Int. J. Heat Mass Transfer. Vol. 32, No. 3, pp. 445–458, 1989
Printed in Great Britain

0017–9310/89 \$3.00+0.00

Saturated flow boiling of water in vertical tubes

D. B. R. KENNING and M. G. COOPER
Department of Engineering Science, Oxford University, Parks Road, Oxford OX1 3PJ, U.K.

(*Received* 22 *March* 1988 *and in final form* 10 *June* 1988)

Abstract—Accurate heat transfer data have been obtained for saturated flow boiling of water at 160–600 kPa in 9.6 and 14.4 mm bore vertical tubes under conditions dominated by convection or by nucleate boiling. Convection in the annular flow regime is well described by a modified Chen correlation; a further modification is required for plug/churn flow. Flow nucleate boiling is shown to be sensitive to surface conditions whereas convective boiling is not. Saturated convective and nucleate boiling are not additive: the larger of α_c or α_{nb} should be used. Appropriate features of general flow boiling correlations are discussed.

1. INTRODUCTION

IT IS GENERALLY accepted that experimental data for saturated flow boiling heat transfer fall between two limiting classes of behaviour:

(a) apparently convective boiling (heat transfer coefficient α_c dependent on mass flux and quality but not on wall superheat);

(b) apparently nucleate boiling (heat transfer coefficient α_{nb} dependent on wall superheat but not on mass flux and quality).

Butterworth and Shock [1] in their 1982 review of flow boiling noted the popularity of prediction methods based on Chen's [2] correlation, having the following features:

(i) superposition of convective and nucleate boiling

$$\alpha = \alpha_c + \alpha_{nb}, \tag{1}$$

(ii) enhancement of liquid-only convection by a factor F dependent on the local thermodynamic quality x, e.g. combined in the Martinelli parameter $1/\chi_{tt}$

$$\alpha_c = \alpha_l F(1/\chi_{tt}), \text{ graphical} \tag{2}$$

$$\alpha_l = 0.023 \frac{\lambda_l}{D} Re_l^{0.8}\, Pr_l^{0.4} \tag{3}$$

$$Re_l = \frac{G(1-x)D}{\mu_l} \tag{4}$$

$$1/\chi_{tt} = \left[\frac{x}{(1-x)}\right]^{0.9} \left[\frac{\rho_l}{\rho_g}\right]^{0.5} \left[\frac{\mu_g}{\mu_l}\right]^{0.1}, \tag{5}$$

(iii) modification of the pool nucleate boiling heat transfer coefficient α_{pb} by a suppression factor $S < 1$ dependent on the flow conditions, e.g.

$$\alpha_{nb} = \alpha_{pb} S(Re_l\, F^{1.25}), \text{ graphical.} \tag{6}$$

Some workers, e.g. Shah [3] replace (i) by the assumption that α is equal to the larger of α_c or α_{nb}:

$$\text{if } \alpha_c > \alpha_{nb}, \quad \alpha = \alpha_c$$

$$\text{if } \alpha_{nb} > \alpha_c, \quad \alpha = \alpha_{nb}. \tag{7}$$

Gungor and Winterton [4] tested nine saturated flow boiling correlations, including their own (GW) development and the Chen and Shah correlations, against a data bank for water, refrigerants and ethylene glycol in vertical (upward and downward) and horizontal flow in tubes and annuli. The GW and Shah correlations performed best, giving 21% mean deviation in α for all fluids and 17% for water in tubes, compared to 58 and 30% for the original Chen correlation. These three correlations apparently employ different formulations for the convective enhancement factor F, Chen's graphical relationship being approximated by

Chen

$$F = 1 + 1.8(1/\chi_{tt})^{0.79}, \quad 1/\chi_{tt} \geqslant 1 \tag{8}$$

Shah

$$F = 1.8[1/\chi_{tt}]^{0.71} \left[\frac{\rho_l}{\rho_g}\right]^{0.04} \left[\frac{\mu_l}{\mu_g}\right]^{0.07} \tag{9}$$

GW

$$F = 1 + 1.37(1/\chi_{tt})^{0.86} + 2.4 \times 10^4 Bo^{1.16}. \tag{10}$$

Gungor and Winterton introduced a dependence on heat flux, through the Boiling number, Bo, because the 'generation of vapour itself in the boiling process results in significant disturbance of the (wall) layer'. For water in the pressure range 160–600 kPa, the conditions of the work to be described in this paper, the three convective correlations are in fact rather similar (Fig. 1). ($F-1$ is chosen as the ordinate, rather than F, to give more linear plots at small $1/\chi_{tt}$.) This suggests that the improved performance of the GW and Shah correlations over the Chen correlation may be due to the treatment of nucleate boiling, for which Chen used a now outdated correlation and GW used the recent correlation in ref. [5]. It is not possible, on the basis of the reported performance of the cor-

NOMENCLATURE

Bo	Boiling number, $\dot{q}/(h_{lg}G)$
c	specific heat [J kg^{-1} K^{-1}]
D	diameter [m]
F	enhancement factor, equation (2), α_c/α_l
G	total mass flux [kg m^{-2} s^{-1}]
h_{lg}	latent heat [J kg^{-1}]
J_g^*	dimensionless superficial vapour velocity, equation (14)
P_R	reduced pressure
Pr	Prandtl number
p	pressure [Pa]
$\dot{q}$	heat flux [W m^{-2}]
Re_l	liquid-only Reynolds number, equation (4)
S	suppression factor, equation (6), α_{nb}/α_{pb}
St	Stanton number, $\alpha/(c_l G)$
T	temperature [K, °C]
ΔT_s	wall superheat [K]
x	thermodynamic quality.

Greek symbols

α	heat transfer coefficient [W m^{-2} K^{-1}]
λ	thermal conductivity [W m^{-1} K^{-1}]
μ	viscosity [N s m^{-2}]
ρ	density [kg m^{-3}]
σ	surface tension [N m^{-1}]
$1/\chi_{tt}$	Martinelli parameter, equation (5).

Subscripts

c	convective
g	vapour
l	liquid
nb	nucleate boiling
pb	pool boiling
s	saturation
w	wall.

relations, to distinguish between Shah's simple treatment of the influence of heat flux by equation (7) and GW's more elaborate treatment in which heat flux affects both α_{nb}, equation (6), and α_c, equation (10), which are then added, equation (1). Bennett and Chen [6] suggested yet another possible dependence of α_c on heat flux. They argued that the original Chen convective correlation should apply only in the complete absence of nucleate boiling, or if $Pr = 1$. Otherwise the disruption of molecular transport at the wall by bubble growth switches on a correction factor which is independent of heat flux (cf. GW):

$$F(Pr \neq 1) = Pr^{0.3}\, F(\text{Chen}). \tag{11}$$

Their analysis contains an error which leads to a result incompatible with their physical model: if the heat transfer is entirely turbulent α_c must be independent of molecular thermal conductivity and the F factor in any correlation using equation (3) for liquid-only transfer must contain a factor $Pr^{0.6}$. Gungor and Winterton found that ref. [6] generally overestimated the experimental data; use of the correct factor would make the agreement even worse. However, Ross *et al.* [7], finding that the uncorrected Bennett and Chen

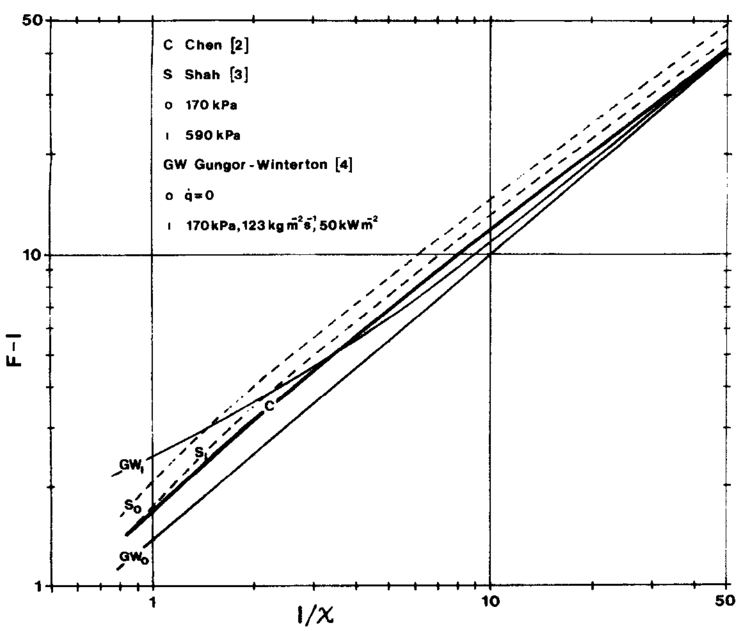

FIG. 1. Comparison of convective boiling correlations.

correlation gave good agreement with convective data for pure refrigerants in *horizontal* tubes, used the assumed dependence on the presence or absence of bubble nucleation to explain the lower convective coefficients which they measured for mixtures.

The significantly larger uncertainty in the correlations for two-phase heat transfer, compared to those for single-phase flow, may be due to inadequacies in the correlations and/or in the experimental data. The assumptions behind Chen-type correlations have been criticized by Mesler [8], who pointed out that experiments performed only with axially uniform heat flux (as most boiling experiments are) cannot separate the effects of upstream history and local quality: they cannot test the hypothesis that the heat transfer coefficient depends only on local conditions. Mesler also argued [9] that conventional deep-pool nucleate boiling is irrelevant for the very thin liquid films of the annular flow regime, in which very efficient heat transfer driven by entrained bubbles (rather than bubbles from stable nucleation sites on the wall) might occur. It is remarkably difficult either to prove or disprove the modelling assumptions embedded in the correlations. The first step must be to confine attention to accurate data for well-defined conditions, preferably falling in the limiting conditions of convection-dominated or nucleate boiling-dominated heat transfer. Gungor and Winterton kindly made available their data bank which was examined and extended by one of us (MGC), confining attention to water in vertical upflow in round tubes. The numbers of data points which could then be classified as 'apparently convective' and as 'apparently nucleate boiling' were small and it was generally impossible to assess the accuracy of the data from the published details. It was concluded that it was essential to extend the range of accurate data, paying particular attention to measurements at low heat flux in the 'convective' mode.

The work described in this paper is a continuation of previous studies of flow boiling of water in vertical tubes [10, 11] with further improvements in experimental technique. The difficulties experienced in achieving high accuracy reinforced the concern over incorporating all available measurements in data banks without regard to their accuracy. The difficulties encountered included accuracy of inner wall temperatures deduced from measurements on the outer surface, measurement of local saturation pressure to an accuracy compatible with the accuracy of wall temperature measurement and long-term flow stability.

Figure 2 shows a sketch of the apparatus used, with water flowing up a vertical tube with two independently-heated sections, a lower 'steam generator' (1 on Fig. 2) and an upper 'test section' (4) separated by an adiabatic section (3). This met Mesler's criticism, in that the thermodynamic quality near the top of the test section (4) could be generated by a variety of upstream conditions. Also it made it possible to measure heat transfer coefficients at low heat flux

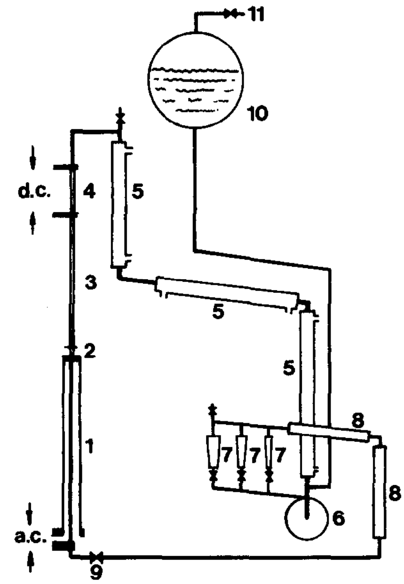

FIG. 2. Flow boiling rig: 1, steam generator (a.c.); 2, transition tube; 3, adiabatic section; 4, upper test section (d.c.); 5, heat exchanger; 6, pump; 7, flow meter; 8, preheater; 9, throttle valve; 10, pressurizer; 11, vacuum.

and high quality without the need for a very long uniformly heated tube.

The range of conditions previously studied in a tube of 9.6 mm bore throughout was extended, and measurements were also made in a tube of 14.4 mm bore for the adiabatic section (3) and test section (4), coupled by a short diffuser (2) to the original steam generator (1) of 9.6 mm bore. The main emphasis was on measurements at different heat fluxes confined to low wall superheats in order to establish the existence of the 'apparently convective' regime and whether it could be correlated in terms of local conditions. Experiments at higher heat fluxes and wall superheats investigated the characteristics of the 'apparently nucleate' boiling regime and its interaction with the convective regime; these experiments included modifications to the surface finish of the test section. The restriction of the experiments to water necessarily excluded the examination of some important parameters, notably Prandtl number which only varied from 1.1 to 1.5. Nevertheless it has been possible to draw useful conclusions about the appropriate form of general flow boiling correlations.

2. EXPERIMENTAL METHODS AND ACCURACY

Figure 2 also shows the remaining components of the Oxford flow boiling rig. Heat is rejected in water-cooled condenser/heat exchangers (5). The closed system contains deionized water initially degassed by boiling under vacuum in a tank (10) fitted with an electrical immersion heater and thermostat, which is subsequently used to maintain the system pressure at any chosen value up to 600 kPa. Degassing is completed by boiling in the steam generator (1) and venting gas periodically from the top of the condenser. To check that it is complete, each batch of tests includes one with zero heat flux at the test section, in which observed pressure and wall temperature are checked against saturation values. If degassing is incomplete, flow instability occurs, probably a consequence of the downflow design of the condenser.

The water is circulated by a centrifugal pump (6) through a variable-area flow meter (7) calibrated to $\pm 1\%$ at its operating temperature of 80°C, a 2 kW electrical preheater (8) with a controller which maintains the exit temperature constant within ± 0.5 K and a throttle valve (9) before entering the bottom of the vertical test section assembly. The assembly comprises a lower steam generator 2.0 m long, an adiabatic section 1.5 m long in which the liquid and vapour develop a near-equilibrium flow pattern and the upper test section with a heated length of 0.5 m. The assembly is thermally insulated. Electrical guard heaters, controlled in four sections, are embedded in the insulation around the adiabatic and upper test sections and guard heaters are attached to the heavy brass electrodes of the upper test section.

The steam generator is a 9.6 mm bore stainless steel tube heated by the passage of a.c. current from a variable-tap transformer. Accurately calibrated instruments are used to determine voltage and current through it, and the power factor was expected to be close to 1.0, but the power calculated from their product is consistently 3% higher than the enthalpy rise calculated from the flow rate and temperature rise in single-phase heating. Experiments at different powers have shown that the very small heat loss from the steam generator does not account for the discrepancy. As the power input is used to calculate the specific enthalpy of fluid entering the upper test section in boiling experiments, the apparent electrical power is corrected by a factor 0.97. The main source of error in specific enthalpy (and hence in thermodynamic quality x and Martinelli parameter $1/\chi_{tt}$) is the uncertainty of $\pm 1\%$ in flow rate, which has an increasing effect as $x \to 0$. At 600 kPa the corresponding uncertainty in $1/\chi_{tt}$ is $\pm 6\%$ at $1/\chi_{tt} = 0.6$, decreasing to $\pm 2\%$ at $1/\chi_{tt} = 2.5$; the uncertainty at 160 kPa is about 20% smaller.

The adiabatic section and the upper test section are formed from a single length of cupronickel tubing, either 9.6 mm bore with 0.2 mm wall or 14.4 mm bore with 0.3 mm wall. The 14.4 mm tube is connected to the 9.6 mm bore steam generator by a conical transition 80 mm long. The test section is heated over a length of 500 mm by current from a d.c. rotary generator with a ripple of less than 0.25%. The current is determined from the voltage across a calibrated resistance and the power is calculated from the current and the test section voltage; the voltages are measured to 0.01% by a digital voltmeter. Inside wall temperatures are calculated from measurements of outside wall temperatures at two stations on the adiabatic section and ten stations on the test section (reduced to eight stations in later tests on the 9.6 mm tube, due to damaged thermocouples). The tube material was chosen to minimize the correction for the temperature difference across the wall (typically 0.35 K at 100 kW m^{-2} for 0.2 mm wall). The temperatures are measured by chromel–constantan thermocouples with cold junctions immersed in crushed ice and water, individually calibrated to ± 0.05 K against standards traceable to NPL. The thermocouple beads were ground flat and cemented to the tube over a pre-cured electrically insulating layer of epoxy cement 25 μm thick. It was found necessary to adjust the temperature of the guard heaters to within ± 5 K of the tube temperature in order to have an acceptably low heat flow through the insulation. Each thermocouple is switched manually to a digital voltmeter (accuracy 0.01%) connected to a microcomputer which takes ten readings over a period of 4 s and displays their average and standard deviation prior to acceptance for storage. Four thermocouples are attached at 90° intervals around the tube at each station on the test section and their readings are averaged. In the convection-dominated regime the circumferential variation is typically $\pm 2\%$ of the wall superheat, attributable in part to circumferential variations in wall thickness and heat flux.

The local pressure, and hence the saturation temperature, is calculated by linear interpolation between readings from pressure tappings at the inlet and outlet to the test section, connected by 3 mm bore U-tubes to absolute pressure transducers operating at room temperature. Early in the programme much difficulty was experienced with drift in the calibration of the transducers, making it impossible to match the accuracy of temperature measurement. This was overcome by using high-stability transducers which could be isolated from the boiling rig for regular *in situ* calibration against a mercury manometer (to 260 kPa) or a deadweight tester (to 600 kPa). The accuracy of calibration is $\pm 0.1\%$ of absolute pressure, corresponding to ± 0.03 K in saturation temperature. The limiting factor then becomes the small variations in system pressure during the measurement of wall temperatures. Each recorded pressure reading is the average of 20 readings taken over 8 s. The inlet and outlet pressures are measured at the beginning, end and two intermediate times during the scan of wall temperatures, and it is assumed that the pressure varies linearly with time between readings. With this procedure the agreement between the independent measurements of adiabatic wall temperature and local saturation temperature is normally better than 0.2 K. This is the best indication of the uncertainty in the measurement of wall superheat and hence of heat transfer coefficient (5% at a wall superheat of 4 K, improving to 1% at a superheat of 20K). The overall reproducibility of the data was tested by comparing run 18 in the convective regime at 160 kPa,

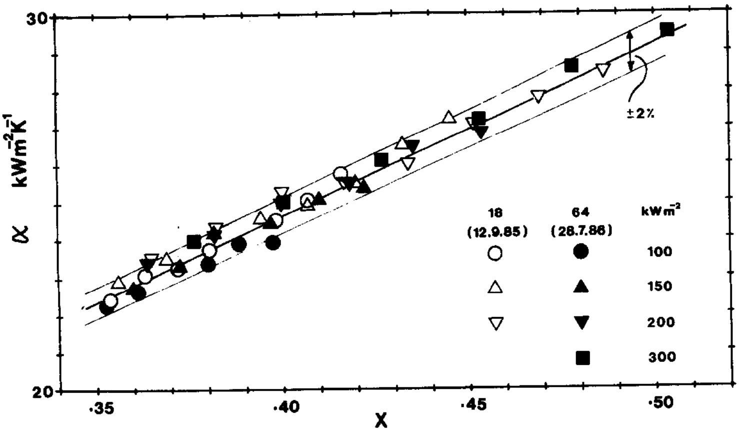

FIG. 3. Reproducibility in convective regime, runs 18 and 64 (modified surface).

123 kg m^{-2} s^{-1} and $x = 0.34$ and run 64 at the same conditions but after intervening experiments over a period of 6 months and modifications to the tube surface finish: the plots of α vs x agreed to within $\pm 2\%$ for fully-developed conditions at superheats exceeding 4 K (Fig. 3). Reproducibility in the nucleate boiling regime is discussed in Section 4.

Any experimental apparatus is constrained by its design to particular ranges of operating conditions. A change in one parameter may necessitate an accompanying change in another and this must be remembered when interpreting data. Experiments on the 9.6 mm bore tube could be performed at mass fluxes in the range 123–630 kg m^{-2} s^{-1} and heat fluxes up to 400 kW m^{-2}. Experiments on the 14.4 mm bore tube were confined to lower mass fluxes in the range 55–334 kg m^{-2} s^{-1} and heat fluxes up to 100 kW m^{-2}.

Each set of experiments was performed at constant mass flux and with constant power input to the steam generator, producing constant quality at the inlet to the test section. The adiabatic section allowed the phase distributions, e.g. resulting from the competing processes of droplet entrainment and deposition in the annular flow regime [10], to approach equilibrium so that the flow pattern entering the test section was not sensitive to the precise conditions of the heating process in the steam generator. The wall temperatures in the test section were measured for a series of increasing values of heat flux, previous experiments having shown that reducing the heat flux again did not cause hysteresis in the convective regime.

3. APPARENTLY CONVECTIVE BOILING

Typical sets of data, presented as plots of heat transfer coefficient α vs quality x, are shown in Fig. 4. We are seeking to establish whether there is a regime in which the heat transfer coefficient depends only on local flow parameters and not on local heat flux or upstream conditions. The interpretation of the data is complicated by thermal entrance effects, by the loss of accuracy at low wall superheats associated with very low heat fluxes, by the change of flow conditions along the test section due to evaporation at higher heat fluxes and by the possible occurrence of nucleate boiling.

At low heat fluxes there is a thermal entrance region over which the heat transfer coefficient decreases by 30–50%. At high inlet quality the entrance region may extend to the first or second thermocouple station (30–50 mm heated length) (Fig. 4(a)). At lower qualities it may extend beyond the fourth station (> 160 mm) (Fig. 4(b)). This is a surprising distance for annular flow with a turbulent liquid film about 0.2 mm thick: by analogy with single-phase turbulent flow the expected length would be about 25 times the film thickness, say 5 mm. It was argued in ref. [11] that these long thermal development lengths are consistent with damping of turbulence at the liquid–vapour interface but there may well be other explanations. Thermal entrance effects are not confined to annular flow. They are observed in the churn flow regime, which additionally exhibits a 'thermal exit' effect (Fig. 4(c)): the heat transfer coefficient rises sharply over the last 60 mm of the test section, possibly because the unsteady flow regime promotes axial mixing with liquid from the unheated region above the test section.

At high heat fluxes, whatever the flow regime, there is no detectable thermal entrance region. The heat transfer coefficient is approximately constant over the first part of the test section, e.g. Fig. 4(b) at 200 and 300 kW m^{-2}, Fig. 4(c) at 100 kW m^{-2}. This is interpreted as nucleate boiling, discussed in Section 4.

The wall superheats in the test section must exceed 4 K if they are to be measured within 5% uncertainty. The requisite heat fluxes generally cause a significant increase in quality along the test section so that it is not possible to maintain the equilibrium two-phase flow developed in the preceding adiabatic section. Since there is continuous hydrodynamic development as well as thermal development, there can be no fully-developed heat transfer coefficient in the sense used in single-phase flow. Nevertheless it is found that the plots of α vs x at different heat fluxes (including those starting in nucleate boiling) merge into a region in which $\alpha(x)$ is nearly independent of heat flux. This is

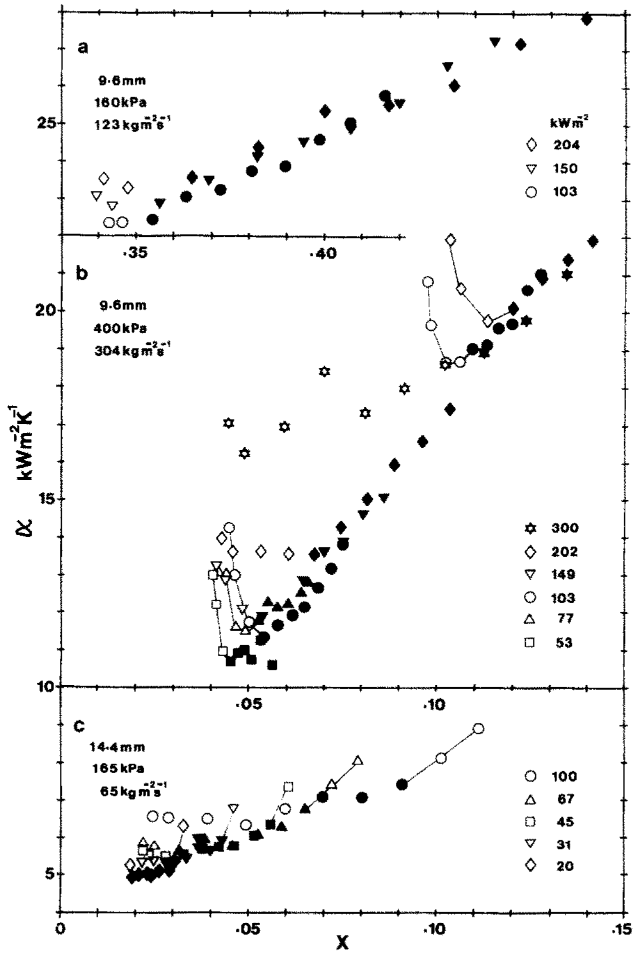

FIG. 4. Typical experimental data for water.

taken to be the fully-developed two-phase convective region, defined by the arbitrary criterion that there should be less than a 10% change in $\alpha(x)$ for a factor 2 change in heat flux, e.g. the closed symbols on Fig. 4. Data from runs with different initial values of x merge into the same convective region, e.g. Fig. 4(b), top right-hand corner.

The fully-developed convective boiling data points, as defined above, obtained in experiments on the 9.6 mm bore tube are presented on a plot of $(F-1)$ vs $1/\chi_{tt}$ in Fig. 5. This form of correlation successfully collapses onto one line, within $\pm 10\%$, data for water at three nominal pressures of 160, 390 and 600 kPa, flow rates of 123, 203, 304, 450 and 630 kg m^{-2} s^{-1} and thermodynamic quality ranging from 0.02 to 0.65. The lowest heat flux is 30 kW m^{-2}; the highest heat flux giving some data in the apparently-convective regime is 400 kW m^{-2}. The correlation is approximated by

$$F = 1+1.8(1/\chi_{tt})^{0.87}, \quad 0.6 \leqslant 1/\chi_{tt} \leqslant 40 \qquad (12)$$

which coincides with Chen's original proposal at $1/\chi_{tt} = 1$ but is 28% higher at $1/\chi_{tt} = 40$. The data are in quite good agreement with Shah's correlation over the range $2 \leqslant 1/\chi_{tt} \leqslant 20$ but there is no evidence of the small systematic variation with pressure proposed by Shah. Gungor and Winterton's correlation for α_c at zero heat flux lies 30% below the data and the experimental evidence does not support their proposal for a significant effect of Boiling number on F at small $1/\chi_{tt}$, equation (10). For the data in Fig. 4(b) at $x = 0.063$, $1/\chi_{tt} = 1.4$, the experimental values of α_c vary by only 5% as the heat flux increases from 77 to 149 kW m^{-2}: GW predict an increase of 22%.

The experiments with the 14.4 mm bore tube were performed at the same three pressures, at flow rates of 55, 65, 90, 135, 175, 200 and 334 kg m^{-2} s^{-1} and heat fluxes up to 100 kW m^{-2}. Some of the convective data sets lie on the same correlation line as the data from the 9.6 mm tube; other data sets lie on a family of parallel lines deviating from the original line at small $1/\chi_{tt}$ (Fig. 6(a)). The lowest liquid Reynolds number at any deviation point is 3100, indicating that laminar–turbulent transition is not the cause. The deviations appear to coincide with the churn–annular flow transition defined by the criterion [12]

$$J_g^* = 1 \qquad (13)$$

$$J_g^* = \frac{Gx}{[gD\rho_g(\rho_l-\rho_g)]^{0.5}}. \qquad (14)$$

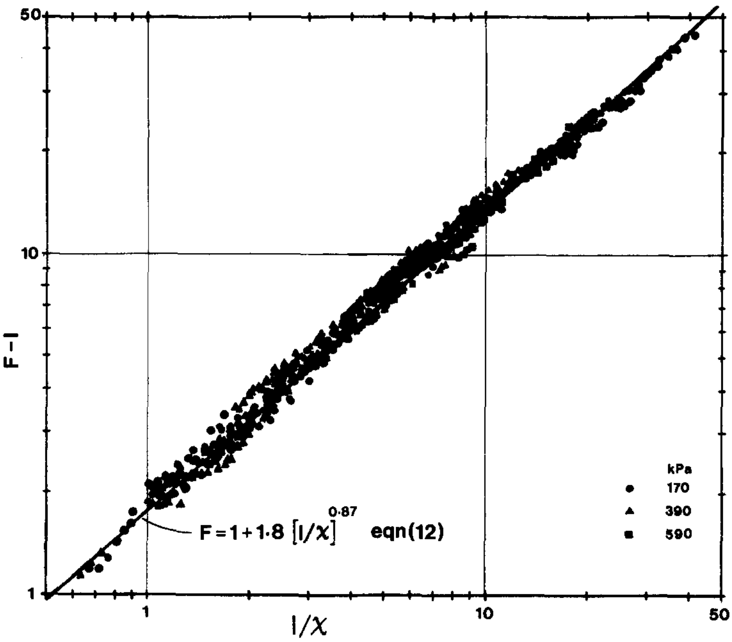

FIG. 5. Correlation of convective heat transfer data for water in 9.6 mm bore tube.

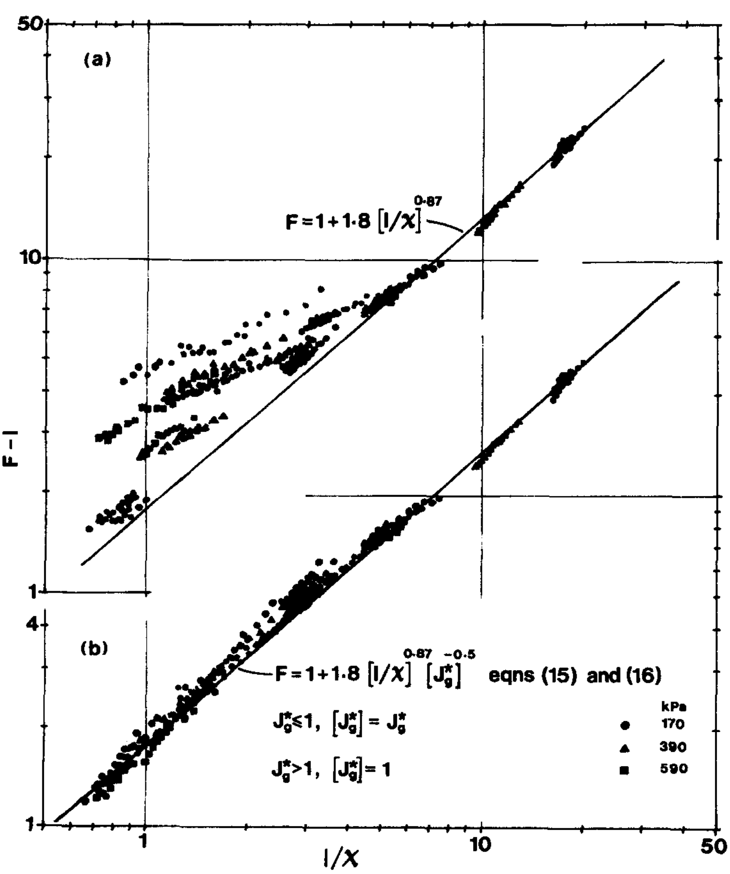

FIG. 6. Correlation of convective heat transfer data for water in 14.4 mm bore tube.

Similar effects have been observed in flow boiling of cryogenic and organic fluids (J. M. Robertson, unpublished work at Harwell Laboratory). A simple modification brings the data for plug/churn flow in the 14.4 mm tube into line with the general correlation (Fig. 6(b))

$$F = 1+1.8(1/\chi_{tt})^{0.87}[J_g^*]^{-0.5} \quad (15)$$

$$[J_g^*] = J_g^* \quad \text{if} \quad J_g^* \leqslant 1$$
$$= 1 \quad \text{if} \quad J_g^* > 1. \quad (16)$$

However, this modification is not required for data from the 9.6 mm tube down to $J_g^* = 0.4$ so equations (15) and (16) do not properly represent the effect of tube diameter. Further investigation is required of convective heat transfer in the plug/churn regime, which coincides with small values of $1/\chi_{tt}$. Given the complications introduced by flow transitions and also the experimental difficulty of determining small values of $1/\chi_{tt}$ accurately, it is not surprising that existing correlations are in poor agreement for $1/\chi_{tt} < 1$. Furthermore, Sekoguchi *et al.* [13], in work based on measurements in uniformly heated tubes, suggest that the influence on α_c of non-equilibrium voidage generated in the subcooled boiling region will persist to a positive thermodynamic quality x_B which may be reformulated as

$$x_B = 0.2\frac{\dot{q}c_l}{\alpha_c h_{lg}} = 0.2\frac{Bo}{St_c} = 0.2\frac{\Delta T_c c_l}{h_{lg}}. \quad (17)$$

For $x < x_B$ they use an empirical expression for the actual quality, predicting an increase in α_c with increasing heat flux, as in the Gungor and Winterton correlation but for different reasons. However, in the experiments with water described in this paper $q/\alpha_c < 25$ K so $x_B < 0.01$, which is below the experimental range of conditions in the convective regime.

In the turbulent annular flow regime the simple form of correlation $F = F(1/\chi_{tt})$ in equation (12) provides a satisfactory correlation of 'fully-developed' convective heat transfer, including the weak effect of tube diameter. The satisfactory performance of a correlation containing several fluid properties in combination does not necessarily mean that the influence of any one property is being modelled correctly. Cooper employed the similarity in the variations of properties with reduced pressure P_R to produce a simple correlation for nucleate pool boiling [5, 14]. Applying a similar technique of least squares fitting to all our convective data in the annular flow regime produced an expression

$$F-1 = \frac{x^{0.874}}{(1-x)^{0.500}}\frac{G^{0.049}\dot{q}^{0.026}}{P_R^{0.412}}\cdot(\text{constant}) \quad (18)$$

so our data confirm directly that mass flux and heat flux have little effect, as implied by the Chen correlation.

In other respects, equation (18) looks very different from expressions such as equation (12), which is also from our data, but uses the Martinelli parameter. However, they are in fact quite similar, as is shown by expressing the Martinelli parameter (equation (5)) in terms of P_R, since $[\rho_l/\rho_g]^{0.5}[\mu_g/\mu_l]^{0.1}$ is closely proportional to $P_R^{0.44}$ for water in this range (and indeed for most fluids—it arises from ρ_g varying roughly in step with pressure). Thus equation (12) becomes

$$F-1 = \frac{x^{0.738}}{(1-x)^{0.738}P_R^{0.364}}\cdot(\text{constant}) \quad (19)$$

which differs from equation (18) principally by having powers of $(1-x)$ differing by 0.238. If data were available for larger x (larger $1/\chi_{tt}$), then it should be possible to decide what that power should be. More data should also determine whether an equation such as equation (18) (in which powers of x and $(1-x)$ are independent, and fluid properties are independently represented by a power of P_R) provides significant improvement over equations such as equation (12) (in which the Martinelli parameter is used, so powers of x and $(1-x)$ are linked equal and opposite, and are also linked to powers of fluid properties). Further accurate convective data are therefore required, ideally involving all of: larger values of $1/\chi_{tt}$, a wider range of pressure, and fluids other than water. Unpublished data (J. M. Robertson and V. V. Wadekar at Harwell Laboratory) indicate that equation (12) is inaccurate for liquids with large values of Prandtl number. The published data for non-aqueous fluids in the Gungor and Winterton data bank are for refrigerants with nucleate boiling occurring and for ethylene glycol [6]. The convective data for ethylene glycol should be disregarded, having been obtained in a rig with two-stage heating in which the upper test section was only 25 mm long, introducing large errors due to the thermal entrance effects described above. The convective data of Ross *et al.* [7] for Refrigerants 152a and 13B1, although suitable in other respects, were obtained in horizontal tubes.

An accuracy of $\pm 10\%$ in a correlation scheme based only on local conditions may well be acceptable for the design of systems with uniform or near-uniform heating, where a major source of uncertainty is the prediction of two-phase pressure gradient and hence of local saturation temperature. Further refinement requires examination of the small dependence of α_c on heat flux $\dot{q}$, as revealed by equation (18). The smallness of the effect does not guarantee that the mechanism of heat transfer is purely convective. The residual effect of $\dot{q}$ may be due to the hydrodynamic flow development accompanying evaporation (which can be examined using an annular flow model such as HANA [15]), or due to some local bubble-related mechanism. Gungor and Winterton's [4] proposal has been shown to greatly overestimate the effect of $\dot{q}$ on α_c. Bennett and Chen's [6] suggestion that any level of bubble nucleation at the wall should cause the same enhancement of α_c might be expected to apply only above some threshold superheat dependent

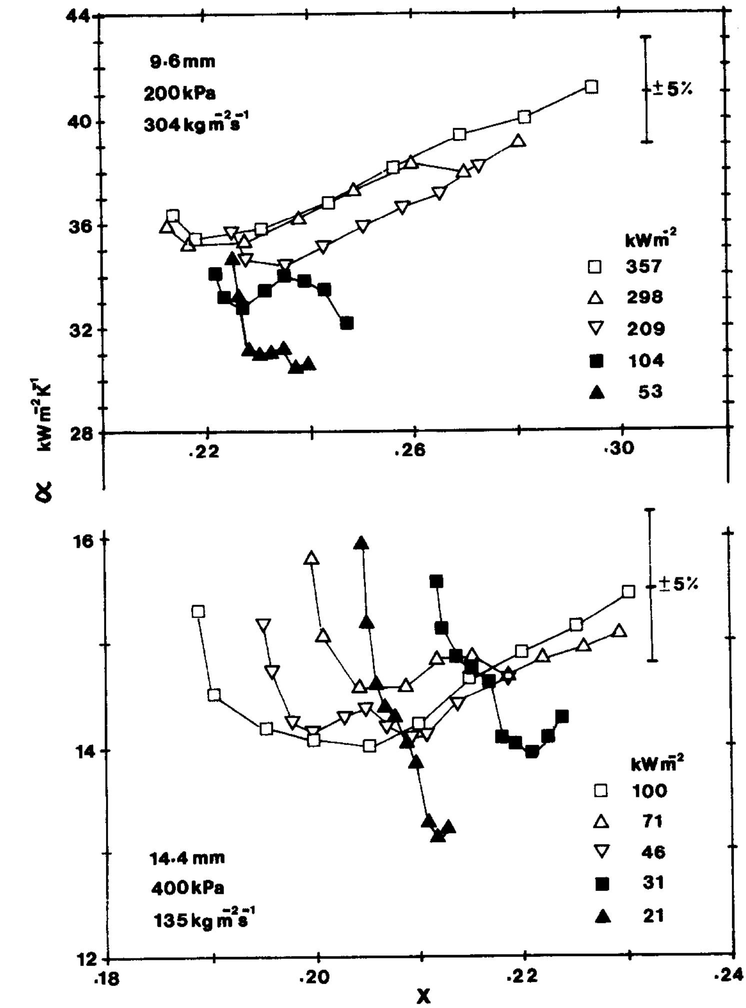

FIG. 7. Anomalous measurements of α_c at low ΔT_s.

on the wall nucleation characteristics: it has been noted in Section 2 that modification of the wall finish (on a scale leaving it still hydrodynamically smooth) had no detectable effect on α_c at superheats above 4 K. Carroll and Mesler [9] suggested mechanisms which would introduce bubbles into the liquid film in annular flow at very low wall superheats or indeed even during condensation. Measurements of α_c at very low superheats hit the problem of experimental accuracy discussed in Section 2. Some anomalously low measurements of α_c (not included in the correlated data) were recorded at wall superheats below 3 K (Fig. 7). The validity of the data and other aspects of the effect of $\dot{q}$ on α_c are being examined further (A. M. Ribeiro, unpublished work at Oxford University). In the next section discussion reverts to bubble nucleation at high wall superheats and the interaction with convection.

4. APPARENTLY NUCLEATE BOILING

As the heat flux is increased, keeping other conditions the same, the heat transfer coefficient becomes dependent on the heat flux. The thermal entrance region characteristic of heat transfer at low heat flux is replaced by random axial variations in α about an approximately constant value; the circumferential variations in wall superheat increase from the $\pm 2\%$ typical of the convective regime to $\pm 10\%$ at some measuring stations. In this 'apparently nucleate boiling' region α is approximately constant along the test section despite increase in quality x, until it meets the 'fully-developed convective' line established in tests at low q (Fig. 8). This constancy of α applies even if the quality increases sufficiently to cause changes in flow regime, e.g. run 13 at 300 kW m^{-2} in Fig. 8, in which α increases by 6% as x increases from 0 (bubbly flow) at station 2 through plug/churn flow to 0.076 (annular flow) at station 9 and α_c/α increases from 0.25 to 0.67. Within the limits of accuracy set by the circumferential and axial variations in α in nucleate boiling and the 10% uncertainty in α_c, there is no evidence to support Chen's assumptions that α_c and α_{nb} are additive with α_{nb} modified by a continuously varying, flow-dependent suppression factor. It appears that α is equal to the larger of α_c or α_{nb}, as in equation (7). The general

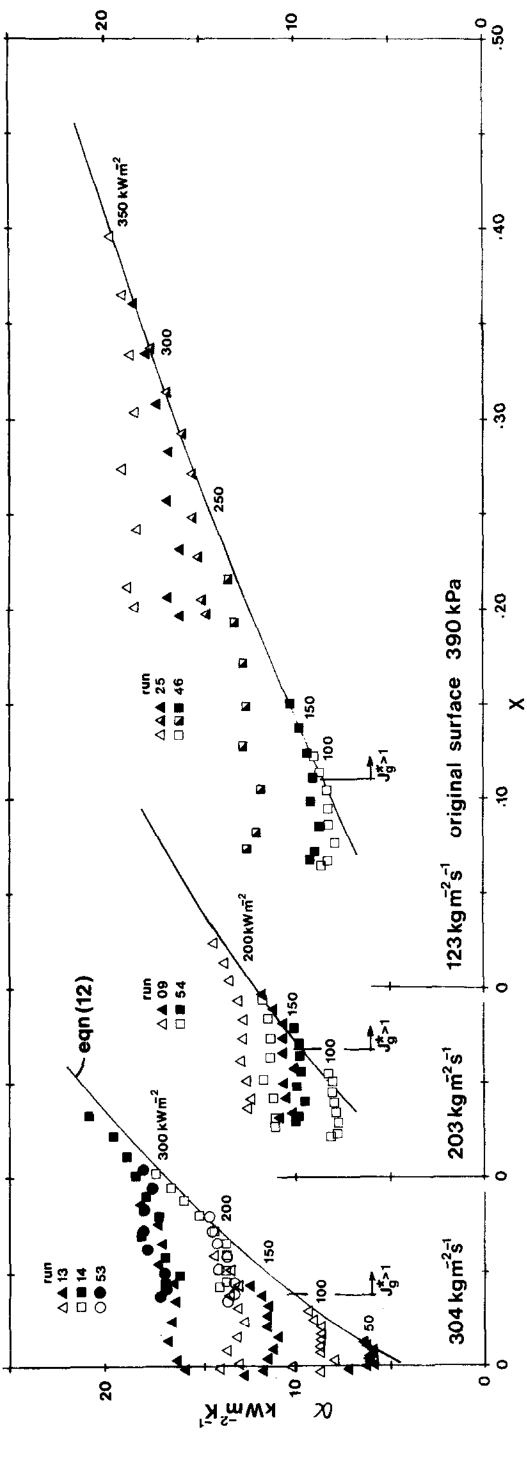

FIG. 8. Data in nucleate boiling regime.

application of this conclusion should be qualified by noting that it is derived from tests on very thin-walled tubing which might modify nucleate boiling: in turbulent convection the local variations in heat flux due to turbulent bursts occur at continually varying positions but in nucleate boiling from stable nucleation sites lateral temperature gradients are established which depress the temperature at the sites, reducing their activity [16]. (Compare this model with the assumption behind Chen's suppression factor that the temperature gradient normal to the wall in the liquid reduces bubble activity, despite the evidence of very effective nucleate boiling in large temperature gradients at high subcooling.) A very thin wall accentuates the temperature depression at active nucleation sites. Shah [17], discussing the application of his correlation to cryogenic fluids, noted three cases in which equation (7) underestimated α by a large margin: all were for thick-walled copper test sections which would minimize the consequences of lateral conduction.

The total heat transfer coefficient α at high heat flux, now identified with α_{nb} if $\alpha > \alpha_c$, has been shown to be independent of x during any particular run at fixed mass flux G. Determining whether α_{nb} is also independent of G depends on the reproducibility between data sets obtained on different days. In contrast to the excellent reproducibility in the convective regime, reproducibility in nucleate flow boiling was generally poor as shown in Fig. 9 which plots α_{nb} against wall superheat ΔT_s. Experimental rigs for pool boiling are fundamentally simpler and smaller, with boiling off the external surface of a heater, so conditions can be more readily controlled. In the most advanced pool rigs, all details have been meticulously refined over many years, with conditions chosen to minimize sensitivity to possible causes of scatter, even surrounding the whole rig in an isothermal box, but nevertheless scatter of order $\pm 10\%$ in ΔT_s seems inevitable, as discussed in ref. [14]. Since $\dot{q}$ is roughly proportional to ΔT_s^3, that corresponds to scatter about $\pm 30\%$ in $\dot{q}$ at given ΔT_s, or $\pm 20\%$ in α_{nb} at given ΔT_s. In Fig. 9 (original surface) the scatter is somewhat larger, being about $\pm 33\%$ in α at given ΔT_s. This increase is not surprising in view of the much greater difficulty of controlling flow boiling experiments. There is no consistent variation with G, or with the chronological order of the measurements (thus ruling out ageing of the surface as the cause of poor reproducibility). In these phenomena in which nucleation is important, data show great scatter, which may long remain unresolved, since they depend on unidentified (perhaps unmeasurable) details of surface conditions which affect nucleation. There may be a tendency for the data to lie towards the extremes of the band, as if there were two boiling curves, but that is not certain. Hysteresis can occur in boiling and lead to two boiling curves, but it is rarely reported with water, being more common with organic fluids which have low values of contact angle, and consequently few stable nucleation sites. The experiments were per-

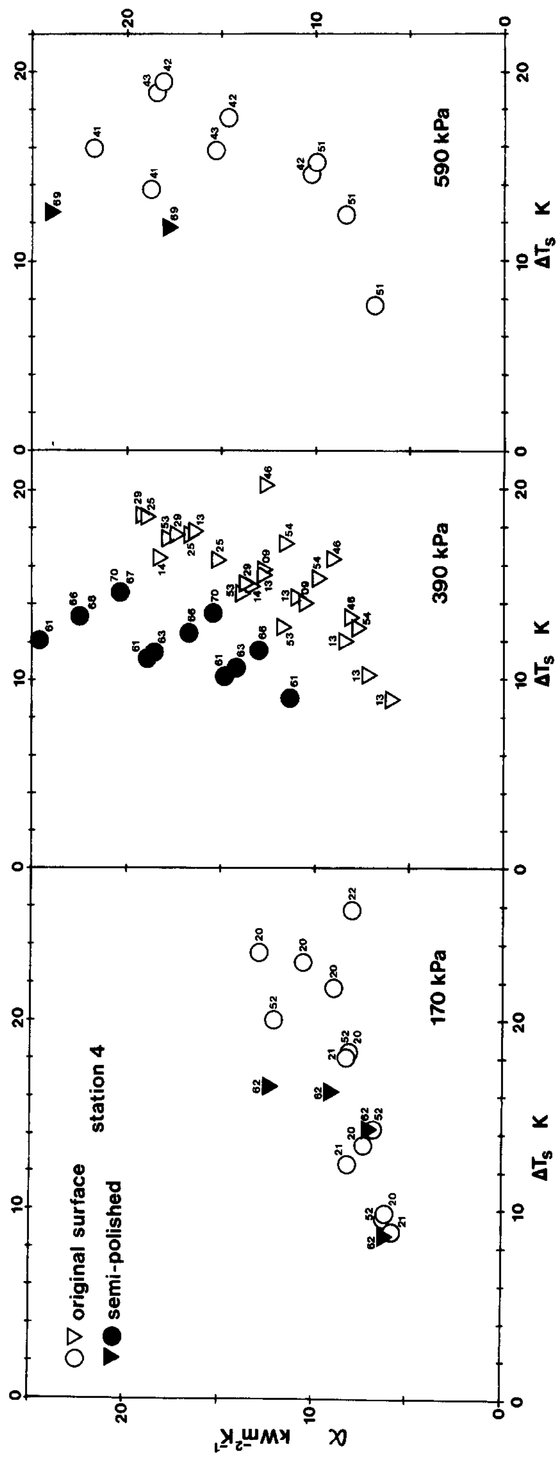

FIG. 9. Nucleate boiling curves, α vs ΔT_s.

formed with increasing steps of heat flux (being a likely mode of operation of boiling equipment) and the system pressure was adjusted to maintain approximately constant pressure at the downstream end of the test section. The adjustments to achieve steady state might have introduced some hysteresis but generally a set of runs on any particular day would follow one boiling curve: there is only one instance in which an increase in heat flux appears to have triggered a jump from one boiling curve to the other. Thus it is more likely that the variability in nucleate boiling depended on the thermal history and degassing (always to a low level) of the water before a set of runs. This could vary from a period of several days at the operating pressure but at ambient temperature, to degassing of a fresh charge of water and pressurization immediately before a set of runs. Gas content is another quantity that is much easier to control in pool boiling.

For a final series of experiments with the 9.6 mm tube its internal surface finish was modified by polishing with a bob rotating at 125 r.p.m. in a stream of water, moving in and out over the last 350 mm of the heated length (a) 100 times with a bob coated with 600 grade 'wet and dry' paper then (b) 120 times loaded with fine jewellers' rouge on felt. This treatment *reduced* the surface roughness from 0.38/0.25 to 0.15 μm CLA but produced significantly *improved* heat transfer in nucleate boiling (Fig. 9). (As noted earlier it had no effect on heat transfer in the convective regime.) This behaviour illustrates the inadequacy of conventional measurements of surface roughness in the prediction of nucleate boiling. Such measurements may be of limited value in charting the progressive stages in a particular form of surface treatment [18].

These experiments have established that the nucleate boiling heat transfer coefficient $\alpha_{nb}(\Delta T_s)$ is independent of local flow conditions. However, α_{nb} is sensitive to surface conditions, which cannot be characterized by measurements of surface roughness. Even on one particular surface α_{nb} may behave inconsistently, probably depending on the thermal history before boiling. Consequently the prospects for accurate correlation of the nucleate boiling regime are poor.

A related parallel study investigated all available data for nucleate flow boiling of water, including that obtained in this apparatus. As reported in ref. [19], the data are remarkably few and very scattered, but they can be seen to lie generally within the broad scatter band of nucleate pool boiling, thus providing general support for Chen's (and others) decision to base α_{nb} on a pool boiling correlation. However, the data do not even enable a choice to be made between, say, the pool boiling correlation used by Chen and more recent ones such as refs. [5, 14]—all must be treated with reserve, as observed values of α_{nb} may differ from any prediction by a factor of 2 or more. An interim recommendation for the correlation of α_{nb} is given in ref. [19].

Our work appears to be the first to study how flow boiling is affected by changing surface conditions inside a tube, but some earlier studies have considered nucleation conditions inside as-received tubes. Aounallah [20] found that samples of this tube material had nucleation site densities one-third of those measured on flat rolled stainless steel plates. Brown [21], measuring nucleation site densities by the gas-bubble technique, observed lower densities inside copper and stainless steel drawn tubes than on polished flat plates. Brown's flow boiling experiments, restricted to artificially roughened surfaces on the outside of tubes, showed no effect of surface finish. His limited study has sometimes been cited as evidence that nucleate flow boiling is generally insensitive to surface conditions. The present study has shown that this is not the case. Further investigation is required to establish whether the inside surfaces of drawn tubes have lower values of α_{nb} than other surfaces, although any effect may be swamped by in-service modification by corrosion and deposition in many industrial applications.

5. CONCLUSIONS

These conclusions are derived from experiments on the saturated flow boiling of water at 160–600 kPa in vertical 9.6 and 14.4 mm bore thin-walled tubes.

(1) The heat transfer coefficient α is equal to the larger of the convective coefficient α_c or the nucleate boiling coefficient α_{nb}. The coefficients are not additive (within experimental error). Further assessment of this conclusion is required for thick-walled tubes of high thermal conductivity.

(2) With extreme care, experimental measurements of α_c are reproducible to $\pm 2\%$. They are unaffected by variations in small-scale surface finish.

(3) In the annular flow regime the 'fully-developed' convective coefficient depends primarily on local parameters $\alpha_c(p, G, x)$ and can be correlated within $\pm 10\%$ by a modification of Chen's expression of $F = \alpha_c/\alpha_l$ as a function of the local Martinelli parameter

$$F = 1 + 1.8(1/\chi_{tt})^{0.87}, \quad 0.6 \leqslant 1/\chi_{tt} \leqslant 40. \quad (12)$$

(4) Because of the similarities in the dependence of saturated fluid properties on reduced pressure, the success of the correlation for data confined to one fluid does not prove that it correctly represents the effects of individual properties. There is evidence that equation (12) does not apply to some organic fluids but the Bennett and Chen correction for Prandtl number should not be used.

(5) In the plug/churn flow regime in the 14.4 mm bore tube α_c lies above the predictions of equation (12) and F is better correlated by

$$F = 1 + 1.8(1/\chi_{tt})^{0.87}[J_g^*]^{-0.5}, \quad J_g^* \leqslant 1. \quad (15)$$

However, this modification is not required for the small amount of data in churn flow in the 9.6 mm bore tube so equation (15) does not provide a generally valid prediction of the effects of tube size and of fluid properties. Further investigation is required of convective heat transfer in the plug/churn flow regime.

(6) The determination of fully-developed values of α_c is complicated by (a) long thermal development lengths, (b) a tendency for α_c to increase slightly with increasing heat flux, (c) occasional anomalies at very low heat flux which may be due to experimental errors at low wall superheats. These effects are the main causes of the uncertainty in the correlation of α_c. If α_c has to be predicted to better than $\pm 10\%$ in heat exchangers with axially varying heat flux, the linked effects of thermal and flow development will have to be taken into account, i.e. heat transfer cannot be correlated entirely on the basis of local conditions.

(7) The nucleate boiling coefficient $\alpha_{nb}(p, \Delta T_s)$ does not depend on flow conditions (flow rate, quality, regime) and thermal development effects are negligible.

(8) α_{nb} is sensitive to surface conditions, causing variations with circumferential and axial position and following surface modification by polishing. The 9.6 mm tube in as-received condition exhibited day-to-day variations, probably due to different thermal histories prior to boiling. These features introduce a large uncertainty into the prediction of α_{nb}, just as in pool boiling.

Acknowledgements—This work was supported by the Science and Engineering Research Council and by the U.K. Atomic Energy Authority Harwell Laboratory. Mr D. Treadwell made the apparatus and Miss M. M. Davidson performed the measurements.

REFERENCES

1. D. Butterworth and R. A. W. Shock, Flow boiling, *Proc. 7th Int. Heat Transfer Conf.*, Vol. 1, pp. 11–30 (1982).
2. J. C. Chen, Correlation for boiling heat transfer to saturated fluids in convective flow, *Ind. Engng Chem. Proc. Des. Dev.* **5**, 322–329 (1966).
3. M. M. Shah, Chart correlation for saturated boiling heat transfer: equations and further study, *ASHRAE Trans.* **88**, 185–196 (1982).
4. K. E. Gungor and R. H. S. Winterton, A general correlation for flow boiling in tubes and annuli, *Int. J. Heat Mass Transfer* **29**, 351–358 (1986).
5. M. G. Cooper, Saturation nucleate pool boiling. A simple correlation, *Proc. 1st U.K. National Conf. on Heat Transfer*, Vol. 2, pp. 785–793 (1984).
6. D. L. Bennett and J. C. Chen, Forced convective boiling in vertical tubes for saturated pure components and binary mixtures, *A.I.Ch.E. Jl* **26**, 454–461 (1980).
7. H. Ross, R. Radermacher, M. di Marzo and D. Didion, Horizontal flow boiling of pure and mixed refrigerants, *Int. J. Heat Mass Transfer* **30**, 979–992 (1987).
8. R. B. Mesler, An alternative to the Dengler and Adams convective concept of forced convective boiling heat transfer, *A.I.Ch.E. Jl* **23**, 448–453 (1973).
9. K. Carroll and R. Mesler, Bubble entrainment by drop-formed vortex rings, *A.I.Ch.E. Jl* **27**, 853–856 (1981).
10. Y. Aounallah, D. B. R. Kenning, P. B. Whalley and G. F. Hewitt, Boiling heat transfer in annular flow, *Proc. 7th Int. Heat Transfer Conf.*, Vol. 4, pp. 193–200 (1982).
11. D. B. R. Kenning and G. F. Hewitt, Boiling heat transfer in the annular flow regime, *Proc. 8th Int. Heat Transfer Conf.*, Vol. 5, pp. 2185–2190 (1986).
12. K. W. McQuillan and P. B. Whalley, Flow patterns in vertical two-phase flow, *Int. J. Multiphase Flow* **11**, 161–175 (1985).
13. K. Sekoguchi, O. Tanaka, T. Ueno, M. Yamashita and S. Esaki, Heat transfer characteristics of boiling flow in subcooled and low quality regions, *Proc. 7th Int. Heat Transfer Conf.*, Vol. 4, pp. 243–248 (1982).
14. M. G. Cooper, Heat flow rates in saturated nucleate pool boiling—a wide-ranging examination using reduced properties. In *Advances in Heat Transfer*, Vol. 16, pp. 157–239. Academic Press, New York (1984).
15. P. B. Whalley, P. Hutchinson and G. F. Hewitt, The calculation of critical heat flux in forced convection boiling, *Proc. 5th Int. Heat Transfer Conf.*, Vol. 4, pp. 290–294 (1974).
16. D. B. R. Kenning, Wall temperatures in nucleate boiling, Report OUEL 1540/84, Department of Engineering Science, Oxford University (1984).
17. M. M. Shah, Prediction of heat transfer during boiling of cryogenic fluids flowing tubes, *Cryogenics* **24**, 231–236 (1984).
18. K. Nishikawa, Y. Fujita, H. Ohta and S. Hidaka, Effect of the surface roughness on the nucleate boiling heat transfer over the wide range of pressure, *Proc. 7th Int. Heat Transfer Conf.*, Vol. 4, pp. 61–66 (1982).
19. M. G. Cooper, Flow boiling—the 'apparently nucleate' regime, *Int. J. Heat Mass Transfer* **32**, 459–464 (1989).
20. Y. Aounallah, Heat transfer in annular two-phase flow, D.Phil. Thesis, Department of Engineering Science, Oxford University (1982).
21. W. T. Brown, Study of flow surface boiling, Ph.D. Thesis, Department of Mechanical Engineering, Massachusetts Institute of Technology (1967).

ECOULEMENT AVEC EBULLITION D'EAU SATUREE DANS DES TUBES VERTICAUX

Résumé—Des données précises de transfert thermique ont été obtenues sur l'écoulement avec ébullition d'eau saturée à 160–600 kPa, dans des tubes verticaux dont les diamètres intérieurs sont 9,6 et 14,4 mm, dans des conditions dominées par la convection ou par l'ébullition nucléée. La convection dans le régime d'écoulement, avec anneau liquide est bien décrite par la corrélation modifiée de Chen; une autre modification est nécessaire pour l'écoulement poche/bouchon. L'écoulement avec ébullition nucléée est sensible aux conditions de surface alors que l'ébullition convective ne l'est pas. Les deux ébullitions convective et nucléée ne sont pas additives: on doit utiliser la plus grande des deux valeurs α_c et α_{nb}. On discute des propriétés respectives des formules pour l'écoulement avec ébullition.

GESÄTTIGTES STRÖMUNGSSIEDEN VON WASSER IN SENKRECHTEN ROHREN

Zusammenfassung—Für gesättigtes Strömungssieden von Wasser in senkrechten Rohren mit 9,6 und 14,4 mm Innendurchmesser wurden im Druckbereich 160 bis 600 kPa genaue Wärmeübergangsmessungen durchgeführt, wobei entweder konvektives oder Blasen-Siedeverhalten vorherrschten. Im Ringströmungsbereich wird das konvektive Verhalten durch eine modifizierte Chen-Korrelation gut beschrieben; für die Pfropfen-Schaum-Strömung wird eine weitere Modifikation benötigt. Im Gegensatz zum konvektiven Sieden erweist sich das Strömungs-Blasensieden bezüglich der Oberflächen-Eigenschaften als empfindlich. Gesättigtes konvektives und Blasen-Sieden sind nicht additiv: der größere der beiden Wärmeübergangskoeffizienten sollte verwendet werden. Für allgemeine Strömungssiedekorrelationen werden zweckmäßige Ansätze diskutiert.

НАСЫЩЕННЫЙ ПОТОК КИПЯЩЕЙ ВОДЫ ПРИ ТЕЧЕНИИИ В ВЕРТИКАЛЬНЫХ ТРУБАХ

Аннотация—Получены точные данные по теплопереносу в насыщенном потоке кипящей воды при давлении 160–600 кПа в вертикальных трубах с внутренним диаметром 9,6 и 14,4 мм в условиях с преобладающим влиянием конвекции либо пузыркового кипения. Конвекция в режиме кольцевого течения хорошо описывается с помощью модифицированного соотношения Чена; в случае дисперсно-стержневого течения необходимо дальнейшее его уточнение. Показано, что условия на поверхности оказывают влияние на пузырьковое кипение при течении и не влияют на кипение в потоке. Кипение в потоке и пузырьковое кипение не являются аддитивными процессами: необходимо использовать наибольший из коэффициентов α_c или α_{nb}. Обсуждаются характерные особенности соотношений, описывающих кипение при течении.