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DEVELOPMENT AND VALIDATION OF BOUNDARIES FOR CIRCUMFERENTIAL ISOTHERMALITY IN HORIZONTAL BOILER TUBES

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Abstract—The extended operating life needed in boiler tubes makes it essential that stressful temperature fluctuations be avoided by selection of the appropriate two-phase flow operating conditions. Physical models and semianalytical Froude number relations for the liquid and vapor velocities, constituting the borders between the isothermal and anisothermal operating zones, are presented. These relations are validated over a wide range of operating conditions using data reported in the literature.

INTRODUCTION

The structure and magnitude of the temperature field in the wall of a steam-generating pipe is largely determined by the prevailing internal and external heat transfer coefficients and close coupling, thus, exists between thermal stress developed in the tube walls and the governing thermal transport processes. Continuous local dry out or repeated dry out and quenching of the internal pipe surfaces, as may well occur in horizontal two-phase flow, can be expected to produce significant spatial and temporal variations (Styrikovich & Miropolski 1950; Rounthwaite 1968; Bar-Cohen *et al.* 1984) which could be highly detrimental to the pipe. Variations in the mass flow rate and the phase distribution (or voidage) can, however, markedly affect the prevailing two-phase flow regime and, hence, the temperature and stress fields. The extended operating life needed in boiler tubes makes it essential that stressful temperature fluctuations be avoided by proper design of the steam-generating tubes and selection of the most benign two-phase flow operating regime. Unfortunately, the dearth of analytical relations and/or reliable empirical data for circumferential isothermality can seriously impair the design of horizontal boilers and, thus, hamper the development of fluidized bed combustors, solar boilers and other systems using this pipe configuration.

Recently, Bar-Cohen *et al.* (1984) formulated physical models and semianalytical Froude number relations for the liquid and vapor velocities constituting the borders between the isothermal and anisothermal operating zones. These relations were shown to adequately describe results obtained in an atmospheric pressure steam-generating pipe. The present study seeks to substantiate and validate these relations over a wide range of operating conditions using data reported in the literature.

It is to be noted that anisothermality resulting from high-quality dry-out in the pipe is not addressed in this effort.

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ESTABLISHING THERMAL BOUNDARIES

Conceptual framework

To insure isothermal operation of a horizontal boiler tube heated by an axially and circumferentially uniform heat flux, it is necessary to maintain a uniform internal heat transfer coefficient. Recalling the flow regime classification for cocurrent, horizontal pipe flow proposed by Taitel & Dukler (1976), namely: stratified (wavy and smooth), intermittent (elongated bubble and slugs), annular (including dispersed liquid) and dispersed bubble flow, it is apparent that pipe isothermality is inherent to the annular and dispersed bubble flow regimes. Alternately, stratified flow can be expected to induce strong anisothermality, while the periodic washing of the upper pipe surface, associated with the intermittent flow regime, can be expected to result in a complex, time-varying temperature field.

A purely hydrodynamic and highly conservative analysis of boiler tube temperature variations would, thus, appear to suggest that the transition from isothermal to anisothermal operation occur along the boundaries between the intermittent flow regime and dispersed bubble and annular flow, respectively. As a consequence, in the superficial velocity coordinates of figure 1, the anisothermal zone could be expected to be bounded from above by the locus T and from the right by the locus W. Alternately, based on the early results of Styrikovich & Miropolski (1950), anisothermal operation is often thought to be limited exclusively to the stratified flow regime, and therefore, to lie below the boundary S in figure 1. More recently, Bar-Cohen *et al.* (1984) observed the isothermal/anisothermal transition to occur at superficial liquid velocities substantially above those associated with stratified flow, but significantly below the dispersed bubble and annular transitions of locus T and W and to lie along locus B in figure 1. It is to be noted that this thermal boundary can be divided

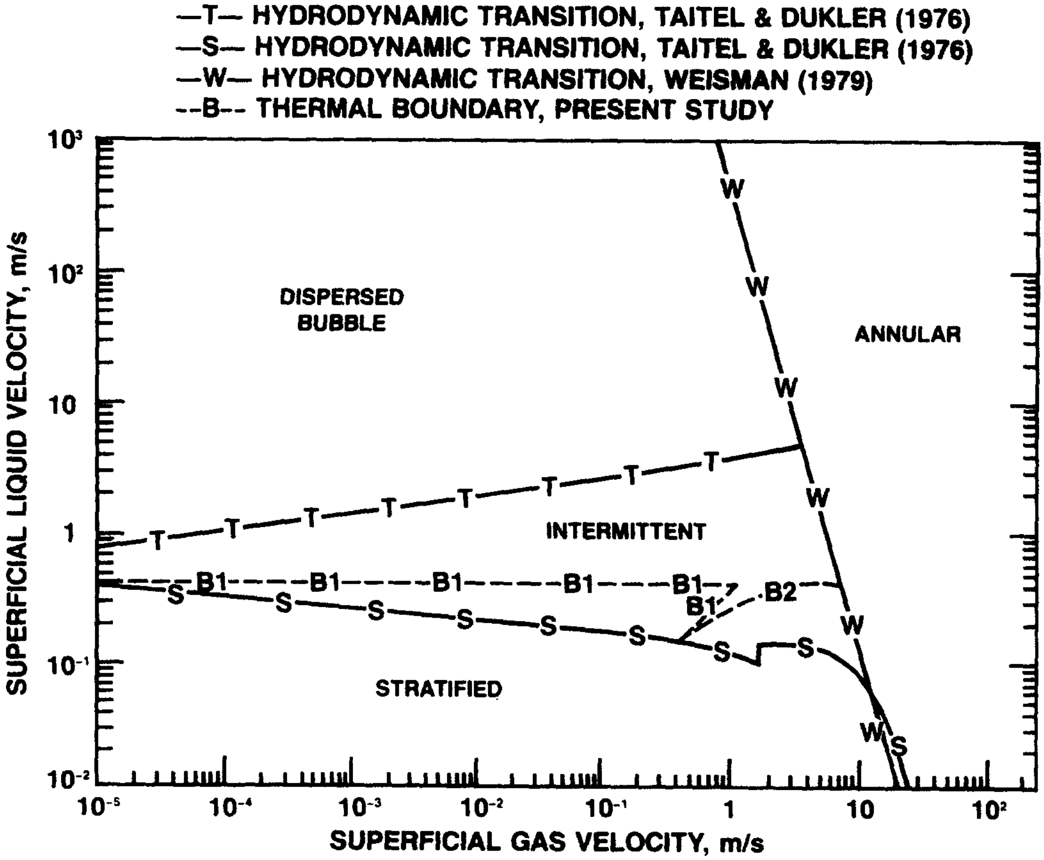

Figure 1. Possible boundaries for the aniosthermal operating zone in a horizontal boiler tube.

into low and moderate quality zones (locii B_1 and B_2), respectively, as explained in subsequent sections.

Low-quality anisothermal zone

The low-quality anisothermal zone was postulated by Bar-Cohen *et al.* (1984) to result from the presence of a stationary vapor bubble producing a locally stratified flow. While such a phenomenon may well be affected by entrance conditions, local stratification can be thought to result from the genesis of the intermittent (slug) pattern, developing from dispersed bubble flow as the buoyancy forces, acting on previously dispersed bubbles, begin to dominate.

In an attempt to define the maximum superficial liquid velocity in the pipe that is compatible with a stationary bubble, it is possible to examine the stability of a single, relatively large, bubble, or a locally stratified region, in a horizontal pipe via potential flow theory.

Referring to figure 2 and assuming a constant property flow, continuity requires that $V_1A_1 = V_2A_2$, where V_1 and V_2 are the liquid velocities at cross section 1 and 2, respectively. Similarly, applying the Bernoulli equation along a streamline, which coincides with the vapor/liquid interface, and assuming an isobaric bubble, yields

$$\frac{V_1^2}{2} + gz_1 = \frac{V_2^2}{2} + gz_2, \qquad [1]$$

where g is the gravitational acceleration and z the liquid level.

While A_1, the cross sectional area occupied by the liquid at the upstream end of the bubble, is simply equal to the cross sectional area of the pipe, A_2 can only be determined from a knowledge of the bubble/liquid interface geometry. Using values derived from Benjamin (1968), the liquid level at the downstream end of the bubble is found to equal 1.126 times the pipe radius and A_2 can then be shown to equal 0.58 of A_1. Using these values in [1], the upstream liquid velocity required for bubble stability is found to equal

$$V_1 = 0.68\,(gD)^{0.5}, \qquad [2]$$

where D is the pipe diameter.

It may thus be postulated that the locally stratified condition can only exist for liquid velocities lower than given by [2]. Defining a superficial liquid velocity Froude number, $\equiv U_L/\sqrt{gD}$, the low quality anisothermal/isothermal boundary can then be given by $Fr_L \simeq 0.68$. Or considering the assumptions employed in deriving this value, the analytic,

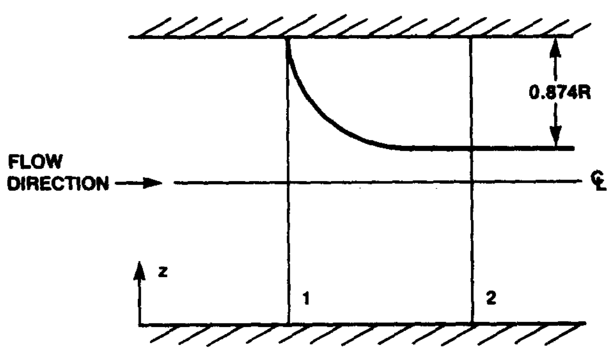

Figure 2. Stationary bubble in a horizontal pipe.

superficial liquid velocity boundary at low qualities could, perhaps, best be approximated by $Fr_L = 1.0$, as indicated by locus B1 in figure 1.

This relation can be expected to apply to the low-quality anisothermal zone associated with the dispersed bubble flow regime. However, it is to be noted that under diabatic conditions, i.e. heat addition at the pipe wall, Taitel (1980) has found the dispersed bubble/intermittent boundary to shift to significantly higher gas velocities than encountered in a purely hydrodynamic transition or sharply to the right on the superficial velocity flow regime map. The line of demarcation between the locus of B1 and B2 in figure 1 is based on Taitel (1980) and represents the dispersed bubble/intermittent boundary for the atmospheric pressure flow of a water/steam mixture with an applied heat flux of 100 kW/m^2.

Moderate-quality anisothermal zone

The transition from stratified to intermittent flow has previously been shown (Bar-Cohen *et al.* 1983) to be insufficient to assure isothermal operation of horizontal boiler tubes and the moderate-quality anisothermal zone can, in fact, penetrate deeply into the intermittent regime. The extent of this penetration would appear to depend on the relationship between the forces working to remove liquid from the surface (i.e. evaporation and drainage) and the forces working to rewet that same surface (i.e. passage of liquid slugs). Anisothermal operation can be expected to persist as long as the frequency of slug passage is insufficient to maintain a continuous liquid film on the surface.

An accurate calculation of the rates of both evaporation and drainage, based on the precise evaluation of the boundary layer behavior and momentum transport phenomena, was first carried out by Coney (1974). For purposes of this discussion, it is convenient to examine drainage and evaporation separately.

Following Bird *et al.* (1960), the velocity profile in a gravitationally draining liquid film is given by

$$v_z = \frac{\rho_L g\,(\delta_L)^2 \cos\beta}{2\mu_L}\left[1 - \left(\frac{x-\delta}{\delta}\right)^2\right], \qquad [3]$$

where v_z is the tangential velocity, ρ_L, μ_L, and δ are the liquid density, viscosity and film thickness, respectively, while x, z, and β are the geometric parameters designated in figure 3.

Integrating the velocity profile across the film thickness, the flow rate per unit width is found to equal

$$Q = \int_0^{\delta} v_z\,\mathrm{d}x = \frac{1}{3}\frac{\rho_L g \delta^3}{\mu_L}\cos\beta. \qquad [4]$$

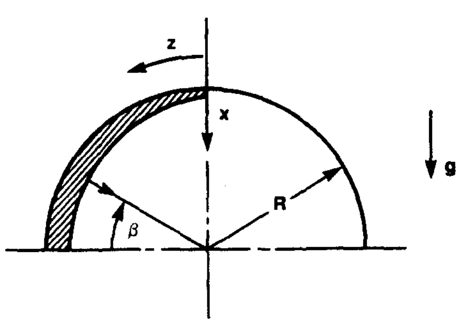

Figure 3. Geometry and coordinate system for draining liquid film.

Continuity considerations dictate that $dQ/dz = -\partial\delta/\partial t$ and, hence,

$$\frac{\partial \delta}{\partial t} = -\frac{g\rho_L}{3\mu_L}\frac{\partial}{\partial z}\left(\delta^3 \frac{z}{R}\right), \quad [5]$$

where $(z/R) = \cos\beta$ (see figure 3).

To find the time-wise variation of the film thickness at the top of the pipe, δ_T, recall that initially the film thickness is δ_0 everywhere and that due to symmetry $\partial\delta/\partial z$ at the top of the pipe (i.e. $z = 0$) equals zero. With these constraints [5] can be shown to yield

$$\frac{1}{\delta_T^2} = \frac{1}{\delta_0^2} + \frac{2g\rho_L}{3\mu_L R}t. \quad [6]$$

It is to be noted that, in the interest of providing an analytically tractable expression, [6] neglects the impact of surface tension and curvature of the liquid/vapor interface on the drainage rate. While these factors can be expected to grow in importance as the pipe diameter decreases, [6] is thought to offer an acceptable estimate of the gravitationally-driven drainage rate in the relevant parametric range.

The contribution of evaporation to this process can be estimated more directly by determining the film thinning rate associated exclusively with evaporation. Performing a simple heat balance,

$$\frac{\partial \delta}{\partial t} = -\frac{q''}{h_{LG}\rho_L}, \quad [7]$$

where q'' is the imposed heat flux and h_{LG} the latent heat. Integrating this relation, the time required to completely evaporate the film, t_{ev}, is found as

$$t_{ev} = h_{LG}\rho_L\delta_0/q''. \quad [8]$$

Based on the empirical values obtained by Fisher & Yu (1975), for dry-out in tubes, the initial water film thickness can be estimated to equal 50–70 μm. For initial film thickness values in this range, boiler tube heat fluxes of 200 kW/sq m, tube diameters of 0.0508m and operating pressures of 1, 36 and 112 bar, [8] yields film evaporation times of approximately 0.54, 0.35 and 0.21 sec, respectively. During comparable time periods, drainage effects, estimated via [6], could be expected to decrease the film thickness by approximately 35%. As the pipe diameter, pressure and heat flux grow, the role of drainage can be shown to diminish even further in importance. It, thus, appears that in the primary operating range of industrial boiler tubes, drainage effects can be neglected relative to evaporation in the determination of the moderate-quality boundary between anisothermal and isothermal operation and [8] used to provide a first-order estimate of the time required to dry out the top of the pipe.

In the intermittent flow regime and especially at the relatively high superficial vapor velocities appropriate to the moderate-quality region, the liquid "slugs" are considerably shorter than the highly elongated vapor "bubbles." As a consequence, the upper surface of the pipe is only very briefly washed by the liquid slug and the time available for evaporation/drainage is nearly equal to the period of time separating the leading edges of two consecutive slugs. The slug period, t_{sl}, can be obtained from the empirical correlation given by Greskovich & Shrier (1971) as

$$t_{sl} = 44.25\left\{\frac{U_L}{gD}\left[\frac{19.75}{U_L + U_G} + U_L + U_G\right]\right\}^{-1.2}, \quad [9]$$

where U_L and U_G are the superficial velocities of liquid and gas, respectively and t_{sl} is given in sec. This correlation, based on experimental data for adiabatic, air/water, horizontal slug flow, is said by the authors to be almost unaffected by tube diameter in the range of 0.03–0.15 m I.D.

Equating t_{sl} from [9] and t_{ev} from [8] yields the operating conditions for which dry-out is just avoided, that is

$$\frac{h_{LG}\rho_L\delta_0}{q''} = 44.25\left[\frac{U_L}{gD}\left(\frac{19.75}{U_L + U_G} + U_L + U_G\right)\right]^{-1.2}. \qquad [10]$$

By solving [10] numerically for U_L and U_G, the boundary separating isothermal from anisothermal operation can be obtained and superimposed on the Taitel & Dukler (1976) flow regime map. The B2 curve in figure 1 obtained via [10] is typical of such a moderate quality boundary. Interestingly, as shown in figure 5 and in Ruder (1984), the moderate-quality boundary determined in this fashion is only slightly below the locus of this boundary and, as expected, can be shown to lie slightly below the locus of this boundary based on the Coney (1974) analysis, which includes both evaporation and gravitational drainage.

While the numerical solution of [10] establishes the desired boundary between anisothermal and isothermal operation, the availability of an analytic expression for the maximum superficial liquid velocity along this boundary could considerably streamline the system design process. Following Ruder (1984) such a relation can be derived from [10] by expressing U_G as a function of U_L and letting $\mathrm{d}U_G/\mathrm{d}U_L$ approach infinity. The maximum liquid superficial velocity, $\hat{U}_L$, is then found to equal

$$\hat{U}_L \simeq B/8.9, \qquad [11]$$

where $B = gD\,(q''/0.0226\, h_{LG}\rho_L\delta_0)^{0.833}$ and U_L is in m/sec. This peak liquid velocity can be expressed in terms of the previously defined Froude number, yielding $\hat{\mathrm{Fr}}_L = B/8.9\,\sqrt{gD}$.

The boundary between anisothermal and isothermal operation, established by the solution of [10], can be expected to apply in the moderate quality zone, extending from the dispersed bubble/intermittent flow regime boundary (appropriate to the relevant diabatic condition) to the intermittent/annular flow boundary. While several different expressions are available for the superficial gas velocities at which this latter transition occurs, Bar-Cohen *et al.* (1983) have previously found the Weisman *et al.* (1979) relation to most closely predict the onset of circumferential isothermality at moderate qualities and it is, thus, this expression that will be used to define the terminal superficial gas velocity for the moderate-quality anisothermal zone.

The Weisman *et al.* (1979) intermittent/annular flow transition is itself expressed in the form of a modified Froude number which, in addition to the superficial gas velocity, tube diameter and gravitational acceleration, attempts to address the secondary influence of the superficial liquid velocity, the gas and liquid densities and the surface tension. When expressed in a form consistent with other Froude numbers in this study, the Weisman-derived bound on the superficial gas velocity is given by

$$\mathrm{Fr}_G \equiv U_G/(gD)^{0.5} = \left(\frac{1.9(U_G/U_L)^{0.125}\,[g(\rho_L - \rho_G)\sigma]^{0.05}}{U_G^{0.2}\,\rho_G^{0.1}}\right)^{2.78}, \qquad [12]$$

where ρ_G is the gas density and σ is the surface tension.

VERIFICATION OF THE THERMAL BOUNDARIES

Introduction

The previous chapter has presented the development of analytical expressions for the boundaries separating isothermal and anisothermal operation of horizontal boiler tubes.

While these boundaries were derived from theoretical considerations, their development was motivated by experimental results for certain operating conditions ($p = 1$ bar, $D = 0.0254$ m, $q'' = 4$–100 kW/m^2) reported by Bar-Cohen *et al.* (1984). To establish the generality of the proposed thermal boundaries, it is necessary to compare the physically based boundary relations with the empirical results available in the literature for different working conditions. In keeping with the preceding discussion, this comparison will be displayed against the background of the superficial velocity flow regime map (in log-log coordinates) for the conditions of each investigation.

It should be noted that while on the flow regime map, the thermal and hydrodynamic boundaries appear as distinct lines, these boundaries are meant to represent separation zones between the various regimes. The exact extent of the separation zones or, in fact, the error band around each boundary is difficult to estimate but may well lie in the ±25% suggested by Weisman *et al.* (1979) for the hydrodynamic intermittent/annular transition.

Styrikovich & Miropolski (1950, 1956)

The 182-bar working pressure results of the Styrikovich & Miropolski study (1950, 1956), analyzed and discussed by Bar-Cohen *et al.* (1983), are plotted in figure 4 along with the thermal boundaries suggested herein. As can be seen, according to the reported maximum and minimum top-to-bottom temperature differences shown on the map, the reported anisothermality for all three investigated liquid superficial velocities of 0.3, 0.6 and 0.9 m/sec, is associated with the moderate-quality or intermittent flow regime zone.

It is interesting to note that while liquid superficial velocities of 0.7–0.8 m/sec are associated with the $Fr_L = 1$ boundary for the low-quality anisothermal region, the peak liquid velocity in the moderate quality zone is calculated via [11] to equal approximately 5.7 m/sec. The transition between dispersed bubble and intermittent flow, appropriate to this diabatic situation and constituting the boundary between the low- and moderate-quality anisothermal zone, was determined on the basis of the Taitel (1980) formulation.

The empirical points of Styrikovich & Miropolski are seen in figure 4 to fall well below the curve defined by [11] even for the highest tested liquid velocity. The experimentally

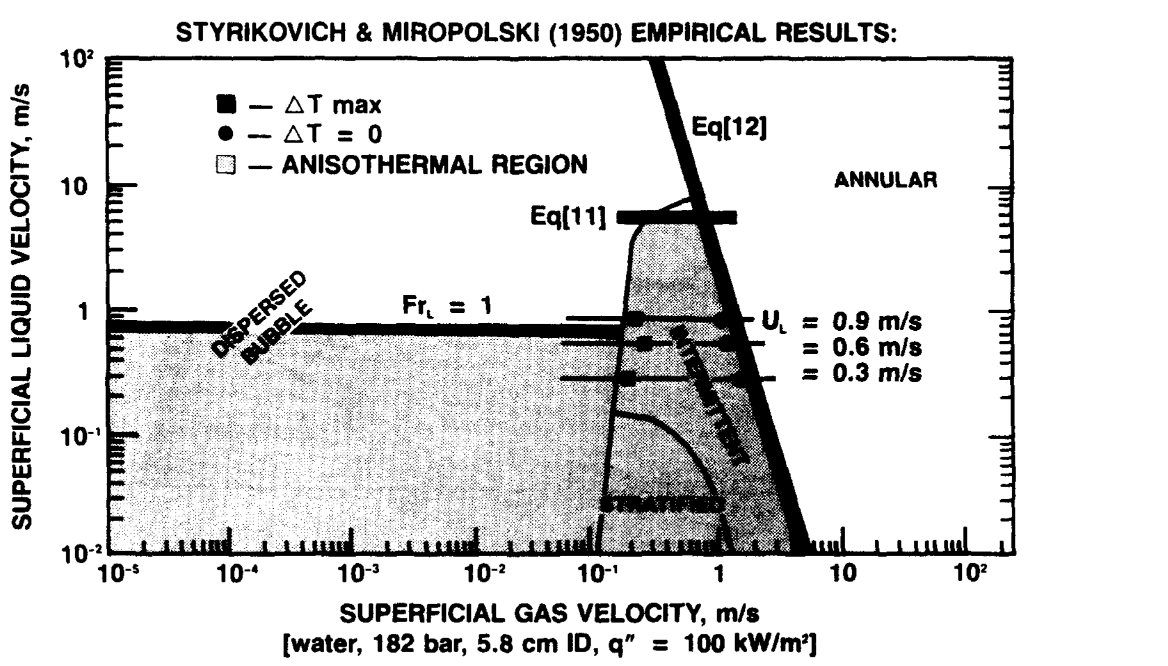

Figure 4. Thermal regime map for the conditions of Styrikovich & Miropolski (1950).

determined minimum ΔT points lie in this figure just to the left of the right-hand thermal boundary represented by the modified (Weisman) Froude number ([12]), i.e. at superficial vapor velocities quite close to the predicted values.

Similar results are obtained at two other pressures tested by Styrikovich & Miropolski (1950, 1956), i.e. 36 and 112 bar, for which the moderate-quality, superficial liquid bounding velocities appear to be 2 and 3.1 m/sec, respectively. It may thus be concluded that the Styrikovich & Miropolski (1950) anisothermal zones lie within the regions defined by the proposed thermal boundaries.

Rounthwaite (1968)

Rounthwaite (1968) measured the circumferential temperature profiles at several axial locations along two 6.2 m long, 4.13 cm I.D. uniformly heated pipes which were connected so as to form a continuous U-tube. His raw data and the corresponding thermal boundary were discussed by Bar-Cohen *et al.* (1983) and also cited by Lis & Strickland (1970).

In the Rounthwaite (1968) investigation circumferentially isothermal operation was encountered at all qualities and pipe locations at the highest mass flux tested of 290 kg/m^2-sec, while in the range of 100 to 240 kg/m^2-sec both isothermal and anisothermal zones were detected along the pipe. The empirical locus of the transition between these two zones is shown in figure 5 and compared to the semianalytic thermal boundaries for both the low- and high-quality regions.

An examination of this figure reveals that the Rounthwaite (1968) boundary is significantly below the semianalytic, moderate-quality liquid velocity boundary (both [10] and based on Coney 1974) but does display the anticipated variation with gas velocity and terminates in close proximity to the intermittent/annular transition given by [12]. The large

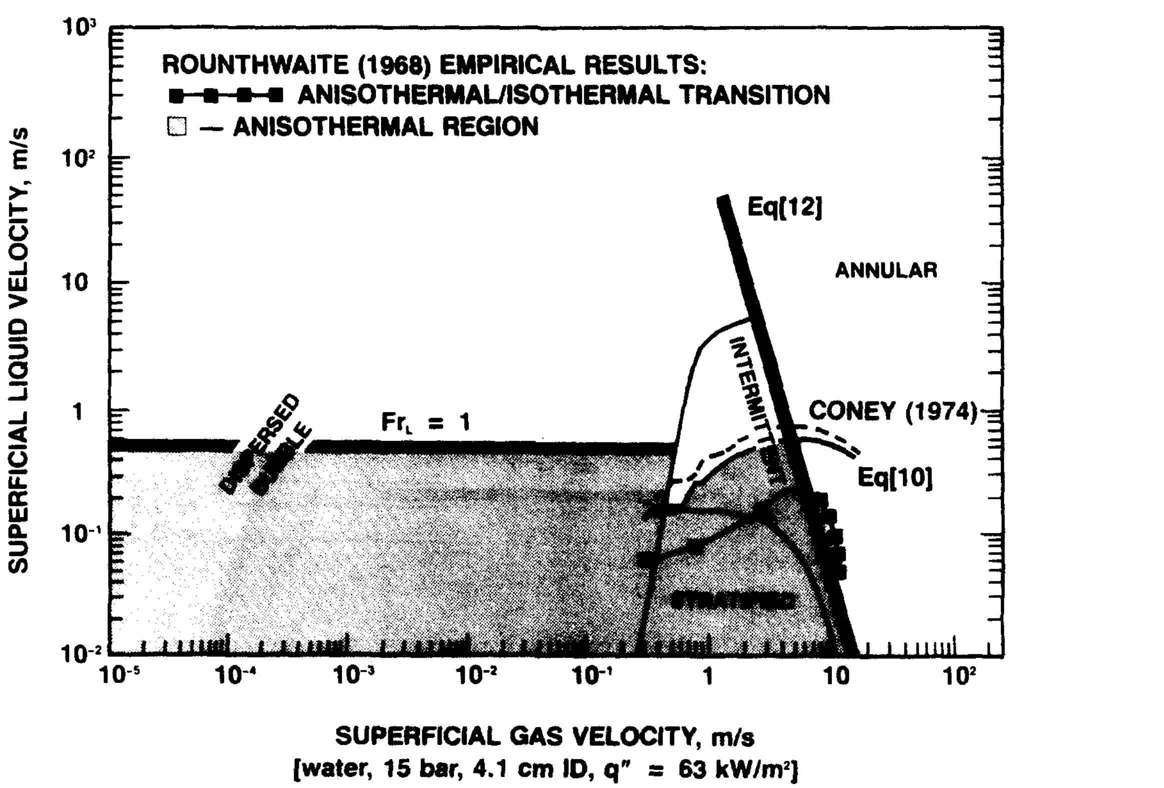

Figure 5. Thermal regime map for the conditions of Rounthwaite (1968).

discrepancy in bounding liquid velocity is reminiscent of the atmospheric pressure results reported by Bar-Cohen *et al.* (1984). The calculated $\hat{U}_L$ for the Rounthwaite condition equals approximately 0.9 m/sec versus the reported value of 0.3 m/sec, while for the Bar-Cohen *et al.* condition (1 bar, 100 kW/m^2) the calculated peak mass flux equalled 600 kg/m^2-sec versus a measured value of approximately 200 kg/m^2-sec. The overly "conservative" nature of [10] and [11] is thought to be associated with shortcomings of the empirical slug frequency correlation of Greskovich & Shrier (1971) and is discussed in Ruder (1984).

The Rounthwaite results do suggest, however, that at near atmospheric ($\leq$15 bar) operating pressures, the low-quality isothermal/anisothermal boundary, i.e. $Fr_L = 1$, may be an acceptable criterion for the moderate-quality zone, as well, and thus, replace the use of [10] and [11] in this operating range. For the conditions of Rounthwaite's investigation, the $Fr_L = 1$ criterion yields a bounding superficial liquid velocity of 0.6 m/sec.

Lis & Strickland (1970)

As in the previously described study by Rounthwaite (1968), Lis & Strickland (1970) report the empirical thermal boundaries for the operating pressures of 35.5, 52.5 and 66.5 bars, heat flux of 63 kw/m^2 and pipe diameter 4.1 cm, along with the raw data for various mass flow rates. As seen in figure 6, for 66.5 bars the experimentally found moderate quality thermal boundary almost coincides with that calculated using [10]. Similar, though somewhat less precise, agreement is found at the two other operating pressures (Ruder 1984).

As for the Lis & Strickland raw data, the extreme anisothermality, for the working conditions of $p = 15$ bars, $q'' = 62$ kw/m^2 and $G = 490$ kg/m^2-sec at a steam quality of 19%,

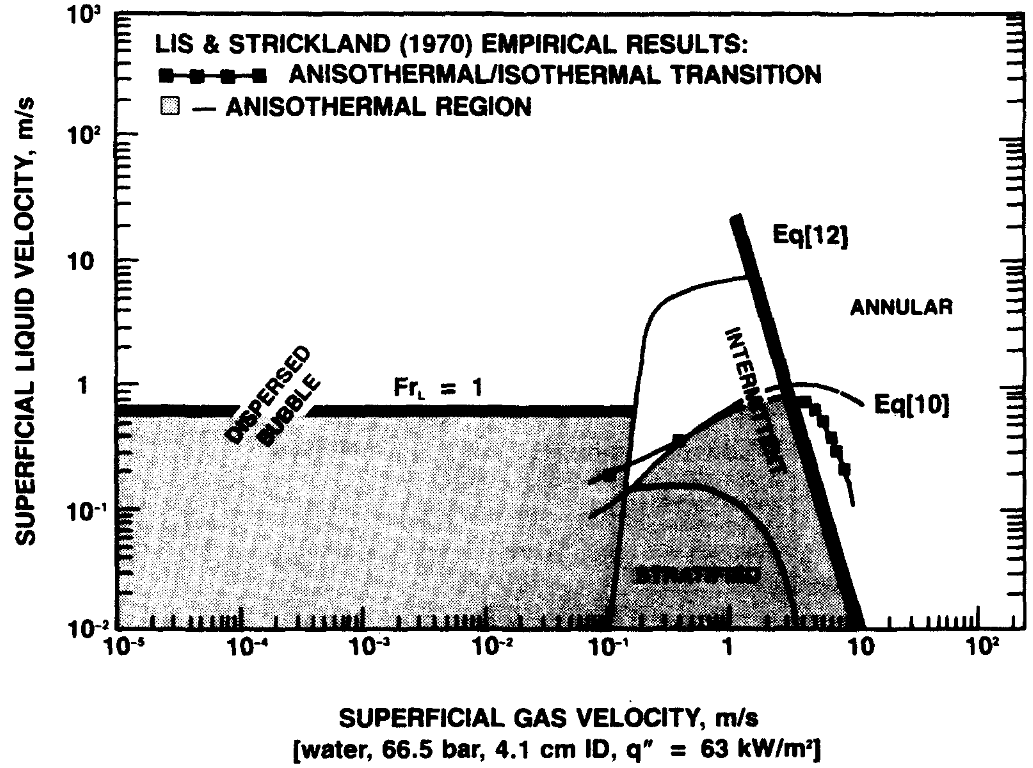

Figure 6. Thermal regime map for the conditions of Lis & Strickland (1970).

can be shown to occur at a point on the thermal regime map well within the anisothermal zone, i.e. below the moderate-quality thermal boundary ($G \simeq 860$ kg/m^2-sec) and only slightly above the low-quality boundary of $\mathrm{Fr}_L = (G \simeq 480$ kg/m^2-sec).

Most of the temperature variations observed by the authors were due, however, to the effects of various inlet bends placed just near the entrance to the test section. In this connection it is of interest to note that the straight calming section, with a length-to-diameter ratio of about 56 between the bends and the test pipe, appeared to be insufficient to prevent the entrance-induced anisothermality. This result is consistent with the conclusion of Bar-Cohen *et al.* (1984) that an L/D ratio not less than 90 is needed to eliminate entrance effects.

Bonn et al. *(1980)*

In the experiments carried out by Bonn *et al.* (1980), the flow of R-12 in a 14-mm I.D. copper tube was found to produce significant circumferential temperature variations for mass fluxes, G, equal to 130 kg/m^2-sec as the heat flux varied from 1.4 to about 80 kW/m^2. The authors related this behavior to progressive dry out of the annular film. The operating locii are, however, shown in figure 7 to fall well below even the $\mathrm{Fr}_L = 1$ defined lower quality anisothermal boundary which would thus imply that dry-out, in this case, resulted from the stratified/intermittent flow pattern rather than that suggested by the authors.

In this case, the criterion $\mathrm{Fr}_L \simeq 1$ would predict anisothermality to occur at any liquid flow rate less the $U_L = 0.4$ m/sec, i.e. $G = 570$ kg/m^2-sec.

Robertson (1972)

In his investigation of dry-out in horizontal hairpin boiler tubes, Robertson (1972) presents pipe temperature data for a water mass flux of 688 kg/m^2-sec, as well as for an

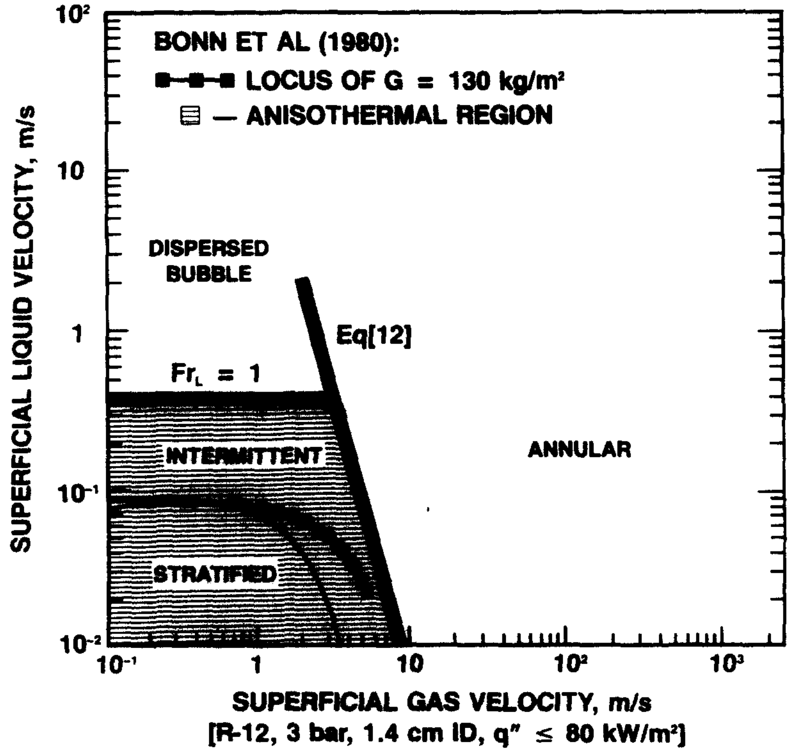

Figure 7. Thermal regime map for the conditions of Bonn *et al.* (1980).

unusually high mass flux of approximately 1375 kg/m^2-sec and a "torrid" imposed heat flux of nearly 1 MW/m^2. Interestingly, while at the low mass flux wall temperature fluctuations were found to occur along the top of the tube for a wide range of qualities, similar fluctuations were limited to qualities in excess of 24% for the high mass flux and imposed heat flux.

Referring to figure 8, it can be seen that although both values of G are lower than the moderate-quality thermal boundary, the highest operating mass flux generates a locus located above the intermittent flow pattern while the locus of $G = 687.5$ kg/m^2-sec traverses the upper end of the moderate-quality (intermittent flow) zone. Thus, [10] can be expected to apply only to the data of the lower mass flux, and no anisothermality should be encountered at $G = 1375$ kg/m^2-sec until well into the annular flow pattern where total dry out may occur.

Tang & Howe (1981)

Echoing the Robertson (1972) results, Tang & Howe (1981) report no tube temperature oscillations to occur for a water mass flow rate of 2200 kg/m^2-sec at the operating pressure of about 15 bar in a 2.54-cm I.D. horizontal pipe under a heat flux of 221 kW/m^2.

The locus, which can be traced out on the corresponding flow and thermal regime map for these operating conditions (figure 9), appears to proceed slightly below the intersection point of the dispersed bubble/intermittent and the intermittent/annular regime boundaries. The point of intersection is associated with an approximate G value of 2800 kg/m^2-sec. The G value calculated according to [11] would, however, equal only 1282 kg/m^2-sec. The operating mass flux, thus, turns out to be within the intermittent regime but above the moderate quality anisothermal zone and, as a consequence, no circumferential temperature variations should be encountered.

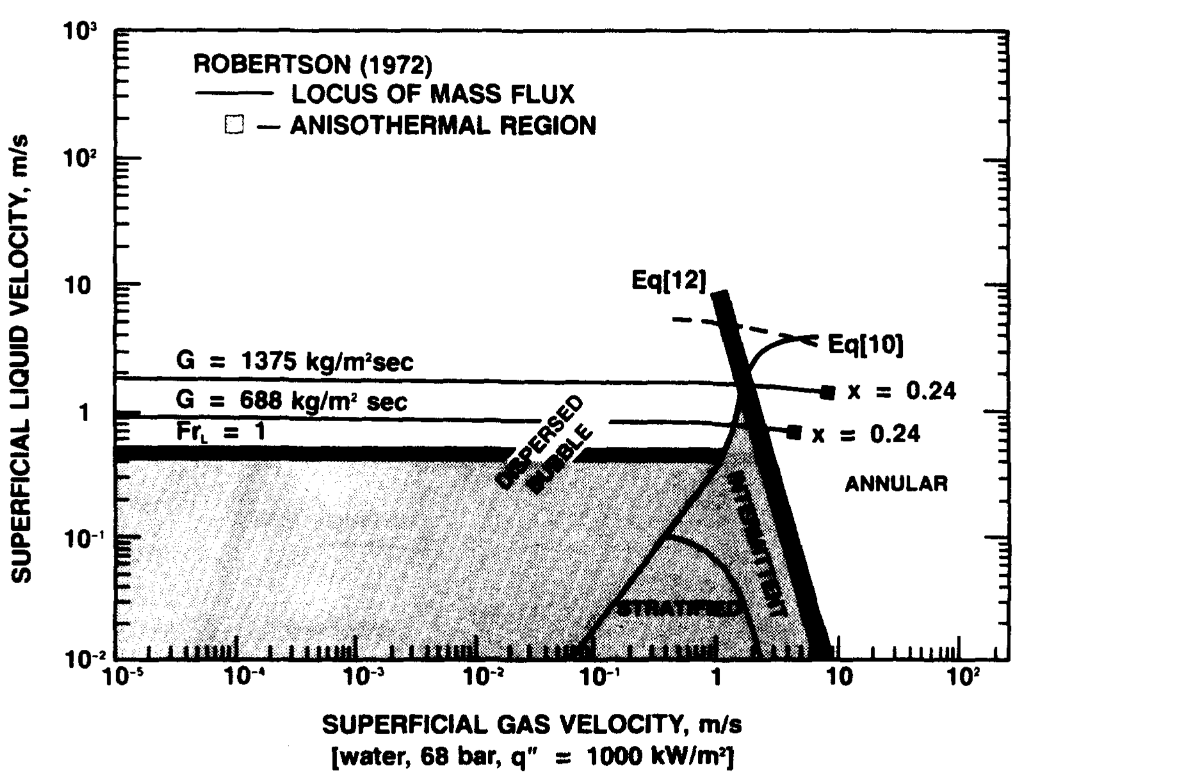

Figure 8. Thermal regime map for the conditions of Robertson (1972).

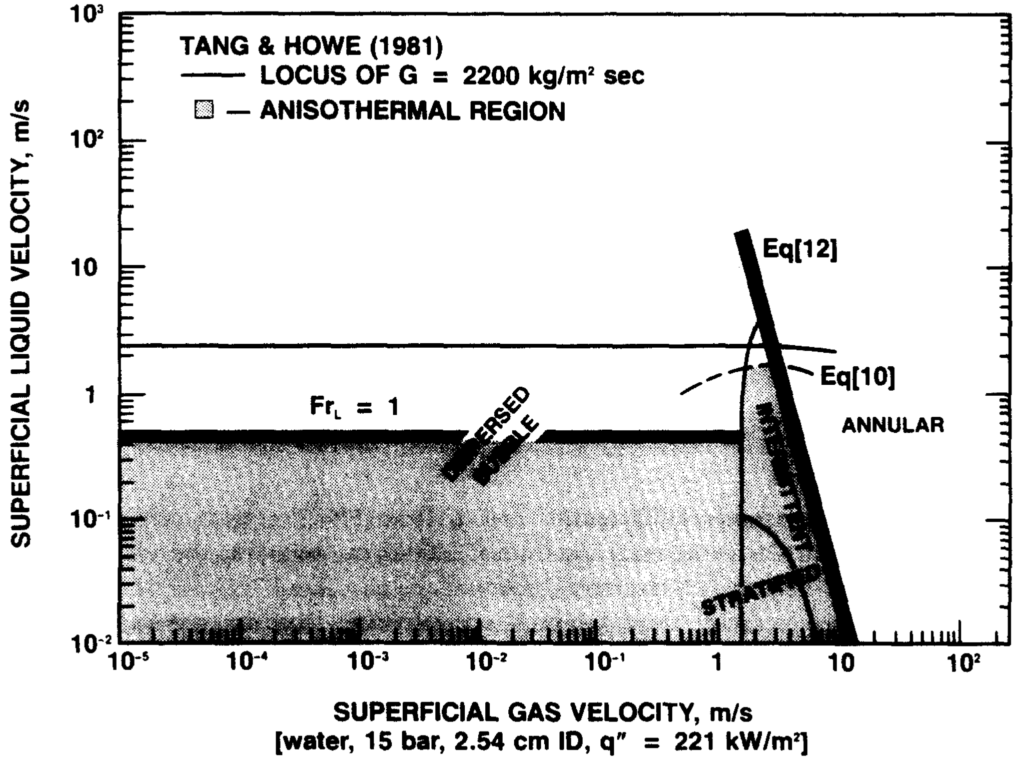

Figure 9. Thermal regime map for the conditions of Tang & Howe (1981).

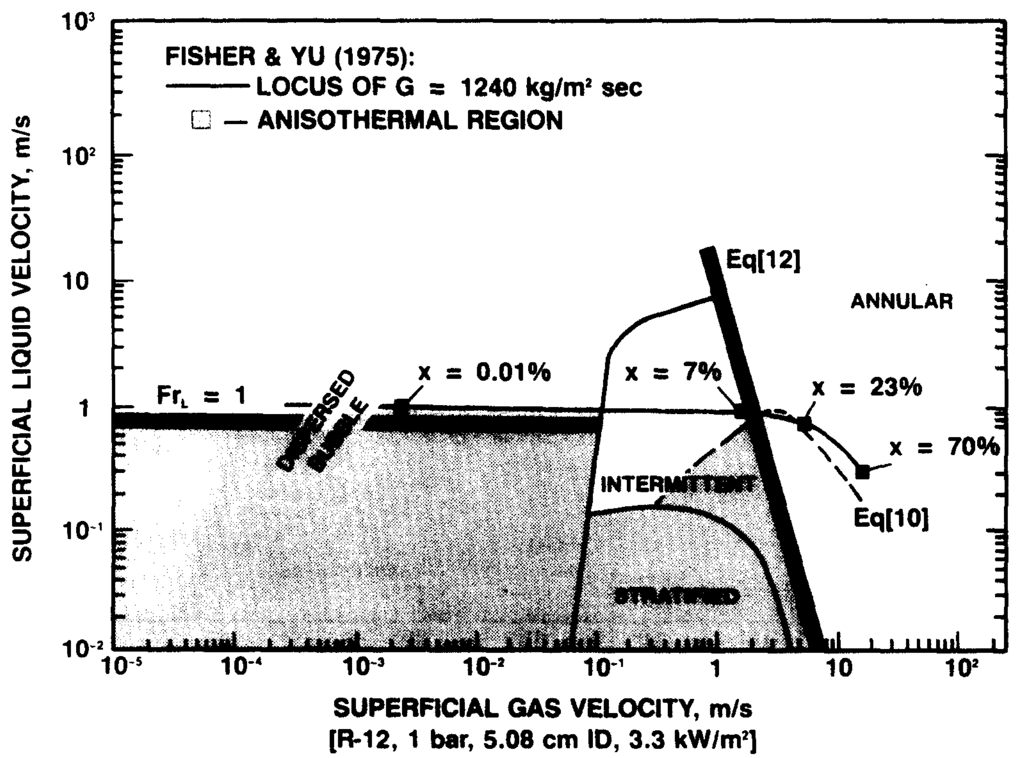

Figure 10. Thermal regime map for the low-pressure, low heat flux conditions of Fisher & Yu (1975).

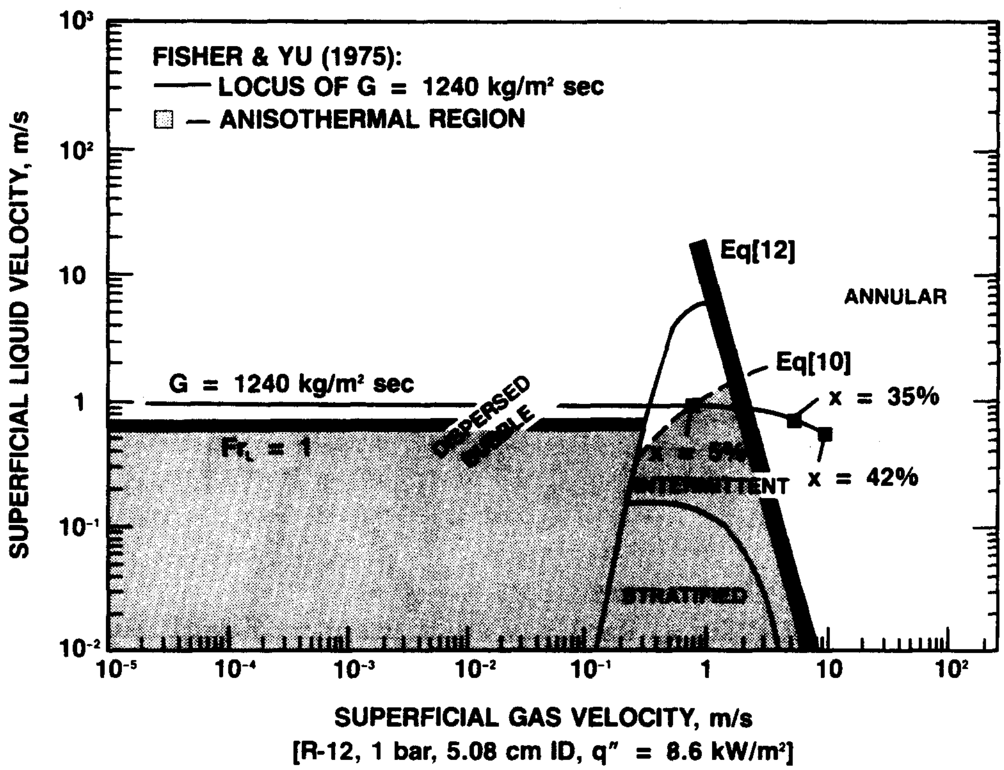

Figure 11. Thermal regime map for the low-pressure, high heat flux conditions of Fisher & Yu (1975).

Fisher & Yu (1975)

These investigators report their results for horizontal R-12 two-phase flow in a 5.08-cm I.D. tube ($p = 1$ bar, $q'' = 3.3$ and 8.6 kW/m^2; and $p = 3.55$ bar, $q'' = 6$–18 kW/m^2). The authors assumed that up to steam qualities of about 40% the observed dry-out was caused by the slug flow regime and that the patterns of dry-out were thus identical in both water/steam (Lis & Strickland 1970) and R-12 systems.

Fisher & Yu's findings, that for the low pressure tests ($G = 1240$ kg/m^2-sec, $q'' = 3.3$ kW/m^2) dry areas usually appeared at qualities between 5% and 10%, disappeared again at qualities of about 20% to 25% and reappeared only at qualities in excess of 70%, are seen in figure 10 to be in obvious agreement with the proposed moderate-quality thermal boundary [10] and even with the right-hand thermal bounding line, if the error band of the Weisman intermittent/annular transition is taken into consideration.

Similarly, for the same pressure and mass flow rate, but a higher heat flux ($q'' = 8.6$ kW/m^2) the first dry-out region was reported by Fisher & Yu (1975) to occur between the qualities of 5% and 10% to 35%, but the second anisothermal zone was encountered at qualities as low as 42%. These results also agree very satisfactorily with the suggested thermal boundaries shown in figure 11.

The same trend is seen in figure 12 for the R-12 high-pressure results obtained by Fisher & Yu (1975). The authors' conclusion that, for these working conditions ($q'' > 6$ kW/m^2), the prevention of dry-out, for qualities of 7% to 37%, requires mass velocities greater than 2530 kg/m^2-sec appears to testify to the validity of [10] and [12] when account is taken of the error band associated with the Weisman *et al.* (1979) boundary.

SUMMATION

The preceding has focused on the analytical development of superficial velocity boundaries which separate circumferentially isothermal and anisothermal operating regions

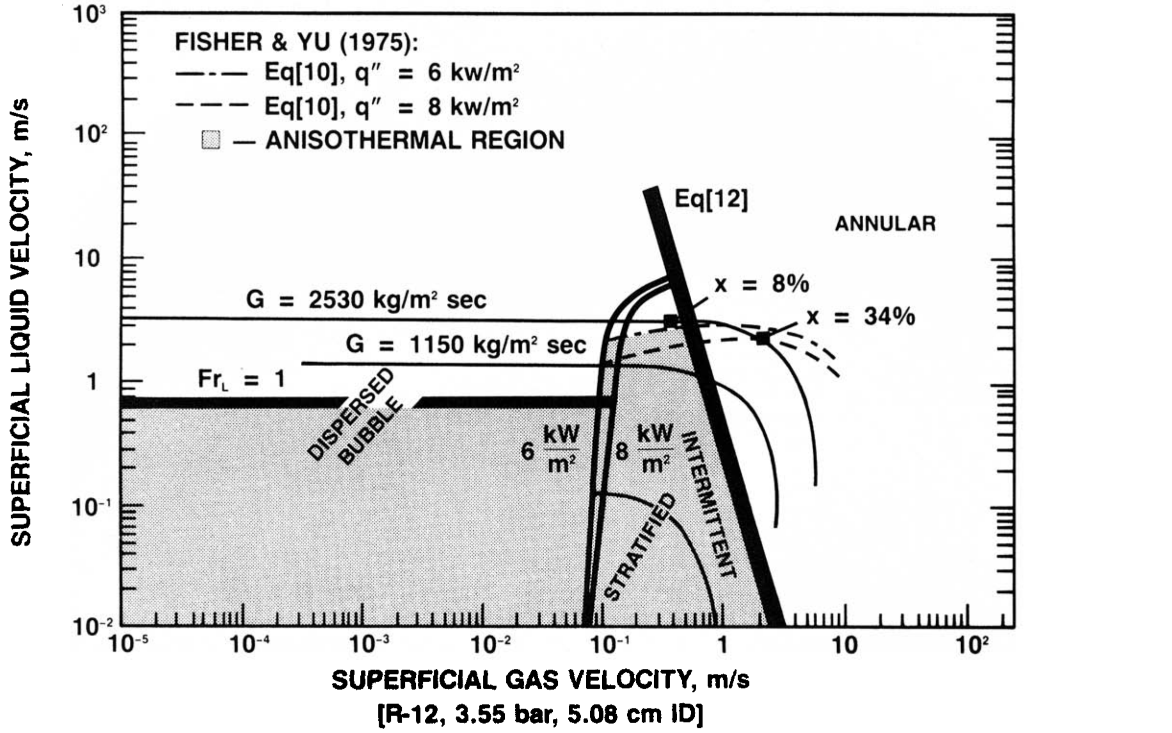

Figure 12. Thermal regime map for the high-pressure conditions of Fisher & Yu (1975).

in horizontal boiler tubes. It appears that these boundaries can be expressed by means of modified Froude numbers based on the superficial liquid (low- and moderate-quality zones) and gas (right-hand bounding line) velocities.

Comparison of the proposed bounding curves with data available in the literature revealed that the right-hand boundary associated with the Weisman intermittent/annular flow regime transition [12] successfully predicts the maximum superficial gas velocity edge of the anisothermal zone for both water/steam and freon two-phase flow for all reported operating conditions. The upper bounding curve of the moderate quality anisothermal zone calculated semianalytically [10] was found to be valid for water/steam and freon data but to overestimate the critical water U_L values for relatively low, industrial pressures. This may be associated with the shortcomings of the existing empirical slug frequency correlation and/or with the insufficiently understood hydrodynamic behavior of the intermittent regime.

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