Int. J. Heat Mass Transfer. Vol. 6, pp. 863–867. Pergamon Press 1963. Printed in Great Britain.

SHORTER COMMUNICATIONS

THE EFFECT OF SURFACE CONFIGURATION ON NUCLEATE BOILING HEAT TRANSFER

B. D. MARCUS* and D. DROPKIN†
Thermal Engineering Department, Cornell University, Ithaca, New York

(*Received* 4 *February* 1963)

NOMENCLATURE

h_θ, coefficient of heat transfer for surface at angle θ from the horizontal, Btu/ft^2 h degF;
q, heat flux, Btu/ft^2 h.

INTRODUCTION

PREVIOUS authors have suggested that the surface configuration is not a factor in nucleate pool boiling heat transfer except perhaps at very low heat fluxes where convection effects are still important, and at the critical heat flux [1, 2, 3]. (The critical heat flux seems to be a hydrodynamically controlled phenomenon and hence geometry will be a variable [4].) The logic behind this thesis is that nucleate boiling is a localized process and thus the overall geometry is unimportant. Nucleate boiling results for horizontal and vertical surfaces, tubes, wires, etc. have all shown similar behavior and this fact has been accepted as proof of this hypothesis.

However, considering the many variables in nucleate boiling heat transfer and the general difficulty encountered in repeating data, even with the same apparatus, it seems unreasonable to discount the effect of surface configuration by making a gross comparison of the results presented by numerous experimenters employing different apparatus.

To resolve this question, experiments were conducted whereby distilled water was boiled under atmospheric saturated conditions with all factors affecting nucleate boiling held constant except the angular position of the hot surface.

DESCRIPTION OF APPARATUS

The apparatus is shown schematically in Figs. 1 and 2. Fig. 1 shows the location of the boiling surface in the test cell. The cell included two nested glass tanks with a common cover. The capacity of the inner and outer tanks was 2 and 12 gal respectively. The heater with its boiling surface was suspended in the inner tank from the cover and could be rotated on pins to any angle. Power and thermocouple leads were brought to and from the unit through three flexible tubes (one shown). The height of the water level above the boiling surface was 4 to 5 in. Both tanks were filled with distilled water and the system was kept at saturation temperature by four immersion heaters in the outer tank (one shown). The bulk liquid temperature within the inner tank was measured by a stainless steel encased thermocouple which could be set at any height. The vapor generated in both tanks was allowed to escape through vents which assured atmospheric conditions. The system was insulated with 2 in of fiberglass except for two viewing ports to permit visual observations.

Fig. 2 is a schematic diagram of the heater and boiling surface. A copper block was machined to $2 \times 2 \times 1\frac{3}{8}$ in. One 2×2 in face served as the boiling surface. The other face, along with a matching compression plate served as a sandwich for the Nichrome heating element. The 0·002 in thick Nichrome element was sandwiched between two 0·01 in thick mica sheets and this assembly was pressed between the two copper surfaces. A $\frac{3}{8}$ in stainless steel fin, $\frac{1}{16}$ in thick, was soldered flush with the boiling surface and finished smooth. This prevented preferential boiling at the edge of the copper block. Four 36 gage copper–constantan thermocouple assemblies, in 0·040 in porcelain insulators, were spring-loaded into position at $\frac{1}{4}$ in intervals from the boiling surface. Refractory spacers and air gaps were used to locate the copper block and to reduce heat losses. The outer case was made of water and heat resistant Micarta.

EXPERIMENTAL PROCEDURE

The method used was to set the heater for a constant power input and an angle θ, wait for equilibrium conditions, and then determine the surface temperature. This procedure was repeated for angles of 0, 22·5, 45, 67·5 and 90° from the horizontal without altering the input power. By continuing the measurements in reverse order and then repeating the entire cycle, the averages of the four readings at each angle minimized any possible effect due to "aging" of the boiling surface.

The heat flux was measured both by correcting the input power for heat losses and by using the temperature gradient obtained from the thermocouples. Agreement between these two values was always very good.

* Graduate Student.
† Professor.

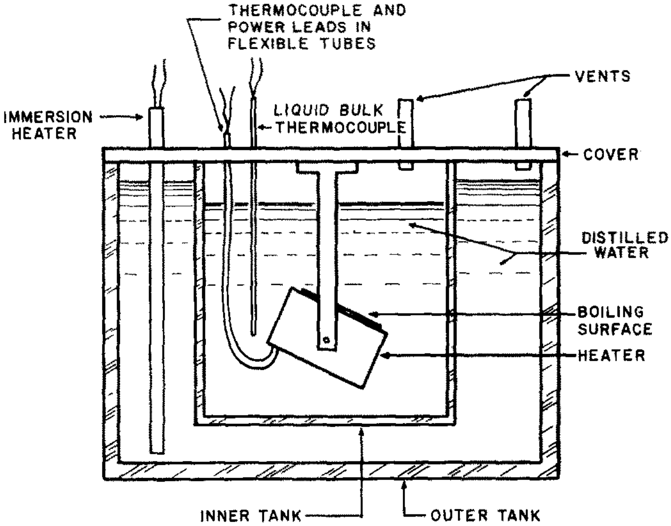

FIG. 1. Schematic diagram of test cell showing the position of the boiling surface in relation to the test vessel.

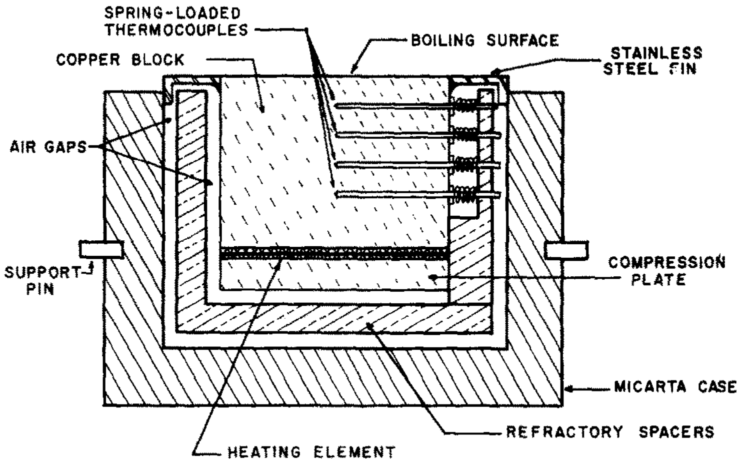

FIG. 2. Schematic detail of the heater and boiling surface showing the spring-loaded thermocouple installation and the heating arrangement.

The surface temperature was measured by extrapolating the observed temperature gradient to the surface. There was very little random scatter in the observed data. A small systematic increase in h due to surface "aging" was eliminated from the results by the averaging procedure previously described.

RESULTS AND CONCLUSIONS

The results of these experiments appear in Figs. 3 and 4. It is apparent that the surface angle is a significant variable in nucleate boiling heat transfer within the range of heat fluxes tested. Variation of the surface angle between 0 and 90° alters h_θ by as much as 25 per cent.

Fig. 3 shows the variation in h_θ with surface angle for boiling heat fluxes from $q = 5265$ to 35 550 Btu/ft^2 h. The curve for $q = 305$ Btu/ft^2 h represents the saturated convection condition. The data of Jakob and Linke [5] show a similar trend; i.e. h_0 (for horizontal surfaces) is substantially greater than h_{90} (for vertical surfaces) at conditions of saturated convection and incipient boiling.

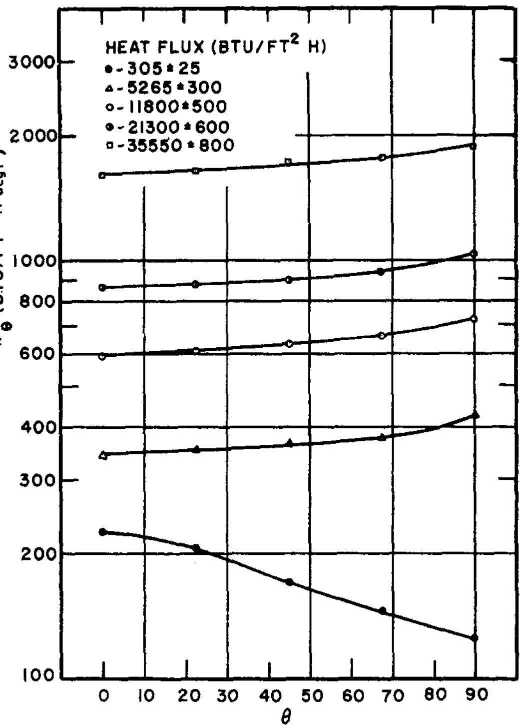

FIG. 3. Coefficient of heat transfer as a function of the boiling surface angle for saturated convection and boiling at several heat fluxes. $\theta = 0°$ represents the horizontal surface.

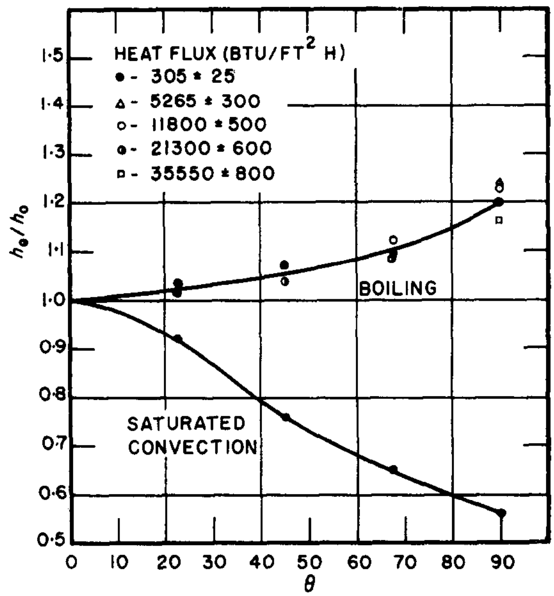

FIG. 4. Ratio of h_θ/h_0 as a function of the boiling surface angle for saturated convection and boiling at several heat fluxes. $\theta = 0°$ represents the horizontal surface.

In the case of vigorous boiling, however, the results indicate a reversal of this trend. It was further observed that the number of nucleating sites was substantially decreased as the surface angle was increased. These results are expected if one accepts the hypothesis that agitation of the super-heated boundary region by the growth and departure of the vapor bubbles contributes significantly to the heat transfer. As the surface angle is increased the departing bubble's path length through this region increases and thus the agitation per bubble is greater. This results in a higher h_θ and smaller bubble density for the same q.

In Fig. 4 the data is plotted as the ratio h_θ/h_0 against the surface angle. Apparently, in the range tested, this quantity does not vary significantly with the heat flux for boiling. Of course, this cannot hold true at all fluxes since for $q = 305$ Btu/ft^2 h (saturated convection) the trend with angle is reversed. The effect of heat flux on this curve must be a continuous function and thus q must be a parameter at lower fluxes. Yamagata *et al.* [6], have observed a transition in nucleate boiling at about 3700 Btu/ft^{2}h. Below this flux the gross convection pattern was undisturbed by the rising bubbles while above it there was general mixing and turbulence within the boundary region. It is possible that this breakdown in the convection pattern also represents the point above which h_θ/h_0 is insensitive to the heat flux. Unfortunately, it was not possible to test this hypothesis with the present apparatus. At fluxes below 5000 Btu/ft^{2}h the active sites would be located primarily at the top edge of the surface for the large angles. This yielded a non-uniform heat flux. Since the method of measuring surface temperature was based on a uniform heat flux, uniform boiling was necessary for this to be a representative value.

The results presented in this paper appear to definitely establish the fact that the angular position of the hot surface is an important factor to consider for both saturated liquid convection and boiling at relatively low heat fluxes. These conclusions are corroborated to some extent by the data of Jakob and Linke [5].

REFERENCES

1. J. W. WESTWATER, *Advances in Chemical Engineering*, T. DREW and J. HOOPES, *Boiling of Liquids*, Vol. 1, p. 58. Academic Press, New York (1956).
2. W. M. ROHSENOW and H. CHOI, *Heat, Mass, and Momentum Transfer*, p. 222. Prentice-Hall, New Jersey (1961).
3. M. A. STYRIKOVICH and G. M. POLYAKOV, On the critical heat flux in pool boiling of liquids, *Izv. Nauk S.S.S.R., Otdel. Teknich. Nauk*, **5**, 652–656 (1951).
4. N. ZUBER, Hydrodynamic aspects of boiling heat transfer. A.E.C. Report AECU-4439 (1959).

5. M. JAKOB and W. LINKE, Der Wärmeübergang beim Verdampfen von Flüssigkeiten an senkrechten und waagerechten Flächen. *Phys. Z.* **36**, 267–280 (1935).

6. K. YAMAGATA, F. HIRANO, K. NISHIKAWA and H. MATSUOKA, Nucleate boiling of water on the horizontal heating surface. *Mem. Fac. Engng. Kyushu*, **15**, 97–163 (1955).

TURBULENT HEAT TRANSFER IN A TUBE WITH CIRCUMFERENTIALLY-VARYING TEMPERATURE OR HEAT FLUX

E. M. SPARROW and S. H. LIN
University of Minnesota, Minneapolis, Minnesota

(*Received* 21 *February* 1963)

ALTHOUGH both experiment and analysis of turbulent heat transfer in a circular tube are generally carried out under the assumption that temperature and heat flux are circumferentially uniform, there are many practical situations where the thermal boundary conditions vary around the tube circumference. For instance, such variations may arise if the tube is externally heated (or cooled) by a forced-convection crossflow, by free convection, by film condensation, or by thermal radiation from a non-uniform environment. This paper presents a brief account of an analysis of turbulent heat transfer with circumferentially-varying temperature or heat flux. It is expected that experiments will be made later to check on the nalysis.assumptions of the a

Consider a hydrodynamically and thermally fully-developed tube flow in which there is a uniform heat-transfer rate per unit length Q, but in which the wall temperature T_w may vary around the circumference in an arbitrary manner as given by a Fourier series

$$\frac{[T_w(\varphi) - \bar{T}_w]}{(\bar{q}r_o/k)} = \sum_{n=1}^{\infty} a_n \cos n\varphi + b_n \sin n\varphi \qquad (1)$$

in which φ is the angular co-ordinate, r_o the tube radius, $\bar{q}$ the average heat flux rate per unit area ($\bar{q} = Q/2\pi r_o$), and $\bar{T}_w$ the circumferentially-averaged wall temperature at some axial position. It will be shown later that the circumferential variation of the heat flux could equally well be prescribed in lieu of $T_w(\varphi)$.

The analysis begins with the energy equation for fully-developed flow and heat transfer ($\partial T/\partial z = -2\bar{q}/r_o\bar{u}\rho c_p$, $\bar{q}$ is positive from fluid to wall)

$$-\left(\frac{4}{r_o^+ RePr}\right) r^+u^+ = \frac{\partial}{\partial r^+}\left[r^+\left(\frac{1}{Pr} + \frac{\epsilon_{h,\,r}}{\nu}\right)\frac{\partial\theta}{\partial r^+}\right] + \frac{1}{r^+}\frac{\partial}{\partial\varphi}\left[\left(\frac{1}{Pr} + \frac{\epsilon_{h,\,\varphi}}{\nu}\right)\frac{\mathrm{d}\theta}{\partial\varphi}\right] \qquad (2)$$

in which

$$\theta = \frac{(T - \bar{T}_w)}{(\bar{q}r_o/k)}, \quad r^+ = \frac{r\sqrt{(\tau_w/\rho)}}{\nu}, \quad u^+ = \frac{u}{\sqrt{(\tau_w/\rho)}} \qquad (3)$$

and Re and Pr are the Reynolds and Prandtl numbers, τ_w is the wall shear. It is the belief of the authors that the turbulent transport is set up by the action of the flow field, which is axisymmetric. Consistent with this, the eddy diffusivities for heat $\epsilon_{h,\,r}$ and $\epsilon_{h,\,\varphi}$ will be taken to be independent of φ.

Although information on the radial eddy diffusivity $\epsilon_{h,\,r}$ is available, there appears to be no information on the circumferential diffusivity, $\epsilon_{h,\,\varphi}$. As an initial assumption, $\epsilon_{h,\,r} = \epsilon_{h,\,\varphi}$ is not unreasonable.* Expressions for the eddy diffusivity and the velocity distribution, previously employed in an analysis of axisymmetric heat transfer [1] which has been checked by experiments for $Pr = 0{\cdot}7$ and $Pr = 7{\cdot}0$, will also be used here.

The solution of equation (2) for θ can be written as

$$\theta(r^+, \varphi) = \theta_{fd}(r^+) + \sum_{n=1}^{\infty} R_n(r^+)\,(a_n \cos n\varphi + b_n \sin n\varphi) \qquad (4)$$

in which θ_{fd} is the fully-developed temperature solution for axisymmetric heating and the R_n functions depend only on r^+. The R_n are found by solving

$$\frac{\mathrm{d}}{\mathrm{d}r^+}\left[r^+\left(\frac{1}{Pr} + \frac{\epsilon}{\nu}\right)\frac{\mathrm{d}R_n}{\mathrm{d}r^+}\right] - \frac{n^2}{r^+}\left(\frac{1}{Pr} + \frac{\epsilon}{\nu}\right)R_n = 0,$$

$$R_n(0) = 0, \; R_n(r_o^+) = 1. \qquad (5)$$

This two point boundary value problem has been solved by applying a numerical integration over most of the cross section and matching this with an analytic series solution near $r^+ = 0$ (where a numerical solution fails). Representative results will be shown later.

The local heat flux may be obtained by differentiating (4)

$$\frac{q}{\bar{q}} = 1 - r_o^+ \sum_{n=1}^{\infty} \left(\frac{\mathrm{d}R_n}{\mathrm{d}r^+}\right)_{r_o^+} (a_n \cos n\varphi + b_n \sin n\varphi). \qquad (6)$$

The variation of q with φ can thus be calculated. Conversely, if $q/\bar{q}$ is prescribed and $\mathrm{d}R_n/\mathrm{d}r^+$ is available from the solutions of equation (5), then the circumferential wall temperature variation may be found by applying

* Personal communication, Professor G. K. Batchelor, Cambridge University, June, 1962.