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INVESTIGATIONS ON POOL BOILING OUTSIDE HORIZONTAL TUBE BUNDLES

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ABSTRACT

As boiling outside horizontal tube bundle designs can result in a seemingly large number of combinations of heat flux, quality and bundle geometry, it needs to be studied as to how the various permutations might effect the flow path of liquid through a tube bundle in a boiling mode. Thus the probable effect on the heat transfer due to a disruption in the wetting (partial dry-out) of upper tubes in a bundle and the possible influence of a variation in the tube pitch so as to affect the wetting and dry-out characteristics of a tube needs to be established. Experiments conducted for pool boiling in 8 x 3 (eight rows and three columns) plain and coated tubes (Ra = 8.279 μm) bundles for three different inter-tube pitch distances (p/d=1.4,1.7 and 2.0) in an equilateral triangular arrangement using distilled water at atmospheric conditions revealed that the local heat transfer coefficient for a tube in a bundle increased with an increase in heat flux as well as with a decrease in the pitch distance. The coated tube bundles with the minimum pitch (p/d=1.4) exhibited the maximum bundle average heat transfer coefficients. The circumferential variation of heat transfer coefficient for the tubes suggest the lowest values for the upper surface of the tube periphery due to a coalescence of bubbles near the top surface while the highest values were observed nearer to the lower surface of the tube periphery due to the striking bubbles from the lower tubes. The experimental data is best fit to suggest a suitable correlation for the enhancement ratio in local pool boiling heat transfer coefficient ($h_{npb,local}/h_{bottom\ tube}$) for the plain and coated tube bundles taking into account the local void fraction and the p/d ratio where the void fraction is calculated using an iterative procedure beginning with the homogeneous void fraction as the initial guess and iterated until the assumed and calculated values agree within a precision of 0.0001 and valid for heat flux ranging from ~ 12 to 45 kW/m2. The present study did not find any conclusive evidence of partial dry-out and deterioration of heat transfer in the upper tubes for an eight-row tube bundle in the heat flux range and pitch-diameter (p/d) ratio considered.

NOMENCLATURE

Cap	[-]	Capillary Number
d	[m]	outer diameter of test tube
g	[m/s^2]	acceleration due to gravity
G	[kg/m^{2}s]	mass velocity
h	[kW/m^2.K]	heat transfer coefficient
N	[-]	tube row
p	[m]	tube pitch
P	[N/m^2]	pressure
q	[kW/m^2]	heat flux
R	[-]	velocity or slip ratio
Ra	[μm]	surface roughness
Ri	[-]	Richardson Number
T	[°C]	temperature
x	[-]	mass vapour quality
Special characters		
ε	[-]	Vapour void fraction
μ	[N/s.m^2]	Dynamic viscosity
θ	[degrees]	Angular position
ρ	[kg/m^3]	Density of liquid phase
σ	[N/m]	surface tension
Subscripts		
bottom		for the bottommost tube in bundle
bundle		tube bundle
l		liquid
npb,local		local nucleate pool boiling coefficient
v		vapor

INTRODUCTION

Boiling outside horizontal tube bundles commonly occurs in applications such as flooded evaporators finding applications in refrigeration systems, kettle and thermosyphon reboilers, waste heat boilers, and fire-tube steam generators. As a process, boiling is a complicated phenomenon because of the large number of variables involved and the complex fluid motion patterns caused by the bubble formation and growth. In a tube bundle, the boiling heat transfer mechanism differs substantially from that on a single tube for same heat and mass flux conditions due to the bulk upward movement of the liquid and vapor mixture, circulation and turbulent effects produced by rising vapor bubbles from the lower tubes as well as due to the static head effects which causes increased saturation temperature in the lower part of a tube bundle and thus reduces the local driving temperature difference. Collier and Thome [1] describe different flow patterns encountered from bottom to top, together with corresponding heat transfer regimes of uniformly heated tubes in a simplified bundle layout.

In the past few years, a significant amount of bundle boiling data has become available for a variety of fluids and types of heat transfer tubes. The test data continue to show that bundle boiling effects are very important in tube bundles, increasing bundle performances. Among the previous studies on tube bundles include those by Leong and Cornwell [2], Cornwell et al. [3], Muller [4], Jensen and Hsu [5], and, Gupta et al. [6]. The previous studies related to boiling outside tube bundles have been reviewed extensively by Brown and Bansal [7], and, Casciaro and Thome [8,9].

While general studies have found an enhancement in overall boiling heat transfer coefficients for the tube bundle as compared to boiling outside a single tube surface, it is possible that during boiling in a tube bundle, a disruption in the wetting of a tube surface or partial dry-out thereby leading to a reduced heat transfer performance. This possibility is due to the peculiarities of the vapour flow specially over the upper rows in a tube bundle, and hence a possibility of the heat transfer characteristics of the upper tubes being affected due to same. Although tube bundle designs can result in a seemingly large number of combinations of heat flux, quality and bundle geometry, it needs to be studied as to how the various permutations might effect the flow path of liquid through a tube bundle in a boiling mode. Thus the effect on the heat transfer due to a disruption in the wetting (partial dry-out) of upper tubes in a bundle and the possible influence of a variation in the tube pitch so as to affect the wetting and dry-out characteristics of a tube needs to be established. Thus the void fraction remains one of the important parameter in building a proper model for the prediction of boiling outside tube bundles. Also it needs to determine whether the effect of tube pitch on boiling heat transfer in tube bundles can be simply ignored or accounted for by validation using experimentation as well as a qualitative assessment [10].

In view of above, an experimental study was undertaken in order to study the heat transfer behavior for local (i.e. on a tube located at a particular position in the bundle) as well as for tube bundle as a whole comprising of plain and coated(rough) tube bundles incorporating the effects of bundle geometry and heat flux.

VOID FRACTION IN TUBE BUNDLES

The void fractions in two-phase flows in tube bundles are much more difficult to measure than those for internal channel flows. Even though shell side void fractions have been studied less than internal channel flows, they are very important for obtaining accurate thermal designs. In particular, for thermosyphon evaporators, the circulation rate depends directly on the two-phase pressure drop across the tube bundle, and, hence, the variation in void fraction is of primary importance. Among the various studies for void fraction include those by Kondo and Nakajima [11], Schrage et al. [12], Dowlati et al. [13], Zuber and Findlay [14], Ishihara et al. [15], etc.

In order to predict the velocity (or slip) ratio R for various vapour qualities, Feenstra et al. [16] developed an empirical expression using a dimensional analysis approach in which the functional parameters influencing R include dynamic viscosity of the liquid, surface tension, two-phase density, liquid-vapor density difference, pitch flow velocity of the fluid, gravitational acceleration, tube pitch , the gap between neighbouring tubes, tube diameter and the frictional pressure gradient. The tube pitch p and tube diameter d were included for their influence on the frictional pressure drop while the two phase density and density difference were included as they are key parameters in void fraction models. The surface tension σ was selected since it affects the bubble size and shape and the liquid dynamic viscosity μ_l was included because of its affect on bubble rise velocities. The following model was suggested for void fraction ε as:

$$\varepsilon = \frac{1}{1 + R\left(\dfrac{(1\text{-x})}{\text{x}}\dfrac{\rho_v}{\rho_l}\right)} \tag{1}$$

where the velocity (or slip) ratio R is calculated in terms of the pitch to diameter ratio (p/d), Richardson number [$Ri = \dfrac{(\rho_l - \rho_v)^2 \; g(p-d)}{G^2}$] and Capillary number [$Cap = \dfrac{\mu_l u_v}{\sigma}$] as:

$$R = 1 + 25.7\ (Ri.\ Cap)^{0.5}\ (p/d)^{-1} \tag{2}$$

Here the mean vapour phase velocity u_v is determined as follows:

$$u_v = (x.G)/(\varepsilon.\rho_v) \tag{3}$$

This method used an iterative procedure to determine the void fraction which begins using the homogenous void fraction as an initial guess. The iterative procedure improves each guess until the difference between successive guesses is less than 0.0001. This method was successfully compared to air-water, R-11, R-113 and water- steam void fraction data obtained from different sources. Consolini et al. [17] also found this method to be the best for predicting static pressure drops at low mass flow rates for an 8-tube row high bundle under evaporating conditions.

EXPERIMENTAL INVESTIGATIONS

An experimental investigation was conducted in order to study boiling heat transfer behaviour for distilled water at near atmospheric pressure for different tube bundle geometries of plain and coated tubes. Earlier investigations with single tube boiling revealed the effect of surface roughness in coated tubes (low-porosity, flame-sprayed) for saturated boiling of water at atmospheric pressure [18]. The parameters varied for the study was heat flux from 12 to 45 kW/m^2, surface roughness (Ra) of 0.3296 µm and 8.279 µm and pitch/diameter ratios of 1.4, 1.7 and 2.0.

The main components of the experimental setup include: the test vessel including the test heaters, preheater, condensing and cooling water system and the measuring instruments including the data logging unit. The test surface comprised of the plain stainless steel surface of commercial finish (Ra=0.3296 µm) and coated tubes of stainless steel (AISI 304) base material. The coatings were created by thermal spraying plain stainless steel tube with a thin coating (0.145 mm thickness) of SS 316 on the outer surface using the wire flame process (wire diameter 3.15 mm). The average roughness (Ra) of the coated test surface was 8.279 µm. Tube bundles (Figure 1 and 2) with three different pitch distances (p/d=1.4, 1.7 and 2.0) were used for plain tubes and for coated tubes. The test heaters comprised of plain stainless steel tubes having an outer diameter of 19.05 mm, an inner diameter of 17.45 mm and an overall length of 175 mm (an effective heated length of 130 mm) resistively heated using A.C. supply through a step down transformer and controlled through a variac. In order to monitor the inner surface temperature of the tube wall, eight, calibrated (28 SWG) copper-constantan thermocouples evenly spaced circumferentially at the mid length of the tube inner surface were used. Only the central column of the bundle was instrumented with thermocouples. A thermocouple was placed near the bottommost tube to monitor the inlet fluid temperature so that only saturated fluid entered the system. A kettle heating element with variac control was used to control the temperature of the same and prevent inlet sub-cooling.

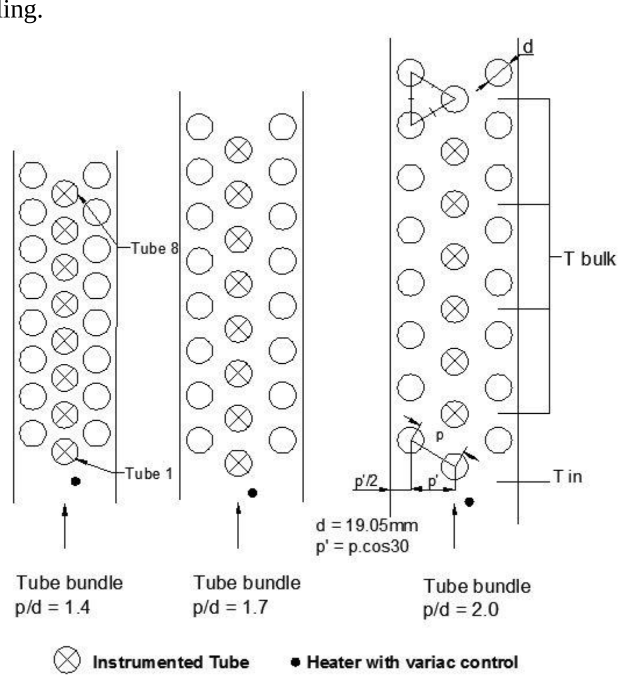


Figure 1 The various arrangements for the experimentation [19]

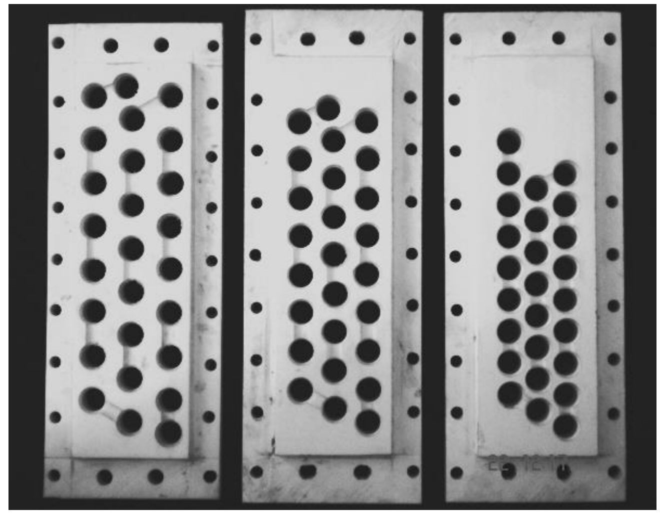


Figure 2 Photograph of tubesheets of different p/d ratios (2.0, 1.7 and 1.4).

The experimental uncertainties estimated through a propagation of error analysis by the Kline and McClintock [20] method for a typical set of data were found to be : T , ±0.2 °C ; q, ±2.37% ; h, ±4.11% ; P , 0.5kPa ; Ra, 12nm.

The pool boiling results for the lowermost tube were found to be similar to those of a single (standalone) tube in a channel and validated with previous studies/correlations. The results for tube bundles as shown in Figure 3 reveal a significant influence of lower tubes on the heat transfer coefficient on the upper tubes. While the heat transfer coefficient of the lowermost tube in the bundle was found to be very near to the results obtained for a standalone single tube experiment, the heat transfer coefficient for the topmost (eighth) tube were found to be slightly lower than those for the seventh tube due to the probable reason that the bubble plumes leaves the upper portion of the tube uninterrupted due to the absence of any other tube above it.

Further, corresponding to same values of pitch distance and heat flux, the heat transfer coefficient on upper tubes of the coated tube bundle was found to be higher than the corresponding tubes of the plain tube bundle. This could be due to comparatively increased vapor bubble activity from the lower tubes and also more bubbles being generated on the tube itself in case of a coated tube bundle. In comparison to the lower tubes, the uppermost tubes did not exhibit much variation in the heat transfer coefficient with a change of heat flux and the reason for this may be that the uppermost tubes of the bundle might have reached a near maximum enhancement possible due to bubbles coming up from the lower tubes (Figure 4).

Figure 5 shows the results for the variation of a tube specific local heat transfer coefficient (considered circumferentially) with respect to the tube average heat transfer coefficient (h/h_{avg}) for the third tube in a plain tube bundle with p/d=1.4. The circumferential variation of heat transfer coefficient for the tubes suggest the lowest values for the upper surface of the tube periphery due to a coalescence of bubbles near the top surface while the highest values were observed nearer to the lower surface of the tube periphery due to the striking bubbles from the lower tubes.

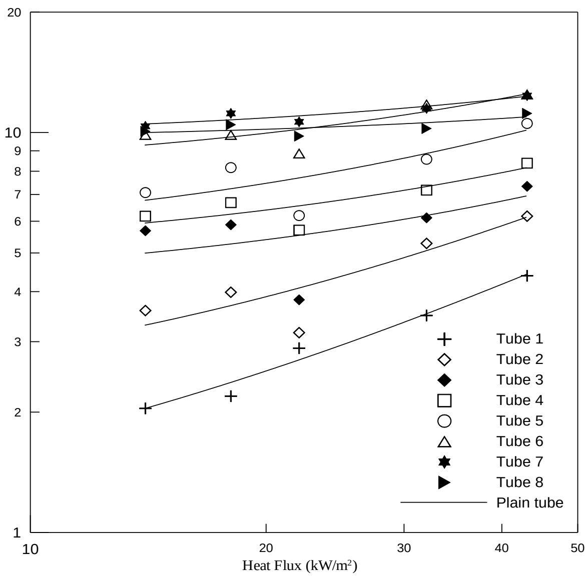
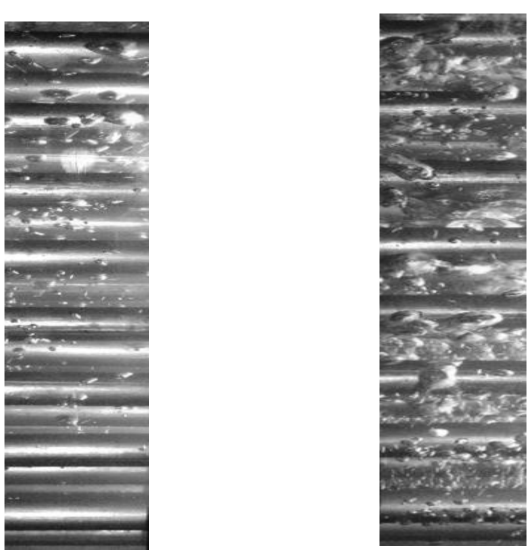


Figure 3 Pool boiling heat transfer coefficient on central column tubes of a plain tube bundle of maximum pitch (p/d=2.0).



q= 22.1 kW/m^2 q= 43.4 kW/m^2

Figure 4 Photographs showing bubble generation on plain tube bundles (p/d= 2.0) at heat flux of 22.1 kW/m^2 and 43.4 kW/m^2 respectively

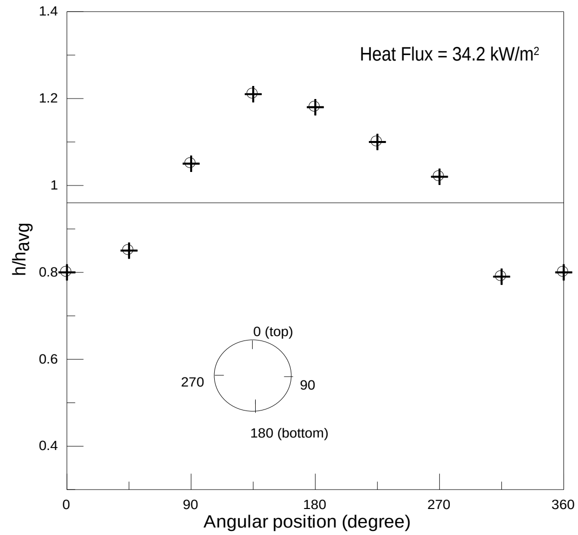


Figure 5 The circumferential variation of heat transfer coefficient (h/h_{avg}) in the tube at third position (from bottom) in a plain tube bundle (p/d=1.4)

Both plain and coated tube bundles with minimum pitch distance gave the maximum value of average bundle heat transfer coefficient at all heat flux values. Further, for the same minimum pitch distance, average bundle heat transfer coefficient was found to be higher for the coated tube bundle than that of the plain tube bundle, reason for this may be generation of more bubbles in case of coated tube bundle creating more turbulence as compared to that in a plain tube bundle. The effect of heat flux on heat transfer coefficient was more apparent on the lower tubes (upto fifth or sixth tube) in comparison to the upper tubes (Figure 6).

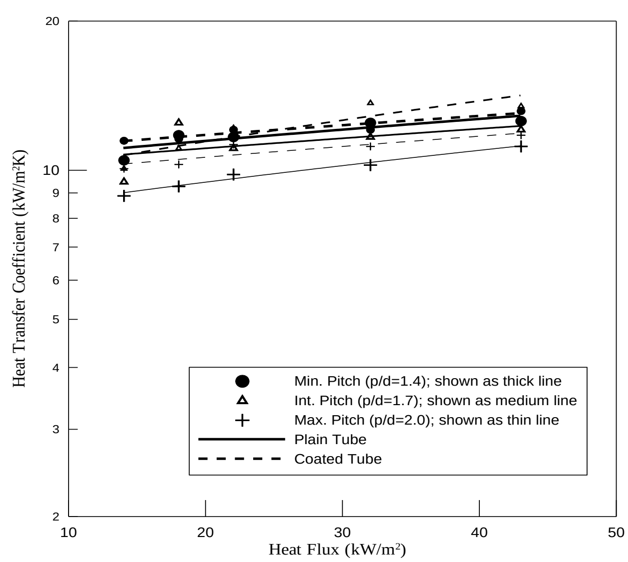


Figure 6 Effect of pitch on pool boiling heat transfer on eighth tube (topmost) from bottom of plain and coated tube bundles

The maximum heat transfer coefficient obtained for a tube surface was 14.7 kW/m^2 K and was found for the tube number seven from bottom at a heat flux of ~ 43 kW/m^2 and corresponding to the coated tube bundle with the minimum pitch distance. The maximum enhancement in the local tube heat transfer coefficient compared to the heat transfer coefficient on the lowermost tube was found to be equal to 5.83 at the seventh tube from bottom in case of plain tube bundle of minimum pitch distance and corresponding to a heat flux of 14.1 kW/m^2, while for the coated tube bundle it was equal to 5.0 corresponding to the seventh tube of the minimum pitch bundle (p/d=1.4) at a heat flux of 14.2 kW/m^2.

The maximum bundle average heat transfer coefficient observed was for the coated tubes bundle of minimum pitch (p/d=1.4) and equal to 10.92 kW/m^{2}K at a heat flux of 43.21 kW/m^2 while the minimum bundle average heat transfer coefficient observed was for the plain tubes bundle of maximum pitch (p/d=2.0) and equal to 5.54 kW/m^{2}K at a heat flux of 14.13 kW/m^2**.** The maximum enhancements in the bundle average heat transfer coefficient as compared to the heat transfer coefficient on the lowermost tube for the plain tube bundle was found to be 3.7 at a heat flux of 14.1 kW/m^2 with minimum pitch distance (p/d=1.4), while for the coated tube bundle the same was equal to 3.0 corresponding for the minimum pitch bundle (p/d=1.4) at a heat flux of 14.2 kW/m^2.

It is therefore concluded that the bundle factor needs to be considered in the design of flooded evaporators for heat fluxes (for the range ~ 45 kW/m^2). Also it was observed that for the range of parameters studied, local (i.e., tube located at a particular location in the bundle) as well as the average bundle heat transfer coefficient under pool boiling condition was found to be maximum at all heat flux values on the coated tube bundle for the minimum pitch distance (p/d=1.4).

DEVELOPMENT OF CORRELATION

Within the tube bundle, the local enhancement ($h_{npb,local}/h_{bottom\ tube}$) is a function of void fraction as well as heat flux. Higher the density of vapour bubbles from lower tubes, greater will be the enhancement. Therefore, the heat transfer coefficient of the upper tubes is found to increase with the increase of heat flux as more bubbles are now rising up from lower tubes. The coated tubes in the bundles exhibit a higher enhancement as the bubbles generated are more than those on a plain tube surface corresponding to the same values of heat flux. Also when the pitch distances are less, the bubbles emanating from the lower tubes form a more denser plume at the upper tubes in comparison to the bundles having larger pitch distances. Thus the void fraction is an important parameter in properly mapping the heat transfer profile of a tube bundle.

As discussed earlier, Feenstra, et al. developed an empirical expression to predict the velocity (or slip) ratio R, for vapor qualities ranging from 0 to 1, using a dimensional analysis approach and suggested a model for predicting the void fraction given by Eqns. 1-3. For the same, an iterative procedure is required to determine the void fraction using this method. As observed from Eqns. 1-3, in order to estimate the void fraction, it is required to work out on three unknowns, viz., G, x and ε. Thus without G, the void fraction ε cannot be estimated. In the present study, G was estimated by considering the vapour generated due to latent energy transfer at each tube level. Here the assumption made is that only the vapour moved in the upwards directions, with more vapour getting added at each tube levels and the cumulative mass moving upwards, thereby disregarding the movement of liquid component. The calculation for the mass velocity G was based on the minimum cross-sectional flow area. A program coded using Borland C language was used to calculate the void fraction based on Feenstra model [21]. Here the void fraction ε was calculated by beginning with homogeneous void fraction as the initial guess and iterating until the assumed and calculated values agree within a precision of 0.00001.

The observed enhancement in local pool boiling heat transfer coefficient ($h_{npb,local}/h_{bottomtube}$) for plain and coated tube bundles best fit the following relation taking into account local void fraction ε, and pitch ratio, p/d.

$$h_{npb,local} / h_{bottomtube} = 2.288\left[1/(1-\varepsilon)\right]^{0.275}\left(p/d\right)^{-0.297} \quad (4)$$

Using the above model, a comparison of experimental and predicted results of local pool boiling heat transfer coefficient showed an average absolute deviation of 20.09 % and a standard deviation of 14.49 %.

CONCLUSIONS

The present study did not find any conclusive evidence of partial dry-out and deterioration of heat transfer performance in the upper tubes for an eight row plain and coated tube bundle with the heat flux range 12-45 kW/m^2 and p/d ratios of 1.4,1.7 and 2.0 (equilateral tube arrangement) during boiling with distilled water at atmospheric pressure. Overall the performance of coated tube bundles with the minimum pitch ratio (p/d=1.4) was found to be the best. Also the maximum enhancements (of local upper tube heat transfer coefficient as well as that of the bundle average heat transfer coefficient) in comparison to the lowermost single tube in the bundle was found to be higher at lower heat fluxes in all the tube bundles. In the design of flooded evaporators, appropriate consideration should be given to the bundle factor involved.

The experimental data for heat transfer enhancement ($h_{npb,local}/h_{bottomtube}$) fitted with a standard deviation of 14.49% into a correlation involving the local void fraction, and pitch ratio where the local void fraction was calculated using an iterative technique developed by Feenstra et al. beginning with the homogenous void fraction as the initial guess.

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