relatively small effect upon pressure drop. The second new flow regime is mist flow in which the liquid film on the wall in mist-annular flow has been evaporated and all the remaining liquid is carried as droplets in the vapor stream.

We may view the flow regime as a consequence of the interaction of two forces, gravity and vapor shear, acting in different directions. At low vapor flow rates, gravity dominates and one obtains stratified, slug-plug, or bubble flow depending upon the relative amount of liquid present. At high vapor velocities, vapor shear dominates, giving rise to wavy, annular, or annular-mist flows. It would be desirable to have some way to predict the flow regime a priori, and many attempts have been made to do this in a general and consistent way. No attempt has succeeded, but the work of Baker (2) is considered to be generally the best available in the open literature even though it is a dimensional representation and defies explanation in fundamental terms. The Baker map is shown in modified form as Fig. 1.9 and is useful in giving a general appreciation of the general kind of flow regime existing under given conditions.

For two-phase flow inside vertical tubes, the stratified and wavy flow regimes cannot exist, and the flow regimes generally recognized in this case are bubble, slug, annular, and annular with mist. The Fair map (3) (Fig. 1.10) is generally recognized as the best for this configuration and is reproduced here in slightly modified form. It was originally developed for the analysis of thermosiphon reboilers and basically refers to a liquid stream entering at the bottom of the tube and boiling as it flows upward.

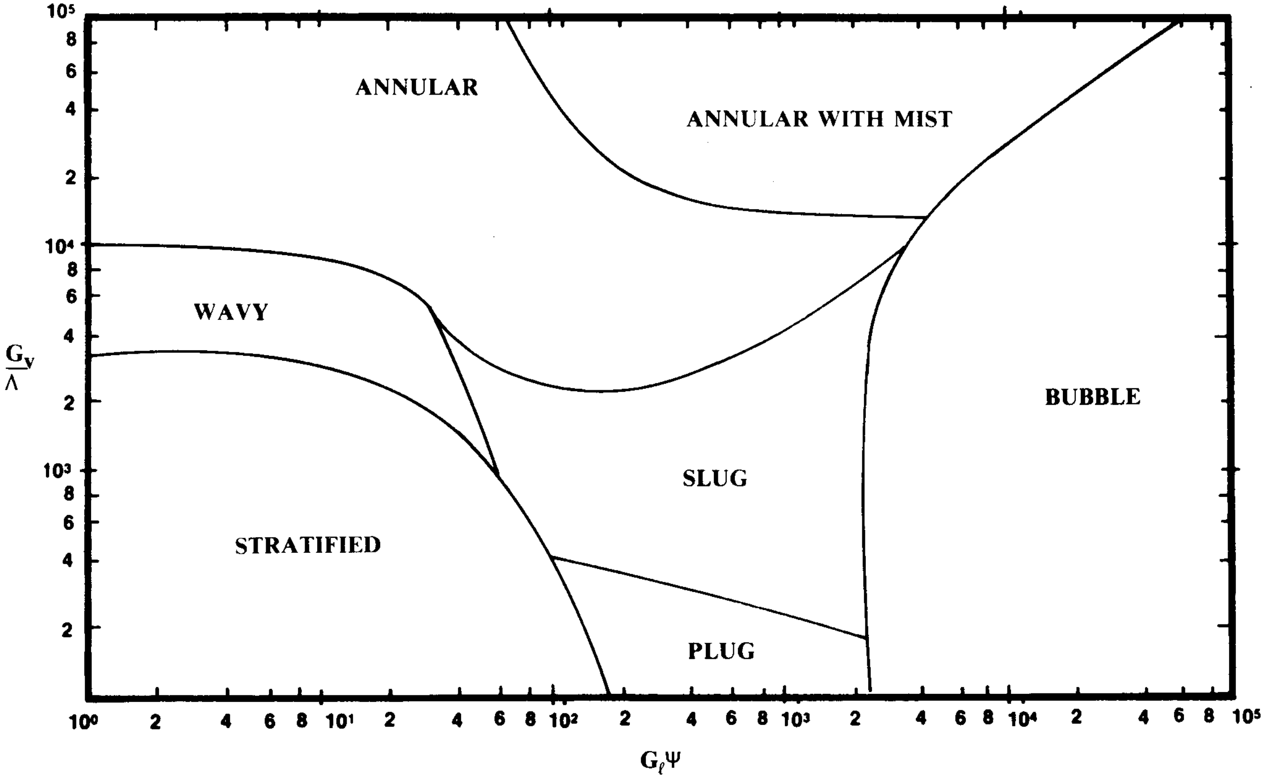

*Fig. 1.9 Transformed Baker Flow Regime Map.*

For condensation inside a vertical tube, the vapor generally enters at the top and the two-phase mixture resulting from condensation flows downward. At high condensing rates, where vapor shear dominates, most of the flow is in the annular flow regime. Fig. 1.10 can be used to estimate when this assumption breaks down.

Our knowledge of two-phase flow patterns across tube banks, with or without baffles, is much more limited and indeed hardly extends beyond what intuition tells us. Fortunately, at least some correlations of pressure drop and related hydrodynamic effects are largely independent of a knowledge of flow pattern, and it is possible to make some quantitative calculations.

**Heat Transfer to a Two-Phase Flow** The analysis of heat transfer to or from a two-phase flow is quite complex, involving the properties, quantities, and fluid mechanics of both phases. The design correlations resulting from these analyses are also subject to greater error than those for single phase heat transfer.

For a gas-liquid flow involving no change of phase as a result of heat transfer, the flow is usually turbulent. The heat is transferred by turbulent eddies within each phase and across the gas-liquid and fluid-solid interfaces by turbulent boundary layer phenomena similar to turbulent flow in a pipe. The rate of heat transfer (and the pressure drop) is relatively high because of the strong turbulence created by the gas-phase shear on the liquid.

Where phase changes are also involved, as in vaporization or condensation, additional heat transfer mechanisms come into play. These are discussed in more detail in subsequent sections.

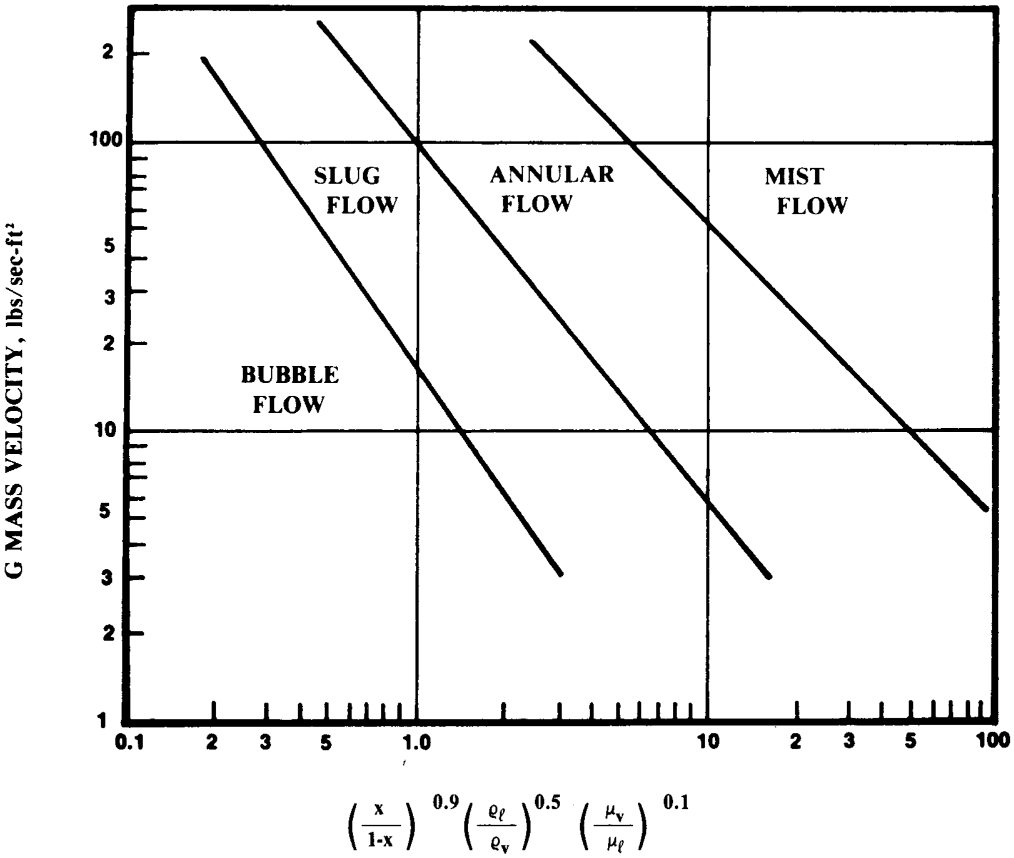

*Fig. 1.10 Modified Fair Flow Regime Map for Vertical In-Tube Flow.*

**Phase Relationships in Two-Phase (Liquid Gas/Vapor) Flow** Two phase flows must also be characterized in terms of the composition and the resulting thermodynamic relationships between the two phases. Four cases can be distinguished:

a. The liquid and the gas are different pure components. The classic example is an air-water mixture, which is not a common industrial problem, but is very important because a great deal of what is known about two-phase flow has been determined on this system. While in general this information can be carried over to condensation and some boiling work, there are important differences that must be recognized and allowed for. Thermodynamically, the pressure and temperature can be independently varied over wide ranges in this system.

b . The liquid and gas (vapor) are the same pure component. This is a common case in condenser design, occurring for example in condensers on columns separating and/or purifying a product. The pressure-temperature relationship in this case is the vapor pressure curve for the component.

c. The liquid and gas (vapor) are multicomponents. The thermodynamic relationships are more complex, the temperature, for example, being variable over a range of values at a given pressure, but with a changing ratio of total liquid to total vapor and with changing composition of each phase. Prediction of the amount and composition of each phase is relatively well understood and easily done in few cases, as for mixtures of light hydrocarbons; other cases require laboratory thermodynamic data.

d. This case is identical to N or (c), but with a non-condensable gas present. e.g. [air in steam, or a solvent in an inert stripping gas]. This case is thermodynamically similar to (c), but the non-condensable gas does not appear in the liquid phase.

### Other Two-Phase Systems

Other two-phase systems occur in industrial processing and are briefly described below. In general, less is known about these systems than for liquid-gas/vapor systems, and the correlations are correspondingly fewer and less accurate.

1. **Solid-gas systems**. The most important application of solid-gas mixtures is in fluidized beds, in which the gas is introduced at the bottom of the bed and flows upwards. If the gas flow rate is great enough, the forces on the solid particles (usually finely divided, such as catalyst particles or sand) cause the bed to expand. Then the particles no longer rest directly on one another and circulate freely through the bed; bubbles of gas rise through the bed and break through the upper surface, giving an appearance similar to boiling of a liquid. Heat transfer surface in the form of tubes may be put into the bed, and heat may also be transferred through the walls that contain the bed. Heat transfer rates in a fluidized bed are generally much higher than for a comparable flow of gas only.

   If the gas rate is increased further, the solid particles are carried along with the gas. This is often done deliberately to give a "transfer line fluidized catalytic reactor." Heat transfer mechanisms in this case are qualitatively similar to those in gas liquid flow in a pipe, except that now the "eddies" in the solid phase are composed of masses of discrete particles all moving together, rather than a macroscopically continuous phase.

2. **Solid-liquid systems**. Liquid fluidized beds are also known; the major difference is that solid and liquid densities are fairly similar, so the solid is more easily fluidized and transported with the liquid. Heat transfer rates are only marginally greater than for the comparable liquid-only flow.

3. **Liquid-liquid systems**. Mixtures of two immiscible liquids are sometimes encountered in heat exchangers and flow patterns similar to those for gas-liquid flows are observed. The heat transfer coefficients for these systems are generally intermediate between the values that would be observed for the single liquids flowing alone at, the same average velocity.

## 1.1.4. Condensation

### Modes of Condensation

Condensation is the process by which a vapor is changed to a liquid by removing the latent heat of condensation from the vapor. There are four basic modes or mechanisms of condensation generally recognized: dropwise, filmwise, direct contact, and homogeneous.

1. **Dropwise condensation**. In dropwise condensation, the drops of liquid form from the vapor at particularly-favored locations, called nucleation sites, on a solid surface. These sites may be pits or scratches or any surface irregularity and there may be many thousands of them per square inch. These drops grow by continued condensation from the vapor and by agglomeration of adjacent drops when they come into contact. Dropwise condensation

occurs only on surfaces which are not strongly wetted by the liquid and so the drops do not spread out over the surface. They grow in place until they become so large that they run off the surface by gravity or are blown off by the flowing vapor.

While dropwise condensation is alluring because of the high coefficients reported, it is not considered at this time to be suitable for deliberate employment in process equipment. Generally, contaminants must be continuously injected into the vapor, or special surface materials (often of low thermal conductivity) employed. Even so, the process is unstable and unpredictable, and of questionable efficacy under conditions of high vapor velocity and industrial practice.

2. **Filmwise condensation.** In filmwise condensation, the drops initially formed quickly coalesce to produce a continuous liquid film on the surface through which heat must be transferred to condense more liquid. The actual heat transfer mechanism that operates in filmwise condensation is closely related to the two-phase flow mechanisms described in the previous section. Filmwise condensation is the usual mode that occurs in practice and that is assumed to exist for condenser design calculations.

3. **Direct contact condensation.** In direct contact condensation, the liquid coolant is sprayed directly into the vapor, which condenses directly onto the surface of the spray drops. Direct contact condensation is a very efficient process, but it results in mixing the condensate and coolant. Therefore it is useful only in those cases where the condensate is easily separated, or where there is no desire to reuse the condensate, or where the coolant and condensate are the same substance.

4. **Homogeneous condensation**. In homogenous condensation, the liquid phase forms directly from supersaturated vapor, away from any macroscopic surface. Since this requires subcooling the vapor on the order of a hundred degrees below the saturation temperature and since homogeneous condensation is actually observed at much smaller subcoolings, it is generally assumed that in practice there are sufficient numbers of dirt or mist particles present in the vapor to serve as nucleation sites. Homogeneous condensation is primarily of concern in fog formation in equipment and is not a design mode.

## Condensation Outside Tubes.

A common condenser design has the vapor outside a bank of tubes, with cold coolant flowing inside the tubes. The

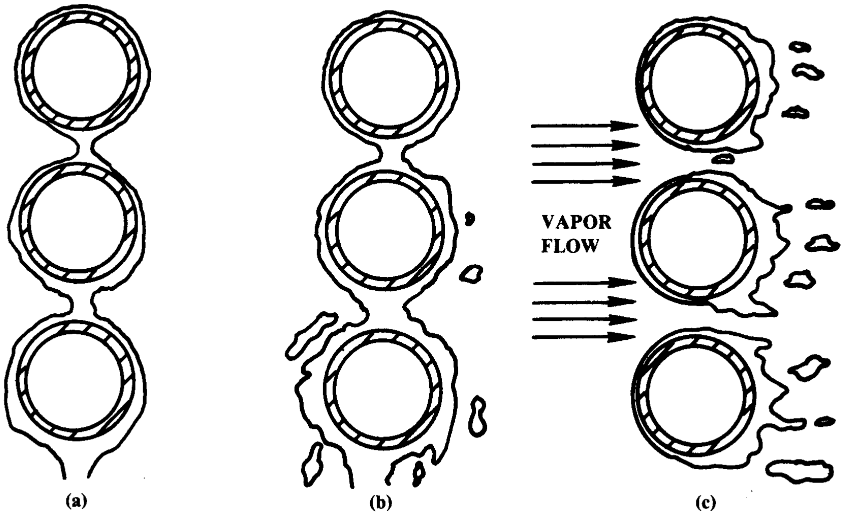

*Fig. 1.11 (a) Diagram of Condensate in Laminar Flow Outside a Bank of Tubes. (b) Diagram of Condensate in Turbulent Flow Outside a Bank of Tubes. (c) Diagram of Condensate in Turbulent Flow with Vapor in Crossflow.*

vapor condenses on the outside of the tubes in the filmwise mode. Then the film flows under the influence of gravity to the bottom of the tube and drips off on to the next tube lower in the bank. If the condensate is very viscous, this filmwise flow may be in the laminar flow regime at least for the first few tubes (Fig. 1. 11a). More commonly, the condensate is relatively inviscid (like water) and the film quickly becomes turbulent (Fig. 1.11b). The two cases lead to quite different predictions of the rate of heat transfer of the latent heat of condensation of the vapor through the liquid film.

If the vapor flow rate is high, vapor shear acting in crossflow on the condensate film becomes significant in blowing off the liquid, carrying it downstream as a spray, and causing the film to become turbulent even earlier than it would have under the influence of gravity alone (Fig.1. 11c).

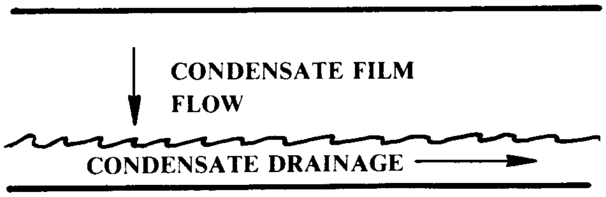

*Fig. 1.12a. Condensation in Horizontal Tubes in Gravity-Controlled Flow.*

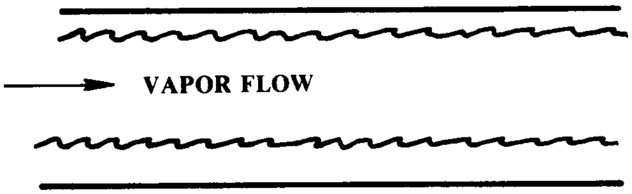

*Fig. 1.12b. Condensation in Horizontal Tubes in Vapor Shear-Controlled Flow.*

## Condensation Inside Tubes.

Another common condenser configuration, especially in air-cooled condensers, has the vapor condensing inside tubes. The tubes are usually horizontal or only slightly slanted downwards to facilitate drainage, but vertical and inclined tube arrangements are also used. If the vapor flow rate is very low, the condensate forms on the cold wall and drains under the influence of gravity into a pool in the bottom of the tube; this pool in turn drains by gravity out the exit end of the tube (Fig. 1.12a). This phenomenon occurs under conditions, which favor the stratified or wavy flow regimes (see Figs. 1.8 and 1.9).

## Condensation of Mixtures.

There are several important differences between the condensation of an essentially pure component and the condensation of a mixture. A basic analysis of the problem for a binary mixture was given by Colburn and Drew (4) who presented the diagram in Fig. 1.13. In a mixture, the heavier, less volatile components condense first, and only as the temperature of the remaining vapor is lowered do the lighter components condense. Thus, there are always sensible heat transfer effects to be considered in both the liquid and vapor phases. Because of the

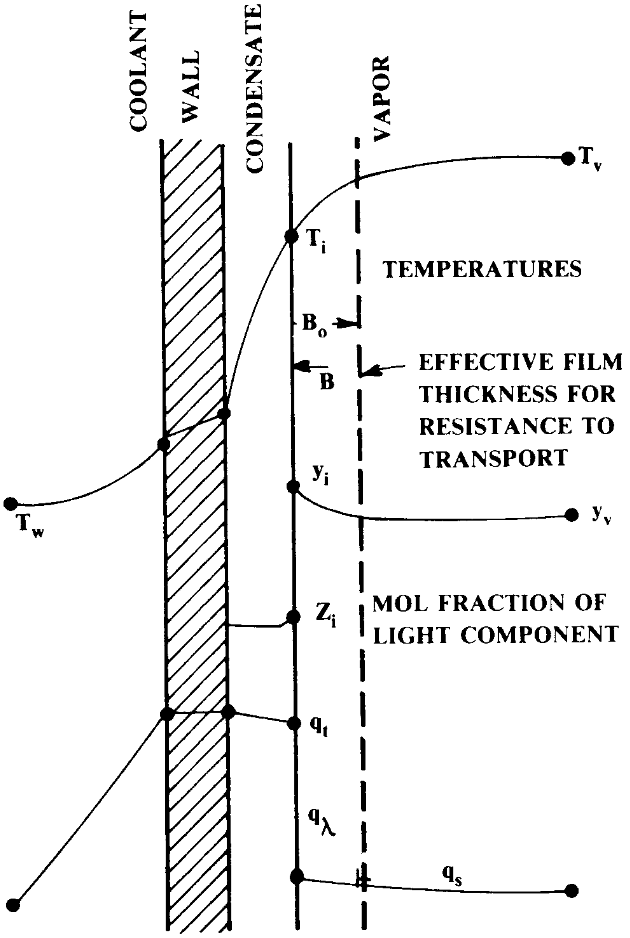

*Fig. 1.13 Temperature, Composition, and Heat Flux Profiles During Multicomponent Condensation.
(After Colburn and Drew (4))*

low heat transfer coefficient associated with cooling the vapor, this process becomes a major resistance to heat transfer in the process, indicated in the figure by the temperature difference $(T_v-T_i)$.

The heavier component is enriched in the vicinity of the interface compared to the bulk (as indicated by the concentration difference $(y_i-y_v)$) and counter-diffuses back into the bulk vapor. Since the compositions of both liquid and vapor phases are continually changing, there is the problem of changing physical properties to consider in evaluating the equations.

The problem becomes even more complex for multicomponent mixtures, and approximation methods must be used to design condensers under these conditions.

## *1.1.5. Vaporization*

### Mechanisms of Pool Boiling.

There are several mechanisms, or processes, through which a liquid at the saturation temperature may be converted to a vapor by the addition of heat. If the boiling or vaporization occurs on a hot surface in a container in which the liquid is confined, the process is called "pool boiling." There are several quite different mechanisms by which pool boiling occurs depending upon the temperature difference between the surface and the liquid, and to a lesser extent, upon the nature of the surface and the liquid. These mechanisms are best discussed in connection with a curve of heat flux to the liquid. The classic curve of heat flux vs. temperature difference between surface and liquid saturation temperature for saturated pool boiling is reproduced in Fig. 1.14. The coordinates are logarithmic and the values shown are typical of a light hydrocarbon.

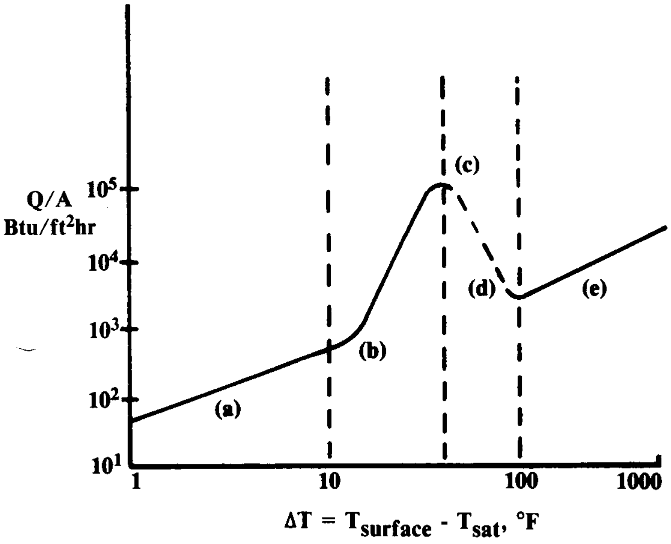

*Fig. 1.14 Typical Saturated Pool Boiling Curve*

The various regimes indicated on Fig. 1. 14 are:

(a) The natural convection regime characterized by a $\Delta T$ less than about 10° F. In this region, the liquid in contact with the hot surface is superheated and rises by natural convection to the surface between the vapor and liquid where the superheat is released by quiescent vaporization of liquid. There is no vapor bubble formation in the bulk of the liquid and the heat transfer coefficients are characteristic of those of natural convection processes.

(b) The nucleate boiling regime, in which vapor bubbles are formed at preferred nucleation sites - typically small pits or scratches - on the hot surface. The liquid is superheated by direct contact with the solid surface. Once a vapor nucleus forms at a nucleation site, the bubbles grow very rapidly by desuperheating the surrounding liquid until buoyant forces pull them free from the surface and cause them to rise to the vapor-liquid interface. There are various correlations that have been proposed in the literature for this region. However, because nucleate boiling phenomena are so strongly affected by the exact nature of the surface and the fluid, it is best to use experimental information when designing in this region.

(c) The point indicated by (c) is variously termed the maximum, peak, critical, or burnout heat flux: the highest attainable heat flux for any reasonable surface temperature. At this point, the release of vapor is so vigorous that the flow of liquid to the surface is just sufficient to supply the vapor. Any further increase in surface temperature results in some of the vapor generated being unable to escape and the heat flux falls off. The peak heat flux phenomenon is essentially a vapor hydrodynamic limit and is nearly independent of the exact nature of the surface. Various fundamentally-derived correlations of the peak heat flux exist.

(d) Transition boiling is an intermediate regime characterized by the occasional generation of a vapor film at the surface, which insulates the surface from the cooling liquid, leading to local hot spots and unstable operation. The film is unstable in the transition boiling regime, and after a short period of time the liquid will flood back to cool the surface and temporarily go back into the nucleate boiling regime. Heat transfer equipment is ordinarily never intended to operate in the transition regime.

(e) Film boiling is the boiling regime that is stable at large temperature differences between the surface and the saturation temperature. In film boiling, a stable, almost quiescent, film of vapor exists between the surface and the liquid pool. Heat transfers by conduction across the vapor to the liquid pool resulting in creation of more vapors. The film eventually becomes unstable and releases large vapor bubbles at relatively infrequent intervals. The bubbles rise through the pool to the interface. Film boiling is characterized by large temperature differences, generally very low heat fluxes, and correspondingly very low heat transfer coefficients. The surface may become hot enough to thermally degrade the substance being boiled. Fouling problems are also strongly accentuated in the film boiling regime because any fouling deposit that forms on the surface cannot be re-dissolved or washed away by liquid. In general, it is considered undesirable to operate in the film boiling regime.

## Vaporization During Flow.

The above boiling processes take place in a container or pool of liquid and are therefore referred to as pool boiling phenomena. Certain classes of vaporization equipment, notably thermosiphon and pump-through reboilers, operate with a net liquid velocity past the heat transfer surface. Under these conditions, the boiling processes are modified by a shear stress operating on the layer of liquid immediately adjacent to the hot surface. In general, natural convection boiling phenomena will be overshadowed by forced convection, and the nucleation process will be suppressed to some degree, possibly completely. With complete suppression, the superheated liquid is transported from the tube wall by turbulent eddies to the vapor-liquid interface, where vaporization takes place to form the vapor. The heat transfer coefficient under these conditions is greater than that which would exist if nucleate boiling only were operative. Film boiling is also possible under forced convection vaporization if the wall temperature is high enough. Frequently in these cases, one encounters mist flow, in which the liquid inventory is carried along in the vapor as tiny droplets, which are heated and vaporized by contact with the superheated vapor. This process is characterized by very low heat transfer coefficients and is never deliberately designed for in vaporization equipment.

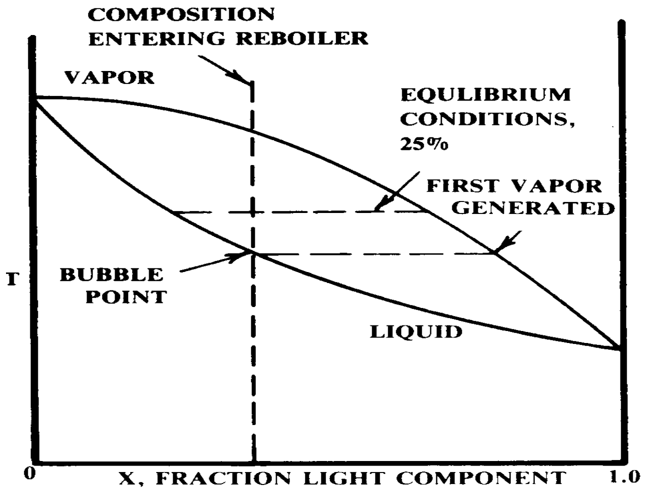

*Fig. 1.15 T-x Diagram for Vaporization of a Binary, Showing Equilibrium Conditions at Start of Vaporization and After 25% Boilup.*

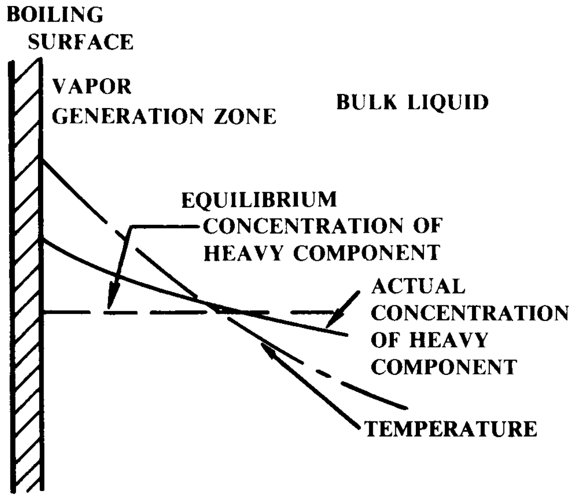

*Fig. 1.16 Temperature and Concentration Profiles After Significant Boilup for a Binary Mixture.*

Correlations exist which allow the designer to estimate the effect of flow phenomena on the boiling process and if these are found to be significant, to calculate the heat transfer coefficient under forced convection conditions.

**Vaporization of Mixtures.**
Mixtures present a serious problem in the design of reboilers. Figs. 1.15 and 1.16 illustrate the problem. When a wide boiling range mixture enters a reboiler, the first vapor to be generated is rich in the low boiling components, leaving behind a liquid which is enriched in the high boiling components, especially in the immediate vicinity of the heat transfer surface.

The light components then must diffuse through this barrier to the surface to vaporize. A greater driving force is required than is indicated by the mixed mean composition of the remaining liquid (the problem is entirely analogous to that of a wide condensing range mixture discussed in the previous section.)

If one measures the heat flux as a function of the boiling surface temperature and compares it to the saturation temperature for the bulk mixture, the apparent result is a decrease in heat transfer coefficient at compositions intermediate to the pure components, as illustrated in Fig 1.17. However, it would seem that the real effect is a distortion of the temperature driving force and that the heat transfer coefficient itself is in fact a monotonically changing function from the pure light component to the pure heavy component. This however does not alter the basic problem which is how does one calculate the "true boiling heat transfer coefficient" - Δ T product, which is the heat flux, and hence gives the amount of area required in the condenser.

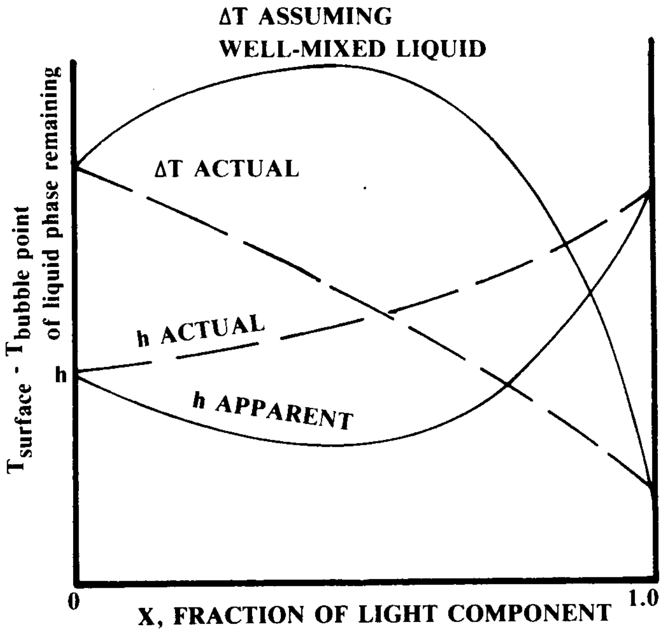

*Fig. 1.17 Typical Diagram of Apparent and Actual h and ΔT for Boiling a Binary, as a Function of Composition.*

## 1.1.6. Radiation

**Source of Radiant Energy.**
All matter constantly radiates energy in the form of electromagnetic waves. The amount of energy emitted depends strongly upon the absolute temperature of the matter and to a lesser extent upon the nature of the surface of the matter. The basic law of radiation was derived by Stefan and Boltzmann and may be written for our purposes as:

$$\frac{Q}{A} = \sigma \varepsilon T_{abs}^{4} \qquad (1.12)$$

where σ is the Stefan-Boltzmann constant (equal to 0. 1714 x $10^{-8}$ Btu/hr $ft^2$ $(°R)^4$) and $T_{abs}$ is the absolute temperature in °R. ε is the emissivity and has a value between 0 and 1. For a perfect reflector ε = 0 and for a perfect emitter, a so-called "black body", ε = 1. Highly polished metals have an emissivity of about 0.02 to 0.05, oxidized aluminum about 0.15, steel from 0.6 to 0.9 whether clean or rusted, and most paints from 0.8 to 0.98, largely independent of color.

Since all surfaces that radiate heat will also absorb heat, it follows that all surfaces that can "see" each other are exchanging heat with one another, the net rate depending upon the absolute temperatures, the emissivities, the areas and the spatial geometric relationships of the surfaces. The complete formulation of the problem is quite complex and will not be developed further here. Among the many excellent texts on the subject is "Radiation Heat Transfer" by L. Siegel and J. R. Howell (5).

## Radiation in Process Heat Transfer Problems

At usual atmospheric temperatures, radiant heat transfer is relatively unimportant compared to most other heat transfer mechanisms, though there are a few areas where it makes a significant contribution, e.g., the loss of heat from non-insulated steam lines. At higher temperatures, radiation becomes relatively more important, and at temperatures above perhaps 1000 - 1500 °F (depending upon the other processes), it is usually essential to take radiation into account.

However, for normal heat exchanger applications such elevated temperatures seldom exist. Therefore, the principles and equations of radiation heat transfer will not be developed further.