Pergamon

Applied Thermal Engineering Vol. 17, Nos. 8–10, pp. 705–715, 1997

Printed in Great Britain
1359-4311/97 $17.00 + 0.00

PII: S1359-4311(96)00071-3

CORRELATIONS FOR THE PREDICTION OF BOILING HEAT TRANSFER IN SMALL-DIAMETER CHANNELS

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Abstract—This paper describes aspects of the work relating to boiling in single, small-diameter tubes as part of a study of compact two-phase heat exchangers. In order to realise the energy-saving potential of compact heat exchangers for evaporating duties it is necessary to establish design procedures. A test facility was commissioned which was used to measure pressure drop and boiling heat transfer coefficients for R141b flowing through tubes 500 mm long with diameters of 1.39–3.69 mm. Established correlations predicted the heat transfer coefficients reasonably well for the largest tube but performed badly when applied to the smaller tubes. It would appear that simple nucleate pool boiling correlations, such as that of Cooper, best predict the data. While under some conditions increasing quality leads to an increasing heat transfer coefficient, it is suggested that intermittent dry-out occurs at very low quality in single narrow channels, thus reducing the average heat transfer coefficient below that expected from the pool boiling correlations. © European Communities 1997. Published by Elsevier Science Ltd.

Keywords—Compact heat exchangers, heat transfer, boiling, evaporation, process intensification.

NOTATION

Bo	boiling number ($q/h_{fg}G$)
Co	confinement Number [Equation (1)]
d_e	hydraulic diameter, m
F	enhancement factor [Equation (2)]
g	acceleration due to gravity, m/s^2
G	mass flux, kg/m^{2}s
k	liquid thermal conductivity, W/mK
M	molecular mass
N	number of data points
Nu	Nusselt number ($\alpha d_e/k$)
Nu$_F$	film Nusselt number ($\alpha\delta/k$)
p_r	reduced pressure
q	heat flux, W/m^2
Re	Reynolds number (Gd_e/μ)
S	suppression factor [Equation (2)]
We$_l$	Weber number ($G^2d_e/\rho_l\sigma$)
x	dryness fraction

Greek letters

α	heat transfer coefficient, W/m^{2}K
δ	film thickness, m
ρ	density, kg/m^3
σ	surface tension, N/m

Subscripts

npb	nucleate pool boiling
L,l	liquid
g	vapour
pred	predicted
exp	experimental

INTRODUCTION

Traditionally, evaporation or boiling processes have occurred in tubular heat exchangers, either within the tubes or on the outside of the tubes. Recently there has been a growing awareness of the benefits of process intensification—the reduction in plant size, for a given capacity, by an order of magnitude or more—and this has led to a requirement for smaller evaporators.

Utilisation of a physically small heat exchanger for a given duty is advantageous for several reasons: often (although not always) small physical size is associated with relatively low capital cost; installation costs are reduced if the size of a component is reduced; a lower fluid inventory can be beneficial on both cost and safety grounds; and the nature of many compact heat exchanger designs is such that close approach temperatures can be achieved. The feasibility of heat recovery schemes, associated with significant energy savings, is enhanced by the small size and low costs associated with compact heat exchangers. Close approach temperatures can also result in significant energy savings in well-integrated plant.

Recognising the importance of this area of research and development the Commission of the European Communities (CEC) has financed collaborative projects in the area of enhanced evaporation and compact heat exchangers under the JOULE I and JOULE II programmes relating to the rational use of energy. The work described in this paper was undertaken as part of a project entitled "Compact Two-Phase Heat Exchangers" [1], a collaborative project involving 18 partners from 5 EU Countries. The laboratories involved in the work package from which this paper is derived were Heriot-Watt University (UK), the University of Oxford (UK) and British Gas plc (UK). Other laboratories involved in the project were IKE (University of Stuttgart), National Technical University of Athens (Greece) and BEHR (Germany) investigating heat transfer and flow phenomena in narrow spaces; AEA Technology (UK), the University of Nottingham (UK) and a group of heat exchanger manufacturers and users studying the design implications for compact heat exchangers; and GRETh (France), together with three heat exchanger manufacturers investigating the performance of enhanced plate evaporators.

The aim of the study was to develop technical expertise to enable new types of compact heat exchangers to be used for evaporation duties and to provide design information required to produce advanced compact heat exchangers. A fundamental requirement of the heat exchanger designer is the ability to predict heat transfer coefficients under the conditions of interest with confidence. In this paper the selection of appropriate design correlations is discussed. It is shown that conventional in-tube flow boiling correlations may be used for tubes having a confinement number [2] of the order of 0.3 and above. Consideration of the flow regimes and heat transfer mechanisms in narrow channels has led to a framework being proposed based on a film flow model [3], which permits the evaluation of heat transfer coefficients in narrow channels in the absence of dry-out. However, at relatively low heat fluxes and qualities, intermittent dry-out of the tube wall can occur during boiling, thus reducing the effective average film coefficient. Until the onset of dry-out can be predicted reliably it is suggested that established nucleate boiling correlations are appropriate for the determination of heat transfer coefficients in small diameter tubes.

TEST FACILITY

The test facility at Heriot-Watt University comprised a rig incorporating two flow loops. One consisted of a loop for the measurement of local heat transfer and pressure drop in single, narrow channels. The other permitted flow visualisation together with the measurement of heat transfer and pressure drop data in geometries with large aspect ratios (76 mm wide by up to 4.5 mm deep) or multi-channel geometries. The rig was operated with water, R141b and Flutec PP1. The results reported here were obtained from the single-tube test section using R141b, which has a boiling point of 32°C at atmospheric pressure.

The rig is shown schematically in Fig. 1. The data logging equipment, fluid reservoir, circulating pump and condenser were common, however, flow meters and instrumentation were specific to one of two loops of the rig. These loops are referred to as the single-tube test loop and the flow visualisation test loop. Liquid was pumped from the reservoir through the appropriate test loop via manual flow control valves and variable orifice flow meters connected in parallel. In each loop an electric liquid preheater was incorporated in the pipe between the flow meters and the test section.

The single-tube test section is illustrated schematically in Fig. 2. All test sections were thin-walled stainless steel and were heated by passage of a DC electric current along the tube. The test section

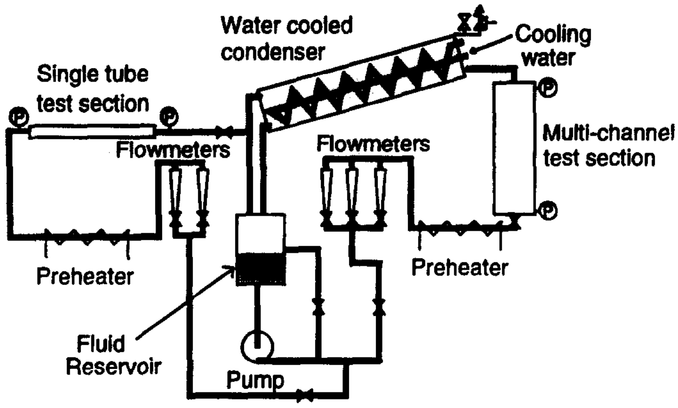

Fig. 1. Schematic diagram of test rig.

was enclosed in a 22 mm diameter copper tube which was heated with three independently controlled guard heaters. The environment immediately surrounding the test section was maintained within 2 K of the average temperature measured by the surface thermocouples. The outer wall temperature along the single-tube test section was measured using 10 equally spaced thermocouples.

Calibrated, Type K thermocouples with an ice reference junction were used for temperature measurement. The thermocouples used on the single-tube test section were made from 0.2 mm diameter wires with spot-welded junctions, while other thermocouples were 1.5 mm diameter and stainless-steel sheathed. Pressures at the inlet and outlet of the test sections were measured using calibrated WIKA type 891.14.525 pressure transducers giving an output signal of 4–20 mA over a 0–2.5 bar range. Flow rates were measured using variable orifice-type flow meters. As supplied these were calibrated for water at 20°C. These were individually calibrated for the working fluids R141b and Flutec PP1. Flow meters having ranges permitting measurement of flows from 10–350 ml/min of water were used on the single-tube circuit and 50–2000 ml/min on the flow visualisation circuit.

Two data acquisition systems were used. A Hewlett Packard 3421A was used for monitoring and recording steady-state and time-averaged values and a Keithley DAS8/EXP16 was used for recording transient values. The Hewlett Packard was a stable instrument with a low susceptibility to noise but a relatively low logging speed. The Keithley was more susceptible to electrical noise when measuring the low voltages associated with thermocouples but was capable of much higher data acquisition rates. The Keithley data acquisition system was connected to a PC running 'EASYLX' data acquisition and analysis software. This arrangement permitted the recording and analysis of rapidly varying signals.

The multi-channel test section incorporated a window which permitted visualisation of the flow within the channels either directly or by video or photographic means. Further details of this test section and the heat transfer results obtained from it are included in ref. [1].

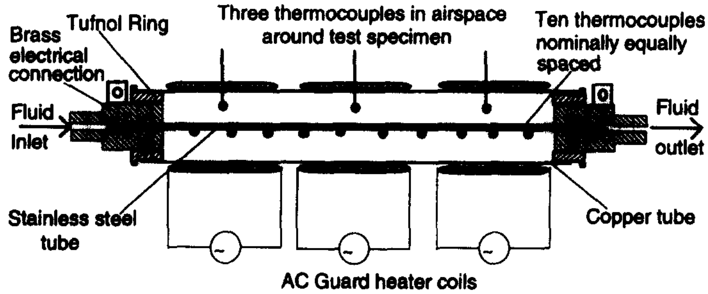

Fig. 2. Schematic diagram of single-tube test section.

Table 1. Correlations used for comparison with measured data

Reference	Reference No.	Equation No.	Correlation
Liu and Winterton (1988)	[6]	2	$\alpha^2 = (S\alpha_{npb})^2 + (F\alpha_L)^2$
Cooper (1984)	[18]	3	$\alpha = 55p_r^{0.12}(-\log_{10}p_r)^{-0.55}M^{-0.5}q^{0.67}$
Cooper (1989)	[19]	4	$\alpha = 35p_r^{0.12}(-\log_{10}p_r)^{-0.55}M^{-0.5}q^{0.67}$
Lazarek and Black (1982)	[12]	5	$\mathrm{Nu} = 30\mathrm{Re}^{0.857}Bo^{0.714}$
Modified Lazarek and Black (1982)		6	$\mathrm{Nu} = 30\mathrm{Re}^{0.857}Bo^{0.714}(1-x)^{-0.143}$
Tran, Wambsganns and France (1995)	[13]	7	$\alpha = 840(Bo^2\mathrm{We}_l)^{0.3}(\rho_l/\rho_g)^{-0.4}$ kW/m^{2}K

BACKGROUND

Widely used flow boiling correlations, for example, those of Chen [4], Shah [5] and Liu and Winterton [6], have been derived using databanks containing few data relating to small diameter passages in which the effects of confinement are significant. Examination of studies of capillary flow [7] and flooding in vertical up-flow [8] and of heat transfer in confined spaces under a variety of conditions [9–11] suggests that the effects of confinement will be significant for channels having hydraulic diameters such that the confinement number, introduced by the present authors [2] and defined by equation (1), is in excess of 0.5.

$$Co = \frac{[\sigma/(g(\rho_l - \rho_g))]^{1/2}}{d_e}. \tag{1}$$

There are few studies concerned with boiling heat transfer in channels of hydraulic diameter of the order of 3 mm and below. Lazarek and Black [12] worked with R113 in a 3.15 mm diameter tube. Wambsganss and co-workers have studied boiling in tubes and rectangular sections having hydraulic diameters of 2.4 to 2.92 mm with various fluids, this work is summarised in ref. [13]. The geometries of Lazarek and Black and Wambsganss and co-workers correspond to confinement numbers of approximately 0.35. It was concluded that, except at very low heat flux, the boiling mechanism was predominantly nucleate, since it was observed that the heat transfer coefficient showed a strong dependence on heat flux, only a weak dependence on mass flux and was not a function of quality. Studies at greater confinement [14, 15], examining heat transfer in very much smaller gaps with confinement numbers of the order of 10 and above, concluded that evaporative heat transfer was predominantly by conduction through a thin layer of liquid on the heated surface and evaporation from that layer.

Film models, involving determination of the annular liquid film thickness and the heat transfer through the film have not been widely used in the determination of boiling heat transfer coefficients in large tubes. Since such an approach involves the determination of film thickness and the film Nusselt Number using empirical methods, it is preferable to use an empirical correlation to determine the heat transfer coefficient directly. However, based on tests and analysis of a typical plate-fin section at low heat and mass fluxes, Robertson [16] demonstrated that the film flow model could be applied in compact heat exchangers. It was shown that the film Nusselt Number, Nu_F, as defined below, was a function of a dimensionless mass flow rate and the relationship was close to that given by Hewitt and Hall–Taylor [17].

$$Nu_F = \frac{\alpha\delta}{k}.$$

The Nusselt Number, Nu_F, is essentially the ratio of the actual rate of heat transfer through the film, thickness δ, to that which would occur due to conduction through a film of the same thickness. For laminar films, Nu_F is unity.

In the study reported in this paper experimental results from a range of single tubes are compared with predicted values of heat transfer coefficient from a number of published correlations. The correlations applied, together with the appropriate equation numbers, are listed in Table 1.

The Liu–Winterton correlation, the form of which is given in equation (2), is widely used in the prediction of flow boiling heat transfer coefficients and is appropriate for use in conventionally sized tubes. It is based upon the assumption that there are two active mechanisms of heat transfer—nucleate boiling and convection. The nucleate boiling contribution is calculated using the correlation due to Cooper [18] with a suppression factor, S, applied. The convective component is determined using the Dittus Boelter equation together with an enhancement factor, F.

Based on the assertion of Lazarek and Black [12] and Wambsganss and co-workers [13] that the boiling in small diameter tubes would be predominantly nucleate, the results were tested against the Cooper [18] correlation, equation (3), which was derived for nucleate pool boiling and the correlation with a reduced constant, equation (4), which Cooper [19] tentatively proposed for predicting the boiling heat transfer coefficient during flow boiling at low quality.

Lazarek and Black [12] proposed a correlation based upon boiling of a single fluid in a 3.19 mm tube, this is presented as equation (5). A modified form has been suggested by the present authors to allow for the observed increase in heat transfer coefficient with quality in the larger tubes tested during the present study. The modified Lazarek and Black equation is given here as equation (6).

Wambsganss and co-workers [13] proposed a correlation, presented here as equation (7), in which the dependency on the mass flux was eliminated by inclusion of the Weber number. It is notable that the dependence of the heat transfer coefficient on the Weber number to the power 0.3 implies

$$\alpha \propto d^{0.3},$$

which suggests that the heat transfer coefficient decreases with confinement. However, the correlation was based upon tests spanning a small range of equivalent diameters.

FLOW REGIMES

Experimental observations have shown that, for boiling flow in small channels, the flow regimes differ slightly from those observed in large channels and described in, for example, ref. [20]. Three flow regimes are sufficient to describe the patterns observed. These have been defined as follows.

Isolated bubble flow

Similar to bubble flow in large channels, bubbles detach from the nucleation sites and flow as discrete units in the liquid, as the flow proceeds further heat addition results in an increase in the number and size of the bubbles.

Confined bubble flow

Bubbles span the gap (in spaces confined in one dimension) or fill the channel (in spaces confined in two dimensions), they are separated from the wall by a layer of liquid which evaporates and causes the bubble to grow exponentially. The bubbles may be formed by isolated bubbles growing or coalescing, alternatively single bubbles may reach a sufficient size to be regarded as confined before becoming detached from their nucleation sites. This latter phenomenon has been observed in relatively large tubes having polished surfaces and has been referred to as cavitation slug flow [21]. A similar phenomenon also occurs in the boiling of liquid metals [22]. The rapid growth of confined bubbles can lead to significant fluctuations in the pressure at entry to the channel and apparent instability in multi-channel arrangements [23].

Annular-slug flow

As the confined bubbles expand, liquid in the slugs between them is deposited on the channel wall and the flow becomes basically annular with random, irregular slugs of liquid interspersed with the vapour.

Determination values of the parameters governing the boundaries between the regimes was not possible using the multi-channel flow visualisation test section since the flow was, as noted above, frequently unstable with constant variations in the local conditions. It was also observed that, in both the confined bubble and annular slug flow regimes, temporary local dry-out of the wall occurred under some conditions. The three flow regimes, together with a representation of the condition of partial dry-out, are illustrated in Fig. 3.

RESULTS

The results presented here relate to tests carried out with R141b working fluid. Less extensive tests using water and Flutec PP1 yield results which indicate the same general characteristics.

Sample results showing the measured variation in heat transfer coefficient with quality are given in Fig. 4. It can be seen that the results in the 3.69 and 2.87 mm channels [Fig. 4(a) and (b)] follow similar trends to those observed in conventionally sized channels. The heat transfer coefficient increases with heat flux at low quality, while at higher qualities the heat transfer coefficient is a function of quality and is essentially independent of heat flux. These trends are less apparent in the smaller tubes. Results obtained at two values of mass flux for the 1.39 mm tube are shown in Fig. 4(b) and (c). It can be seen that at high mass flux the heat transfer coefficient falls rapidly with increasing quality.

The percentage deviations between the experimental values and the heat transfer coefficients predicted by the six correlations listed in Table 1 are given in Table 2. It can be seen that all correlations, with the exception of equation (4), perform reasonably well for the 3.69 mm diameter tube. The deviation for all correlations increases with increasing confinement. The Liu and Winterton correlation performs very poorly when applied to the 2.05 and 1.39 mm diameter tubes. Comparisons of the predicted and measured heat transfer coefficients are presented for selected cases as Fig. 5(a)–(f). Clearly the correlations which do take no account of mass flux or quality result in a single value for each heat flux, while the experimental results span a considerable range at each heat flux. Since a variation of heat transfer coefficient with quality has been observed, neither equation (3) nor equation (7) can represent the physical reality under all conditions. The modified Lazarek and Black correlation [equation (6)] does take into account an increase in heat transfer coefficient with quality. Examination of Fig. 5(a) and (e) suggests that while there is considerable scatter of the data around the predicted value for equation (3), equation (7) tends to underpredict the data. The value predicted by equation (7) appears to form a lower bound to the measured value. This may imply that the correlation predicts the nucleate boiling component of the heat transfer but does not take into account an additional convective component.

Considerable scatter is apparent in the measured data when compared with all of the correlations applied to the 1.39 mm diameter tube. It should be noted that since overheating of the tube occurred at relatively low heat fluxes and qualities, all tests with this tube were at low quality and low to moderate heat flux. These conditions are not generally conducive to obtaining high heat transfer coefficients. The tendency to overheat combined with the observation that the heat transfer coefficient fell rapidly with quality under some conditions, as illustrated in Fig. 4(c), suggests that local dry-out occurred. This was due to the expulsion of the liquid in the tube by a rapidly expanding bubble which ultimately filled the tube, dry-out then occurred and there was a rapid increase in tube wall temperature until the liquid was replenished.

The relatively good performance of simple nucleate boiling correlations in the prediction of heat transfer coefficients in single narrow tubes may be explained by consideration of the mechanisms involved in pool boiling and in boiling in narrow tubes. Flow visualisation has shown that in narrow channels discrete bubbles form which then grow to fill the width of the channel. These

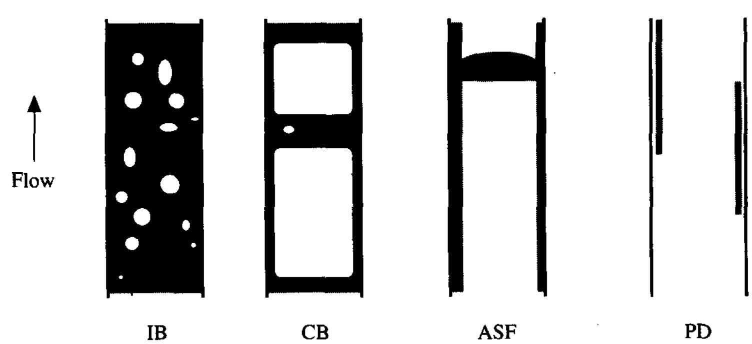

Fig. 3. Schematic diagram showing flow regimes: □, vapour; ■, liquid; IB, isolated bubble; CB, confined bubble; ASF, annular slug flow; PD, partial dry-out.

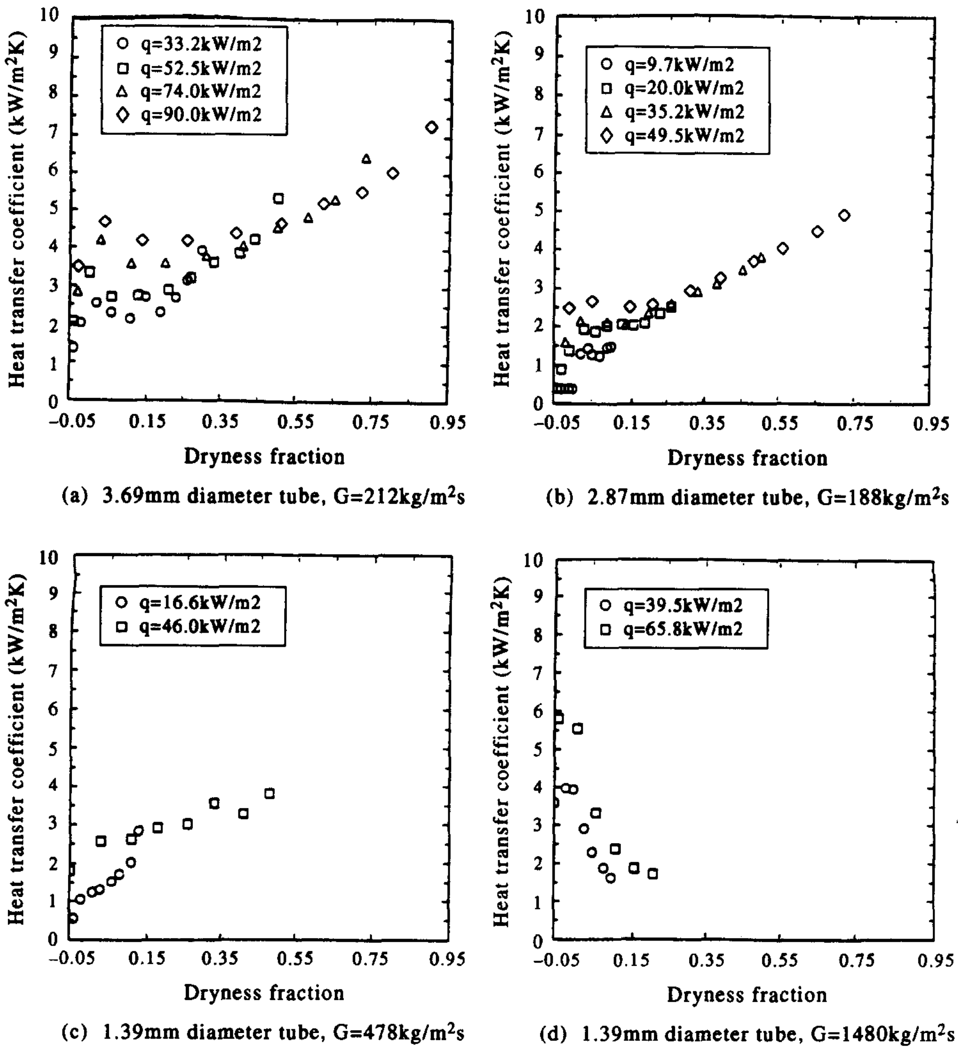

Fig. 4. Variation of heat transfer coefficient with dryness fraction. (a) 3.69 mm diameter tube, $G = 212$ kg/m²s. (b) 2.87 mm diameter tube, $G = 188$ kg/m²s. (c) 1.39 mm diameter tube, $G = 478$ kg/m²s. (d) 1.39 mm diameter tube, $G = 1480$ kg/m²s.

confined bubbles then grow and move along the tube. The work of Aligoodarz and Kenning [24] involved measurement of the variation in wall superheat during the passage of a sliding bubble through a narrow channel. They observed that the temperature variations were similar to those due to microlayer evaporation under pool boiling bubbles.

It is thus concluded that if the liquid film between the wall does not break down, a substantial portion of the heat transferred is by conduction through this film and evaporation from the surface of the film. The proportion of the tube length which is occupied by bubbles at any one time is

Table 2. Comparison of measured single-tube heat transfer data with selected correlations

Tube description	*Co*	Number of points	Percentage Deviation Equation (2)	Equation (3)	Equation (4)	Equation (5)	Equation (6)	Equation (7)
3.69 mm ϕ	0.33	146	21	25	36	22	23	24.9
2.87 mm ϕ	0.42	141	42	21	46	19	19	32.7
2.05 mm ϕ	0.59	182	118	30	43	47	48	37.3
1.39 mm ϕ	0.87	79	250	36	69	69	71	39.3
2.10 mm □	0.57	149	72	33	54	39	39	
2.92 mm ϕ*	0.34		15					

ϕCircular section tube.
□ Square section channel.
* Data of ref. [13]

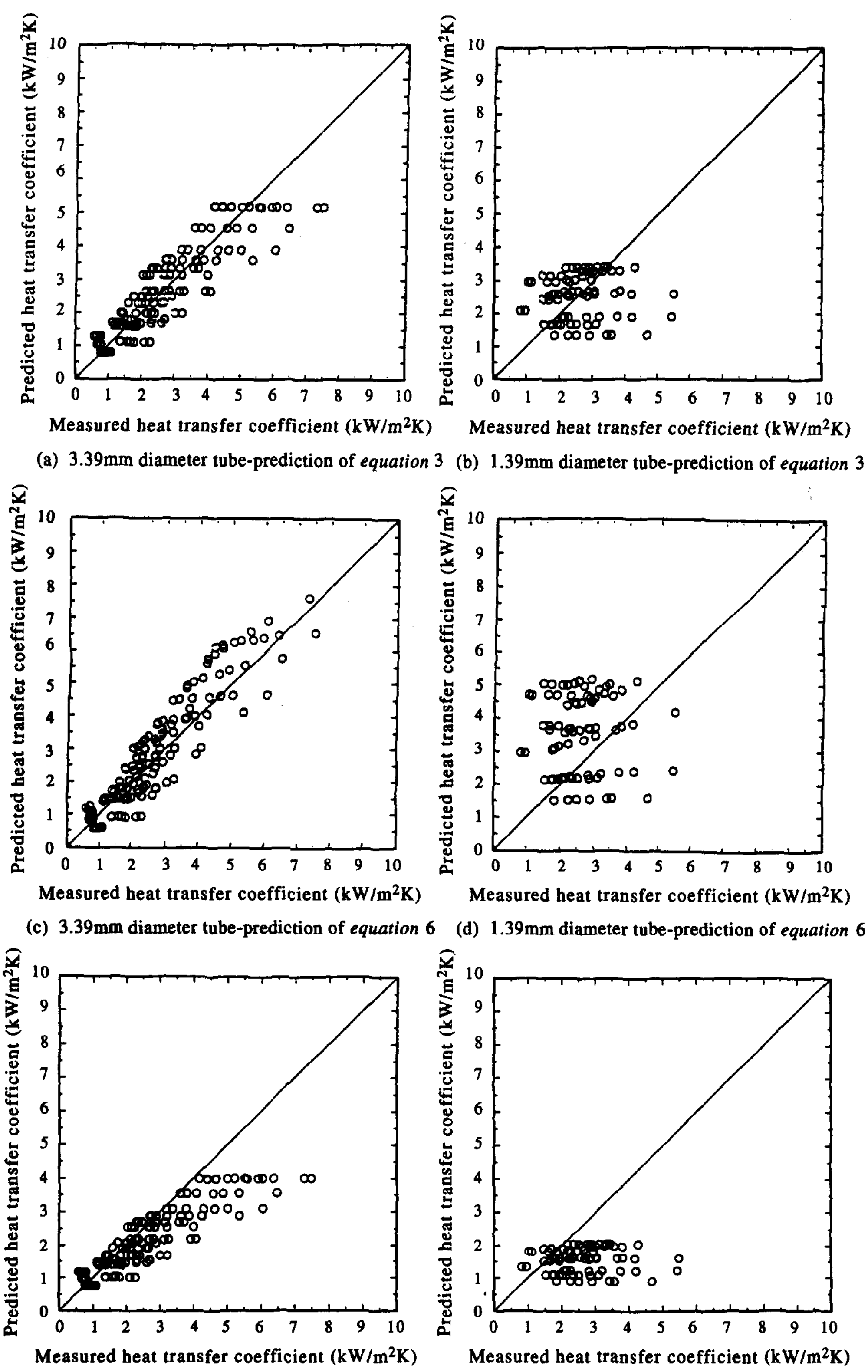

Fig. 5. Comparison of measured data with predictions. (a) 3.69 mm diameter tube—prediction of equation (3). (b) 1.39 mm diameter tube—prediction of equation (3). (c) 3.69 mm diameter tube—prediction of equation (6). (d) 1.39 mm diameter tube—prediction of equation (6). (e) 3.69 mm diameter tube—prediction of equation (7). (f) 1.39 mm diameter tube—prediction of equation (7).

a function of the heat flux, which determines the propensity of bubbles to nucleate on the wall and the net rate of vapour generation. The processes are thus similar to those occurring during nucleate pool boiling, except that the bubbles are constrained during their expansion. It is therefore logical that nucleate pool boiling correlations give reasonable results when applied to narrow tubes.

A NUCLEATE BOILING AND FILM FLOW MODEL

A model for the prediction of heat transfer coefficients to a fluid evaporating in a single channel has been formulated. The model is based upon the observation that heat transfer to a fluid evaporating in a narrow channel may be through one of four mechanisms:

1. Nucleate boiling.
2. Confined bubble boiling.
3. Convective boiling.
4. Partial dry-out

Note that these mechanisms do not correspond exactly with the flow regimes described in Section 4. In the isolated bubble regime it is reasonable to assume that heat transfer occurs entirely through nucleate boiling, this is similar to the situation suggested by Cooper [18] for larger tubes. At any point in the channel occupied by the confined bubble regime then either nucleate boiling or confined bubble evaporation dominates. In the annular slug flow regime nucleate boiling or convective boiling determines the heat transfer. The contribution of single-phase convection to liquid slugs is assumed to be small throughout.

In testing the proposed model the heat transfer coefficient attributable to mechanism 1 was obtained from an appropriate nucleate boiling correlation. Mechanism 2 was considered to be active in the confined bubble regime and heat transfer is by conduction through the layer deposited on the wall by the passing bubble and evaporation of the film. The film thickness was determined using the data of ref. [25] as reported in ref. [7]. The Nusselt number was assumed to be unity and the proportion of the surface covered by the liquid layer was taken to be equal to the void fraction calculated for homogeneous flow.

Mechanism 3 was considered when the flow was in the annular slug flow regime with the liquid phase deposited on the wall of the channel or passing through as liquid slugs; heat transfer was assumed to be by conduction and convection through and evaporation of the liquid film. Calculation of the heat transfer by mechanism 4 was similar to that for mechanisms 2 or 3, except that at any given time a portion of the wall is dry and therefore heat transfer from that portion is negligible. The film thickness in the annular slug flow regime was estimated using the CISE correlation for void fraction [26], neglecting the volume of liquid in the slugs. The film Nusselt number was then determined from the data presented in ref. [17].

In testing the model it was assumed that there was a minimum film thickness which could exist on the wall. In the annular slug flow regime as the film thickness reduced there was a tendency for the film to break up and dry areas form, this led to the occurrence of partial dry-out. Partial dry-out also occurred if the film evaporated fully before being replenished, this has been observed in the confined bubble regime.

The criterion for transition from confined bubble to annular slug flow was set as the dryness fraction at which the film thickness calculated for confined bubble flow, which increased with dryness fraction, was equal to the film thickness calculated for annular slug flow, which decreased with increasing dryness fraction. Thus, the selected flow regime was that which yielded the thinner film of liquid on the wall.

The approach used in the annular slug flow regime would lead to a film thickness approaching zero at as the dryness fraction approached unity. This in turn would result in the heat transfer coefficient approaching infinity. In practice, during evaporation at high values of dryness fraction, the heat transfer coefficient must reach a peak and then sharply declines, due to partial dry-out of the walls. During the flow visualisation studies [1] intermittent local dry-out was noted at low dryness fraction and moderate heat flux in the confined bubble regime. This was due to complete evaporation of areas of the liquid film deposited beneath the confined bubbles. The film flow model proposed has not yet been modified to account for the phenomenon of partial dry-out in either

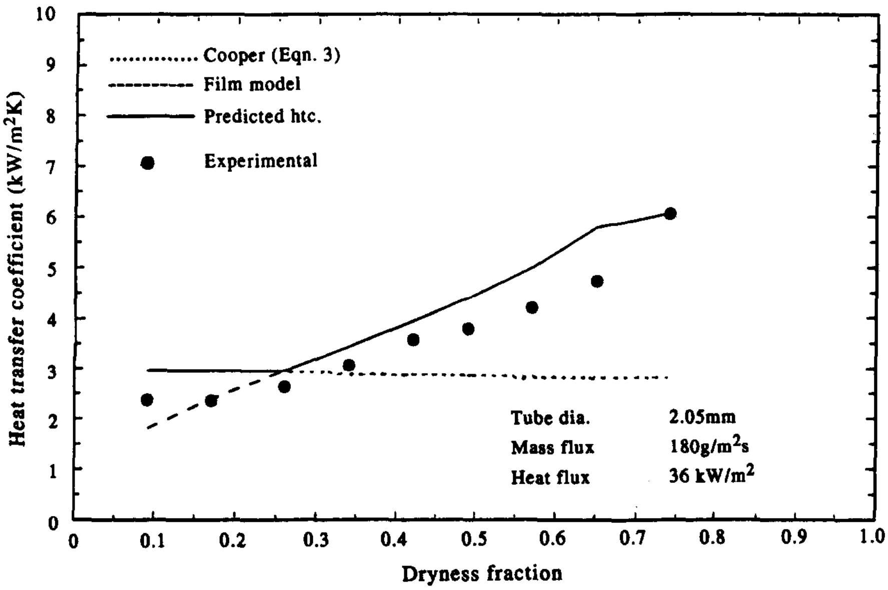

Fig. 6. Application of the proposed model to sample set of data.

regime. A simple approach involving specification of a minimum film thickness, below which the film would be expected to break up, leaving dry areas through which there would be negligible heat transfer, has not yielded satisfactory results.

This model correctly explained the trends observed for many of the measured data sets, an example of which is shown as Fig. 6. However, in many cases, particularly in the smaller tubes, the model significantly overpredicted the heat transfer coefficient. Examination of the experimental results, examples of which are presented Fig. 5, shows that in the smaller diameter tubes which were tested, the measured heat transfer coefficient frequently fell below that predicted for nucleate boiling alone, even at low dryness fraction, implying that intermittent dry-out occurred. It is suggested that the model and the correlations applied provide a basis for the prediction of boiling heat transfer coefficients in narrow tubes, however, intermittent dry-out occurs under a wider range of conditions than expected and the model must be refined to account for this.

CONCLUSIONS

Existing flow boiling heat transfer correlations do not perform well when applied to narrow channels having a confinement number of the order of 0.5 and above. Nucleate boiling type correlations yield better results in spite of not taking account of the convective component of heat transfer or the tendency for intermittent local dry-out to occur in confined tubes. A model based upon film evaporation and nucleate boiling has yielded promising results but does not yet account satisfactorily for intermittent dry-out.

Acknowledgements—The authors gratefully acknowledge the financial support received from DG-XII of the European Commission through Contract No. JOU2-CT92-0045, in the framework of the JOULE II Programme.

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