



5.3. Boiling Heat Transfer

The prediction of the heat transfer coefficient to boiling liquids is subject to large errors due to the inability to specify, manufacture, and maintain the nucleation characteristics of surfaces, as was illustrated by Figures 5.2 and 5.5. However, the boiling heat transfer coefficient is only part of the overall coefficient and the effect of these large errors is reduced in the overall design but must be considered in selection of safety factors or the selection of operating parameters; e.g., using only a fraction of the available steam pressure. Boiling heat transfer has been studied extensively and good summaries of these researches are found in (9, 10, 21, 22).

The design procedures and equations are different for boiling outside of tubes or pool boiling and for boiling inside of tubes. A single tube boiling heat flux curve is a basic starting point and the development of this curve is as follows. A natural convection coefficient equation is used to generate the natural convection heat flux curve by

$$q_{nc} = h_{nc}\Delta T \quad (5.6)$$

and plotted in Figure 5.17 as line OA. Then a nucleate boiling heat flux curve is calculated and plotted as line AB. The intersection of these lines is point A. A calculation of the maximum flux determines point B on the nucleate curve. Usually these two curves are sufficient but if the critical ΔT at point B can be exceeded, then the minimum flux and the corresponding temperature difference is calculated and establishes point C and the film boiling flux curve CD drawn. No predictive equations exist for the intermediate region (BC) but a straight line is drawn between these points. What we now have is an approximation of the curve in Figure 5.1 by a series of straight lines. The real curve (as in Figure 5.1) has a smooth transition between these straight lines but in design these transition curves are ignored and results in a slight conservatism.

5.3.1. Pool Boiling - Single Tube

(a) Natural Convection

When the wall temperatures are too low to initiate nucleate boiling the heat transfer coefficient is based on the liquid natural convection coefficient where the ΔT is based on the difference between the wall and the liquid temperatures. For horizontal tubes the following equation is used

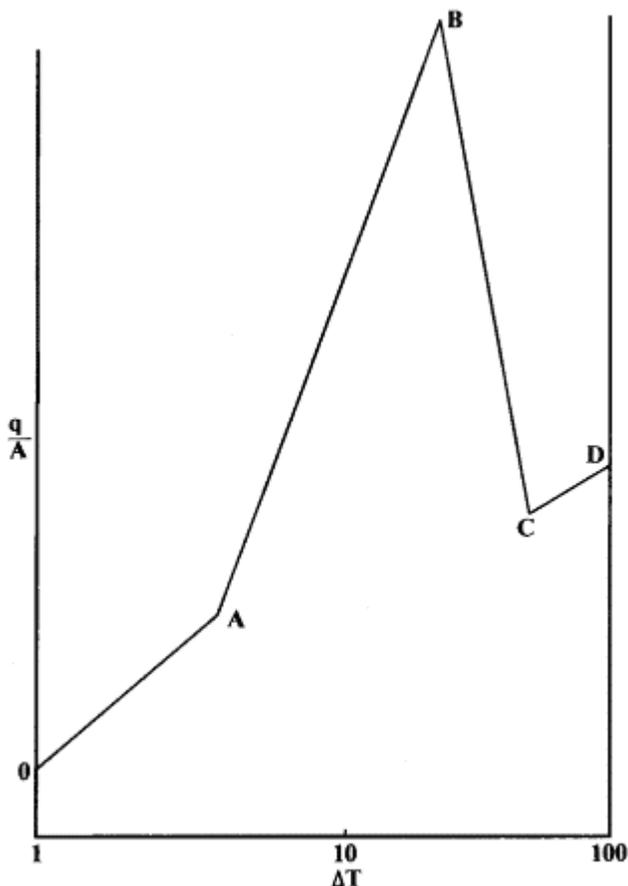


Fig. 5.17 Development of a boiling curve.



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$$\frac{h_{nc} d_o}{k_\ell} = 0.53 \left[\left(\frac{d_o^3 \rho^2 g \beta \Delta T}{\mu^2} \right) \left(\frac{\mu c_p}{k} \right) \right]_\ell^{.25} \quad (5.7)$$

(b) Nucleate Boiling

Although several attempts (23, 24) of theoretical type equations utilizing the fluid properties have been proposed, they are impractical because of the required physical property data (often unavailable for the designer's problem), their complicated evaluation, and their inherent uncertainty due to the surface conditions. A simpler approach, widely used by designers, is based on the work of Borishanski (25) who utilized the law of corresponding states and, as modified by Mostinski (15) and Collier (26), is given as

$$h_{nb} = A^* q^{0.7} F(P_r) \quad (5.8)$$

where A^* is a constant evaluated at a reference reduced pressure of $P_r = 0.0294$ and $F(P_r)$ is a function of reduced pressure as shown in Figure 5.18.

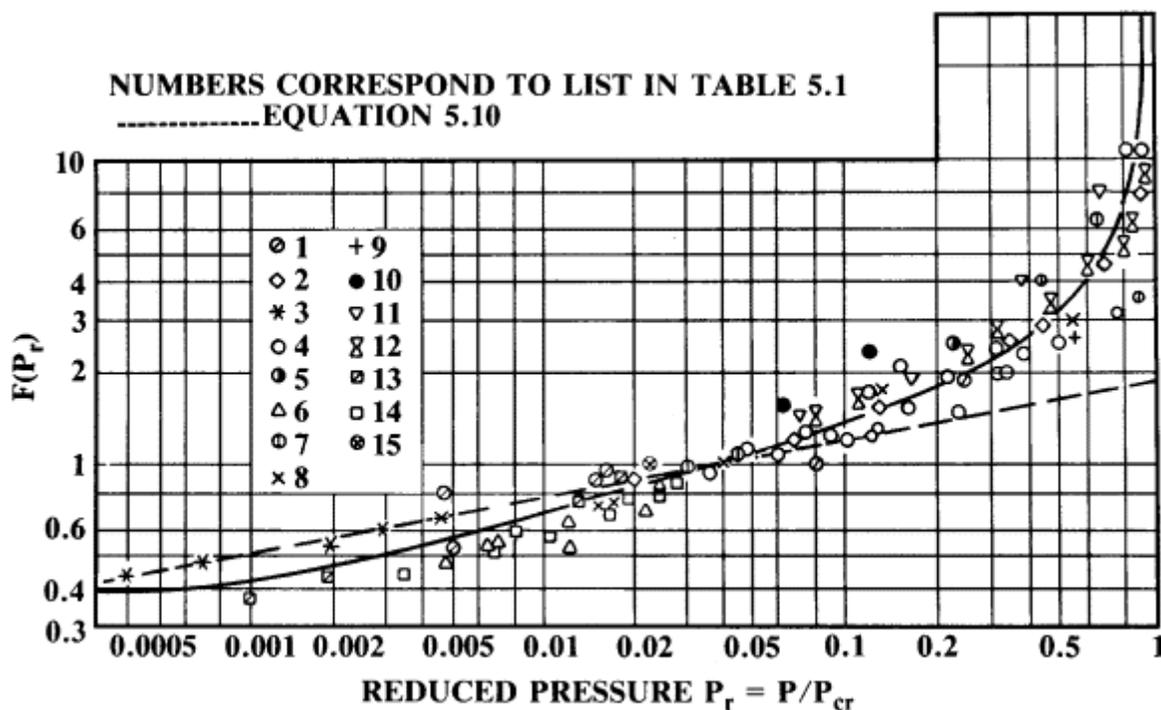


Fig. 5.18 Variation of $F(P_r)$ with reduced pressure. [26]

$$A^* = 0.00658 \underline{P}_c^{0.69} \quad (5.8a)$$

$$F(P_r)_1 = 1.8 P_r^{0.17} + 4 P_r^{1.2} + 10 P_r^{10} \quad (5.9)$$

However, dropping the last two terms is a safe design (26) for pool boiling hence



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$$F(P_r)_2 = 1.8 P_r^{0.17} \quad (5.10)$$

However, for refrigerants (R11, R12, R113, R115, etc.), $F(P_r)$ should be evaluated as

$$F(P_r)_3 = 0.7 + 2 P_r [4 + 1/(1 - P_r)] \quad (5.11)$$

and the values of A^* for these refrigerants given in the square brackets in Table 5.1 be used in equation 5.8. (26)

It should be emphasized that the above equation was developed for single component liquids and that the system pressure, tube surface condition, presence of non-condensable gases, hysteresis of the boiling curve, size and orientation of the surface, subcooling, wettability, and gravitation acceleration are some of the variables that can affect the result.

(c) Critical or Maximum Heat Flux.

Cichelli and Bonilla (13) found that the maximum flux was a function of reduced pressure, $P_r = P/P_c$, and that this curve also had a maximum at a reduced pressure of 0.3, Figure 5.4. Based on the assumption that the maximum flux was limited by the hydrodynamic flow pattern at the surface, Zuber (24), developed the following theoretical equation which seems to correlate the data and with a minor adjustment of the theoretical constant, $\pi / 24$, is

$$q_{cr} = 0.18 \lambda \rho_v \left[\frac{g \sigma g_c (\rho_l - \rho_v)}{\rho_v^2} \right]^{1/4} \quad (5.12)$$

This equation was for flat plates and some effect of geometry is found which shows the constant ranging from 0.12 to 0.2 (27) depending upon a dimensionless parameter

$$L \left[\frac{g(\rho_l - \rho_v)}{g_c \sigma} \right]^{1/2}$$

where L is radius or a length of plate. Curves are given for several different shapes, sphere, plates, and cylinders, Figure 5.19. In addition, other factors such as liquid viscosity, subcooling, and surface conditions can affect the values given by equation 5.12. However, equation 5.12 is generally used in commercial design as there are other contributing factors in an actual exchanger that influence the maximum.

(d) Minimum Film Boiling Flux.

The minimum heat flux, q_f , for film boiling, point C in Figure 5.17, occurs when the minimum rate of vapor formation to sustain a stable vapor film is reached. The Zuber theory for the minimum heat flux in film boiling was improved by Berghmans (28) who considered second order perturbations and included the

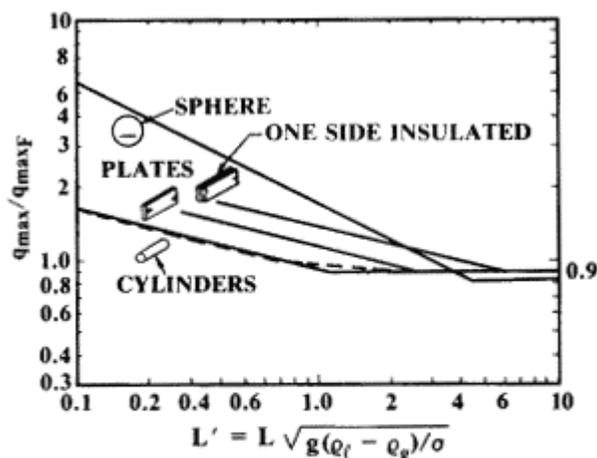


Fig. 5.19 Collected predictions of q_{max} for four heater configurations [27]. (q_{max} is from equation 5.12 but with a constant of $\pi/24$ instead of 0.18.)



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effect of vapor film thickness into the analysis. From an analysis by Berenson (29) for flat plates and the maximum flux equation 5.12, we find that

$$\frac{q_{cr}}{q_{mf}} = \frac{0.18}{0.09} \left[\frac{\rho_\ell + \rho_v}{\rho_v} \right]^{1/2} \quad (5.13)$$

and that the corresponding ΔT_{\min} is

$$\Delta T_{\min} = 0.127 \frac{\rho_v \lambda}{k_v} \left[\frac{g(\rho_\ell - \rho_v)}{\rho_\ell + \rho_v} \right]^{2/3} \left(\frac{g_c \sigma}{g(\rho_\ell - \rho_v)} \right)^{1/2} \left(\frac{\mu_v}{g_c(\rho_\ell - \rho_v)} \right)^{1/3} \quad (5.14)$$

A straight line between points B and C in Figure 5.17 then is used for the so-called transition region. Through point C a film boiling flux curve CD can be drawn using the heat flux calculated from the film boiling coefficients.

(e) Film Boiling Heat Flux.

At high temperature differences a continuous vapor film covers the surface and analytic analysis has followed an analogy to film condensation. For large horizontal plates

$$h_f = 0.425 \left[\frac{k^3 \rho_v (\rho_\ell - \rho_v) g \lambda_e}{\mu_v \Delta T [L_c / 2\pi]} \right]^{1/4} \quad (5.15)$$

where L_c is the shortest unstable wave length for the Taylor instability given by

$$L_c = 2\pi \left[\frac{g_c \sigma}{g[\rho_\ell - \rho_v]} \right]^{1/2} \quad (5.16)$$

and λ_e is an effective latent heat including the superheat effect

$$\lambda_e = \lambda [1 + 0.4(c_{pv} \Delta T / \lambda)] \quad (5.17)$$

For tubes the equations are

$$h_f = [0.59 + .069 L_c / d] \left[\frac{k^3 \rho_v (\rho_\ell - \rho_v) g \lambda_e}{\mu_v \Delta T L_c} \right]^{1/4} \quad (5.18)$$

but here



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$$\lambda_e' = \lambda [1 + 0.34(c_{pv} \Delta T/\lambda)]^2 \quad (5.19)$$

These equations give the conduction heat transfer but in addition at these temperature levels radiation becomes important. Hence, the total film coefficient as suggested by Bromley (30) is

$$h_{ft} = h_f + 0.75 h_r \quad (5.20)$$

where h_r is the coefficient for radiation transfer assuming the liquid is a black body and radiation is between infinite parallel plates.

The flux is then calculated by equation 5.6.

5.3.2. Single Tube in Cross Flow

The effect of cross flow on the boiling coefficients is shown in Figure 5.20. Basically the forced convection coefficient may exceed the nucleate boiling coefficient at low temperature differences but at higher ΔT the nucleate boiling coefficient becomes dominant. The simple rule is to use the highest coefficient.

The critical or maximum heat flux is also affected by cross flow velocity but the magnitude of this effect is a function of tube size and seems to disappear when approaching industrially used dimensions as shown in Figure 5.21.

In the film boiling region equation 5.18 will apply for low cross flow velocities but for higher velocities and for less than 45°C (81°F) subcooling the following equation applies

$$h_c = 2.7 \left[\frac{V_\infty k_v \rho_v \lambda}{d_o \Delta T_{sat}} \right]^{1/2} \quad (5.21)$$

5.3.3. Boiling on Outside of Tubes in a Bundle

In spite of the wide use of horizontal tube bundles there is very little data and only elementary suggestions for predicting its heat transfer. The very early work of Abbot and Comley (31) showed the coefficients for a

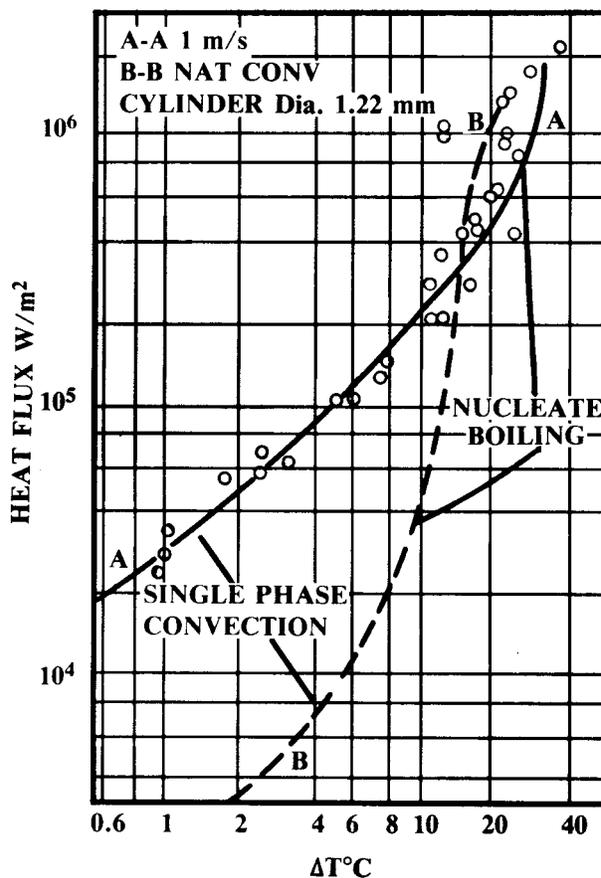


Fig. 5.20 Effect of cross flow on nucleate boiling curve for water. [74]

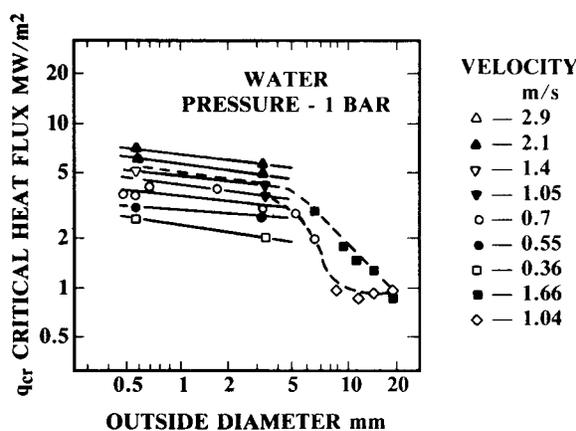


Fig. 5.21 Effect of tube diameter on critical heat flux in cross flow. [75]

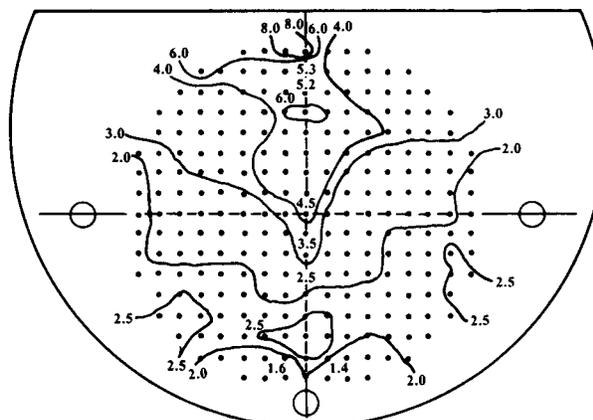


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bundle and a single tube were essentially the same. Other reports by Palen et al., (32) showed that bundles may perform better than single tubes due to the circulation induced through the bundle. Although circulation models are being developed, very little has been published on these models. As shown in Collier (26) and in Figure 5.22 the coefficients vary in a haphazard fashion throughout the bundle but, in general, increase from the bottom to top. Palen (34) recommends using

$$h_b = h_{nbl} F_b F_m + h_{nc} \quad (5.22)$$

where $F_b = 1.5$, h_{nc} = natural convection coefficient, and h_{nbl} = the single tube nucleate coefficient. F_m is the mixture correction. The 1.5 factor is a conservative approximation as it could range up to 3 depending on a bundle layout, size, and heat flux.



Figures in kW/m²K

Fig. 5.22 Heat transfer coefficient contours in kettle reboiler [33].

There is a maximum flux for a bundle that is different, and lower, than the single tube maximum flux. Based on some plant experience Palen and Small (35) proposed a model assuming a vapor blanketing effect. They developed a correction term, Φ_b , which is used to multiply the q_{max} as calculated by equation 5.12 and corrected for mixture effects, eqn. 5.38. Their result can be simplified to

$$\Phi_b = 2.2 \left(\frac{\pi D_B L}{A_s} \right) = \frac{K L_{tp}}{d_o (D_B / L_{tp})^{1.1}} \quad (5.23)$$

where $K = 4.12$ for square pitch
 $K = 3.56$ for triangular pitch
 $(\Phi_b)_{min} = 0.1$

This result is reported to be conservative by Palen et al. (32).

5.3.4. Boiling Inside Tubes

As shown in Figure 5.6 vaporization inside tubes involves a number of different flow regimes each of which requires a different evaluation of the coefficient plus a local temperature difference which in turn requires corresponding pressure drop calculations. Since most are of a natural circulation design, only the available liquid head is known thus resulting in a trial and error series of calculations to determine the feed rate per tube. The calculated procedure is, thus, too tedious for hand calculation and computers are utilized. However, the computer programs are also complex and expensive to develop and, therefore, become proprietary. We list below the various equations used for the design of vertical in tube vaporizers. Although horizontal in-tube vaporization is also used, the heat transfer equations and methods are the same but the flow pattern now includes a stratified two-layer region. The major difference between horizontal and vertical in-tube vaporization is the definition of and the flow criteria used to define the limits of each regime. The appropriate heat transfer equation is then used for each regime.



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(a) Single Phase Liquid Region.

In a circulating vaporizer the temperature of the liquid entering the tube is below the local boiling point due to the effect of the hydrostatic head on the saturation temperature. This liquid zone extends to the point where the temperature has increased and the local pressure decreased such that the local saturation point has been reached. Actually some further superheat is required to initiate nucleation. The liquid zone heat transfer coefficients are calculated from

$$\frac{h_c d_i}{k_\ell} = 0.17 \left(\frac{d_i G}{\mu_\ell} \right)^{.33} \left(\frac{\mu_\ell}{k_\ell} \right)^{.43} \left(\frac{\text{Pr}_\ell}{\text{Pr}_w} \right)^{.25} \left(\frac{d_i^3 \rho_\ell^2 g \beta \Delta T}{\mu_\ell^2} \right)^{.1} \quad (5.24)$$

for $L/d_i > 50$ and $d_i G/\mu < 2000$.

For turbulent flow and $d_i G/\mu > 10,000$ use

$$\frac{h_c d_i}{k_\ell} = 0.023 \left(\frac{d_i G}{\mu_\ell} \right)^{.8} \left(\frac{c_p \mu_\ell}{k_\ell} \right)^{1/3} \quad (5.25)$$

and interpolate between these two equations on a Re number basis for $2000 < d_i G/\mu < 10,000$.

(b) Boiling Region.

The boiling region can be further subdivided into a sub cooled boiling, saturated boiling, and two-phase boiling regions with predictive equations for each (9,26). Another approach taken by Chen (36) is to combine the saturated and two-phase regions into one, with an equation combining the convective and nucleate boiling mechanisms

$$h_b = s h_{nb} + h_{cb} \quad (5.26)$$

where h_b = the boiling coefficient
 h_{nb} = the nucleate boiling coefficient
 h_{cb} = the convective coefficient
 s = Chen suppression factor

The convective coefficient is a function of the Martinelli two-phase flow parameter, X_{tt} , and the Chen correlation using this factor is

$$\frac{h_{cb}}{h_c} = f(x_{tt}) = F_{ch} \quad (5.27)$$

$$F_{ch} = 2.35 \left[\frac{1}{x_{tt}} + 0.213 \right]^{0.73} \quad (5.28)$$

$$x_{tt} = \left(\frac{1-x}{x} \right) \left(\frac{\rho_v}{\rho_\ell} \right)^{0.57} \left(\frac{\mu_\ell}{\mu_v} \right)^{0.11} \quad (5.29)$$



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x = weight fraction of vapor

h_c = liquid phase heat transfer coefficient based on the amount of liquid present, Equation 5.25.

The nucleate boiling coefficient, h_{nb} , is determined as

$$h_{nb} = h_{nbl} F_m \quad (5.30)$$

where F_m is a correction applied for mixtures (discussed later) and h_{nbl} is the coefficient determined from equation 5.8. The suppression factor, s , is determined as follows:

1. Calculate liquid phase $Re_\ell = d_i G / \mu_\ell$

2. Calculate two-phase $Re_{tp} = Re_\ell F_{ch}^{1.25}$ (5.31)

3. Calculate $s = 1/[1 + 2.53(10^{-6}) Re_{tp}^{1.17}]$

Now equation 5.26 can be solved for h_b .

The subcooled boiling coefficient can be obtained by again using equation 5.26 but with $s = (\Delta T_b / \Delta T_o)$ where ΔT_b is the temperature difference between the tube wall and the saturation temperature of the liquid at the given local pressure and ΔT_o is the difference between the tube wall and subcooled bulk temperature. Instead of the convective coefficient, h_{nbl} , the liquid coefficient (eqn. 5.24 or 5.25) is used. The nucleate coefficient, h_{nbl} , is obtained from the transformed equation 5.8 as

$$h_{nbl} = 5.43(10^{-8})(\underline{P}_c)^{2.3}(\Delta T)^{2.33}[F(P)]^{3.33} \quad (5.32)$$

and equation 5.32 changed to

$$h_{nb} = h_{nbl} F_m \quad (5.33)$$

where F_m is from equation 5.38.

(c) Mist Flow.

In mist flow the small amount of remaining liquid is entrained as droplets and the tube wall is essentially dry. The coefficient drops rapidly and approaches that of heat transfer to gas. In this regime sensible heat is transferred to the gas which in turn transfers some of the heat to the droplets until they are completely evaporated after which only sensible heat transfer to gas occurs. The main problem is the determination of the vapor temperature, hence, temperature difference. Two extreme conditions are: (1) no heat is transferred to the droplets hence the vapor temperature rises rapidly; and (2) heat is rapidly transferred to the droplets until they disappear and during this evaporation phase the vapor is at saturation temperature. Condition 1 is approached at low pressures and velocities and condition 2 at high pressure and velocities. The actual case is somewhere between 1 and 2. Some attempts to develop empirical and theory based equations are reported (26) but the range of data seem too limited. We would recommend to use an equation like 5.25 based on gas properties and then make an engineering judgment guess of the fraction of the sensible heat transferred to the gas that is used up as latent heat for the evaporation of drops. The



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resulting effect on vapor temperature could be used to calculate an LMTD for the mist region and with the calculated gas coefficient used to determine the heat flux.

The mist region can be determined from a Fair (37) map or from the simple equation derived from this map

$$G_{mm} = 1.8(10^6) X_{tt} \text{ lb/hr ft}^2 \quad (5.34)$$

where G_{mm} is the maximum mass velocity before mist flow begins.

(d) Film Boiling.

This type of boiling should be avoided, if possible, due to control problems, possible fouling, and lack of data on pressure drop calculations. But if the temperature difference is high enough over the entire tube length, then the heat transfer coefficient can be calculated by the Glickstein and Whiteside (38) correlation

$$Nu = 0.106 Re^{0.64} Pr^{0.4} (\rho_b/\rho_v)^{0.5} \quad (5.35)$$

where the bulk average density on a no slip basis is

$$\rho_b = \rho_\ell / \left[x \left(\frac{\rho_\ell}{\rho_v} - 1 \right) + 1 \right] \quad (5.36)$$

Properties in eqn. 5.35 are based on the liquid. However, the main problem is to determine the mass velocity, G . In film boiling inside a tube we have a core of liquid surrounded by an annular layer of gas which is of very low viscosity. No data in the open literature exist for this case and, thus, determining the circulation rate is a real problem. An alternative estimate of the film coefficient could be made based on pool boiling correlations.

(e) Maximum Heat Flux for Stable Operation.

In vertical tube thermosyphons there are several limits to operation such as surging, critical heat flux, and mist flow. The surging instability occurs as ΔT is increased beyond a limit but this surging is dependent upon the hydraulic layout and is controllable by the physical arrangement. Blumenkrantz and Taborek (39) discuss the phenomenon. The surging can be controlled by increasing the frictional resistance in the inlet piping.

Usual recommended design for thermosyphons has the outlet pipe cross-section area equal to the total cross section area of the tubes. The inlet liquid line is usually smaller and in the range of 25 to 50% of the outlet pipe area.

As the temperature difference increases, the evaporation rate of a given tube will increase, pass through a maximum, and then decrease. This was investigated by Lee (40) and confirmed by Palen et al. (41) and both presented correlations for this effect. The Palen correlation is preferred due to its simplicity

$$q_{max} = 16066(d/L)^{0.35} (P_c)^{0.61} (Pr)^{0.25} (1 - Pd) \quad (5.37)$$



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While most mixtures follow the curves as in Figure 5.23 where one component is more volatile over the entire concentration range, there are other systems where one component is more volatile over only a portion of the concentration range and less volatile over the remaining portion. These systems form azeotropes where the azeotropic composition is one where the composition of the vapor and liquid are identical. This azeotropic mixture boils as though it were a single component liquid. Examples of such systems (42) are shown in Figures 5.27 and 5.28. These figures show a minimum boiling point azeotropic mixture but maximum boiling point azeotropic mixtures also exist.

Recent papers by Stephan (42) and by Thomas (43) give short reviews of some current theories for heat transfer to boiling mixtures. When we have multi-component mixtures the theories become too complicated for design purposes. A simple empirical approach to calculating mixture boiling coefficients is based on the paper of Palen and Small (35) which was later confirmed as suitable for equipment design in (32) and recommended by Palen (34). Here a mixture correction factor, F_m , is used to modify the calculated coefficient of the volatile component, h_{nbl} , determined from equation 5.32 so that the mixture coefficient = $h_{nbl}F_m$ where

$$F_m = \exp(-0.015 BR) \quad (5.38)$$

where BR = boiling range, dew point-bubble point, °F. with a lower limit of $F_m = 0.1$. This relation is shown in Figure 5.29. This equation is recommended (34) as a reasonable approach for multicomponent systems. This empirical equation is based on the boiling range which is the spread between the vapor and liquid curves as shown in Figures 5.23, 5.25, and 5.26 and is close to some of the theoretical methods (42).

(b) Mixture Maximum Heat Flux.

Although studies of maximum heat flux for mixtures have shown some instances, Figure 5.30, where the maximum flux of a mixture can be greater than the maximum flux of the components, (44) there is some disagreement

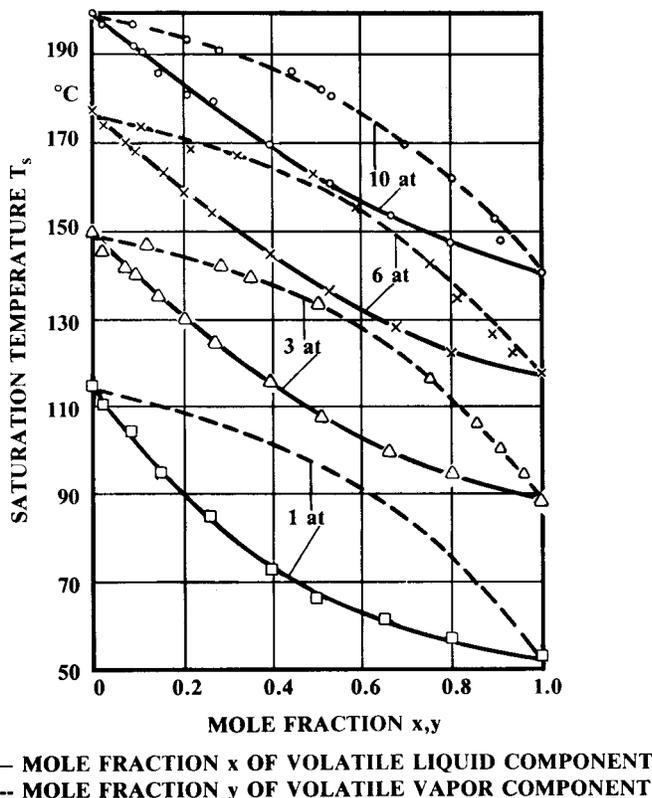


Fig. 5.25 Equilibrium curves for acetone- n-butanol [42].

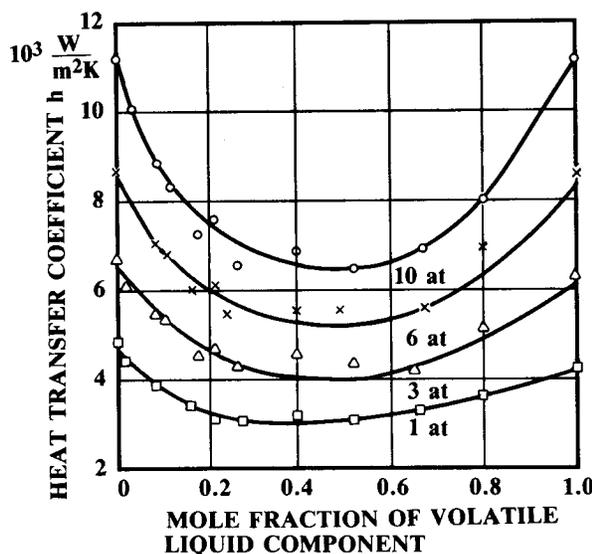


Fig. 5.26 Heat transfer coefficients for boiling acetone- n-butanol [42].



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on the cause for some of the higher values and some question on the adequacy of some theories, see e.g., (44,45,46) for details. For design purposes it is recommended (34) that equation 5.5 be used with the critical pressure of the mixture based on the molar average. This will give maximum fluxes lying between those of the components and will be conservative.

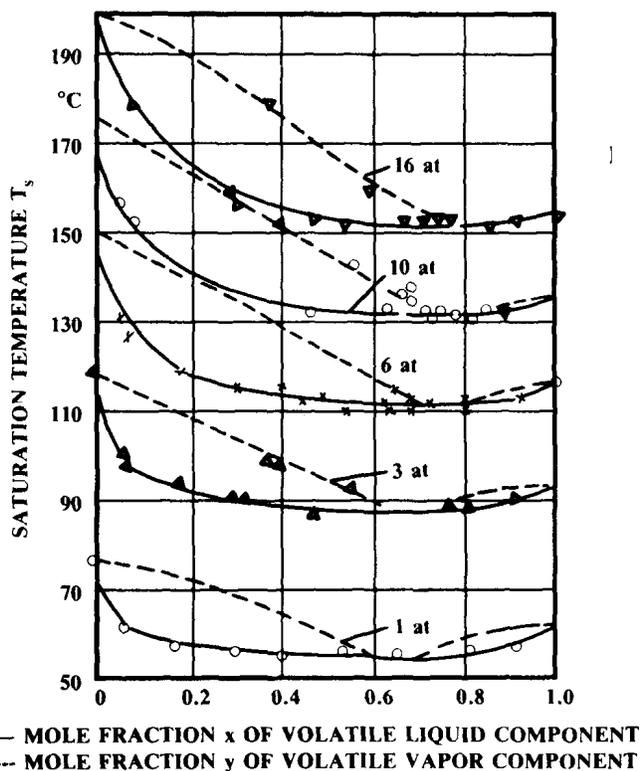


Fig. 5.27 Equilibrium curve for methanol-benzene [42].

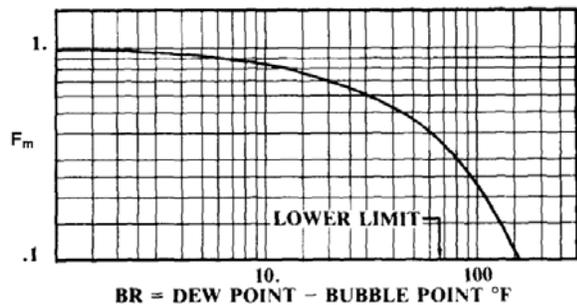


Fig. 5.29 Correction factor, F_m , for boiling mixtures.

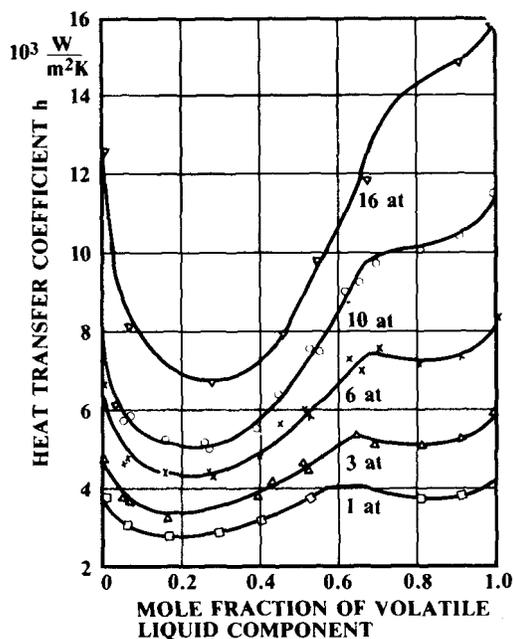


Fig. 5.28 Heat transfer coefficients for boiling methanol-benzene [42].

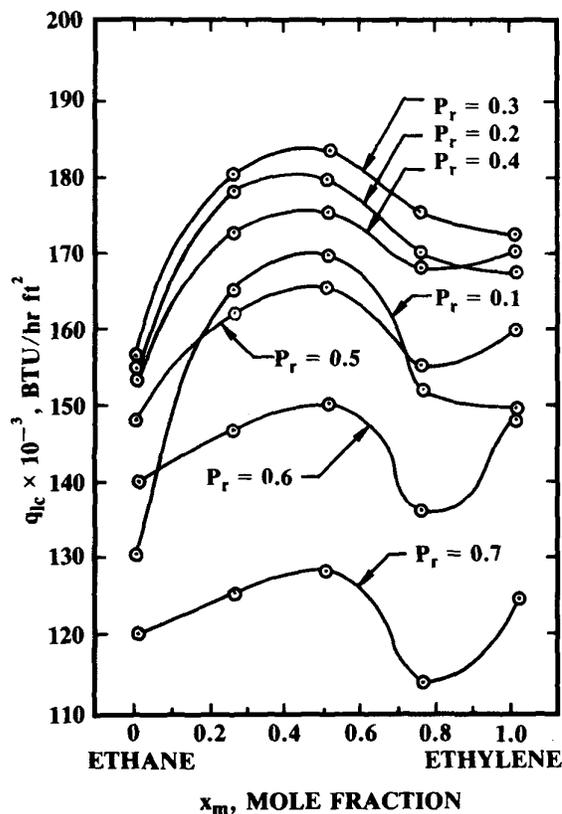


Fig. 5.30 Burn-out data for ethane-ethylene [44].