

0017-9310(94)00282-7

Boiling heat transfer in small horizontal tube bundles at low cross-flow velocities

AKHILESH GUPTA, J. S. SAINI and H. K. VARMA†

Department of Mechanical and Industrial Engineering, University of Roorkee, Roorkee 247 667,
India

(Received 9 January 1992 and in final form 18 August 1994)

Abstract—An experimental investigation has been carried out to determine the local forced convective boiling heat transfer coefficient in small tube bundles consisting of horizontal tubes in a vertical column arranged in a large channel under low cross-flow velocities in saturated distilled water at atmospheric pressure. The effects of the impressed heat flux, cross-flow velocity and tube geometry on the heat transfer characteristics have been investigated. While no significant effect of the bundle geometry was observed on the heat transfer coefficient of the bottom tube compared to that of the single tube, vapour bubbles rising from the lower tubes and agitating the liquid around the upper tubes were found to yield higher values for heat transfer coefficients on the upper tubes. A Chen-type relation has been used to correlate the data on local forced convective heat transfer coefficients of upper tubes with reasonably accuracy.

INTRODUCTION

The phenomena of cross-flow boiling in tube bundles occurs in a variety of industrial appliances such as kettle reboilers, flooded evaporators and chemical process equipment. Historically these appliances have been designed on the basis of heat transfer results obtained from a single tube. However, it has now been well established that the heat transfer characteristics of a tube in a heated tube bundle are considerably different from those of a single tube due to vapour bubbles generated on lower tubes rising and causing turbulence around the upper tubes. Very little information is available in the literature concerning the local (i.e. tube position within the tube bundle) boiling heat transfer coefficient in tube bundles. Chan and Shoukri [1] analyzed in detail the boiling characteristics of small multitube bundles under pool boiling condition. Bennett *et al.* [2], Cornwell and Leong [3], Polley *et al.* [4], Hwang and Yao [5], Hsu *et al.* [6], Jensen and Hsu [7] and Cornwell and Scoones [8] are a few who have studied local boiling heat transfer coefficients in tube bundles under cross-flow velocity conditions. These investigators have used ideal tube bundles at low pitch-to-tube diameter ratios ($P/d = 1.3\text{--}1.5$) and a small gap between the outer tube of the bundle and the channel wall. Assuming a uniform distribution of vapour bubbles over the entire cross-section of the channel, heat transfer relations were expressed in terms of the void fraction or the Martinelli parameter. These relations, however, are not suitable for a small tube bundle in a large channel as the vapour bubbles in such system are confined to a small fraction of the channel area. An accurate

prediction of the local void fraction in such systems is therefore not possible. A large number of systems, such as kettle reboilers and flooded evaporators, are found to have small tube bundles in a large shell under low cross-flow velocity conditions. In view of this, an experimental investigation was planned to determine the local boiling heat transfer coefficients using small bundles of horizontal tubes arranged one above the other at different pitch distances in a large channel ($d/H = 0.064$) under low cross-flow velocity conditions. The effect of the heat flux, bulk fluid velocity and tube geometry is presented in this paper. A correlation for local convective boiling heat transfer coefficient based on Chen's [9] relation is proposed.

EXPERIMENTAL APPARATUS AND PROCEDURE

Experimental apparatus

A schematic diagram of the experimental set-up used for this purpose is shown in Fig. 1. The main components of this set-up include a test vessel, test heaters, a preheater, a receiver unit, a condensing and cooling water system, and the necessary measuring instruments. The test vessel was fabricated from stainless steel sheet and measured $300 \times 200 \times 425$ mm. It had a perforated $300 \times 200 \times 1$ mm aluminum sheet fitted at its bottom in order to obtain flow uniformity at the entrance to the vessel. In order to facilitate the flow visualization and photography, sight glasses were provided on three sides of this vessel. One side of the test vessel was fitted with a $250 \times 250 \times 25$ mm Bakelite sheet. Holes of 20 mm diameter at desired locations were drilled into it where the test heaters could be fixed. An overflow pipe was provided to maintain a constant liquid level (100 mm above the top tube of

† Author to whom correspondence should be addressed.

NOMENCLATURE

Bo	boiling number, $q/h_{fg}G$	N	number of lower tubes
d	tube diameter [m]	P	pitch distance [m]
E	enhancement factor	Pr	Prandtl number
g	acceleration due to gravity [$m\ s^{-2}$]	q	heat flux [$W\ m^{-2}$]
g_c	proportionality constant used in Newton's second law [$kg\ m\ N^{-1}\ s^{-2}$]	Re	Reynolds number, Gd/μ
G	mass flux based on minimum flow cross-section area [$kg\ m^{-2}\ s^{-1}$]	T_s	saturation temperature [K]
h	heat transfer coefficient [$W\ m^{-2}\ K^{-1}$]	T_w	temperature at heating wall-liquid interface [K]
h_{fg}	latent heat of vapourization [$J\ kg^{-1}$]	ΔT_e	effective wall superheat [K].
h_L	forced convection heat transfer coefficient of liquid [$W\ m^{-2}\ K^{-1}$]	Greek symbols	
h_{mic}	microconvective heat transfer coefficient associated with the bubble nucleation and growth [$W\ m^{-2}\ K^{-1}$]	μ	fluid viscosity [$N\ s\ m^{-2}$]
h_{PNB}	nucleate pool boiling heat transfer coefficient [$W\ m^{-2}\ K^{-1}$]	ρ	fluid density [$kg\ m^{-3}$]
k	thermal conductivity of fluid [$W\ m^{-2}\ K^{-1}$]	σ	surface tension [$N\ m^{-1}$]
		θ	angular position [$^\circ$].
		Subscripts	
		L	liquid
		v	vapour.

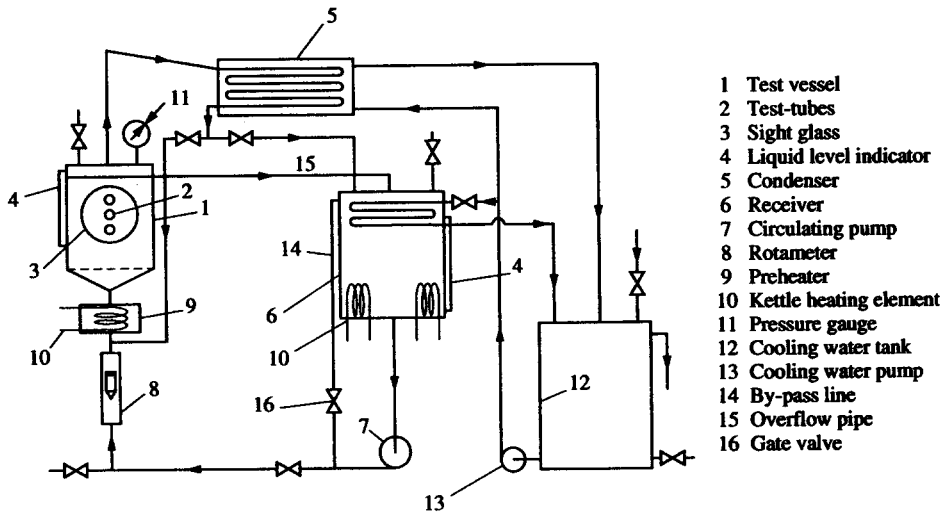


Fig. 1. Schematic diagram of experimental set-up.

the bundle) in the vessel and the top of the test vessel was connected to the condenser through a pipe. A pressure gauge and a mercury thermometer fitted at the top of the vessel were used to monitor the system pressure and bulk fluid temperature, respectively. The vessel was fitted with a water level indicator and pockets to insert thermocouples to measure the bulk fluid temperature. A 1 h.p. (0.736 kW) pump was used to circulate liquid from the receiver unit to the test vessel, and the rate of circulation was regulated through a gate valve and a by-pass line; this rate was measured by means of a rotameter. The fluid entering the test vessel was saturated and its temperature was controlled by a heating element provided in the preheater. Power to this heating element was regulated with a

variable transformer. The system components and piping were well insulated to prevent heat losses to the surroundings.

A stainless steel (AISI 304) tube of commercial finish having a 19.05 mm o.d. and 0.89 mm wall thickness formed the test heater, the details of which are shown in Fig. 2. Each tube was 245 mm long and, with an effective heating length of 190 mm, provided resistive heating by means of a high alternating current fed to it through a step-down transformer. All tubes of the bundle were connected in series and the amount of current passing through the bundle was controlled through a variable-voltage transformer. Power supplied to test heaters was measured with the help of a digital voltmeter and digital ammeter with a lowest

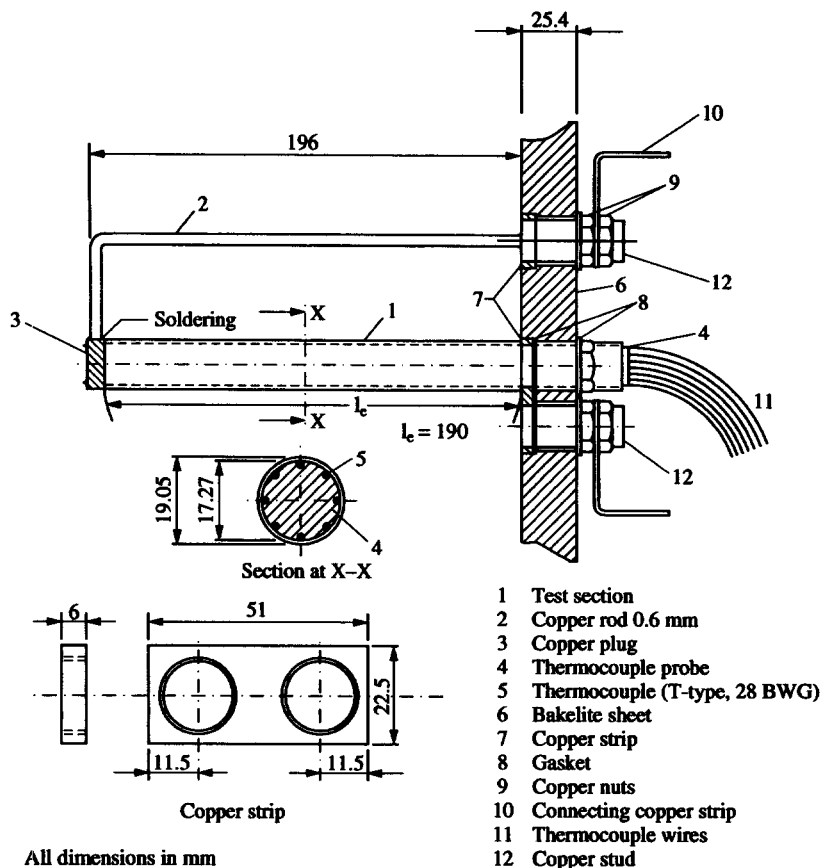


Fig. 2. Single-tube test section details.

count of 0.001 V and 0.1 A, respectively. Eight 28 BWG copper-constantan thermocouples (T-type) evenly spaced at 45° on the inner wall and midway between the heated length of each tube were used to measure the average wall temperature of the tube. Thermocouple beads were held firmly against the inner wall of the tube through a specially designed cylindrical fire-clay thermocouple probe as shown in Fig. 2. The thermocouples were connected to a 100 channel Orion Data Logging System to record the temperature with a lowest count of 0.01°C.

Experimental procedure

The experiment was planned to investigate three distinct sets of bundle arrangements, namely: (i) a single tube in a channel to represent a basic building block, (ii) two horizontal tubes placed one above the other at different pitch distances, and (iii) three horizontal tubes one above the other at a constant pitch distance. The system was run for about 10 h daily for 1 week at near maximum heat flux in order to eliminate the effect of aging prior to actual data collection. The air vents were kept open during the initial stages of heating and were closed only when steam in sufficient quantity started coming out, to ensure complete air removal from the system. The heat flux and fluid flow rate were adjusted to the desired levels and

the data pertaining to the rotameter, ammeter, voltmeter, thermocouples, pressure gauge and barometer were recorded when the system had attained a steady state. A similar procedure was adopted for each set of system and operating parameters investigated.

The range of system and operating parameters of this study is listed below:

Working fluid: distilled water

Inlet fluid temperature: saturation temperature

System pressure: 1.0019 bar

Tube geometry: (i) single tube in a channel ($d/H = 0.064$)

(ii) two tubes one above the other ($P/d = 6.0, 4.5, 3.0, 1.5$)

(iii) three tubes one above the other ($P/d = 3.0$)

Heat flux range, q : 10–40 kW m⁻²

Mass flux range, G : 0–10 kg m⁻² s⁻¹

Uncertainties in the experimental data, as estimated through a propagation of error analysis [10] for a typical set of data are: q , ±0.54%; G , ±3.62%; h , ±2.7%; p , 1.23 kPa.

RESULTS AND DISCUSSION

The experimental results for tube bundles were analyzed and it was found that the heat transfer coefficient

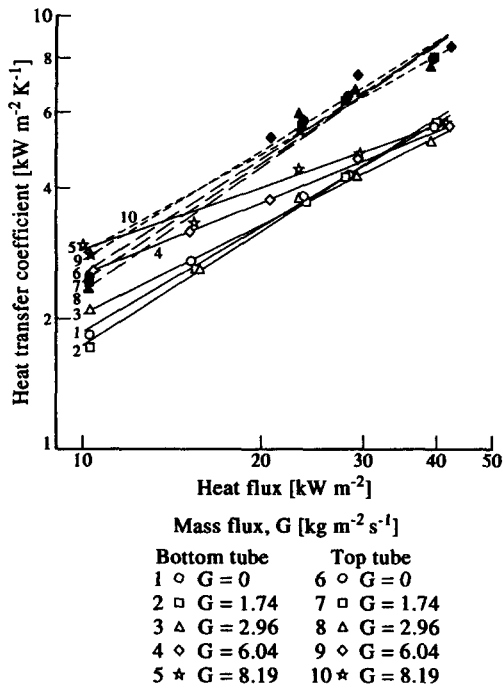


Fig. 3. Heat transfer coefficient for the bottom and top tubes of a two-tube bundle (1×2 , $P/d = 3.0$).

of the bottom tube in all tube bundle configurations was nearly the same as that on a single tube, and it was found to increase with the increase in heat flux and cross-flow velocity, the enhancement being more at low heat flux values; however, it diminishes as the heat flux increases. Heat transfer coefficients for the bottom and top tubes of a typical two-tube bundle are shown in Fig. 3. It is seen that the heat transfer coefficient of the upper tube is enhanced as a result of turbulence created by vapor bubbles rising up from the lower tube and by the impingement of these bubbles on its surface. Further, this enhancement is found to decrease with an increase in flow velocity, presumably due to suppression of nucleation over the lower tube under these conditions, which in turn generates a lower degree of turbulence around the upper tube.

The effect of pitch distance on the heat transfer coefficient of the upper tube in a two-tube bundle geometry for pool boiling and a typical case of flow boiling are shown in Figs. 4 and 5 respectively. It is found that the heat transfer coefficient of this tube increases as the pitch distance decreases, apparently due to the larger number of bubbles intercepted by the upper tube when the pitch is lowered.

The heat transfer coefficients of bottom, middle and top tubes under pool boiling, and a typical case of forced convection boiling, are shown in Figs. 6 and 7, respectively. The heat transfer coefficient is found to increase from the bottom tube upwards, the top tube having the maximum heat transfer coefficient as it experiences the maximum agitation due to the combined effect of vapour bubbles rising from the bottom as well as the middle tube. Moreover, this enhance-

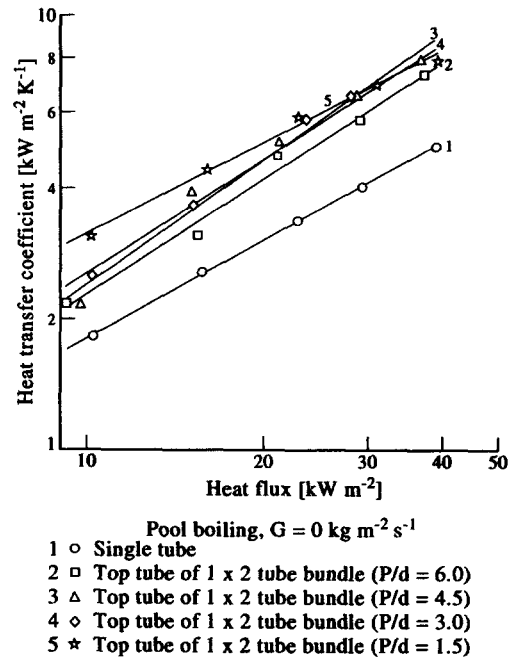


Fig. 4. Effect of P/d on average heat transfer coefficient of top tube for two-tube bundles under pool boiling conditions.

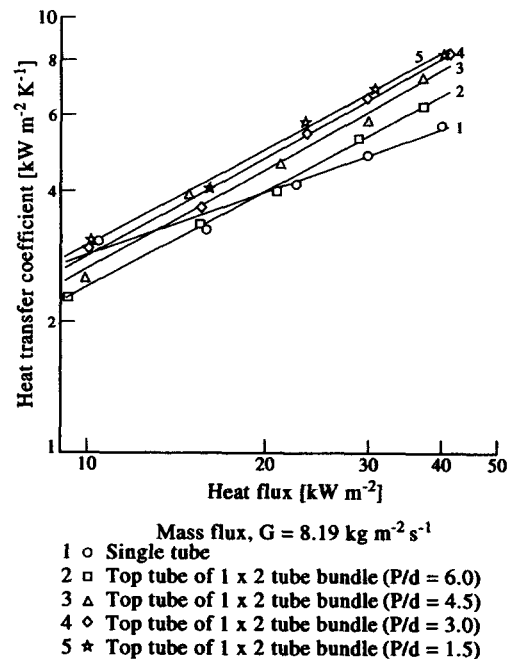


Fig. 5. Effect of P/d on heat transfer coefficient of top tube for two-tube bundles under maximum cross-flow velocity condition.

ment is found to be higher under low-velocity condition due to the higher density of bubble generation in the bottom tube, which causes a higher enhancement of the heat transfer coefficient for the middle and top tubes, respectively.

The angular distribution of the local heat transfer coefficients of the tubes revealed that, in all cases,

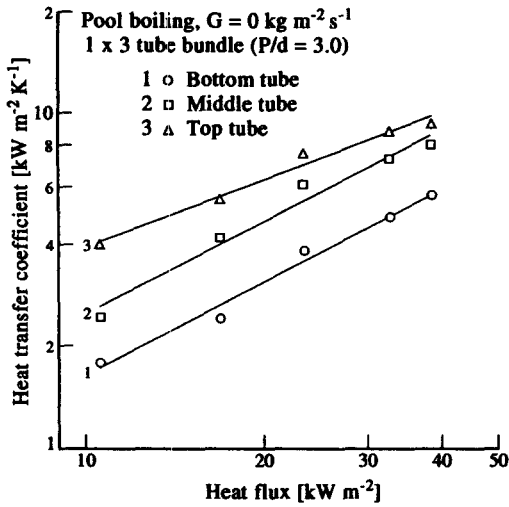


Fig. 6. Heat transfer coefficient from different tubes of a three-tube bundle under pool boiling condition (1×3 , $P/d = 3.0$).

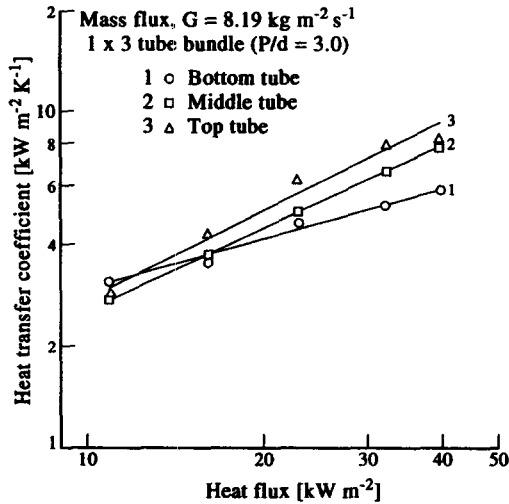


Fig. 7. Heat transfer coefficient from different tubes of a three-tube bundle under maximum cross-flow velocity condition (1×3 , $P/d = 3.0$).

maxima and minima occurred at the bottom ($\theta = 0^\circ$) and top ($\theta = 180^\circ$), respectively. Angular variation of the local heat transfer coefficient over the surface of the bottom and top tubes for a typical case is shown in Figs. 8 and 9, respectively.

CORRELATION OF DATA

Bottom tube of the bundle

The data for the convective boiling heat transfer coefficient (h) of the single tube and the bottom tube of all tube bundle configurations presently investigated were compared with the results proposed by Kutateladze [11], as given below :

$$\frac{h}{h_L} = \left[1 + \left(\frac{h_{PNB}}{h_L} \right)^n \right]^{1/n} \quad (1)$$

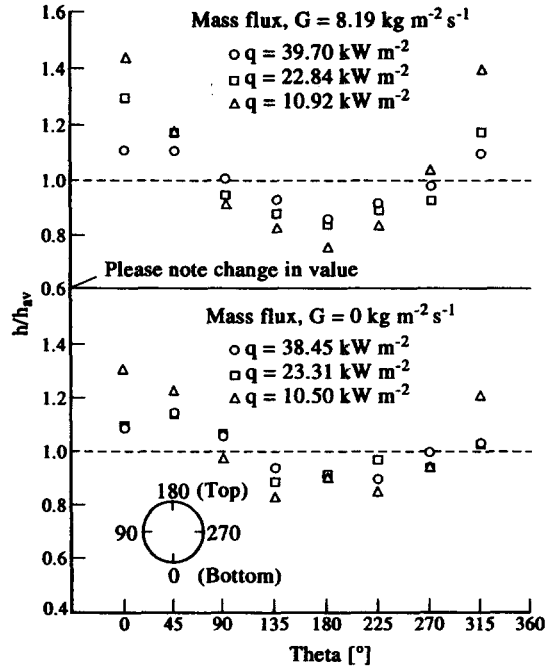


Fig. 8. Circumferential variation of heat transfer coefficient on bottom tube of a three-tube bundle (1×3 , $P/d = 3.0$).

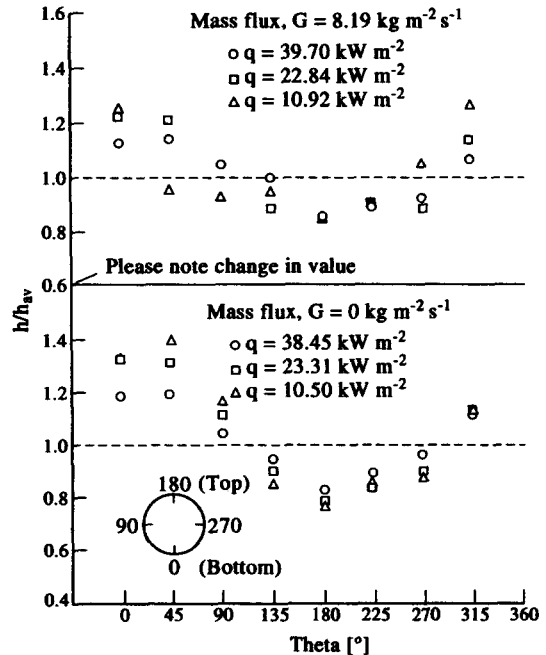


Fig. 9. Circumferential variation of heat transfer coefficient on top tube of a three-tube bundle (1×3 , $P/d = 3.0$).

The value of the liquid phase forced convection heat transfer coefficient (h_L) has been determined from the correlation proposed by Whitaker [12] for a cross-flow tube, as given below :

$$h_L = \frac{k_L}{d} (0.4Re_L^{1/2} + 0.06Re_L^{2/3})(Pr_L)^{0.36} (\mu_L/\mu_v)^{0.25} \quad (2)$$

The value of the pool boiling heat transfer coefficient (h_{PNB}) was obtained from the following relation obtained from the pool boiling data for a single tube:

$$h_{\text{PNB}} = 13.035(T_w - T_s)^{2.881}. \quad (3)$$

With $n = 1$, equation (1) was found to correlate the convection boiling data reasonably well and had an average absolute deviation of $\pm 6.8\%$.

Upper tubes of the bundle

The experimental data pertaining to the heat transfer on the upper tubes of the bundles investigated in the present work have been correlated using the following Chen-type relationship:

$$h = Eh_L + h_{\text{mic}} \quad (4)$$

where Eh_L represents the heat transfer coefficient associated with the bulk movement of vapour and liquid. E , the enhancement factor accounts for the enhancement of the single-phase liquid convective heat transfer coefficient (h_L) due to turbulence created by rising vapour bubbles and impinging on the tube surface. The microconvective component (h_{mic}) relates to the heat transfer associated with the bubble nucleation and growth. To evaluate the microconvective term of forced convection boiling, as suggested by Bennett *et al.* [2], the most valid pool boiling correlation in terms of the wall superheat [$\Delta T_s(T_w - T_s)$] was identified and then the term ΔT_s was replaced by ΔT_e , the effective wall superheat. Replacing ΔT_s of equation (3) by ΔT_e , h_{mic} may be written as

$$h_{\text{mic}} = 13.035(\Delta T_e)^{2.881}. \quad (5)$$

ΔT_e is evaluated using the expression developed by Bennett *et al.* [2], as given below:

$$\frac{\Delta T_e}{\Delta T_s} = \frac{k_L}{(Eh_L)X_o} [1 - e^{-(Eh_L X_o/k_L)}] \quad (6)$$

where

$$X_o = 0.041 \left[\frac{g_c \sigma}{g(\rho_L - \rho_v)} \right]^{1/2}. \quad (7)$$

ENHANCEMENT FACTOR

Several investigators [2–8] developed correlations for the enhancement factor (E) using nearly ideal tube bundles in terms of the void fraction or the Martinelli parameter χ_{tt} . However, these relations, as explained earlier, are not suitable for systems using a small tube bundle in a large channel. Hence, there is a need to develop a relationship between the enhancement factor and the physical and operating parameters of the system. The enhancement factor depends on the agitation due to vapour bubbles rising from the lower tubes and impinging on the tube under consideration; the density of bubbles depending upon the impressed heat flux, cross-flow velocity, pitch distance and the number of tubes below it. It can therefore be assumed

that this factor is a function of the boiling number [$Bo(q/h_{fg}G)$], pitch-to-diameter ratio (P/d) and the number of tubes below the tube under consideration (N). A functional relationship can therefore be written:

$$E = c(Bo)^{m_1}(P/d)^{m_2}(N)^{m_3}. \quad (8)$$

Using experimental data and equations (4)–(7), the constants c , and the indices m_1 , m_2 and m_3 , of equation (8) were determined, and the following relationship for the enhancement factor (E) for the upper tubes of all tube bundles was obtained:

$$E = 938.53(Bo)^{0.710}(P/d)^{-0.186}(N)^{0.463}. \quad (9)$$

The values of the heat transfer coefficients for upper tubes were calculated using equations (4)–(9) and compared with those obtained from experimental data as shown in Fig. 10. Experimental data were found to correlate well, most of the points lie within the $\pm 20\%$ lines, and the heat transfer coefficient values have been found to have an average absolute deviation of 12.1%.

Polley *et al.* [4], in their investigation of cross-flow boiling on tube bundles, used the actual wall superheat to evaluate the microconvective component (h_{mic}) and computed flow boiling heat transfer coefficient using a relation similar to the Chen type. Following Polley's approach of assuming the actual wall superheat and the effective superheat to be the same, the boiling component (h_{mic}) of the flow boiling heat transfer coefficient has also been computed using a similar procedure to that mentioned above and the following correlation was found in this case:

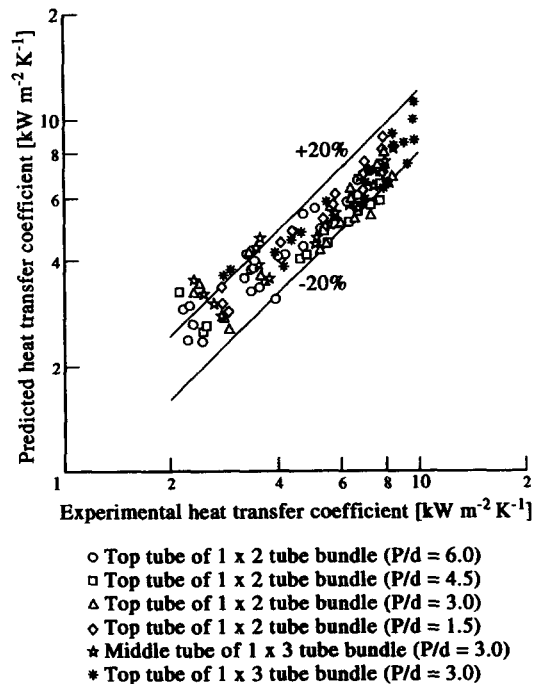


Fig. 10. Experimental vs predicted heat transfer coefficient for upper tubes of various tube bundles.

$$E = 851.38(Bo)^{0.707}(P/d)^{-0.245}(N)^{0.577} \quad (10)$$

A comparison of the experimental values of the heat transfer coefficient with those determined using equation (10) were found to give an average absolute deviation of 12.8%.

It is therefore concluded that the use of the Chen-type relation [equation (4)] can be used to correlate the data on forced convective cross-flow boiling reasonably well, and the use of the effective superheat in place of the actual wall superheat yields slightly better results.

CONCLUSIONS

The following conclusions are drawn from this experimental investigation of cross-flow boiling under low heat flux and low cross-flow velocity conditions carried out on small tube bundles in a large channel:

- (1) Heat transfer characteristics of the lowermost tubes in a tube bundle have been found to be independent of the presence of the bundle.
- (2) As a result of the enhanced turbulence generated by vapour bubbles rising from lower tubes, the heat transfer coefficient for the tubes located at higher levels in the flow direction is found to experience substantial enhancement. The maximum enhancement of the order of 100% was observed for the top tube of a 1 × 3 tube bundle under pool boiling and low heat flux conditions.
- (3) A Chen-type relation can be used to correlate the experimental data in forced convective flow boiling with reasonable accuracy.

REFERENCES

1. A. M. C. Chan and M. Shoukri, Boiling characteristics of small multitube bundles, *ASME J. Heat Transfer* **109**, 753–760 (1987).
2. D. L. Bennett, M. W. Davis and B. L. Hertzler, The suppression of saturated nucleate boiling by forced convection flow, *A.I.Ch.E. Symp. Ser.* **76**, 91–103 (1980).
3. K. Cornwell and L. S. Leong, On the analysis of low quality flow boiling, *Int. J. Heat Fluid Flow* **1**(2), 63–69 (1979).
4. G. T. Polley, T. Ralston and I. D. R. Grant, Forced cross flow boiling in an ideal in-line tube bundle, Paper presented at 19th National Heat Transfer Conference, ASME-AIChE, Orlando, FL (1980).
5. T. H. Hwang and S. C. Yao, Forced convective boiling in horizontal tube bundles, *Int. J. Heat Mass Transfer* **29**, 785–795 (1986).
6. J. T. Hsu, G. Kocamustafaogullari and M. K. Jensen, correlation of local cross flow boiling heat transfer coefficients in a horizontal tube bundle. In *Experimental Heat Transfer Fluid Mechanics & Thermodynamics* (Edited by R. K. Shah, E. N. Ganic and K. T. Yang), pp. 1634–1641. Elsevier Science (1988).
7. M. K. Jensen and J. T. Hsu, A parametric study of boiling heat transfer in horizontal tube bundle, *ASME J. Heat Transfer* **110**, 976–981 (1988).
8. K. Cornwell and D. J. Scoones, Analysis of low quality boiling on plain and low finned tube bundles, *2nd U.K. National Conference on Heat Transfer*, Vol. 1, pp. 21–32 (1988).
9. J. C. Chen, Correlation for boiling heat transfer to saturated fluids in convective flow, *Ind. Engng Chem. Process Des. Dev* **5**(3), 322–329 (1966).
10. A. Gupta, Investigation of boiling heat transfer from tube bundles in cross flow, Ph.D. Thesis, Mechanical and Industrial Engineering Department, University of Roorkee (1992).
11. S. S. Kutateladze, Boiling heat transfer, *Int. J. Heat Mass Transfer* **4**, 31–45 (1961).
12. S. Whitaker, Forced convection heat transfer correlations for flow in pipes, past flat plates, single cylinders, single spheres, and flow in packed beds and tube bundles, *A.I.Ch.E. J.* **18**, 361–371 (1972).