Int. J. Multiphase Flow Vol. 9, No. 1, pp. 1-12, 1983
Printed in Great Britain.

0301-9322/83/010001-12$03.00/0
Pergamon/Elsevier

CIRCUMFERENTIAL WALL TEMPERATURE VARIATIONS IN HORIZONTAL BOILER TUBES

AVRAM BAR-COHEN and ZVI RUDER
Department of Mechanical Engineering, Ben-Gurion University of the Negev, Beer Sheva, Israel

and

PETER GRIFFITH
Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, MA 02139, U.S.A.

(*Received* 20 *September* 1981; *in revised form* 19 *April* 1982)

Abstract—The design of horizontal boiler-tube arrays, for both fluidized bed combustors and conventional power plants, is today constrained by the uncertainty in the circumferential wall temperature profile prevailing in the tubes. High circumferential anisothermality at elevated temperatures can lead to high thermal stress and threaten both the load carrying and operating life of steam-generating boiler tubes.

A critical review of the available results and a detailed analysis of two-phase flow regimes in boiler tubes appear to indicate that a reduced circumferential temperature difference is associated with transition from intermittent to annular flow, rather than the termination of stratified flow. The two-phase flow regime analysis is, furthermore, shown to successfully predict the axial variation and flow rate dependence of circumferential anisothermality.

INTRODUCTION

The growing interest in boiler applications of fluidized bed combustors (FBC) has redirected attention to the thermal characteristics of horizontal, steam-generating pipes. In this configuration, partial or complete stratification of the two-phase (steam–water) flow may occur, leading to circumferential anisothermality and elevated thermal stresses. The design of horizontal boiler tube arrays, for FBC's as well as for conventional fossil-fuel boilers, is today constrained by the uncertainty in the circumferential wall temperature profile prevailing in the pipe and the associated ambiguity in the thermal stresses, which may threaten the load carrying the operating life of the steam-generating tubes (Glicksman *et al.* 1977).

While the literature abounds with studies of two phase flow and flow boiling, only limited data are available on the circumferential temperature variation in a horizontal tube. These data, notably as reported by Styrikovich & Miropolski (1950) for high pressure water/steam flow, suggest that there is a distinct two-phase velocity above which the tube can be considered circumferentially isothermal. The oft quoted Styrikovich & Miropolski (1950) result has, however, remained a nearly-isolated empirical finding and a detailed literature search reveals no attempt to critically evaluate this result in light of today's understanding of two-phase flow regimes nor any attempts to generalize these findings to other working fluids, operating conditions and/or tube diameters.

The present effort represents just such an attempt and commences with a brief review of flow regimes in two-phase, horizontal pipe flow. This is followed by a critical analysis of the reported axial and flow rate dependence of the circumferential temperature profile, leading to physically-based criteria for both the peak anisothermality and circumferentially isothermal conditions. It is to be noted that circumferential anisothermality, associated with "dryout" of the liquid film in high-quality annular flow, is not addressed in this study.

FLOW REGIMES IN HORIZONTAL TWO-PHASE FLOW

Adiabatic flow

The thermal and hydrodynamic characteristics encountered in the adiabatic flow of gas-liquid mixtures in pipes have been the subject of many investigations and numerous flow regimes have been identified. In general, as the gas fraction increases, transition is made from flow regimes in which the gas is dispersed in the liquid to regimes, at high gas fraction, where the liquid droplets are the dispersed phase. The presence of gravity and shear forces between the two phases leads to various sub-categories of these flow regimes, but until recently prediction of the precise flow regimes, even for concurrent gas/liquid flow in horizontal pipes, was beyond the state-of-the-art.

One of the earliest flow regime maps was generated by Baker in 1954, but, despite its initial success in rationalizing some of the observed phenomena, attempts to develop general two-phase flow maps proved unsuccessful. More recently, however, superficial gas and liquid velocity coordinates were found by Mandhane *et al.* (1974) to provide satisfactory mapping of a large data base. The use of these coordinates in horizontal pipe flow was subsequently rationalized by use of a mechanistic model developed by Taitel & Dukler (1976).

In this model, Taitel & Dukler (1976) examine stratified flow and focus on the mechanism by which a flow transition occurs to a new flow pattern. Proceeding in this manner they define 4 flow regimes: stratified (wavy and smooth), intermittent, annular-dispersed liquid and dispersed bubble. These flow regimes are illustrated in figure 1 for a water/air mixture flowing in a 2.5 cm smooth, horizontal pipe.

As can be seen in figure 1, stratified flow is associated with low values of both liquid and gas superficial velocity. Increasing liquid velocity results in transition to intermittent flow and at higher velocities to dispersed bubdle flow. Alternatively, increasing gas superficial velocities appears to lead to stratified wavy flow and the flow pattern progresses into annular-dispersed liquid at very high gas velocities.

The flow regime boundaries in the Taitel & Dukler model are derived from distinct physical phenomena. Thus, transition from stratified to intermittent and annular dispersed liquid flow is defined to occur when the conditions are such that a finite amplitude wave on the stratified liquid will grow. Similarly, the transition from smooth stratified to wavy stratified flow is ascribed to gas velocities sufficient to cause wave formation but less than required for the unstable growth of the previous transition.

When the liquid superficial velocity is high, gas bubbles present in the stream are acted upon by turbulent fluctuations that overcome the buoyant forces tending to stratify the flow. Following Taitel & Dukler (1976), dispersion of the gas and transition between the intermittent and the dispersed bubble regimes can be related to a derivable liquid velocity.

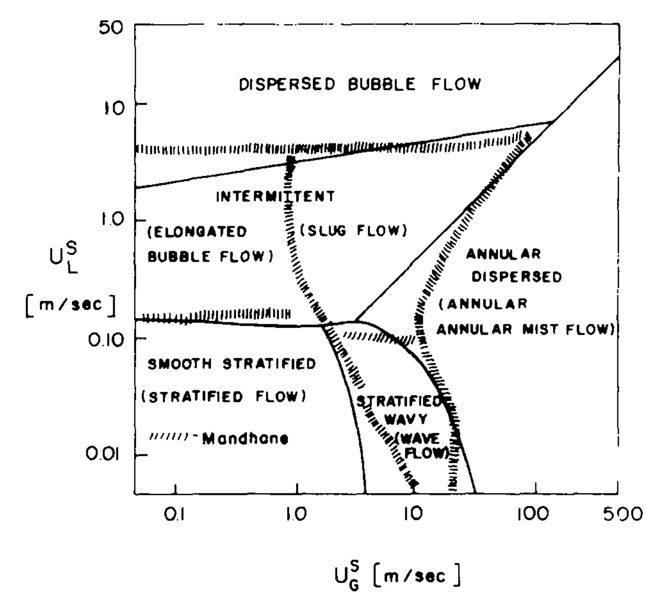

Figure 1. Flow regimes in horizontal pipe air/water flow (2.5 cm diameter, 25°C, 1 bar) (Taitel & Dukler 1976, Mandhane *et al.* 1974).

The final flow regime boundary, separating intermittent and annular-dispersed liquid flow, is, according to Taitel & Dukler (1976), simply related to the liquid level in the pipe. When the liquid level is below the pipe centerline, unstable growth on the liquid level results in the formation of an annular film, whereas for $h_L/D > 0.5$ a stable liquid bridge and intermittent flow is obtained. Thus $h_L/D = 0.5$ serves as the requisite flow regime boundary, where h_L is the liquid level in the pipe and D the pipe diameter.

Calculation of the specific flow regime boundaries requires simultaneous solution of the Navier–Stokes equation for each of the two phases to determine the liquid level in the pipe. This parameter and appropriate continuity relations then allow comparison of the gas or liquid velocity to the various transition criteria. The details of these calculations can be found in Taitel & Dukler (1976).

In keeping with the Taitel & Dukler model, a detailed analysis of extensive, new data performed by Weisman *et al.* (1979) revealed that the major influences on the flow patterns in horizontal pipes are the liquid and gas superficial velocities and that fluid properties and pipe diameter exert only a moderate influence. Furthermore, these investigators found that for air/water and related systems the Taitel/Dukler formulations adequately represent the flow regime boundaries for all except the intermittent/annular flow transition. This latter transition was found by Weisman *et al.* (1979) to be given by

$$1.9(U_G/U_L)^{1/8} = (\mathrm{Ku})^{0.2}(\mathrm{Fr})^{0.18}$$

where

$$Ku \equiv U_G \rho_G^{1/2}/[g(\rho_L - \rho_G)\sigma]^{1/4}$$

$$\mathrm{Fr} \equiv (U_G^2/gD) \qquad [1]$$

U_G and U_L are the superficial gas and liquid velocities, respectively; ρ_G and ρ_L the gas and liquid densities; and σ is the liquid/vapor surface tension.

Application to steam/water flow

The adiabatic flow regime maps and transitions discussed above necessarily neglect thermal interactions between the phases, as well as between the pipe and the environment, and are thus not strictly valid for the boiling or condensing flow of water under the influence of an external heat source/sink. Nevertheless, under saturation conditions and for modest axial changes in the vapor fraction or steam quality, the adiabatic flow-regime maps can be expected to offer a first guide to the thermal/hydraulic behavior of boiling or condensing flows, with possible variations in the precise location of the regime boundaries due to the influence of the applied surface heat flux.

This assertion has been recently confirmed by the successful application of the Taitel & Dukler (1976) flow regime map to the prediction of horizontal, tube side condensation by Breber *et al.* (1979) and new diabatic flow calculation by Taitel (1980). The latter results, incorporating inter-phase momentum transfer, bubble generation in the liquid and axial variations in the flow parameters, appear to suggest that for water at boiling heat fluxes as high as 100 KW/m^2, the most significant perturbation in the flow regime map is the shift in the dispersed bubble/intermittent boundary to the right, towards U_G values of between 0.1 and 1 m/sec and comparable U_L values.

While the adiabatic flow of a gas/liquid mixture can be represented by a single point on the superficial velocity flow regime map, the flow of water or refrigerant in boiler tubes can be expected to trace a locus of points on the map, as the quality, i.e. vapor fraction, increases in the flow direction. This locus may be wholly within a single flow regime or, more generally, pass from one flow regime to the next, as the ratio of vapor to liquid flow rate increases. These trends are displayed in figure 2, where the loci traced out by 5×10^2 kPa (5 Bar) steam/water flows at four different mass flow rates are shown.

Interestingly, in this and similar maps, the stratified flow regime appears to be of importance only for very low mass flow rates. This theoretical result suggests that it is highly unlikely for stratified flow to be encountered in either conventional or FBC horizontal boiler tubes which typically operate at mass flow rates in excess of 140 kg/sec m^2 (Pushkin *et al.* 1977).

EMPIRICAL CIRCUMFERENTIAL TEMPERATURE PROFILES

Styrikovich & Miropolski

Styrikovich & Miropolski (1950) were the first to document a study of circumferential temperature variations around a horizontal boiler tube. In their study the temperature difference between the pipe top and bottom was found to vary with the superficial liquid velocity, pressure and the two-phase mixture velocity U_{L+G} (equal to the sum of the superficial gas and liquid velocities). As can be seen in figure 3, for a fixed value of U_L (superficial liquid velocity), the relevant ΔT was found to increase abruptly and reach its peak value at very low superficial gas

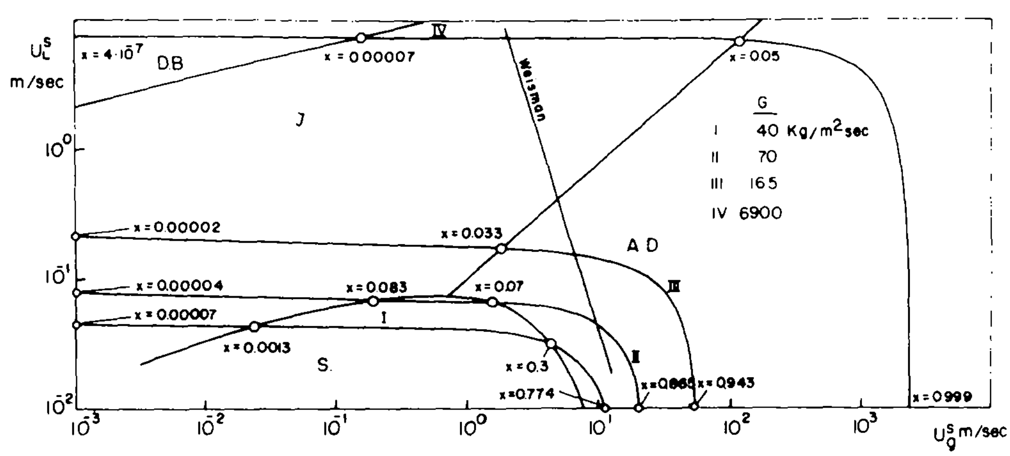

Figure 2. Constant mass flux locii on flow regime map for steam/water at 5 bars in 2.5 cm diameter pipe.

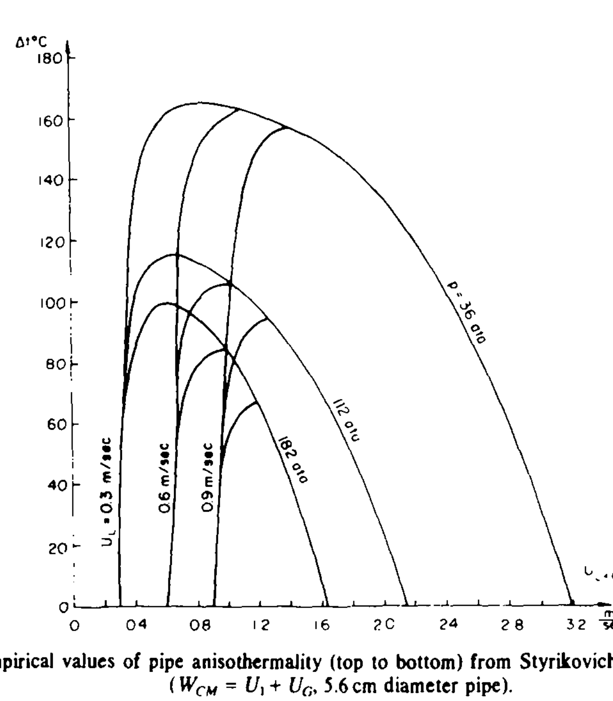

Figure 3. Empirical values of pipe anisothermality (top to bottom) from Styrikovich & Miropolski (1956) ($W_{CM} = U_l + U_G$, 5.6 cm diameter pipe).

velocities and then decrease gradually toward zero as the mixture velocity increased. The U_{L+G} values at the maximum anisothermality condition are seen to be somewhat dependent on pressure while U_{L+G} values for the isothermal condition appear to be independent of U_L and to depend only on the operating pressure.

The U_{L+G} value at which the circumferential temperature difference vanishes, was identified by Styrikovich & Miropolski (1950) as the "critical velocity" and its detailed dependence on pressure and heat flux in their experimental apparatus is presented in figure 4. The indicated critical velocities were interpreted to represent the boundary of the stratified flow regime and were considered by Styrikovich & Miropolski (1950) as the lower limit for safe boiler tube operation. Collier (1981) has recently shown these results to agree quite well with the semi-empirical "critical velocity" obtained by Gardner & Kubie (1976) in the flow of an alcohol-water mixture simulating high pressure steam–water flow.

Rounthwaite (1968)

In the course of the 1968 investigation by Rounthwaite, the circumferential temperature profile was measured at several axial locations along two 6.2 m long, 41.3 mm dia. uniformly heated pipes connected so as to form a continuous U-tube. Unfortunately, Rounthwaite (1968) does not report the raw temperature data but presents instead the local heat transfer coefficient at the pipe top and bottom. With the assumption of uniform heat flux, these results can be used to generate a graphical representation of the top-to-bottom temperature difference in the pipe as a function of the two-phase flow velocity, as is shown in figure 5.

The region of large differences in the heat transfer coefficients (and hence wall temperatures) between the tube top and bottom was thought by Rounthwaite (1968) to reflect the effects of stratified flow, while the region of near-equality in the coefficients was believed to be associated with annular flow.

Other studies

Among the more recently published studies of heat transfer in boiler tubes there is none that explicitly addresses the development of circumferential anisothermality in horizontal pipes for the flow regimes examined herein. In many of these studies, notably Ricque & Roumy (1970), Lis & Strickland (1970), Robertson (1972) and Bonn *et al.* (1980), primary emphasis was placed on operation in the annular flow regime and the determination of parameters influencing "dryout" of the liquid annulus. Nevertheless, a critical review of this data can shed some additional light on the circumferential temperature variation in the regions of interest.

In the Lis & Strickland (1970) study, vertical inlet bends were found to induce substantial circumferential and longitudinal temperature variations on the outside wall of a horizontal steam evaporator tube. Interestingly, however, these variations were totally absent at very low

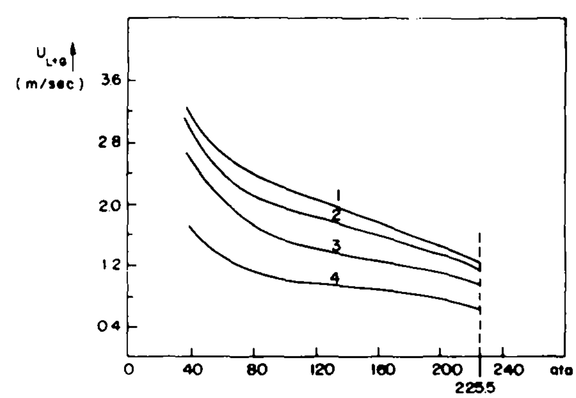

Figure 4. Empirical critical mixture velocity according to Styrikovich & Miropolski (1950) (1: $q - 50 \times 10^3$ kcal/m^{2}hr, 2: $q = 40 \times 10^3$ kcal/m^{2}hr, 3: $q = 30 \times 10^3$ kcal/m^{2}hr, 4: $q = 20 \times 10^3$ kcal/m^{2}hr).

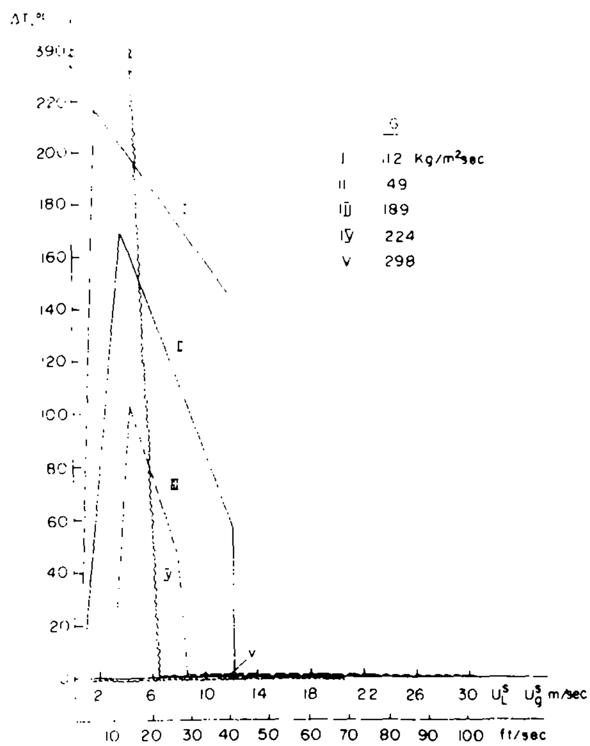

Figure 5. Empirical values of pipe anisothermality (top to bottom) from Rounthwaite (1968) (4.1 cm diameter pipe, 15 bars).

steam qualities (≤3%); began to appear in random fashion at the top of the tube at small qualities; gradually increased in magnitude as the quality increased and disappeared altogether for still higher steam qualities.

The Robertson (1972) investigation of dryout in horizontal hairpin waste-heat boiler tubes presents pipe temperature data for an unusually high water mass flux (approx. 1375 kg/m^2 sec) and imposed heat flux (approx. 1 MW/m^3). Under these conditions, wall temperature fluctuations were found to occur along the top of the tube *only* at steam qualities in excess of approximately 20% and to be nearly independent of the inlet geometry. The absence of large circumferential temperature variations, at low-to-moderate steam qualities, is in sharp contrast to the results of previously described studies by Styrikovich & Miropolski (1950), Rounthwaite (1968) and Lis & Strickland (1970).

Clear and most precise circumferential temperature profiles for horizontal boiler tubes appear in the 1980 study by Bonn *et al.* In their experiments with R-12 flowing in a 14 mm i.d. copper tube, top-to-bottom temperature differences were found to vary significantly from near-zero, for an imposed heat flux of 1.4 kW/m^2, to approximately 9°C at heat fluxes varying from 80 to 69 to 57 kW/m^2 for vapor quality of 0.3, 0.5 and 0.7, respectively. While the authors related this behavior to progressive dryout of the annular film, the high thermal conductivity, small tube diameter and large tube wall thickness (2.2 mm) may well have masked the existence of stratified and/or intermittent flow for nearly all the conditions tested.

EVALUATION AND INTERPRETATION OF EMPIRICAL RESULTS

Theoretical ΔT *variation in a boiler pipe*

In a previous section, the axial variation in vapor content along a horizontal boiler tube was shown to trace out a locus of points on the superficial velocity flow regime map. Examination of figure 2 reveals that for modest-to-high mass flux values and near-zero inlet quality the locus of interest commences in the dispersed bubble regime, enters immediately into the intermittent flow regime and at high flow qualities, or high superficial gas velocities, crosses into the annular regime.

It might thus be anticipated that for a uniformly heated tube, the circumferential or tube top-to-bottom temperature difference would be near zero at the pipe entrance, where liquid heat transfer coefficients prevail along the entire circumference of the pipe, but increase almost immediately (and at low superficial gas velocities) upon transition from dispersed bubble to intermittent flow. In intermittent flow at low gas velocities, slowly moving vapor packets along the upper surfaces of the pipe and the resulting low heat transfer coefficients in this region can be expected to yield large circumferential antisothermality. For progressively higher gas velocities, leading to more frequent liquid bridging or wetting of the upper surfaces, the more vigorous nature of the intermittent flow can be expected to reduce the average circumferential temperature difference. Finally, the establishment of a liquid film around the pipe wall in the annular flow regime could be anticipated to result in the near-elimination of circumferential temperature differences. However, a further, substantial increase in superficial gas velocity, associated with high vapor quality, can be expected to erode the liquid annulus and lead to a second parametric range of pipe anisothermality, as dry patches develop on the upper pipe surface. Such post-annular behavior is beyond the scope of this study.

Styrikovich & Miropolski

In interpreting the Styrikovich & Miropolski (1950) data presented in figure 3, it is convenient to note that, for the lines of constant superficial liquid velocity shown, the variation in U_{L+G} along the ordinate is equivalent to a variation in the superficial gas velocity, U_G. Consequently, the reported temperature differences can be examined by reference to a superficial velocity flow regime map, as for example in figure 6 for water/steam at 112 bars.

In figure 6 an adiabatic and two diabatic dispersed bubble/intermittent boundaries are illustrated. The latter are for applied heat fluxes of $q'' = 1\ \mathrm{W/cm^2}$ and $q'' = 100\ \mathrm{W/cm^2}$, respectively, according to the preliminary results of Taitel (1980). Although the precise location of this transition for the Styrikovich & Miropolski (1951) operating conditions ($q'' = 10\ \mathrm{W/cm^2}$, see figure 3) is unknown as yet, it may be expected to lie between the two diabatic boundaries on the map.

In figure 6 the locii appropriate to the Styrikovich & Miropolski study, i.e. $U_L = 0.3$, 0.6, and 0.9 m/sec, respectively, are seen to lie substantially above the stratified flow regime. Consequently, the theoretical ΔT variation described above for modest-to-high superficial liquid

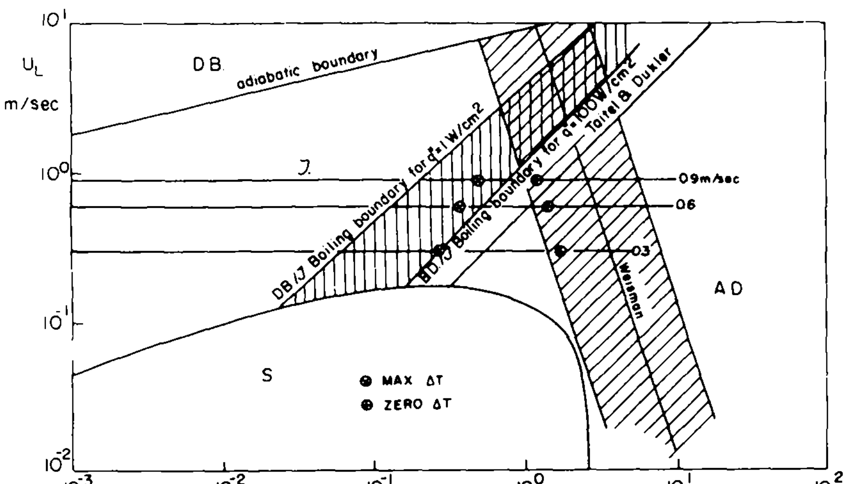

Figure 6. Constant superficial liquid velocity locii on flow regime map for conditions of Styrikovich & Miropolski (1950) at 112 bar (I, $U_L = 0.3$ m/sec; II, $U_L = 0.6$ m/sec; III, $U_L = 0.9$ m/sec).

velocities can be anticipated to apply to the Styrikovich and Miropolski (1950) results, i.e. the peak ΔT value occurring upon transition from the dispersed bubble to the intermittent regime and the critical two-phase velocity reflecting transition to Annular flow rather than the termination of Stratified flow, as originally suggested by Styrikovich & Miropolski (1950). This interpretation is supported by a comparison of the mixture velocities at the peak ΔT and isothermal condition shown in figure 3 and the values of U_{L+G} at the relevant flow regime boundaries shown in figure 6. Such a comparison reveals the Styrikovich & Miropolski (1950) peak ΔT points to occur at mixture velocities appropriate to the estimated location of the dispersed bubble/intermittent boundary for boiling flow. Similarly, as anticipated, the "critical mixture velocities" are seen to fall in the vicinity of the intermittent/annular flow at the specified condition of 112 bar.

It is of interest to note that the peak ΔT points occur at progressively larger two-phase flow velocities and U_G values as the liquid superficial velocity increases. This trend is consistent with the slope of the dispersed bubble/intermittent flow regime boundary and may explain, as well, the reduced magnitude (at a fixed pressure) of the peak ΔT at higher superficial liquid velocities, due to improved pipe-top heat transfer rates at higher gas velocities. Similarly, improved vapor heat transfer coefficients may explain the significant influence of pressure on the peak circumferential temperature difference. As previously mentioned, Styrikovich & Miropolski (1950) found the critical sum of U_L and U_G to be independent of the superficial liquid velocity. This is clearly not the case for the Taitel & Dukler intermittent/annular boundary, as is evident from its positive slope, yielding progressively larger values of U_{L+G} as the superficial liquid velocity increases.

Alternately, this condition is more nearly met when the Weisman *et al.* (1979) relation is used to define the intermittent/annular boundary. This boundary can be seen in figure 6 where the error band, as presented by Weisman *et al.* (1979), is taken into account (the hatched region on both sides of the transition).

Thus, while for 112 bar [1] yields mixture velocities nearly 50% higher than the empirical Styrikovich & Miropolski (1950) result, i.e., 3.8, 3.6, 3.4 m/sec at superficial liquid velocities of 0.3, 0.6, 0.9 m/sec, respectively, rather than the reported value of 2.2 m/sec, this relation appears preferable to the Taitel/Dukler expression for the intermittent/annular transition which yields average values approximately 30% lower than the empirical value but which vary from 0.85 m/sec to 1.7 m/sec to 2.55 m/sec as the superficial velocity increases from 0.3 to 0.9 m/sec. Unfortunately, the strong influence of surface heat flux on the critical velocity and especially at $q'' < 50\ \mathrm{KW/m^2}$ reported by Styrikovich & Miropolski (1950) cannot be account for in the quasi-adiabatic analysis presented herein, nor by the preliminary results of the Taitel (1981) diabatic study.

While, in the interest of brevity, the detailed exploration of the Styrikovich & Miropolski (1950) data is limited to the 112 bar values, nearly identical results have been obtained with the 36 and 182 bar data (Ruder 1981).

Rounthwaite (1968)

Figure 7 displays the locii appropriate to the 15 bar data obtained by Rounthwaite (1968) on the superficial velocity flow regime map. In contrast to the Styrikovich & Miropolski (1950) results, the locus of the lowest mass flux ($G = 94\ \mathrm{kg/m^2\,sec}$) is seen to cross from intermittent into the stratified smooth and stratified wavy regimes and to remain close to the annular flow boundary at high qualities. The locii appropriate to larger G values lie above the stratified regime and progress from dispersed bubble to intermittent to annular flow in the previously described manner.

A detailed examination of the figure 7 flow regime map serves to explain much of the thermal behavior observed by Rounthwaite (1968). His assertion that circumferential isothermality is attained upon transition to annular flow and his empirical "critical" mixture velocities

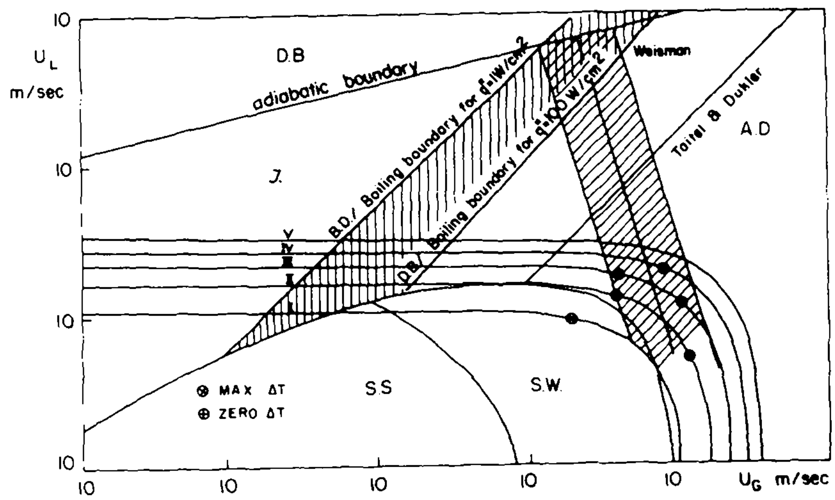

Figure 7. Constant mass flux locii on flow regime map for conditions of Rounthwaite (1968) (I, $G = 100\,kg/m^2sec$; II, $G = 150\,kg/m^2sec$; III, $G = 190\,kg/m^2sec$; IV, $G = 240\,kg/m^2sec$, V, $G = 290\,kg/m^2sec$).

of 5.5 to 12.7 m/sec (for the conditions of figures 5 and 7) are generally consistent with the Weisman intermittent/annular flow boundary, which yields mixture velocities in the range of 3.9 to 14.8 m/sec when considering the error band appropriate to this boundary (Weisman 1979). Use of the Taitel & Dukler boundary for the intermittent/annular flow transition results in substantially lower U_{L+G} values than observed and a trend towards reduced "critical" mixture velocities at lower mass flow rates that is contrary to the empirical findings.

Returning to figure 5, it may be noted that the pipe top-to-bottom temperature difference for the lowest mass flow rate fails to display the "critical" mixture velocity at which circumferential isothermality is attained. This apparently anomolous behavior may be related to the locus of this flow rate on the flow regime map which suggests that even at near-unity qualities clear transition to annular flow is unlikely. Consequently, it might be anticipated that at this flow rate only a modest reduction in circumferential anisothermality would be encountered at increasing values of the mixture velocity, due to occasional wetting of the upper pipe surface in the Stratified Wavy regime.

With the exception of the results for the mass flux of $290\,kg/sec\text{-}m^2$, the decrease in the magnitude of the pipe top-to-bottom temperature difference with increasing G can be traced to the influence of the dispersed bubble/intermittent flow boundary slope, as previously discussed in examining the Styrikovich & Miropolski (1950) results. The decreasing circumferential temperature difference and shift in the peak ΔT value of the mixture velocity with increasing mass flow rate, for all but $G = 290\,kg/m^2\,sec$, can be similarly explained but it must be noted that inexplicably the empirical values for this parameter are substantially larger than suggested by the flow regime map of figure 7.

This and other noted discrepancies can be partially explained by the difficulty in obtaining precise data in two-phase flow and the resulting scatter in both the data and empirically-derived flow regime boundaries, and may reflect, as well, the need to more fully address the influence of surface heat flux.

Other studies

The verbal description of the quality-dependence of circumferential anisothermality, appearing in Lis & Strickland (1970), is remarkably similar to that of Styrikovich & Miropolski (1950) and Rounthwaite (1968). Furthermore, construction and examination of a Taitel & Dukler flow regime map for the Lis & Strickland conditions suggests that, for flow boiling, in purely horizontal tubes, transition from dispersed bubble to intermittent flow, and thus from initial isothermality to the anisothermal situation, should occur in the range of 1 to several per cent flow quality. This result is in general agreement with the Lis & Strickland observation and, together with the "familiar" verbal description, appears to imply that the presence of a vertical inlet bend merely enhances, rather than gives rise to, significant wall temperature variations in horizontal evaporator tubes operating in the Intermittent flow regime.

A flow regime map for the conditions of the Robertson (1972) investigation, including the locus of the 1375 kg/m^2 sec mass flux and the approximate diabatic boundary for transition from dispersed bubble to intermittent flow, is shown in figure 8. Most significantly, this latter transition is seen to fall within the error band of the Weisman criteria for transition from intermittent to annular flow. It would thus appear possible that Robertson may have avoided operation in the thermally undesirable intermittent regime and progressed directly from the dispersed bubble to the annular regime. In complete agreement with the Robertson (1972) results, when operating in this fashion, circumferential isothermality can be maintained up to the moderately high qualities at which post-annular dryout may occur.

CONCLUSIONS

The preceding analysis of the limited data available for the circumferential temperature variation in horizontal boiler tubes has revealed the importance and validity of interpreting these results in light of the hydrodynamic phenomena accompanying boiling two-phase flow. The parametric dependence and axial variation of the tube top-to-bottom temperature difference have been found to reflect the influence of the two-phase flow regimes encountered along the appropriate locus in a flow regime map.

In contrast to the oft quoted conclusion by Styrikovich & Miropolski (1950) that minimization of the pipe top-to-bottom temperature difference is associated with termination of stratified flow, the present study appears to suggest that it is transition from the intermittent to the annular flow regime that is generally responsible for circumferential isothermality. Use of the flow regime maps appropriate to the specified operating conditions further suggests that the peak value of the circumferential temperature difference occurs along the dispersed bubble/intermittent flow regime boundary.

Application of the present results to the design of FBC and conventional boilers suggests that the avoidance of stratified flow is insufficient to insure circumferentially isothermal, horizontal boiler tubes and that care must be taken to avoid, as well, operation in the intermittent flow regime. For zero inlet quality flow and/or whenever pumping power costs are

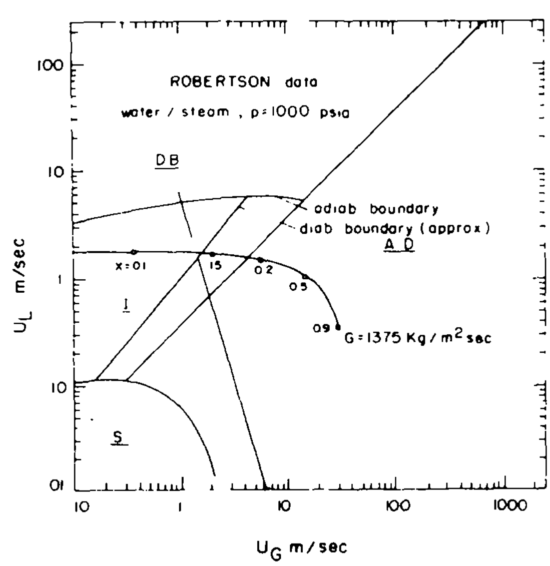

Figure 8. Steam/water flow regime map and operating locus for conditions of Robertson (1972).

relatively unconstrained this goal may be attained by use of uncommonly high mass flow rates, permitting direct transition from the dispersed bubble to the annular flow regime. Alternately, for more modest flow rates, superficial mixture velocities appropriate to the Weisman *et al.* (1979) intermittent/annular flow transition should be prescribed as close as possible to the pipe inlet. In this configuration, much of the pipe can be expected to operate isothermally in the annular flow regime and the anisothermal inlet zone of the pipe, where elevated circumferential temperature gradients associated with intermittent flow are likely to prevail, can be taken into account in the design process. Care must, of course, be taken to avoid operation at qualities known to cause post-annular dryout.

It is interesting to note that Styrikovich & Miropolski (1956) suggest that inclination of the pipe upwards in excess of 9.5° can similarly be used to assure pipe isothermality.

Acknowledgements—The subject study was funded by the Israel-U.S.A. Binational Science Foundation via grant number 2059/79. The authors are grateful to Prof. Yehuda Taitel for his helpful comments and for providing initial results of his diabatic flow regime analyses. The aid of Mrs. Orna Nurenberg, in preparing the illustrations, and Mr. Yoram Klein, in performing the flow regime calculations, is similarly acknowledged.

REFERENCES

BAKER, O. 1954 Simultaneous Flow of Oil and Gas. *Oil Gas J.*, **53**, 185–195.

BONN, W., IWICKI, J., KREBS, R., STEINER, D. & SCHLUNDER, E. R. 1980. Über die Auswirkungen der Ungleichverteilung des Wärmeübergangs am Rohrumfang bei der Verdampfung im durchströmten waagerechten Rohr. *Wärme- und Stoffüberstragung* **13**, 265–274.

BREBER, G., PALEN, J. W. & TABOREK, J. 1979 Prediction of Horizontal Tubeside Condensation of Pure Components Using Flow Regime Criteria. NHTC-Condensation Heat Transfer. *ASME Publication No.* 100123.

COLLIER, J. G. 1981 Introduction to two-phase flow problems in the power industry. *Two-Phase Flow and Heat Transfer in the Power and Process Industries* (Edited by BERGLES, A. E., COLLIER, J. G., DELHAYE, J. M., HEWITT, G. F. & MAYINGER, F.) Chap. 19. Hemisphere & McGraw Hill, Washington.

GARDNER, G. C. & KUBIE, J. 1976 Flow of Two Liquids in Sloping Tubes: An Analogue of High Pressure Steam and Water. *Int. J. Multiphase Flow* **2**, 435–451.

GLICKSMAN, L. R., PELLOUX, R., DECKER, N. & SHEN, T. 1977 Thermal stresses and fatigue of heat transfer tubes immersed in a fluidized bed combustor. *Proc. 5th Int. Conf. on Fluidized Bed Combustion*, Washington, D.C.

LIS, J. & STRICKLAND, J. A. 1970 Local variations of heat transfer in a horizontal steam evaporator tube. *Proc. 4th Int. Heat Transfer Conf.*, Paris, V, Paper No. B4.6.

MANDHANE, J. M., GREGORY, G. A., & AZIZ, K. 1974 A flow pattern map for gas–liquid flow in horizontal pipes. *Int. J. Multiphase Flow* **1**, 537–553.

PUSHKIN, N. J. 1977 *Marine Steam Generators*. Sudostroyenie Press, Leningrad, (in Russian).

RICQUE, R. & ROUMY, R. 1970. Exchange thermique en double phase dans des tubes verticaux ou horizontaux. *Proc. 4th Int. Heat Transfer Conf.*, Paris, V, Paper No. B5.12.

ROBERTSON, J. M. 1972. Dryout in horizontal hairpin waste-heat boiler tubes. *A.I.Ch.E. Symp. Ser.* **69**, 55–62.

ROUNTHWAITE, C. 1968. Two-phase heat transfer in horizontal tubes. *J. Inst. Fuel* **41**, 66–76.

RUDER, Z. 1981. Heat transfer to boiling fluids in an inclined pipe in a large particle fluidized bed. M.Sc. Thesis, Dept. of Mech. Eng., Ben-Gurion University of the Negev, Israel.

STYRIKOVICH, M. A. & MIROPOLSKI, Z. L. 1950. Stratification in vapor–water mixture flow at high pressures in the heated horizontal tube. *Dokl. Akad. Nauk. SSSR* **LXXI**, 2, (in Russian).

STYRIKOVICH, M. A. & MIROPOLSKI, Z. L. 1956. Report No. IGRL-T/R4 (Translation).

TAITEL, Y. & DUKLER, A. E. 1976. A model for predicting flow regime transitions in horizontal and near horizontal gas–liquid flow. *A.I.Ch.E. J.* **22**, 47–55.

TAITEL, Y. 1980. Class notes for flow with phase change. Dept. of Chemical Engineering, University of Houston.

WEISMAN, J., DUNCAN, D., GIBSON, J. & CRAWFORD, T. 1979. Effects of fluid properties and pipe diameter in two-phase flow patterns in horizontal lines. *Int. J. Multiphase Flow* **5**, 437–462.