Peripheral variation of heat transfer under pool boiling on tubes

K. Cornwell and J. G. Einarsson*

Boiling heat transfer measurements on a tube designed to yield the peripheral variation of heat transfer coefficient without interfering with the nucleation site density are presented. A variation of up to 25% around the tube is found with a maximum at the base. High speed cine photography was used to estimate the variation of mean bubble layer thickness and mean velocities with angle. An iterative heat balance around the periphery indicated a voidage decrease from about unity at the base to 0.3–0.6 at 90°

Key words: *heat transfer, boiling, two-phase flow*

The main experimental problem when designing a rig to determine h variation with θ on horizontal tubes under boiling conditions lies in obtaining a sufficiently localised value of surface temperature without affecting the local nucleation site density. A number of workers[1–4] have used essentially Method (a) in Fig 1 and their results suffer from the disadvantage that an undetermined amount of heat flows tangentially through the metal tube leading to a dampening of the characteristic $h = f(\theta)$. In an attempt to correct this, tests were undertaken in our laboratory using Method (b), but this proved unsatisfactory owing to spurious nucleation at the resin surface. Tests essentially using Method (c) were recently reported by Bier *et al*[5] who analysed the two-dimensional effects of heat flow in the metal. The work reported here is based on Method (d) in an attempt to isolate the boiling characteristic from tangential conduction effects within the surface.

The observation and basic analysis of bubbles which slide around the circumference from the base of the tube have been reported previously[6,7]. Study of individual sliding bubbles is, however, unlikely to lead to evaluation of the time-averaged variation of h with θ owing to the unpredictabilities of nucleation site distribution and bubble frequency. Conventional analysis of the two-phase tangential flow around the tube, although somewhat limited, does throw some light on the form of flow in the bubble layer.

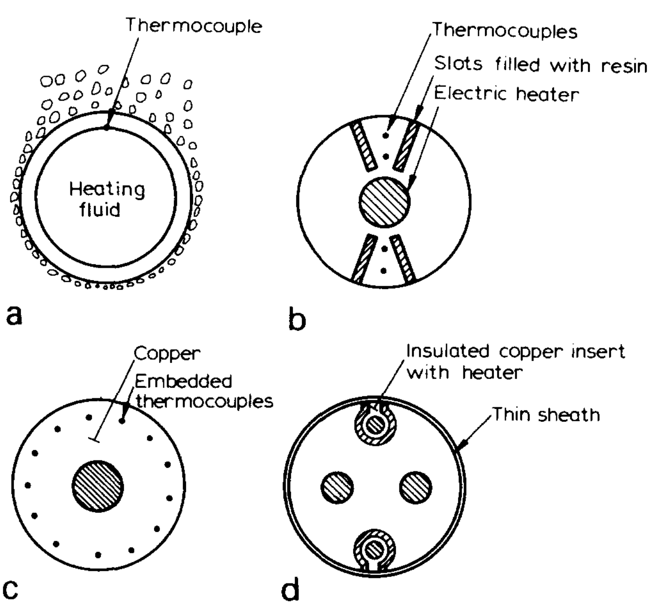

Fig 1 Possible methods of measuring h = f(θ)

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Experimentation

The rig consists of a rectangular test section with glass sides in which nucleate boiling occurs on a 27 mm diameter cylinder over a length of 76 mm. Internally the cylinder is arranged as in Fig 2 and, by balancing

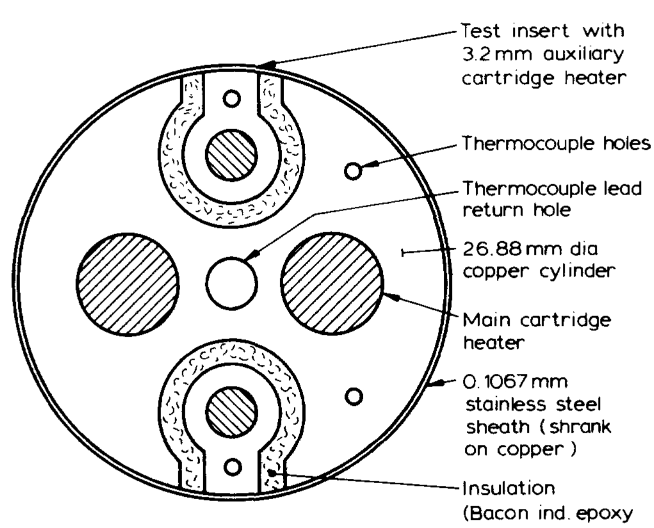

Fig 2 Cross section of the tube

the 4 cartridge heaters, a nominally uniform surface heat flux can be maintained, ie the mean surface heat flux at the insert is the same as the mean heat flux for the rest of the tube. The surface temperature at the test locations can be found from a knowledge of the power input and temperature at the thermocouple position. The tube surface was finished with 400 grit emery paper and aged for over 2 hours before each test.

Fig 3 shows the average results from 10 tests at each heat flux. Readings at a particular angle and heat flux varied by up to 30%. The results are in general agreement with Lance and Myers[1] and Bier *et al*[5] with the high temperature at the top decreasing rapidly at the separation point (~60°, see Ref 8).

In order to analyse the two-phase flow in the bubble stream, cine film at 1000 frames/s was taken at each heat flux. Rough indication of the mean vapour velocity variation with angle was found by measuring the progress of many large (~2 mm) bubbles on the films. There were also a number of micro-bubbles (<0.2 mm) in the stream and these were used to indicate mean liquid velocity by assuming negligible slip for this bubble size and the small distances involved. The vapour velocities obtained for the case of $q = 25 \text{kW/m}^2$ are shown in Fig 4. The bubble layer thickness is fairly well defined in the film and mean values at each heat flux are shown in Fig 5.

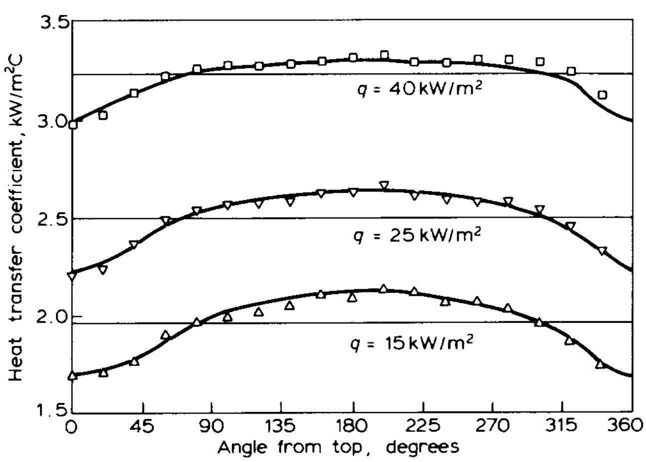

Fig 3 Variation of h *along the circumference of the tube*

Analysis of bubble layer

Voidage from two phase flow theory

If the bubble layer thickness δ is taken as small compared to the cylinder radius r, the governing factor in the tangentially diverging layer is the pressure drop

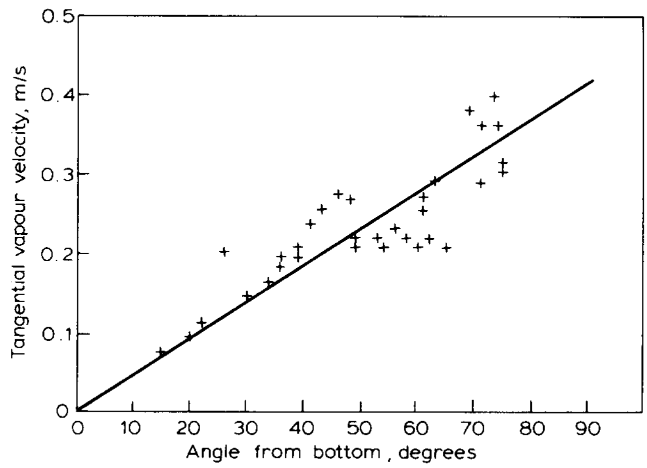

Fig 4 Velocity distribution of large bubbles underneath the tube

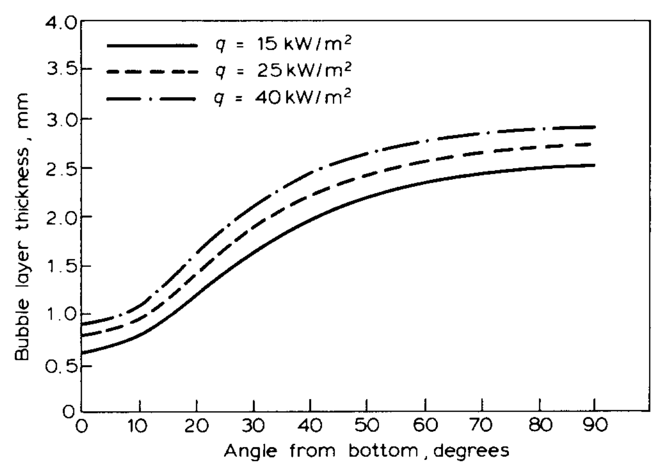

Fig 5 Bubble layer thickness for lower half of tube

Nomenclature

A	Area
C	Friction factor
g	Acceleration due to gravity
h	Heat transfer coefficient
h_{fg}	Latent heat of evaporation
$\dot{m}$	Mass flow rate
p	Pressure
q_b	Boiling component of heat flux
q_c	Convective component of heat flux
Q_{total}	Total heat flux ($q\Delta A_s$)
r	Radius of cylinder
U	Velocity
X_{tt}^2	Martinelli parameter
α	Voidage
δ	Bubble layer thickness
θ	Angle
ρ	Density
ΔA_s	Surface area of segment
ΔT	Temperature difference

Subscripts

f	Liquid
g	Vapour
r	Radial direction
t	Tangential direction
s	Tube surface

in the tangential direction. Application of separated flow theory is possible by defining the Martinelli parameter as:

$$X_{tt}^2=\frac{\left(\frac{1}{r}\frac{\mathrm{d}p}{\mathrm{d}\theta}\right)_\mathrm{f}}{\left(\frac{1}{r}\frac{\mathrm{d}p}{\mathrm{d}\theta}\right)_\mathrm{g}}
=\frac{\left(\frac{1}{r}\frac{\mathrm{d}p}{\mathrm{d}\theta}F\right)_\mathrm{f}+\left(\frac{1}{r}\frac{\mathrm{d}p}{\mathrm{d}\theta}\frac{\dot{m}u}{A}\right)_\mathrm{f}+\left(\frac{1}{r}\frac{\mathrm{d}p}{\mathrm{d}\theta}z\right)_\mathrm{f}}{\left(\frac{1}{r}\frac{\mathrm{d}p}{\mathrm{d}\theta}F\right)_\mathrm{g}+\left(\frac{1}{r}\frac{\mathrm{d}p}{\mathrm{d}\theta}\frac{\dot{m}u}{A}\right)_\mathrm{g}+\left(\frac{1}{r}\frac{\mathrm{d}p}{\mathrm{d}\theta}z\right)_\mathrm{g}}$$

that is, in terms of the frictional accelerational and gravitational forces. Substitution of the relevant forces, taking the frictional component to act on the wall-side only, yields:

$$X_\mathrm{tt}^2=\frac{\frac{C_\mathrm{f}\rho_\mathrm{f}U_\mathrm{f}^2}{2\delta}+\frac{2}{r}\rho_\mathrm{f}U_\mathrm{f}\frac{\mathrm{d}U_\mathrm{f}}{\mathrm{d}\theta}+\rho_\mathrm{f}g\sin\theta}{\frac{C_\mathrm{g}\rho_\mathrm{g}U_\mathrm{g}}{2\delta}+\frac{2}{r}\rho_\mathrm{g}U_\mathrm{g}\frac{\mathrm{d}U_\mathrm{g}}{\mathrm{d}\theta}+\rho_\mathrm{g}g\sin\theta} \quad (1)$$

Substitution of laminar flow frictional coefficients and values of $U_\mathrm{g}(\theta)$, $U_\mathrm{f}(\theta)$, $\mathrm{d}U_\mathrm{g}/\mathrm{d}\theta$, $\mathrm{d}U_\mathrm{f}/\mathrm{d}\theta$ and $\delta(\theta)$ from the experiments allows approximate experimental estimation of $X_\mathrm{tt}(\theta)$. Use of:

$$\alpha=(1-0.28X_\mathrm{tt}^{0.71})^{-1} \quad (2)$$

for the bubbly flow[9] then yields $\alpha(\theta)$ as shown in Fig 6.

Voidage from a heat balance

An alternative method is to estimate the voidage distribution by a heat balance. It is assumed that, for this case, straight addition of convective and boiling heat fluxes is appropriate (ie adaptive factors are unity) thus:

$$q(\theta)=q_\mathrm{c}(\theta)+q_\mathrm{b}(\theta) \quad (3)$$

The term $q_\mathrm{c}(\theta)$ is estimated from single phase flow (with $m=1$ for this case[5]).

Since bubbles grow as they slide, the mass vapour flowrate has a radial velocity component as well as tangential component. The radial component is given by:

$$U_\mathrm{r}(\theta)=U_\mathrm{t}(\theta)\frac{\mathrm{d}\delta(\theta)}{r\,\mathrm{d}\theta} \quad (4)$$

The vapour flow rate for surface area ΔA_s is:

$$\dot{m}_\mathrm{gr}(\theta)+\dot{m}_\mathrm{gt}(\theta)=\frac{Q_\mathrm{total}-q_\mathrm{c}(\theta)\,\Delta A_\mathrm{s}}{h_\mathrm{fg}} \quad (5)$$

where $\dot{m}_\mathrm{g}$ is vapour flowrate produced adjacent to ΔA_s. The vapour flowrate components in terms of velocities are:

$$\dot{m}_\mathrm{gt}(\theta)=\rho_\mathrm{g}A_\mathrm{c}(\theta)U_\mathrm{t}(\theta)\alpha(\theta) \quad (6)$$

$$\dot{m}_\mathrm{gr}(\theta)=\rho_\mathrm{g}\Delta A_\mathrm{s}U_\mathrm{r}(\theta)\alpha(\theta) \quad (7)$$

Solving Eqs (5), (6) and (7) in terms of $\alpha(\theta)$, for progressive steps i around the tube gives:

$$\alpha_i(\theta)=\frac{Q_\mathrm{total}-q_\mathrm{c}(\theta)\Delta A_\mathrm{s}+\sum_{n=1}^{i-i}(\dot{m}_\mathrm{gn}h_\mathrm{fg})}{\rho_\mathrm{g}h_\mathrm{fg}(U_\mathrm{ri}\Delta A_\mathrm{s}+U_\mathrm{ti}A_\mathrm{ci})}$$

By iterative computation, the voidage for the lower half of the tube can be obtained and the results are shown in Fig 7.

The asterisks in Fig 7 show the integrated value of α at 90° when it is assumed that all the heat is turned into vapour by this point. They thus indicate the maximum voidages at 90° for the three heat fluxes.

Discussion and conclusion

The general form of the variation of h around a cylinder in pool boiling (Fig 3) is in agreement with that of lance and Myers[4] and Bier *et al*[5]. However, the magnitude of the variation in our case is larger. This is because our measurement method prevents conduction of heat from the part of the surface being tested to the rest of the tube and the associated effect on nucleation site density.

The heat transfer coefficients are much higher than those expected from single-phase convection

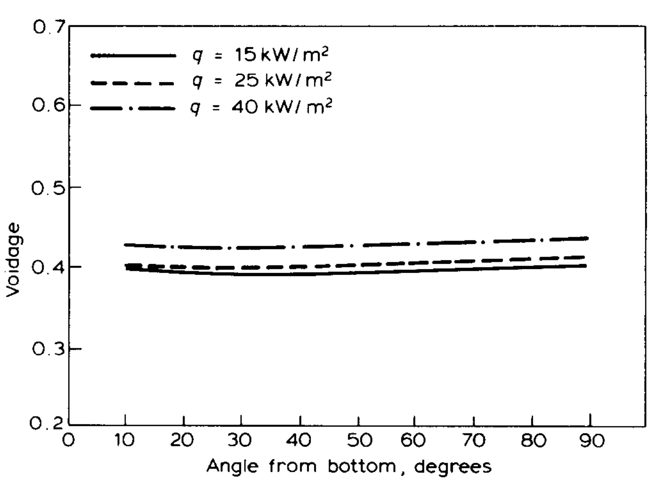

Fig 6 Voidage from two phase flow theory

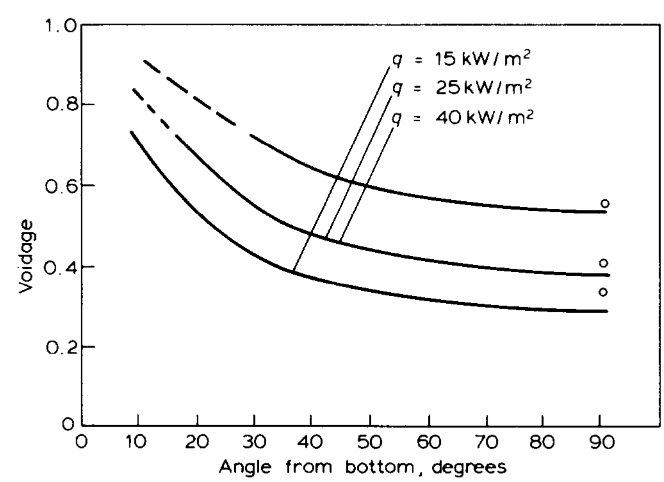

Fig 7 Voidage from a heat balance

alone and the predominant mechanism is therefore bubble induced. Observation of the tube showed vigorous nucleate boiling underneath and resulted in the highest h values being in this position. On the sides of the tube, the films showed very few active nucleation sites while Fig 3 indicates h values almost as high as those underneath. This can only be due to the strong effect of the sliding bubbles which were very evident in the films (and have been reported on other tubes[7]). The low h value recorded above the tube corresponds with a low nucleation site density in this area.

The mean tangential bubble velocity around the tube at some value of θ is obviously a rather crude concept as confirmed by the scatter in Fig 4. However, the indicated value of about 0.4 m/s at 90° for this tube does correlate well with the results on various diameters given elsewhere[10]. The free rise velocity for 2 mm diameter bubbles under these conditions is 0.14 m/s.

Conventional two-phase separated flow theory leads to incorrect voidage predictions (Fig 6). In particular visual inspection shows that at the higher heat fluxes the voidage underneath the tube approaches unity. Iterative heat balance around the circumference leads to a voidage variation (Fig 7) which more nearly fits the boundary conditions of tendency towards high values at $\theta = 0$ and the indicated voidage from the total heat flux at $\theta = 90°$.

It would appear that the experimentally measured value of h varies considerably around the circumference when there is no surface conduction. h is dependent on the mechanisms of bubble initiation, growth, sliding and induced luquid movement in a complex way which is peculiar to this geometry. The practice of applying results from boiling on flat plates or thin wires to pool boiling on tubes is suspect and was shown in previous work[10] to yield considerable errors.

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