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A CORRELATION FOR HEAT TRANSFER IN STRATIFIED TWO-PHASE FLOW WITH VAPORIZATION

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Abstract—A mechanism of vaporization is suggested for the case in which a saturated fluid flows vertically upwards through a heated annulus. Visual observations and measurements of vapour fraction on an experimental apparatus are recorded, and it is noted particularly that an annulus of vapour was seen to surround a thin liquid layer on the heater surface in the upper regions of the heated system. No nucleate boiling occurred in the liquid layer. The experiment was carried out in the temperature-difference range 18° to 40°F, and the liquid mass flow-rate was varied from 1 to 5 lb per minute.

At high heat fluxes in this stratified flow zone it is demonstrated that it may be possible to consider the heat transfer to be entirely convective. A hypothesis is developed analytically, and is substantiated in this experiment by the good correlation of the heat and mass transfer results by means of a standard expression for forced convection.

Résumé—Un processus de vaporisation est proposé dans le cas d'un fluide saturé s'élevant dans un espace annulaire chauffé. Des observations et des mesures de la quantité de vapeur ont été enregistrées sur un dispositif expérimental; on a remarqué, en particulier, que dans la partie supérieure du dispositif un anneau de vapeur entourait le film liquide à la surface du réchauffeur. Aucune ébullition ne se produit dans le liquide. Les expériences ont été faites pour un domaine de différence de température compris entre 18 et 40°F et un débit massique de liquide variant de 1 à 5 lb/min.

Dans cette zone d'écoulement stratifié, il est démontré que, pour des flux de chaleur élevés, on peut considérer que la transmission de chaleur se fait uniquement par convection. Une hypothèse a été étudiée par le calcul et s'est trouvée étayée dans cette expérience par un bon accord entre les résultats concernant le transport de masse et de chaleur donnés par l'équation classique de la convection forcée.

Zusammenfassung—Für eine gesättigte Flüssigkeit, die durch einen beheizten Ringraum senkrecht nach oben strömt, wird der Verdampfungsvorgang beschrieben. An einer Versuchseinrichtung wurden visuelle Beobachtungen angestellt und der Dampfgehalt gemessen. Bemerkenswert war im oberen Teil der beheizten Apparatur eine Dampfschicht, die eine dünne Flüssigkeitsschicht auf der Heizfläche ringförmig umgab. In der Flüssigkeitsschicht trat keine Blasenverdampfung auf. Bei den Versuchen wurden die Temperaturdifferenzen von 10 bis 22 grd und der Mengenstrom der Flüssigkeit von 0,45 bis 2,3 kg/min variiert.

Für grosse Wärmeströme darf im Bereich der Schichten der Wärmeübergang als rein konvektiv angesehen werden. Die Versuche bestätigen eine analytisch entwickelte Hypothese, denn bei Benutzung eines gebräuchlichen Ausdrucks für die erzwungene Konvektion stimmen die Ergebnisse des Wärme- und Stoffaustausches gut überein.

Аннотация—Предложена гипотеза для объяснения механизма парообразования при течении насыщенного потока вертикально вверх через нагретый кольцевой зазор. На экспериментальной установке проводились визуальные наблюдения и измерения компоненты пара; в частности, было отмечено, что кольцо пара окружало тонкий слой жидкости на поверхности нагревателя в верхних участках нагреваемой системы. В слое жидкости не наблюдалось пузырькового кипения. Опыт проводился в интервале температурных разностей от 18° до 40°, весовой расход жидкости изменялся от 1 до 5 фунтов в минуту.

Показано, что при больщих тепловых потоках в зоне расслоённого течения теплообмен можно считать полностью конвективным. Дано аналитическое объяснение гипотезы; гипотеза подтверждается хорошим соответствием экспериментальных результатов по тепло—и массопереносу с общепринятым выражениэм для вынужденной конвекции.

NOTATION

- A, cross-sectional area of annulus (ft^2);
- α, vapour volume fraction (—);
- C, dimensionless constant (—) ;
- c_p, specific heat (Btu/lb °F);
- d, hydraulic diameter of annulus (ft);
- D, diameter (ft);
- h, heat transfer coefficient (Btu/ft^2 hr °F);
- k, thermal conductivity (Btu/ft hr °F);
- K, empirical constant (ft/hr);
- λ, latent heat of vaporization (Btu/lb);
- L, total length of heated wall (ft);
- μ, dynamic viscosity (lb.ft/hr);
- Q, heat input to heated wall (Btu/hr);
- q'', heat flux density (Btu/ft^2 hr);
- ρ, density (lb/ft^3);
- r, vapour mass fraction (—);
- T, temperature (°F);
- ΔT, temperature difference (°F);
- V, mean local velocity (ft/hr);
- z, Height measured from beginning of heated section (ft).

Dimensionless numbers

- Nu, Nusselt number $= \dfrac{h \, . \, d}{k}$;
- Pr, Prandtl number $= \dfrac{c_p \, . \, \mu}{k}$;
- Re, Reynolds number $= \dfrac{\rho \, . \, V \, . \, d}{\mu}$.

Subscripts

- 1, outer wall of annulus;
- 2, inner wall of annulus;
- *f.f.*, forced flow;
- o, natural circulation flow;
- L, liquid;
- v, vapour;
- s, saturated liquid;
- w, heater wall.

1. INTRODUCTION

RECENT papers [1–9] on heat transfer to boiling liquids moving past a heated wall show that all correlations are based on some physical consideration of bubble growth.

Staley and Baker [1], on boiling heat transfer rates in heated tubes with forced convection, review the correlations for such a system and show that they are unreliable, having been derived in general from expressions applicable to pool boiling.

The qualitative description of forced-convection boiling in tubes and annuli is well known [10], and its mechanism, observed in an experiment with a fluorinated hydrocarbon refrigerant boiling in a short vertical annulus, is recapitulated below.

Pool boiling describes the nucleation phenomenon which occurs when a heated surface is immersed in a stagnant fluid at or near its saturation temperature. No purposely imposed fluid flow over the heated surface exists. It is noticeable, however, that at high heat fluxes the stirring action of rapid bubble formation influences heat transfer, and experiments show a tendency towards correlation by an expression relating Nusselt number to heat input, of the form

$$Nu \propto (q'')^n$$

Jakob [2] uses the expression

$$Nu = \frac{h}{k}\sqrt{\frac{\sigma}{\rho_L}} = 30\left\{\frac{q''}{\rho_v \lambda K}\right\}^{0\cdot 8}$$

to describe the results of experiments by Jakob and Linke on water and carbon-tetrachloride, and a modified expression with the same index of q'' to take into account the influence of pressure. Bonilla and Perry [3] made a further modification to correlate results for a wider range of fluids, lowering the index of q'' to 0·73, adjusting the constant and adding $Pr^{0\cdot 5}$ as a factor. McAdams and Akin, Rachko, Braunlich, and Cichelli and Bonilla [4] have reported an index of q'' less than 0·7 and Jens and Leppert a value of 0·75. Thus, in pool boiling the relationship between q'' and the temperature difference $\Delta T_{w,s}$ is considered to lie between about $q'' \alpha (\Delta T_{w,s})^3$ and $q'' \alpha (\Delta T_{w,s})^5$, although recent work indicates that even this large range of correlating indices may prove to be insufficient as more information becomes available on effects of surface roughness on pool boiling.

The above remarks apply, however, to boiling without forced convection. The effect of fluid velocity in promoting both additional general turbulence and an increase in dynamical action on bubble separation has been investigated by

Styushchin and Sterman [5]; they found, for the ratio of the heat transfer coefficients with and without forced flow, that

$$\frac{h_{f.f.}}{h_0} = \left[\frac{V_L + (q''/\rho_L\lambda)}{(q''/\rho_L\lambda)}\right]^{0\cdot18}$$

where V_L = liquid velocity (ft/hr). The form of the equation confirms many similar findings: that at high heat flux densities in the nucleate boiling region, where $(q''/\rho_L\lambda) \gg V_L$, the flow velocity has little effect on heat transfer. Levy [6] has argued that fluid velocities of the magnitude normally used exert a minor effect in comparison with the local turbulence produced by the bubble motions. For a fluid containing vapour Levy proposed a correlation of the form

$$q'' \propto (1 - r)(\Delta T_{w,s})^3$$

and demonstrated its validity at low vapour qualities. Rohsenow {[8], Discussion} has pointed to the general experience that, in forced convection in a vertical tube with uniform q'', the temperature difference $\Delta T_{w,s}$ decreases with increase in vapour quality, citing for example Rohsenow and Clark's results [7]. Thus the heat transfer rate at constant $\Delta T_{w,s}$ increases with quality. Levy replied that, at high qualities, a majority of the cross-sectional area of flow is occupied by the vapour, and the remaining saturated liquid can attain very high velocities. Thus convective effects are no longer negligible, especially at low pressure where, due to the lower density of the vapour, a larger portion of the cross-sectional area of flow is occupied by vapour. He went on, "It is even possible, as pointed out by Dengler [8], to suppress boiling under these conditions from excessive convective effects."

It is thought that the present work throws further light on the latter statement, and describes the magnitude and extent of the "suppressive effect" as being such as to inhibit nucleate boiling completely over the major part of tube length; the convective heat transfer effect under these circumstances must be at least equal to the pool boiling heat transfer effect, were the latter allowed to take place.

2. EXPERIMENT AND RESULTS

Refrigerant 11 ($C.Cl_3.F$) was passed through a vertical annulus in which an electrically heated stainless-steel tube, 12 in. long and $\frac{1}{2}$ in. in diameter, was surrounded by a Perspex tube with an inside diameter of $\frac{31}{32}$ in. Thermocouples were brazed on the inside wall of the heater at various positions and others were placed opposite to them in the annular space; temperatures were measured by a Honeywell-Brown recorder.

The volume vapour fraction of the fluid at the uppermost thermocouple position was evaluated by measuring the electrical capacitance between the heater tube and an electrode round the Perspex tube. The electrode was a brass clip, 1 in. wide, which surrounded the tube completely, and had four small holes drilled at equal intervals along its width for calibration purposes. The capacity of this condenser was proportional to the dielectric constant of the fluid between the electrodes, and therefore proportional to the volume liquid fraction, and this was checked by bringing the liquid to each calibration point in turn. The resultant graph between capacity and liquid level was a straight line.

Although it is shown below that the flow configuration at this point consists of a vapour annulus surrounded by a liquid annulus, with considerable "slip" between the phases, it was nevertheless assumed that the fluid was homogeneously mixed for the purpose of capacity

Table 1. *Heat transfer coefficient, h, and vapour volume fraction* α; *for flow of Refrigerant* 11 ($C.Cl_3.F$) *in a vertical annulus* ($\frac{31}{32}$ *in.*/$\frac{1}{2}$ *in. diameter*). *Measurements taken at* 11 *in. from entry to heated section: post-coalescence, stratified flow in all cases*

Heat flux density Btu/ft²hr	Coolant mass flow rate, lb/min					
	5		3		1	
	h	α	h	α	h	α
5,087	272	0·48	268	0·53	298	0·59
6,644	289	0·58	283	0·62	295	0·68
8,410	295	0·63	300	0·67	317	0·72
10,381	358	0·68	335	0·73	341	0·79
12,562	419	0·74	387	0·79	392	0·84
14,950	490	0·77	446	0·83	446	0·86
17,542	557	0·82	524	0·86	502	0·92
20,348	626	0·84	598	0·91	550	0·945
23,359	687	0·88	661	0·93	592	0·97

measurement. This assumption was partly justified by the extreme turbulence of the system, and would in the perfectly stratified case give an error of only 4 per cent at a volume vapour fraction of 65 per cent, decreasing with increasing vapour fraction.

Each result shown in Table 1 is the average of three sets of readings whose divergence was never greater than ± 4 per cent in temperature, velocity or volume vapour fraction measurements. The experiment was performed throughout with the liquid phase at entry at a saturation pressure equivalent to ambient temperature, in order to obviate vaporization in the delivery tube and to reduce the transfer of heat through the outer annular wall to insignificant proportions.

Figure 1 illustrates the mechanism of boiling which was observed in the vertical annulus. In stage A no boiling occurred even if the liquid were slightly superheated on entry. Stage B indicates the narrow band in which nucleate boiling took place on the heater wall. After flowing in the liquid stream for a short interval (C), the bubbles combined to form "slugs" of vapour—large bubbles totally enclosed by liquid (D). In region E the slugs appeared to push aside the liquid and form an annular chimney, leaving a thin film of liquid on the heater wall and a thick annulus of liquid surrounding the vapour. Coalescence of the vapour bubbles to form slugs occurred at a low mass vapour fraction ($\sim$0·01), so that the momentum of the vapour had thereafter little effect on the liquid velocity. Calculation of the vapour and liquid velocities at the thermocouple positions indicated that the liquid velocity changed little after coalescence, at which it had reached approximately three times its value at inlet.

3. DISCUSSION AND HYPOTHESIS

Considering first the curves plotted in Fig. 2, it is noticeable that a transition from one mode of flow to another appears to take place in the region $20 < \Delta T_{w,s} < 30$ approximately. The so-called "typical curves" for forced convection,

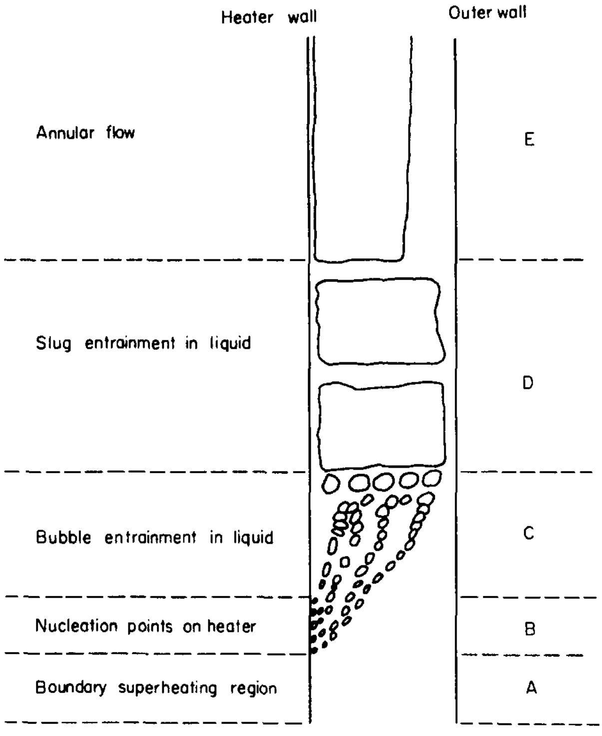

FIG. 1. Mechanism of boiling in annulus with one heated wall.

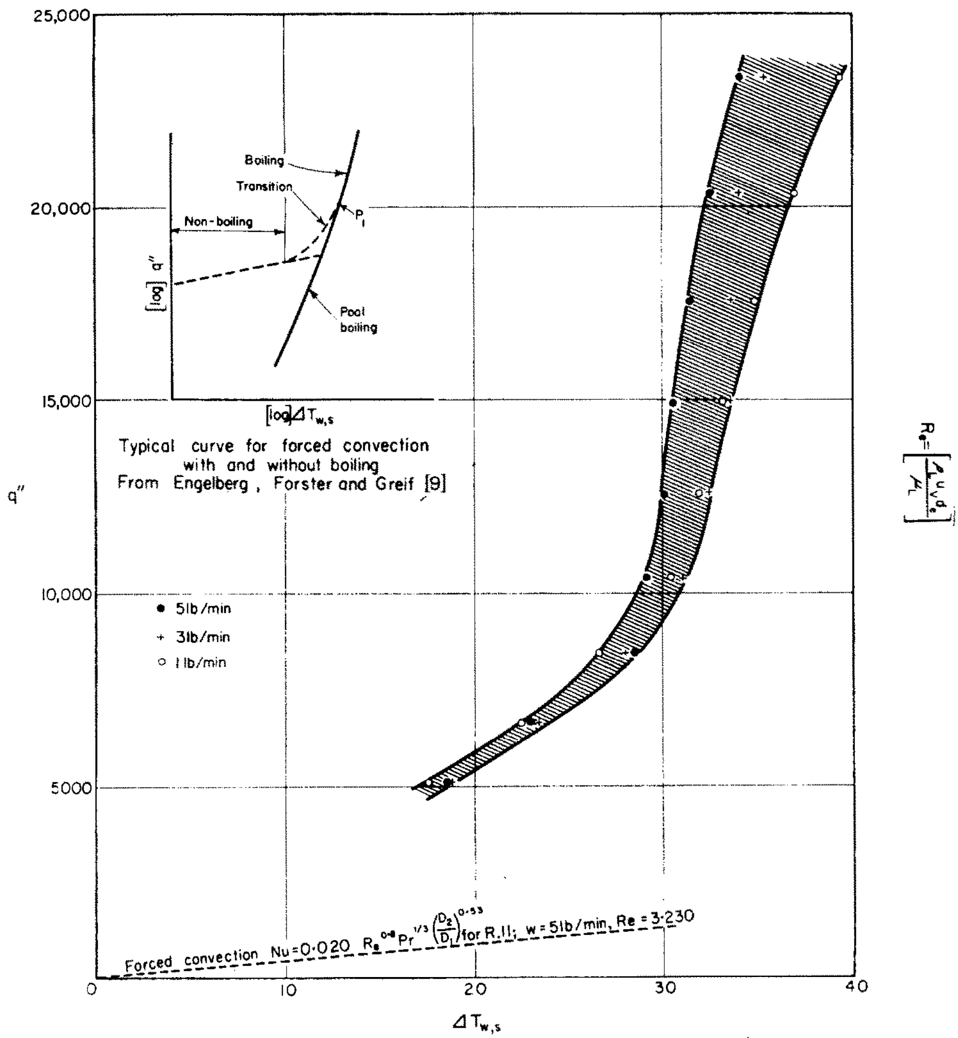

FIG. 2. Heat flux density versus surface/saturation temperature difference.

with and without boiling, from Engelberg-Forster and Greif [9] are shown inset. (It is assumed that it was the intention in the original to plot to a log–log scale, since neither the "non-boiling" nor the "pool boiling" curves pass through the origin.) Re-plotting the experimental results as log q'' versus log $\Delta T_{w,s}$ (Fig. 3) and adding the forced convection curve calculated from Monrad and Pelton's empirical correlation [11] for flow in annuli of the single liquid phase, it is at once clear that a mechanism different from the conventionally accepted one of nucleate boiling plus forced convection may be in operation at the high vapour fractions of the experiment.

As the experiment was performed under forced-flow conditions, the forced convection expression has been chosen in preference to the natural convection correlation, although heat transfer coefficients given by both are of the same order.

It is, however, of interest to note that Engelberg-Forster and Greif state,

"For every convective velocity W and temperature difference (T wall $-$ T liquid) there

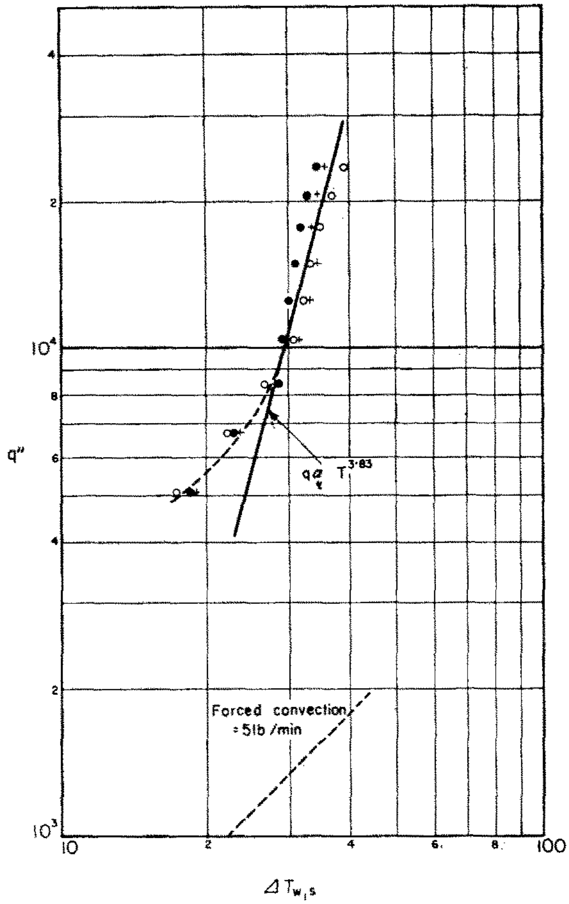

FIG. 3. Heat flux density versus temperature difference to logarithmic co-ordinates.

is a certain point P_1 (see inset, Fig. 2) on the heat-flux curve above which the relation between heat flux and superheat is essentially the same as for pool boiling."

Superposition of McAdams' suggested correlation [12] for pool boiling with forced convection ($q'' \alpha \Delta T_{w,s}{}^{3.83}$) (see also Ref. 9), shows that the usual approximate proportionality is still maintained in this case. The ordinate shift, to about six times that which would be expected in order to fit the transition curve to the forced convection curve, is characteristic of the curves for natural convection vs. pool boiling. Thus the mass flow rates of the experiment are probably lower than those described by Engelberg-Forster's typical smooth transition, which can only occur if the forced convection and pool boiling curves intersect at a reasonably high q'' value—that is above the onset of nucleation in pool boiling.

In the experiment, nucleate boiling (the emission of streams of discrete bubbles from nucleation points on the heated surface) was readily observed to take place over a length of heated surface shortly beyond entry, irrespective of either the flow rate or the heat flux within the ranges investigated. Beyond this nucleation region there followed the well-known coalescence and "slugging" zones, in which the bubbles unite and form large, turbulent "slugs" of vapour.

The net vapour generated at this stage then formed an annulus between two layers of liquid, one on the central heated rod and the other on the outer tube.

It appears to be customary to assume that nucleate boiling, once initiated, occurs over the whole length of any fully wetted, heated surface which exhibits an appropriate temperature relative to the saturation temperature of the surrounding liquid. Close examination of the heated surface, through the transparent tube forming the outer wall of the annulus, revealed that no further nucleation took place on any part of the heater following the coalescence region. Of the zones described, the "slugging" and "stratified flow" zones may therefore be of greatest practical interest, since they are found to extend over the greatest proportion of the heated tube length, in a system in which operation with a high exit vapour quality is required. The nucleate boiling zone occupied only a very small fraction of the total length of the tube.

Although there was no nucleate boiling in these later flow stages, the temperature of the electrically heated surface generally diminished from the coalescence zone to exit. Similarly the average liquid temperature diminished slightly (about 2°F) over the length of the heated section, as might be expected from the pressure drop associated with frictional loss, change in potential head and work done to accelerate the fluid during the phase change process. In fact it was found from a number of temperature measurements over the length of the heater that the effective heat transfer coefficient increased somewhat in the direction of flow, in confirmation of Rohsenow's statement {[8], Discussion}. It might be expected, therefore, that heat transfer observations alone, taken on a system in which the flow was not visible, might give rise to the assumption that nucleate boiling takes place over the entire heated length.

Further consideration of the observed phenomena led to a hypothesis that the heat transfer is entirely convective in character in the slugging and stratified flow zones of a vertical, forced-convection heat transfer system with change of phase—that heat is transferred through a boundary layer of the liquid phase by conduction, and is thence carried away by a process due to forced convection characteristics determined by the velocity profile of the vapour core.

This hypothesis will be examined in the light of both the experimental results and the deductions from observation of the modes of flow.

4. ANALYSIS

The two equations which can be applied in general to the flow conditions are those of continuity (1) and energy (2).

$$\rho_L A(1 - \alpha) V_L + \rho_v A \alpha V_v = w \tag{1}$$

$$\frac{Qz}{L} = \lambda \rho_v \,.\, \alpha \,.\, A \,.\, V_v. \tag{2}$$

The changes in the kinetic and potential energies of the fluid are considered to be comparatively insignificant, and the sensible heat content of the liquid is taken as remaining unchanged once boiling has begun.

The experimental determination of vapour fractions enables V_v, V_L, d_v and d_L to be evaluated for the flow rates and heat fluxes at which the temperatures were measured, and it emerges that the heat transfer coefficient and vapour velocity are approximately constant for a particular heat flux, and independent of mass flow rate. A theoretical explanation for this behaviour is given below, and falls into two parts.

4.1. *Nucleation and growth of the bubble*

Let it be assumed (as has often been observed in the literature) that, for all mass flows and heat fluxes within the reasonably narrow range investigated, the number of active nucleation sites in stage B (Fig. 1) remains constant. The diameter of a bubble when it breaks away from the wall is probably determined by the thickness of the boundary layer, which, if taken as being fully developed in this stage, is in inverse proportion to the Reynolds number of the liquid flow. Thus, when the bubble breaks away, its diameter may be said to vary inversely as the liquid velocity. From experiments by Jakob [13], the frequency of bubble emission is observed to be approximately inversely proportional to the bubble diameter, which, taking into account the pressure variation within bubbles of different size, maintains continuity of mass evaporation for a given heat flux density.

The bubbles emitted from the surface are entrained in the liquid until coalescence takes place (stage D in Fig. 1). It is now assumed that coalescence occurs at a vapour fraction which is constant and independent of heat flux or mass flow rate; note that, for a fluid with a high liquid to vapour density ratio, the volumetric vapour fraction approaches unity very rapidly for only a very small change in mass fraction, hence the occurrence of coalescence very close to the onset of nucleate boiling. From these arguments and assumptions, the quantity of vapour entering the coalescence zone in unit time will be proportional to the mass flow rate. The liquid velocity will therefore be increased in the same ratio for all inlet velocities, assuming negligible vapour–liquid slip. After zone D the vapour and liquid velocities are unaffected by each other, and the volume vapour fraction will be similar for all flows; this is found to be approximately the case.

The liquid in the nucleate boiling stage is accelerated by the inertial force of the growing bubble, but the energy required is negligible. Forster and Zuber [14], in an elegant inequality analysis, have determined that, for large bubbles ($>10^{-5}$ in. dia.) rising through the liquid phase of the same fluid, the energy available for acceleration of the liquid has an upper limit of $(\Delta T)/T$ times the total energy input, where ΔT is the superheat of the liquid surrounding the bubble ($\sim 2°$F), and T is the absolute temperature. This energy quantity is insignificantly small compared with the heat energy transferred, but nevertheless it is found to be sufficient to increase the liquid velocity to many times its original value.

4.2. *Stratified annular flow*

In the stratified flow stage no further nucleation could be observed on the heated surface. If heat is transferred through a liquid boundary layer by conduction and then carried away by mass transfer in the vapour annulus, the mode of heat transfer is analogous to that in single-phase forced convection, where the heat may be said to diffuse through a thermal boundary layer by conduction, and is then transported in the turbulent mixture of the main stream. The similarity between velocity and temperature fields encourages the use of the Reynolds number, which determines the extent of the regions in which the two processes take place. In the slugging and annular stages of evaporative flow, the liquid appears to be vaporized at the interfaces between liquid and vapour, and the main stream mass transfer is thus carried out entirely by the vapour. It is suggested that in this respect the vapour column behaves as if it were a highly turbulent liquid moving at the vapour velocity, and that the liquid in the main stream (between slugs) and on the outer wall of the annulus only replenishes the boundary layer and provides evaporative fluid. When this supply is exhausted the liquid boundary layer itself diminishes rapidly by evaporation and the dry condition with "burn-out" of the heater surface is approached.

The possible convective nature of the heat transfer has thus been described and the main hypothesis may now be stated in the form of the Nusselt similarity relationships: that the heat transfer coefficient is given by an expression of the form

$$[Nu] = \frac{hd}{k_L} = C\,[Re]^n\,[Pr]^m \tag{3}$$

where the Reynolds number, Re is defined as $[(V_v d_v \rho_L)/\mu_L]$, and the Prandtl number for the liquid $[Pr] = [(c_p \mu)/k]_L$.

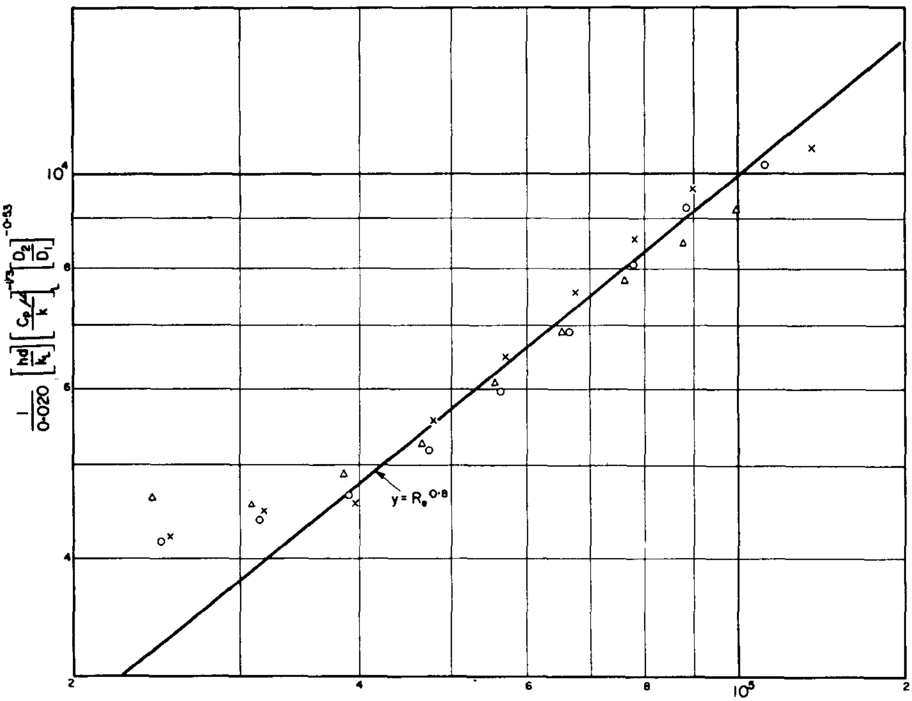

FIG. 4. Correlation of experimental results for forced convection, phase-change heat transfer at high vapour fractions—stratified flow of Refrigerant 11 in a heated annulus.

The physical properties of the liquid are used, as liquid forms the convective boundary layer. The effective diameter in determining the action of the vapour on the liquid is the hydraulic diameter of the vapour cross-section—or of approximately the total flow cross-section in a plain tube. The typical dimension of length in the Nusselt number is that which is representative of the heated surface, for instance the heater diameter in an annulus, in order to have general applicability.

5. EXPERIMENTAL CORRELATION

For the annular arrangement of the experimental apparatus the modified empirical equation due to Monrad and Pelton [11] appeared appropriate:

$$[Nu] = 0{\cdot}020\ [Re]^{0\cdot 8}\ [Pr]^{0\cdot 33} \left(\frac{D_2}{D_1}\right)^{0\cdot 53} \qquad (4)$$

where D_2 is the diameter of the outer tube, D_1 the diameter of the central heater.

In Fig. 4 the quantity

$$\frac{1}{0{\cdot}020}\, Nu\ [Pr]^{-0\cdot 33} \left(\frac{D_2}{D_1}\right)^{-0\cdot 53}$$

is plotted against Reynolds number for the observed experimental values, which are shown to be correlated well by raising the Reynolds number to the power 0·8, with retention of the identical numerical constant of 0·020 as is used for forced convection.

6. CONCLUSION

It has been found that nucleate boiling in a vertical tubular heat transfer apparatus may occupy a very small proportion of the total heated length in the direction of flow.

Observation of the flow regimes obtaining with net vapour generation has led to the hypothesis that convective processes, rather than nucleate boiling phenomena, apply to the extensive region subsequent to coalescence of the vapour bubbles produced in the nucleate boiling zone. A possible explanation of this has been discussed and a standard forced convection expression has been shown to correlate the experimental heat transfer and flow data, using a simple modification to the Reynolds number.

Many expressions have been developed to correlate boiling heat transfer phenomena, and have been used outside their field of application on the supposition that nucleate boiling heat transfer generally predominates in a practical system involving change of phase. It is believed that the present description of heat transfer in the stratified flow region subsequent to the nucleate boiling zone may assist in interpretation of future results.

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