Major parametric effects on isothermality in horizontal steam-generating tubes at low- and moderate-steam qualities

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In some special boiler installations (for example, once-through boilers and fluidized-bed type boilers), horizontal or slightly inclined steam-generating tubes are used for technological or other reasons. One of the problems of these installations, leading to frequent failures, appears to be the circumferential anisothermality that occurs at certain operating and geometric conditions in the horizontal boiler tubes. This is a serious constraint for successful boiler design and maintenance.

Although we have studied the subject both experimentally and theoretically in previous works, major parametric effects on the tube top-to-bottom isothermality resulting from distinct two-phase flow phenomena are identified in this paper. The anisothermality creates high temporal and spatial variations in thermal stresses, which is of utmost importance to engineers involved in the design of industrial applications containing horizontal tubes. Approximate relationships, design recommendations, and a practical example of the proposed approach are presented.

Keywords: *two-phase flow; steam-generating tubes; circumferential anisothermality; flow regimes*

Introduction

At present, the circumferentially anisothermal operation of horizontal steam-generating tubes represents a considerable constraint in designing certain types of industrial boilers (fluidized bed types, once-through steam generators, and so forth). Frequent contacts and private communications with representatives of industrial companies in both the United States and Israel led us to conclude it was the undesirable, nonuniform temperature field in the tube walls that created severe thermal stresses, leading in most cases, to a sharp decrease of boiler operating life or even in the sudden failure of entire boiler systems.

Continuous local dryout or repeated dryout and quenching of the internal pipe surface, as well as entrance-affected local flow stratification and other two-phase flow phenomena, were previously found[1-4] to directly influence the tube wall temperature field.

It would, therefore, be of great practical importance if the detrimental wall temperature profiles could be minimized by the proper design of the steam-generating tubes. Indeed, the identification of specific operating conditions associated with minimum temperature variations—which is possible only with a deep understanding of the thermal and hydrodynamic phenomena occurring and correct interpretation of major parametric effects on these conditions—can make it possible to establish proper design criteria.

This paper seeks to provide boiler designers with additional information on the behavior of anisothermal steam-generating tubes operating at low and moderate qualities as a function of possible variation of major operating and geometric parameters; thus we focus not on experimental or theoretical aspects of the problem, as in previously published studies, but rather on parametric sensitivity of isothermal tube operation. This subject, along with the broad recommendations to boiler designers based on laboratory investigation and practical verification of the results, should be of value to the modern boiler industry.

Finally, this work focuses only on the problem of circumferential anisothermality at low and moderate qualities; top-to-bottom ΔT variations at high qualities (annular flow, mist flow) are not discussed here.

Previous experimental and theoretical results

It was analytically shown by Dukler and Taitel[5] that the primary flow regimes—stratified, intermittent (slug), dispersed bubble, and annular—may be successfully plotted on a map in the coordinates of liquid and gas superficial velocities. All the transitions between these regimes were shown to be identified by analytically calculated curves. The practical use of such a map, with the addition of Weisman's empirical boundary[6] for diabatic cases was explained by, for example, Bar-Cohen, Ruder, and Griffith.[1] Figure 1(a) illustrates a typical diabatic flow regime map.

As is obvious, from the thermal viewpoint, the stratified flow pattern is the most undesirable, whereas the dispersed bubble and annular regimes seem the most benign. It was, however, theoretically shown (Bar-Cohen, Ruder, and Griffith[1]) that to avoid circumferential anisothermality, merely providing nonstratified flow in a pipe (as suggested by Styrikovich and Miropolski[7]) is insufficient, and the intermittent (slug) pattern is commonly responsible for the moderate-quality dryout.

Moreover, further experimental research showed that not the entire intermittent regime region on the flow map is associated with anisothermality but only a sector of it. A description of the experimental apparati used, results obtained, and conclusions drawn are given in detail by Bar-Cohen, Ruder, and Griffith[2] and by Ruder.[3] Figure 1(b) shows an example of the top-to-

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Received 17 September 1986 and accepted for publication 4 April 1987

0142-727X/87/030218-10$3.00

bottom temperature differences measured along a 6-m long, 1-in. diameter steam-generating tube.

The empirical values of superficial gas and liquid velocities associated with tube anisothermal operation were plotted on the respective flow regime maps and separated from the measured isothermal U_G and U_L values by means of curves that could, with good reason, be called thermal boundaries. Examples of thermal boundaries experimentally obtained for the various heat fluxes tested are seen in Figures 6(a) and 6(b). The dry regions located below these curves on the maps appeared to consist of two parts: that at low-steam qualities (low U_G values) and that at moderate-steam qualities (moderate U_G values). The anisothermality was found to cease in the vicinity of the hydrodynamic intermittent/annular flow transition.

It also appeared possible to develop semiempirical relationships for each thermal boundary and to define all of them by means of modified Froude numbers. The lower-quality thermal boundary found to be associated with entrance phenomena was semiempirically determined merely by

$$\mathrm{Fr}_L = \frac{U_L}{\sqrt{gd}} = 1 \tag{1}$$

For the industrially important range of heat fluxes, the peak of the thermal boundary for the moderate-quality dry zone was analytically shown to be

$$\mathrm{Fr}_L = 3.2\,10^4\, d^{0.5}\left(\frac{q''}{h_{fg}\rho_L}\right)^{0.833} \tag{2}$$

where all values are in SI units.

Superficial velocities corresponding to the termination of the low-quality anisothermal zone and the onset of the moderate-quality anisothermal zone were found to be associated with the hydrodynamic boundary between the dispersed bubble and intermittent flow regimes. Unfortunately, precisely determining the location of the hydrodynamic boundary requires numerical solution of the diabatic momentum equations developed by Dukler and Taitel,[5] which is cumbersome; on the other hand, this information is generally of less practical importance than the bounding liquid velocity values. Following the theory of Dukler and Taitel, for very high heat fluxes, the position of the boundary on the map corresponds to the liquid holdup value of 0.25. This value is associated with maximum rhombohedraulic packing of bubbles beyond which bubbly configuration is impossible.

The relationship determining the onset of the annular dispersed transition coinciding with the right-hand thermal boundary appeared to be

$$\mathrm{Fr}_G = \left[\frac{1.9(U_G/U_L)^{0.125}[g(\rho_L-\rho_G)\sigma]^{0.05}}{U_G^{0.2}\rho_G^{0.1}}\right]^{2.78} = \frac{U_G}{\sqrt{gd}} \tag{3}$$

A flow regime map with designated thermal boundaries would, thus, represent a hydrodynamic-thermal map, where zones of isothermal and anisothermal operations of boiler tubes are separated. A typical example is shown in Figure 1(a).

It was shown by Bar-Cohen, Ruder, and Griffith[2,4] that the liquid and gas superficial velocity values associated with the area below the low- and moderate-quality thermal boundaries (Equations 1 and 2, respectively) and to the left of the right-hand thermal boundary (Equation 3) should be avoided during the operation of steam-generating tubes, since those yield the detrimental thermal conditions of the pipe wall.

Discussion—major parametric effects

Pressure

The thorough analysis of the boundary relationships and comparison of the data available performed by Bar-Cohen, Ruder, and Griffith,[4] suggests pressure does affect the right-hand thermal boundary defining the onset of the annular flow regime. The increase of pressure would result in a shift to the left—that is, toward the lower vapor quality region on the flow map—of the boundary line defined by Equation 3. This occurs because as pressure grows, the gas density increases considerably, whereas the liquid density only slightly decreases, and the liquid surface tension also decreases somewhat. The decrease of the numerator and, especially, the considerable increase of the denominator in Equation 3 yield the decrease of the boundary gas superficial velocity values for fixed values of U_L. This pressure effect is illustrated in Figure 2, where two extreme operating conditions of Styrikovich and Miropolski (36 bar and 182 bar) are compared. (The operating conditions of the literature data quoted in this paper are presented in detail by Bar-Cohen, Ruder, and Griffith.[4]) As is seen, this thermal boundary appears to be shifted on the map by 55% as a result of pressure increase within this range.

The effect of pressure on the moderate-quality thermal

Notation

d	Tube diameter
g	Acceleration of gravity
G	Mass flux
h	Heat transfer coefficient
h_{fg}	Latent heat
h_L	Liquid level in the pipe
L	Length
p	Pressure
q''	Heat flux
ΔT	Temperature difference
U	Superficial velocity
x	Axial coordinate
α	Angle of tube inclination
μ	Dynamic viscosity
ρ	Density
σ	Surface tension

Subscripts

ad	Adiabatic approach
crit	Critical
d	Diabatic approach
G	Gas phase
in	Inner, inside
L	Liquid phase
max	Maximum value
s	Superficial value

Superscripts

^	Bounding value of the moderate-quality anisothermal zone
*	Upper bounding value of the mass flux associated with the intermittent (slug) flow pattern
-	Average value

Dimensionless criteria

Fr Froude number $= \dfrac{U}{\sqrt{gd}}$

Q Dimensionless heat flux $= \dfrac{q''\pi U_G}{h_{fg}\left(\dfrac{dP}{dx}\right)^s_G}$

X Lockhart-Martinelli parameter $= \left[\dfrac{\left(\dfrac{dP}{dx}\right)^s_L}{\left(\dfrac{dP}{dx}\right)^s_G}\right]^{1/2}$

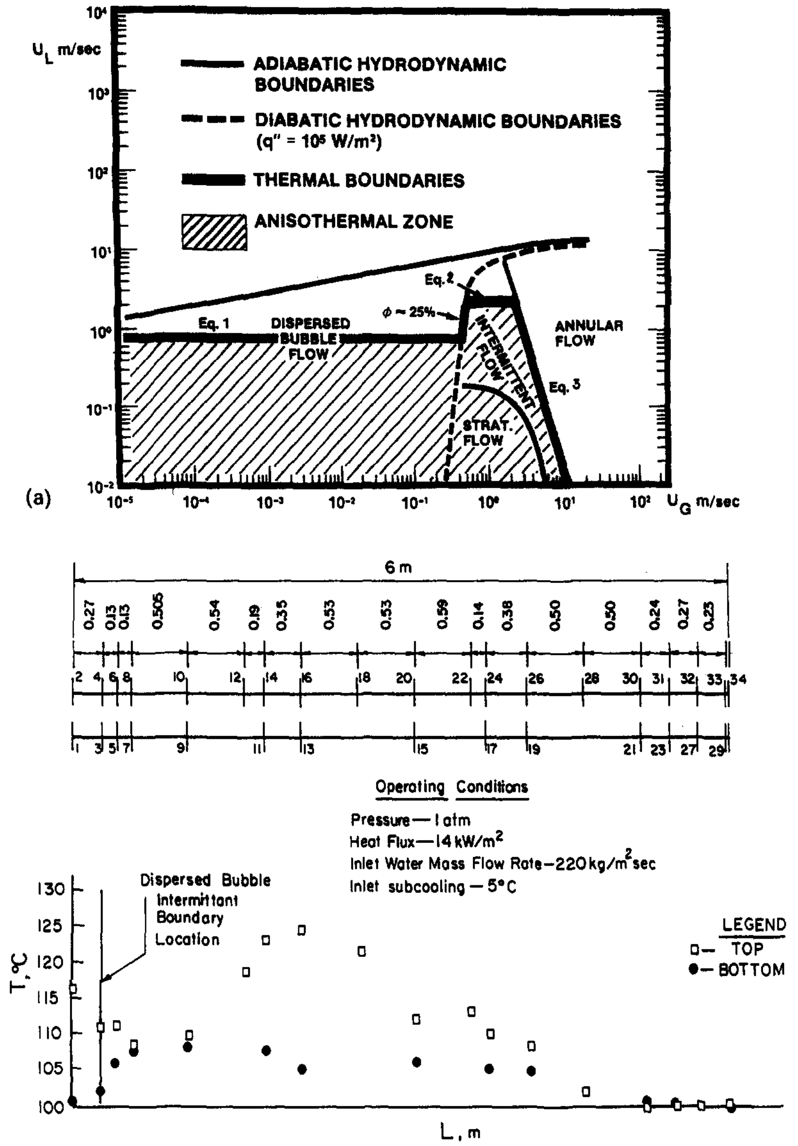

Figure 1 (a) Typical thermal/hydrodynamic map for a horizontal tube; (b) typical empirical temperature distribution for horizontal tube (6 m long, 1 in. I.D., $q''=14$ kW/m^2, $G=220$ kg/m^2 s, $p=1$ bar)

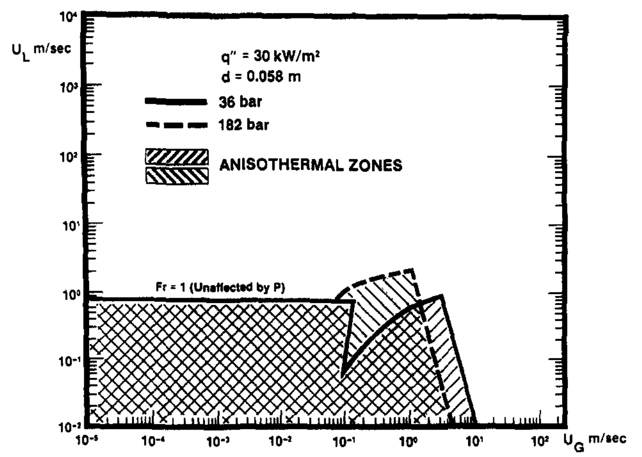

Figure 2 Effect of pressure on the thermal boundaries

boundary defined by Equation 2 appears to be as follows. For a fixed applied heat flux and tube internal diameter, the critical Froude number associated with the peak of the boundary curve grows as pressure is increased (due to the decrease of the liquid density and latent heat). For example, for water and steam flow in a 0.058 m I.D. tube under the applied heat flux of 30 kW/m^2, the critical Froude number increases from 0.69 to 2.8, as the operating pressure is raised from 1 bar to 182 bar. The trend can also be seen in Figure 2. Alternatively, the low-quality thermal bounding line of $\mathrm{Fr}_L=1$ is apparently unaffected by pressure.

Heat flux

Because of the strictly hydrodynamic nature of both the low-quality thermal bounding line and the right-hand thermal transition (Ruder[3]), the only boundary affected by heat flux turns out to be that of the moderate-quality region. This effect seems to be rather complicated.

As was empirically found for the atmospheric operating conditions reported by Bar-Cohen, Ruder, and Griffith,[2] the peak of the bounding curve somewhat decreased as the applied heat flux was raised from 4 kW/m^2 to about 100 kW/m^2. Equation 2, however, shows the opposite trend: the obvious increase of the critical Froude number with q''. Figure 3 shows both trends. Moreover, applying Equation 2 to the data obtained for high heat fluxes and elevated pressures and reported in the literature reveals no disagreement with the reported experimental results, as was shown by Bar-Cohen, Ruder, and Griffith.[4]

Dukler and Taitel[5] showed that for high heat fluxes and near-atmospheric conditions, the diabatic transition from the

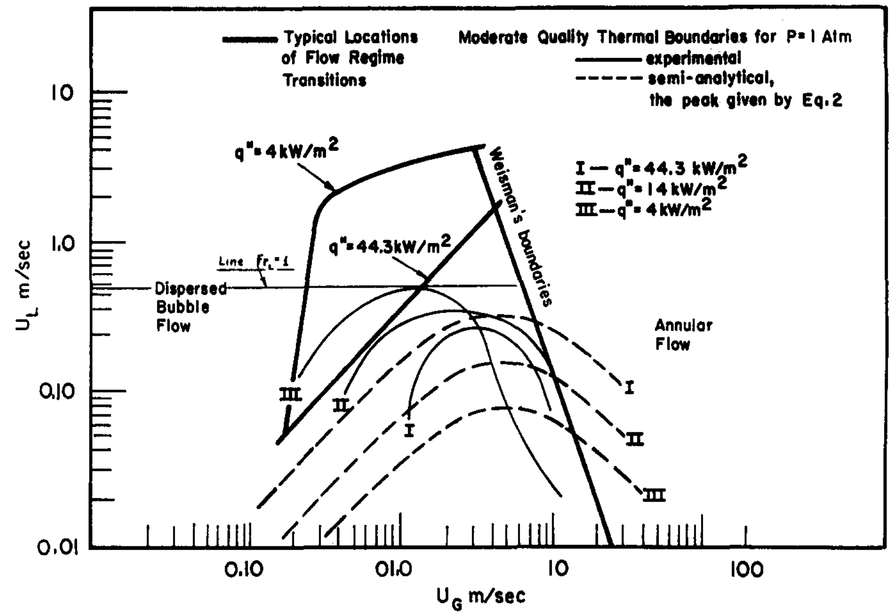

Figure 3 Heat flux effect on the moderate-quality thermal boundary—empirical and semianalytical results for $p=1$ bar, saturated inlet conditions

dispersed bubble to the intermittent flow is shifted to higher-steam qualities, which is related to the transient effects of generation, growth, and coalescence of small vapor bubbles in the liquid layer. Since the transition to the annular flow was found to be independent of heat flux (Bar-Cohen, Ruder, and Griffith[4]), this phenomenon results in the considerable diminishing of the entire slug flow region in the diabatic map and, hence, in the "shrinkage" of the moderate-quality anisothermal zone. Therefore, this latter effect seems responsible for the decrease of the Froude number associated with the moderate-quality thermal boundary. Since the dominant mechanism for the anisothermality in this zone had been shown by Bar-Cohen, Ruder, and Griffith[2,4] to be the evaporation of the after-slug film on the top occurring between consecutive slugs, the logical explanation of the phenomenon would be a "counterintuitive" dependence of the slug frequency on the heat flux: the higher the heat flux, the shorter the period between two consecutive slugs. This, oddly enough, would imply that quenching of the top of the tube is more frequent for high heat loading for the near-atmospheric pressure. The slug frequency correlation proposed by Greskovich and Shrier[8] and used to obtain Equation 2 thus appears somewhat inaccurate for atmospheric diabatic operating conditions. It might be anticipated that a more general correlation would include, in some form, an applied heat flux term to account for the influence of this parameter on the two-phase flow patterns. For now, this assertion should be considered highly tentative, although it seems to be the most logical result of the careful treatment of both theoretical approaches and experimental data. However, as pressure grows, the size of the intermittent region on the flow map appears to be practically unaffected by the heat flux. This fact may, in turn, mean at elevated pressures, the slug frequency is unaffected by the rise of external heat loading, which supports the validity of the Greskovich and Shrier[8] slug frequency correlation used in developing a relationship (Equation 2) for *elevated pressures*. In other words, the obtained trend of the direct proportionality of $\mathrm{Fr_L}$ with respect to q'' for $p \gg 1$ bar appears to be physically based.

On the basis of the careful study of the data available in the literature (Bar-Cohen, Ruder, and Griffith[4]), it may be concluded (Ruder[3]) that, at least up to $p=15$ bar, the low-quality bounding line $\mathrm{Fr_L}=1$ does, in fact, cover both anisothermal zones, regardless of the results of Equation 2 which, for these operating conditions, appears to be too conservative. This conservatism is thought to be a direct result of the inherent shortcomings of the slug frequency correlation.

Diameter

Tube internal diameter affects all three thermal bounding velocity values. Its effect can sometimes be considerable. The critical liquid velocity value for the lower-quality anisothermal region is defined by Equation 1 to be directly proportional to the square root from the tube diameter. This would imply that, for the commonly used diameter range of 0.0254 to 0.1 m (1–4 in.), the bounding superficial liquid velocity would be within the range of 0.5 to 0.99 m/s.

Unlike the lower-quality dry region, the moderate-quality anisothermal zone was found to be bounded by the liquid velocity proportional to the tube diameter. This effect seems to be a serious constraint for designers seeking to use larger tube internal diameters to avoid high hydraulic energy losses because the increase of d from 0.0254 m (1 in.) to 0.1 m (4 in.) would yield a fourfold rise of the bounding U_L values.

Alternatively, the increase of diameter would lead to the shift of the transition to the annular flow regime associated with the right-hand thermal boundary, toward higher-quality values on the flow map. This shift is brought about by the rise of the critical values of U_G as the tube internal diameter is increased. Following Equation 3, the relationship between U_G and d, for fixed U_L values, appears in the form

$$U_G \sim d^{0.631} \tag{4}$$

Overall, from the standpoint of tube overheating, the increase of diameter *negatively* affects tube circumferential isothermal operation because of the broadening of the entire anisothermal zone on the flow regime map. Figure 4 illustrates this trend.

Angle of tube inclination

The dearth of available data on the effect of the inclination of steam-generating tubes on their circumferential isothermality suggests the need for more research in this field. In their study, Styrikovich and Miropolski[10] identified the upward inclination angle of 9.5° as critical for avoiding anisothermality, but this value seems to be somewhat arbitrary and does not represent the true boundary, which is believed to depend on many factors, such as pressure, heat and mass fluxes, and tube geometry.

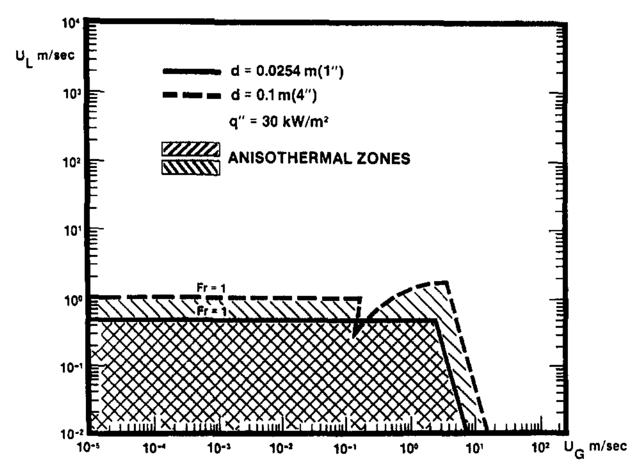

Figure 4 Effect of tube diameter on the thermal boundaries

The experimental apparatus described by Bar-Cohen, Ruder, and Griffith[2] was used to carry out tests with the pipe inclined upwardly by 10° and 2°. The results obtained for the entire range of the operating and geometric parameters ($G = 73$–$730\,kg/m^2s$, $q'' = 4$–$50\,kW/m^2$) at atmospheric pressure conditions revealed complete disappearance of circumferential anisothermality of the 0.0254 m I.D. tube even for the angle of 2° (Figures 5a and 5b). At the present, the only explanation seems to be the increase of the slug frequency as the angle is raised. This assertion is, however, hypothetical, and more experimental or theoretical results are needed to substantiate this idea. The flow maps show the hydraulic stratified/intermittent boundary utterly disappears for a 2° upward inclination, not to mention a 9.5° upward inclination.

The results for 2° are not in variance with those obtained by Styrikovich and Miropolski[5] for high pressures of 36, 112, 142, and 182 bar, since they do suggest the critical angle is expected to rise with pressure. According to the two investigators, for their operating conditions, "for high pressures, an angle of slope of 10° no longer appears sufficient for the elimination of separation." The effect of the angle of tube inclination on circumferential anisothermality is discussed in more detail by Bar-Cohen *et al.*[11] and by Ruder.[3]

Entrance effects

The research of some investigators focused not merely on the thermal effect of two-phase flow in horizontal tubes but primarily on studying the wall-drying influences of entrance sections, horizontal and vertical inlet bends, and so forth. Lis and Strickland,[12] for example, investigated vertical inlet bends found to induce large amplitude temperature oscillations and even "dry patches" along the top of the 41-mm I.D., 3-m long evaporator tube for a wide range of operating conditions ($p = 65$–67 bar, $G = 200$–$600\,kg/m^2s$, $q'' = 25$–$62\,kW/m^2$). The authors offered no satisfactory explanation for the effects of bend orientation on tube thermal behavior that was found to be a function of steam pressure, mass and heat fluxes, and steam quality.

The entrance phenomena, which may be regarded as a secondary (with respect to a developed two-phase flow pattern) effect, is believed to result in the appearance of an additional low-quality anisothermal zone along a horizontal steam-generating pipe. Bar-Cohen, Ruder, and Griffith[2] found it possible to bound this zone on the flow map by the line $Fr_L = 1$.

The low-quality anisothermal zone bounding line defined by this relationship turns out to be valid but only for certain operating conditions, namely, when the flow is undisturbed by artificial entrance effects (such as bends and fittings) and the local stratification resulting in circumferential anisothermality is, thus, produced exclusively by the genesis of the intermittent flow pattern. Since the hydrodynamic diabatic dispersed bubble/intermittent boundary is associated with very low values of vapor quality, the high top-to-bottom ΔT is measured very close to the tube inlet. Hence, these conditions may with good reason be called pure entrance conditions.

The situation may be much more severe in the case of artificial entrance conditions, that is, when various bends or fittings are located just before the pipe inlet. This phenomenon was discussed by Ruder.[3] Actually, even for the calming section L/d ratio of 30, the measured circumferential anisothermality at low qualities was influenced not only by the hydraulic phenomenon associated with the transition to slug flow but also by the general "undeveloped" nature of the flow due to the horizontal or vertical bend before the pipe inlet. Bends like these can, according to Lis and Strickland,[12] make local dryout very pronounced and usually bring about tube top overheating, as measured by Lis and Strickland,[12] Rique and Roumy,[13] and others. Moreover, for the experiments without the calming section, these two thermally problematic phenomena were found to be superimposed on the third phenomenon, the so-called jet effect, which was caused by the use of a short fitting tube, about 10 cm long, with an internal diameter slightly smaller than that of the test section (0.019 m versus 0.0254 m).

The essence of the jet effect phenomenon can be explained as follows. The one-phase liquid leaving the fitting, whose diameter is smaller than that of the heated pipe, cannot expand immediately on entering the test section. This results in the appearance of dry spots around the pipe near the inlet. So, as the point of the net vapor generation was shifted upstream along the test section (from the axial position associated with $L/d = 90$ through $L/d = 30$ through $L/d = 0$), the lower-quality anisothermal zone was found to be characterized not only by increasingly higher ΔT values but also to expand on the flow regime map to higher U_L values. The critical Fr_L number was, then, experimentally found to increase from about 0.6 at $L/d = 90$ to as high as 1.6–2 at $L/d = 0$, whereas the maximum top-to-bottom temperature difference was found to change from 30°C–50°C to 70°C–80°C, respectively. These effects can be seen by comparing the low-quality thermal boundaries obtained empirically for the "pure" (calming $L/d = 90$) and "artificial" ($L/d = 0$) entrance conditions, as shown in Figures 6(a) and 6(b).

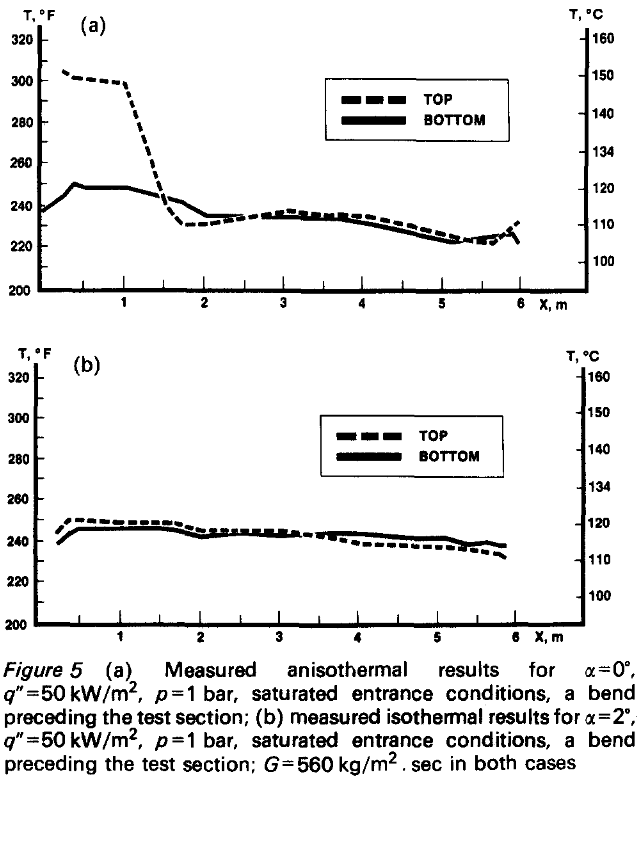

Figure 5 (a) Measured anisothermal results for $\alpha = 0°$, $q'' = 50\,kW/m^2$, $p = 1$ bar, saturated entrance conditions, a bend preceding the test section; (b) measured isothermal results for $\alpha = 2°$, $q'' = 50\,kW/m^2$, $p = 1$ bar, saturated entrance conditions, a bend preceding the test section; $G = 560\,kg/m^2 \cdot sec$ in both cases

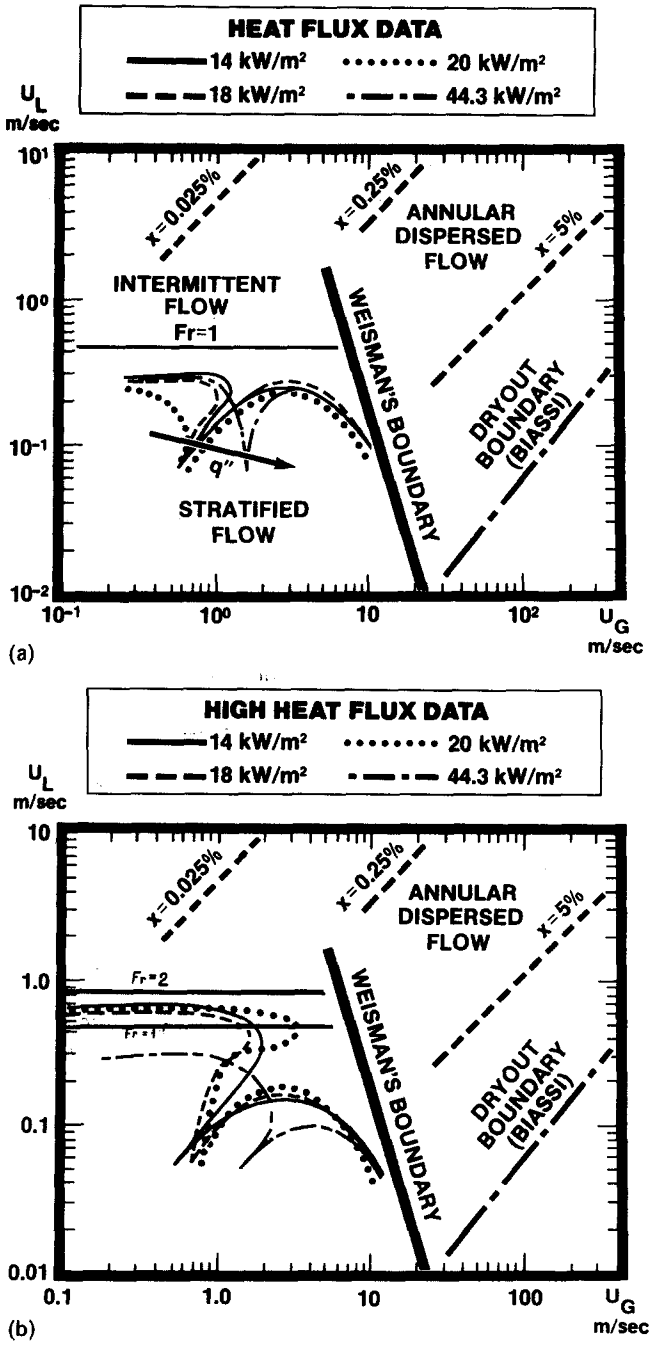

Figure 6 (a) Entrance effect – $p=1$ bar, water subcooled at the entrance, saturated conditions at $L/d=90$, (b) entrance effect – $p=1$ bar, saturated entrance conditions ($L/d=0$)

Finally, for the tested flow conditions severely affected by the entrance phenomena, it was experimentally found that the anisothermal zone never exceeded the line $Fr_L=2$ (Figure 6b). Thus the latter Froude value appeared to be sufficiently high to ensure low-quality isothermality. Although this result was obtained for water and steam at atmospheric pressure, whether other operating conditions would seriously change this value seems doubtful.

It was also found that such a negative, artificial entrance effect could be avoided by providing test section with a preceding straight pipe of the same diameter and having the length-to-diameter ratio not less than 90. If this high L/d value appears to be practically unachievable, $Fr_L=2$ value would, at this time, be recommended. This latter conclusion is also somewhat indirectly supported by the empirical results of Lis and Strickland,[12] who found a postbend calming length of $L/d=56$ to be insufficient to provide circumferential tube isothermality.

Nonuniform heat flux

Although almost all available experimental results reported in the literature deal with uniformly heated pipes, most applications involve a circumferentially nonuniform heat flux. In this latter case, the phenomenon of tube overheating is physically identical to that produced by a uniform heat flux and, as such, cannot be expected to affect the thermal boundaries in a way other than the uniform heating under the same operating and geometric conditions. When the imposed heat flux is circumferentially (or axially) nonuniform, local heat and flow parameters, rather than averaged characteristics, must be considered, as explained later. Equally important is understanding that the severity of the problem of tube circumferential anisothermality depends on the site of q''_{max} imposition. The most thermally undesirable situation can be anticipated when the peak heat loading is applied to the top of the tube where the liquid film is the thinnest. The most thermally desirable situation arises when the maximum heat flux is imposed at the bottom of the tube, since when $q''_{max}<q''_{crit}$, no local dryout can be expected to occur at the bottom. As for the high q'' values imposed at the side of the horizontal steam-generating tube, they either result in local overheating (as in the first case) or produce no local dryout on the pipe. The particular thermal effect of the heat loading applied to the side of the pipe depends on the local liquid level in it (Ruder[3]).

Since in most practical situations, the peak heat loading is imposed on the tube side, it is usually required to evaluate the level of liquid in the tube and to compare the level with the vertical position of the peak heat flux. If the former value is lower than the latter, a possibility of tube overheating exists at this particular site. The dimensionless minimum liquid level parameter associated with the transition to the annular flow regime can be found for operating conditions of interest using LIQLEV, a program developed and listed by Ruder.[3] LIQLEV, written in FORTRAN for the CDC computer, calculates, along with minimum h_L/d values, the gas and liquid superficial velocities and vapor quality associated with the intersection point between an operating locus and the Weisman-proposed intermittent/annular transition. However, a simpler, first-order approach to estimate h_L/d is explained below.

Approximate relationships for the liquid level in a horizontal boiler tube

The dimensionless liquid level parameter (h_L/d) can be calculated for local conditions by solving numerically the two-phase momentum equations derived by Dukler and Taitel.[5] Because this numerical procedure is rather cumbersome and complicated, it is appropriate to suggest a more simple, approximate approach that, with certain reasonable assumptions and simplifications, is principally based on these same equations. These assumptions and the procedure for developing the proposed relationships are discussed in detail by Ruder.[3]

As shown by Ruder,[3] for very high average heat fluxes imposed on the tube (which fall above the curve in Figure 7a), the h_L/d value can be found by means of Figure 7(b) (Dukler and Taitel[5]) using the following relationships:

$$X=0.125\left(\frac{G-10\rho_G}{\rho_L+2.85\rho_G-0.285G}\right)^{0.9}\left(\frac{\rho_L+2.85\rho_G}{\rho_G}\right)^{0.4}\cdot\left(0.26\frac{\mu_L}{\mu_G}+0.74\right)^{0.1} \tag{5}$$

$$Q=5.4\frac{q''}{h_{fg}}\left[\frac{\rho_L}{\rho_G(\rho_L+2.85\rho_G-0.285G)}\right]^{0.8}\left(\frac{d}{\mu_G}\right)^{0.2} \tag{6}$$

where all values in Equations 5 and 6 are in SI units. If, alternatively, the applied q'' is relatively low (below the curve in Figure 7a), Figure 7(c) is applicable for estimating h_L/d by

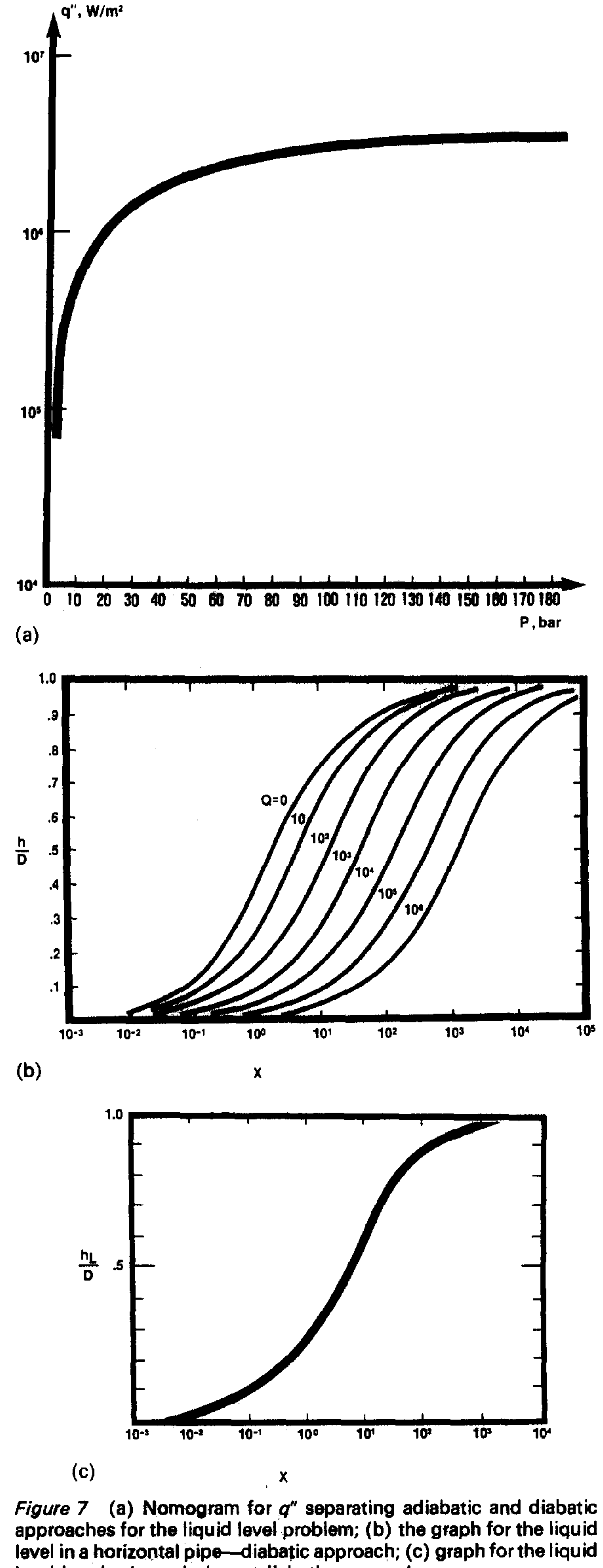

Figure 7 (a) Nomogram for q'' separating adiabatic and diabatic approaches for the liquid level problem; (b) the graph for the liquid level in a horizontal pipe—diabatic approach; (c) graph for the liquid level in a horizontal pipe—adiabatic approach

means of the expression

$$X=\rho_L^{-0.5}\rho_G^{-0.4}\left(\frac{\mu_L}{\rho_G}\right)^{0.1}(0.1G-\rho_G)^{0.9} \tag{7}$$

where all values are in SI units.

If the h_L value thus found turns out to be higher than the spot where the peak heat flux is imposed, no anisothermality problem can be expected to develop. If, on the other hand, this limiting liquid level appears to be lower than the site of q''_{max} application, there is a possibility of tube overheating at this particular site, in which case the bounding relationships (Equations 1 and 2) with local thermal and flow parameters are to be applied. Moreover, if the spot where the peak of heat loading is lower than about $0.563d$, the lower-quality anisothermal zone cannot be expected to appear because, as shown by Benjamin,[14] the height of the stationary bubble, or a locally stratified zone, can be expected to be only $0.437d$ for a circular tube. Thus the only criterion remaining to be tested is that of the moderate-quality anisothermality (Equation 2). This conclusion, however, should be further substantiated by more systematic data.

In keeping with this approach, four additional points should be emphasized:

1. Although the liquid level in the pipe and the heat flux values of Figure 9(a) are to be estimated using the averaged (over the entire pipe) heat flux, the peak of the anisothermal zone (Equation 2), must be found by means of the local, maximum value of q''.
2. Since in the suggested approach for estimating h_L/d in a horizontal pipe, the problem of conductive circumferential heat smoothing along the pipe walls is not taken into account, all q'' values (both averaged and local) should be those of the internal heat flux. When the pipe is characterized by very low thermal conductance values (as in the tests reported here), the internal q'' values may be assumed equal to the externally imposed heat loadings.
3. In the case of side heating, Equation 2 may be expected to somewhat overestimate the severity of the situation because of a small but thermally benign effect of the liquid drainage down the wall from the higher, neighboring areas where the local q'' values are lower. This phenomenon, which has a relatively small (for moderate to high heat fluxes) but certainly negative effect on the development of the dry zone at the top of the tube in the case of side heating, could contribute to moderating the trend toward local dryout.
4. In the case of the maximum q'' occurring below the minimum liquid level (estimated as shown above), it may still happen that even $q''<q''_{max}$ is sufficient to dry out the wall above the h_L value found. And as mentioned, if the locally imposed heat flux at the site below h_L is within the range $q''_{crit}<q''<q''_{max}$, the tube overheating can arise below the liquid level as well. These cases would require some additonal verification.

Comparing the h_L/d values obtained using the approximate approach presented by Ruder[3] with the results of the numerical solution of the precise momentum transfer equation (program LIQLEV) reveals the surprisingly high accuracy of the h_L/d values. For example, for the atmospheric water and steam operating conditions described by Bar-Cohen, Ruder, and Griffith[2] for $G=440\,kg/m^2s$, $q''=45\,kW/m^2$, and h_L/d parameter appears to be 0.52 versus 0.546 (approximate versus accurate); for the Rounthwaite[9] conditions at 15 bar, h_L/d equals 0.31 versus 0.311; and for the Styrikovich and Miropolski[7] conditions at 36 bar, h_L/d turns out to be 0.31 versus 0.360. The diabatic approach yields for the water and steam conditions reproted by Bar-Cohen, Ruder, and Griffith[2] for $G=440\,kg/m^2s$, $q''=10\,kW/m^2$, give the h_L/h ratio of 0.24 versus 0.26 (approximate versus accurate).

The approximate approach ceases to work at very high pressures and relatively low mass fluxes because then the term $(0.1G-\rho_G)$ in Equations 5 and 7 becomes negative. The striking example of this situaton is the case of Styrikovich and Miropolski's[7] working conditions ($p=112\,bar$ and $G=474\,kg/m^2s$). In this case, only LIQLEV can be used to evaluate the minimum h_L values.

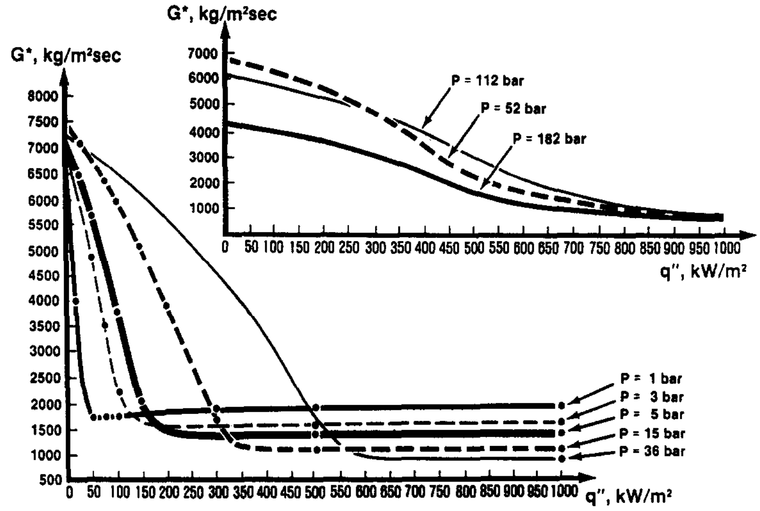

Figure 8 Typical effect of pressure, tube diameter, and heat flux on the shift of the intersection point between the dispersed bubble/intermittent and intermittent/annular hydrodynamic transitions

Fluids other than water

Other than water, various kinds of refrigerants are the most commonly used working fluids in two-phase flow situations. Since the thermo-physical and heat transfer parameters of freons differ considerably from each other, no simple trend of the fluid effects on the Fr-defined thermal boundaries can be specified.

Freon 12, for example, as reported by Fisher and Yu[15] and examined in detail by Bar-Cohen, Ruder, and Griffith,[4] exhibits a rather high value of the moderate-quality bounding mass flow rate (G) of 2530 kg/m^{2}s (confirmed both experimentally and using Equation 2), for 3.52 bar pressure and heat fluxes of 6–18 kW/m^2. This value is obviously much higher than that for water, which at the same operating conditions, would constitute only about 220 kg/m^{2}s. This striking difference between the water and R-12 thermal behavior is explained by the fact that the liquid freon density and especially its latent heat are lower than those of water (865 kg/m^3 versus 930 and $5.22 \cdot 10^4$ versus $2.15 \cdot 10^6$ J/kg°C, respectively). In this particular case, the R-12 right-hand thermal boundary is nearly identical to that of water.

The change of the thermo-physical properties (in this case, from R-12 to water) results in a change in the relative position of the low- and moderate-quality boundaries. In the case of R-12, the dominant thermal boundary is that of the moderate-quality region (Equation 2), whereas for water, it is the bounding line of the low-quality zone (Equation 1).

In contrast to R-12, certain properties of refrigerant-22 are more similar to those of water, and the thermal behavior of the latter freon is thus more reminiscent of the phenomena observed for water for the same operating conditions.

Increasing liquid flow rate

This is a conventional way of avoiding the tube anisothermality by shifting the operating locus to higher values of U_L on the flow map. The loci are thus expected to fall above the entire anisothermal zone on the map defined by the dominant-, low-, or moderate-quality thermal boundary, that is, by Equations 1 or 2. In the extreme case, the loci would lie above the intersection point between the diabatic dispersed bubble/intermittent boundary and the transition to the annular regime, thus avoiding the thermally problematic slug flow pattern. The results of Robertson,[16] critically discussed in detail by Bar-Cohen, Ruder, and Griffith,[4] are good examples of such a situation.

It may well happen, however, that at elevated pressures and very high heat fluxes (for example, Robertson's data), the peak of the moderate-quality thermal boundary calculated in accordance with Equation 2 would be located higher on the map than the intersection point. When an operating locus also proceeds above this point, the situation may lead a designer to an incorrect conclusion of possible anisothermality. Clearly, a criterion is needed to test if the operating mass flux is within the region of validity of Equation 2.

Unfortunately, a way has not yet been found to provide an analytic expression of G^*, above which the operating mass fluxes are high enough to make the use of these equations physically irrelevant. This would require a simultaneous solution of both Equation 3 and the two-phase momentum equation derived by Dukler and Taitel;[5] thus far, the latter has been solved only numerically. Figure 8 gives an example of the graphical results of such a numerical solution for various pressures and heat fluxes. Designers should thus first examine (by means of Figure 8) if their operating mass fluxes are below corresponding G^* values and, if they are, test their conditions using Equations 1, 2, and 3.

Finally, although increase of liquid flow rate is widely used in practice as an effective way of avoiding thermally undesirable operating flow conditions, the significant increase in pumping power can be rather expensive and, at times, may be totally unacceptable.

Conclusions

The occurrence and extent of circumferential anisothermality in horizontal boiler tubes is found to be affected in a complex but predictable manner by major operating parameters and tube geometry.

The increase of pipe diameter, rise of pressure and heat flux (at elevated pressures), as well as the insufficient length of the entrance section ($L/d < 90$) and the location of bends and fittings near the inlet are shown to *negatively* affect the tube thermal field because of broadening of the entire anisothermal zone on the flow map. On the other hand, upward tube inclination and increasing heat loading (for near-atmospheric pressures) are shown to be *positive* factors leading to the decrease or, at certain conditions, even disappearance of the anisothermal zone on the map.

The thorough study of these major parametric effects, as well as possible thermally negative consequences of nonuniform external heating, yield practical recommendations and broad initial design guidelines to avoid anisothermality of horizontal steam-generating tubes. The principal criterion to be used while designing such a horizontal boiler tube could be reduced to the five main expressions shown in Table 1. The Appendix presents a practical example of some of these expressions in calculations.

Table 1 Principal design criteria[a]

Low-quality anisothermal zone ($L/d>90$)	Peak of moderate-quality thermal boundary (at elevated pressures)	Right thermal boundary (onset of annular flow pattern)	Dimensionless local liquid level parameter (needed in nonuniform heat loading)
$Fr_L \equiv \frac{U_L}{\sqrt{gd}} = 1$	$\hat{Fr}_L = 3.2 \cdot 10^4 d^{0.5} \left(\frac{q''}{h_{fg}\rho_L}\right)^{0.833}$	$Fr_G = \left[\frac{1.9(U_G/U_L)^{0.125}[g(\rho_L-\rho_G)\sigma]^{0.05}}{U_G^{0.2}\rho_G^{0.1}}\right]^{2.78}$	For high $\bar{q}''_{int}$ values Figures 9(a) and 9(b), where $X_d = 0.215\left(\frac{G-10\rho_g}{\rho_L+2.85\rho_G-0.285G}\right)^{0.9} \times \left(\frac{\rho_L+2.85\rho_G}{\rho_G}\right)^{0.4}\left(0.26\frac{\mu_L}{\mu_G}+0.74\right)^{0.2}$ $Q = 5.4\frac{q''_{int}}{h_{fg}}\left(\frac{\rho_L}{\rho_G(\rho_L+2.85\rho_G-0.285G}\right)^{0.8} \times \left(\frac{d}{\mu_G}\right)^{0.2}$ or using LIQLEV
$U_L = \sqrt{gd}$	$\hat{U}_L = 10^5 d\left(\frac{q''}{h_{fg}\rho_L}\right)^{0.833}$	$U_G = Fr_G\sqrt{gd}$	For low $\bar{q}''_{int}$ values Figures 9(a) and 9(c), where $X_{ad} = \rho_L^{-0.5}\rho_G^{-0.4}\left(\frac{\mu_L}{\mu_G}\right)^{0.1}(0.1G-\rho_G)^{0.9}$

[a] All relationships are expressed in SI units.

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Appendix

Practical example

An example of the proper selection of an isothermal liquid mass flux following the conclusion and recommendations of this paper is given.

Statement of the problem

A horizontal, uniformly heated, long, steam-generating tube of 0.0508 m (2 in.) I.D. is operated under a pressure of 36 bar. The tube material has a low thermal conductivity value. The imposed heat flux is $100\,KW/m^2$. The tube is preceded by a vertical bend and various fittings. It is required to determine the lowest water mass flow rate value ensuring circumferentially isothermal operation within both the low- and moderate-quality ranges.

Solution

In this case, the procedure of selecting an isothermal operating mass flux is as follows. One should turn to Figure 8 and, according to the p and q'' values given, find the required flow rate value as the lowest G falling above the relevant curve: $G = 6520\,kg/m^2s$. It may, however, be sufficient to select a mass flux higher than that associated with the peak of the moderate-quality anisothermal zone. This condition can be verified using Equation 2, which yields the minimum isothermal G value of about $1500\,kg/m^2s$.

It must now be checked if this mass flow is isothermal in the low-quality entrance-affected region as well. Since the entrance conditions considered are "artificial" (due to the inlet bend and fittings), the lower-quality anisothermal zone bounding criteria of $Fr_L = 2$ may be accepted as a more relevant bounding relation. For the present operating conditions, the latter relationship gives the lower-quality bounding flow rate of about 1230 kg/m^{2}s. In this particular case, this means it is the moderate-quality peak that actually determines the upper boundary of the entire anisothermal zone of tube operation. It may, thus, be concluded that the required minimum isothermal mass flux equals, for this case, 1500 kg/m^{2}s.

Once again, the criteria discussed in this paper deal only with the low- and moderate-quality anisothermal zones; anisothermal effects related to the operation in the annular flow pattern are beyond the scope of this investigation.