Boiling Heat Transfer on the Outside of A Miniature Diameter Compact Tube Bundle

Ebenezer Adom, Sheikh Z Islam, Peter Kew

Abstract— **The advancement of process intensification has necessitated the establishment of an appropriate approach into the boiling outside tubes. This paper present knowledge on the boiling heat transfer outside small diameter tubes and bundles and elucidates the effect of diameter in a compact tube bundle. It is shown that the effect of tube clearance between the tubes restrict the movement of bubbles up the bundle. The confinement number Co is used to investigate the boiling heat transfer inside a tube bundle and it is shown the correlations developed for boiling inside micro channels is appropriate for analysis for compact tube bundles.**

Index Terms— **boiling heat transfer, small diameter tubes, tube bundles.**

I. INTRODUCTION

The earliest work to consider the effect of diameter on nucleate pool boiling outside a horizontal tube was done by Cornwell [1] in which they investigated the heat transfer coefficient for horizontal tube diameter in the range of 6-30mm for water, refrigerants and organics at nominal atmospheric pressure. It was concluded that the heat transfer coefficient decreases as the diameter increases. Later studies by Cornwell and Houston [2] investigated the effect of tube diameter using a convection-based correlation, which is applicable to water, refrigerant and organics. For tubes in the industrial range the equation's applicable and it was concluded that for tubes of diameter more than three times the departure size diameter, the diameter affects the heat transfer coefficient and for tubes with diameter less than about 6mm there was a decrease in the heat transfer coefficient. Moreover the correlation deduced was shown not to be satisfactory with the data set at lower diameter tubes. Recent studies by Kew and Houston [3] showed that correlations that indicate inverse relationship between the heat transfer coefficient and the diameter cannot be applied to diameter less than 6mm. Thus data for 8mm tube diameter over predict the heat transfer coefficient for diameter below 3mm. Further experiments carried out on tube diameter ranging from 1-6mm revealed that there was no systematic variation of heat transfer coefficient with the diameters used. Kaupman, Gorenflo et al [4] investigated the effect of pool boiling heat transfer on horizontal tubes of different diameters (4-30mm). Their results showed that at constant normalized pressure the heat transfer coefficient do not vary much with the tube diameter at medium to high heat fluxes. They argued that the influence of tube diameter in their investigations were comparatively small and less compared with correlations such as those of Cornwell and Houston [2]. Studies by Cornwell and Kew [5] on a column of horizontal wires using distilled water and R-113 at nominal atmospheric pressure indicated that there is an increase in heat transfer coefficient upwards but this tends to decrease at the higher position of the wires.

In 2004, Das, Putra and Kabelec [6] investigated the effect of diameter for pool boiling for tubes in the range of 4-8mm using distilled water and R-123 as boiling fluids at near atmospheric pressure. It was found out that the usual correlation such as Cooper [7], Stephan [8] developed for the horizontal plate's underestimates the heat transfer coefficient for narrow tubes because of neglecting sliding bubble. However the correlation for large tubes overestimates heat transfer due to considering fully developed sliding bubbles mechanisms.

A reduction of the tube diameter reduced the bubble size as such sliding bubbles effect becomes minimal which results in a decrease in the heat transfer coefficient. Boiling on tube bundles, at low quality increase in heat transfer coefficient due to sliding bubbles in large bundle will be absent for small tubes, but the effect of turbulence due to rising bubbles will still lead to an increase in the heat transfer coefficient. At high vapour quality convective effect are dominant and the small diameter tubes will give higher values of heat transfer coefficient. Dryout at intermediate heat flux are due to the narrow gaps between the tubes which impedes the vapour flow. Dry out has also been reported by Schuller et al [9] for large tube bundles. In a compact tube bundle, dryout would be problematic as it is likely to occur at low vapour qualities as shown in Fig 1.

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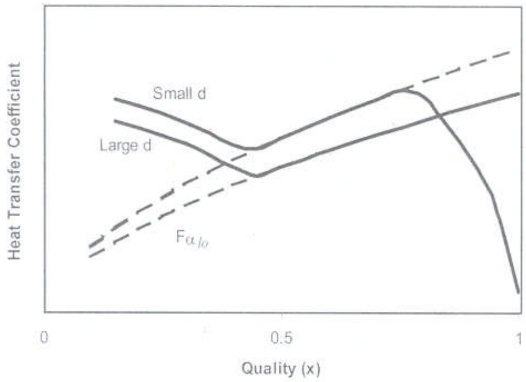

Figure 1 heat transfer coefficient with quality for tube bundles with large and small diameter tubes, Cornwell, Kew et al [10]

Confinement number Co has been used by Kew and Cornwell [11] for the heat transfer associated with a small channel. They suggested that confinement has effect for channel dimension having hydraulic diameter in the excess of 0.5. This criterion enabled the determination of the critical size of the channel defining the shift from isolated bubble regime to confined bubble regime for a given fluid pressure. The confinement number C_o defined by Cornwell and Kew [12] for flow in tubes was given as ;

$$C_o = \frac{\sqrt{\sigma / g(\rho_l - \rho_g)}}{d} \qquad (1)$$

The C_o has been used to differentiate the transition from large tubes from micro-channels by several investigators such as Thome [13].

II. Heat transfer mechanisms associated with large tube bundles

Heat transfer mechanism occurring in a tube bundle has been shown to be made up of nucleate and a convective component occurring simultaneously on a tube surface. Investigation into the anomalous increase in heat transfer coefficient in the upper tubes has resulted in the general conclusion that sliding bubble contributes to the enhancement that has been observed and reported in literature. Cornwell and Schuller [14] investigated the increase in heat transfer coefficient in the upper tubes of a bundle. A photographic study of boiling outside the tubes near the top reveals multiple bubbles sliding up the side of the tube as shown in figure 2.

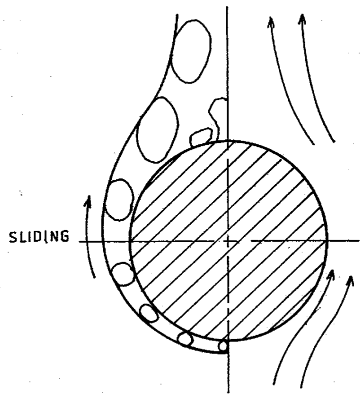

Fig 2: Sliding bubbles on a horizontal tube, Cornwell and Schuller [14]

It was found out that sliding bubbles account for the heat transfer in tube and bundles. A model based on the evaporation of the film layer beneath the bubble was used to explain the enhancement in heat transfer coefficient. A force balance analysis of the bubble reveals that the velocity of the sliding bubble is sensitive to the liquid properties and the contact angle.

In a later studies, Cornwell [15] carried out an experimental investigation to distinguish between the mechanisms that increases heat transfer coefficient in a tube bundle during bubbly flow using R-113. The heat transfer mechanism observed is a combination of forced convection, sliding bubbles and nucleate boiling, which is illustrated by the expression;

$$\alpha = \alpha_c + \alpha_{sb} + \alpha_{nb} \qquad (2)$$

The liquid convection term was obtained by using the appropriate Nusselt number equation, whereas the sliding bubbles was due to the heat flow to a tube as a results of bubbles existing in the free stream. The nucleate boiling terms was due to bubbles which nucleate and grow on the surface of the tube. Moreover the boiling heat transfer was determined on the surface of the tubes and the sides of the tubes were shown to exhibit high heat transfer due to sliding bubbles. In a similar studies, Cornwell [16] investigated the role of sliding bubbles in boiling on tube bundle using R113 at atmospheric pressure. He argued that the heat transfer coefficient within a bundle is determined from;

$$\alpha_{total} = \alpha_b \text{ for } \alpha_{total} > F\alpha_c \qquad (3)$$

or

$$\alpha_{total} = F\alpha_c \text{ for } \alpha_{total} > \alpha_b \qquad (4)$$

where α_b was the summation of the nucleate and sliding bubbles. Moreover it was inferred that tubes in the upper section of the bundle does not have any enhancement due to nucleation rather it is due to liquid convection and sliding bubbles alone, but tubes in the lower column experienced nucleation due to low quality and voidage of the fluid. They concluded that sliding bubbles play an important role in heat transfer in tube bundles and simple addition of boiling and convective mechanism are inadequate and a proper model should incorporate the quality. In another work, Cornwell and Houston [17] carried out experimental studies on a test tube within a bundle to investigate the effect of sliding bubbles during the heat transfer mechanism that occur at the outside tubes. He concluded that the enhancement was due to sliding bubbles rather than nucleation. Moreover, there was conduction through a thin film of liquid on the surface of the tube.

III. Boiling Heat transfer models in confined spaces

In compact tube bundles where there is effect of gap size, heat transfer correlations are complicated due to the dependence of gap size on the flow regimes. For large channels or gap sizes established correlations are available

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which predicts experimental results fairly. Experimental investigations such as those of Chen [18] and several others inside channels have distinguished two types of mechanisms governing the heat transfer coefficient namely nucleate and convective boiling. Correlations for flow boiling in conventional large tube have been based on nucleate and convective component heat transfer coefficient. There are three main models that exist in literature as reviewed by Gupte et al [19], superposition, asymptotic and enhancement.

Recent interests in the development of a model to predict the local boiling heat transfer coefficient in small circular tube has led to the development of a two-state model by Jacobi and Thome [20]. Jacobi and Thome [20] modeled and developed a method to predict the heat transfer coefficient in small circular tube. It was assumed that the heat transfer coefficient observed in flow boiling was due to an elongated bubble in which transient evaporation was taking place. They concluded that the heat transfer coefficient in the laminar flow of the liquid was negligible compared to the thin film coefficient thus making the model a one–zone model. Parametric studies showed that the model predicted several independent data quite well when it was assumed that the film thickness lies in the range of 10-20μm for channel size of 2.5mm.

Thome, Dupont et al [21] developed a three–zone model shown in Figure 3 which was an extension of the Jacobi [20] model to predict the heat transfer coefficient inside micro tubes. Their model predicted the heat transfer coefficient at fixed location in a channel.

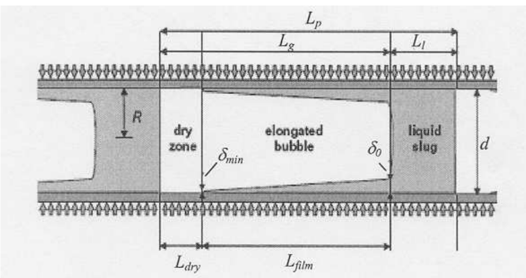

Fig 3; Diagram illustrating the three zone model of Thome, Dupont et al [21]

The model was given for the time averaged local heat transfer coefficient was given as;

$$\alpha(z) = \frac{t_l}{\tau}\alpha_l + \frac{t_{film}}{\tau}\alpha_{film}(z) + \frac{t_{dry}}{\tau}\alpha_v(z) \qquad (5)$$

The time periods used in the above model were given as;

$$t_l = \frac{\tau}{1 + \frac{\rho_l}{\rho_g}\frac{x}{1-x}}, \qquad (6)$$

$$t_g = \frac{\tau}{1 + \frac{\rho_g}{\rho_l}\frac{1-x}{x}} \qquad (7)$$

and $t_{dry\ film}(z) = \frac{\rho_l h_{fg}}{q}\left[\delta_o(z) - \delta_{min}\right]$ (8)

$$f = \frac{1}{\tau} \qquad (9)$$

The average heat transfer coefficient through the elongated bubble was obtained using the following set of equations:

$$\alpha_{film}(z) = \frac{k_l}{\delta_o - \delta_{end}}\ln\left(\frac{\delta_o}{\delta_{end}}\right) \qquad (10)$$

$$\delta(z,t) = \delta_o(z) - \frac{q}{\rho_l h_{fg}}t \qquad (11)$$

$$\frac{\delta_o}{d} = C_{\delta_o}\left(3\sqrt{\frac{\nu_l}{U_p d}}\right)^{0.84}\left[\left(0.07Bo^{0.41}\right)^{-8} + .1^{-8}\right]^{-1/8} \qquad (12)$$

$$U_p = G_{total}\left[\frac{x}{\rho_g} + \frac{1-x}{\rho_l}\right] \qquad (13)$$

IV. EXPERIMENTAL STUDY AND DISCUSSIONS OF RESULTS

An experimental study has been carried out by the authors Adom, [22] et al using a 3mm stainless steel outside diameter in a compact tube bundle. The set up consisted of a 30 tubes with the pitch to diameter ratio of 1.5. The central tubes were instrumented with to give temperature readings.

Typical results obtained from the set up with the working fluid as distilled water are shown in the Figure 4-6 for various heat fluxes.

The results showed a variation of heat transfer of heat transfer with tube position. Also photographic studies indicated that the heat transfer enhancement observed was due to the effect of confined bubbles in the space between the tubes. The effect of mass flux was negligible within the conditions.

The results presented for distilled water shows a variation of heat transfer coefficient with vapour quality as there is a cyclical variation. Visual observation shows and photographic studies shows that at heat low heat flux of 10kW/m^2, nucleate bubbles were observed forming on the tubes from tube 1 to 10. These bubbles are of similar size to the tube diameter and as a result the sliding bubbles mechanism that was responsible for the enhancement on large tube bundle could not be responsible. At high heat flux of 21 kW/m^2 , full nucleate boiling was observed on the tubes and these bubbles grow and deform in size as it passed through the spaces between the tubes. This disruption of the bubbles causes the heat transfer coefficient of the upper

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tubes to decrease as there were also intermittent partial dry out. At high vapour qualities, the heat transfer coefficient is expected to increase with quality but this is quite opposite in the results presented. The effect of mass flux on the distilled water results is relatively marginally as only a small increase. Thus bubbles are confined within the gaps of the tubes, which led to a the confinement number extended to this compact bundle in a related paper by the authors (Kew et al [20]).

Traditional kettle reboilers shows an increase in bundle effect at low to medium heat fluxes, contrary the small tube bundle shows a rise from the lower to mid sections of the bundle. In general it is suggested that in the confined bundle bubbles generated on the lowest tubes are impeded and therefore the area of the tube covered by an evaporating micro layer is greater than that which would be observed on a single isolated tube. Moreover the superimposed mass flux at the inlet of the bundle also increases the forced convective component of the heat transfer.

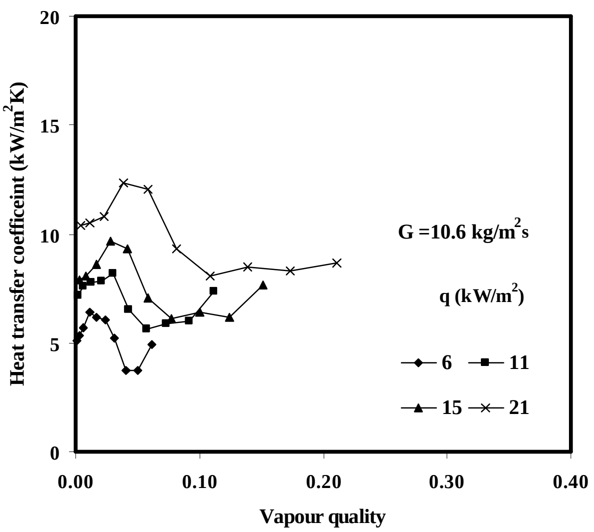

Fig 4: Heat transfer coefficient against vapor quality for mass flux of 10.6kg/m^{2}s

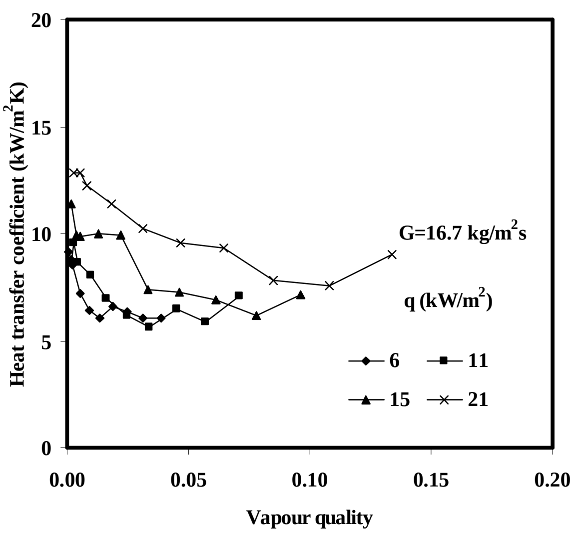

Figure 5: Heat transfer coefficient against vapor quality for mass flux of 16.7 kg/m^{2}s

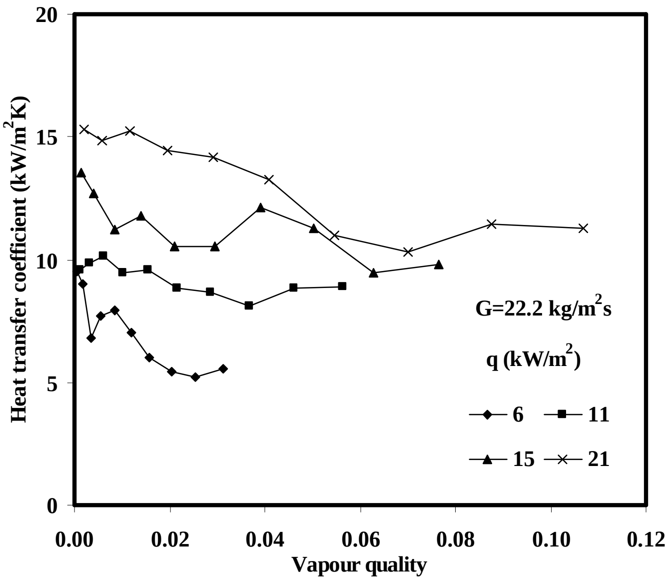

Figure 6: Heat transfer coefficient against vapor quality for mass flux of 22.7 kg/m^{2}s

This model showed that the heat transfer coefficient due to the liquid and vapor slug are negligible, but the dominant mechanism is evaporation through the elongated bubble. The above model in predicting the experimental results yielded a deviation of 35 % with experimental results which is shown in Fig 7-9.

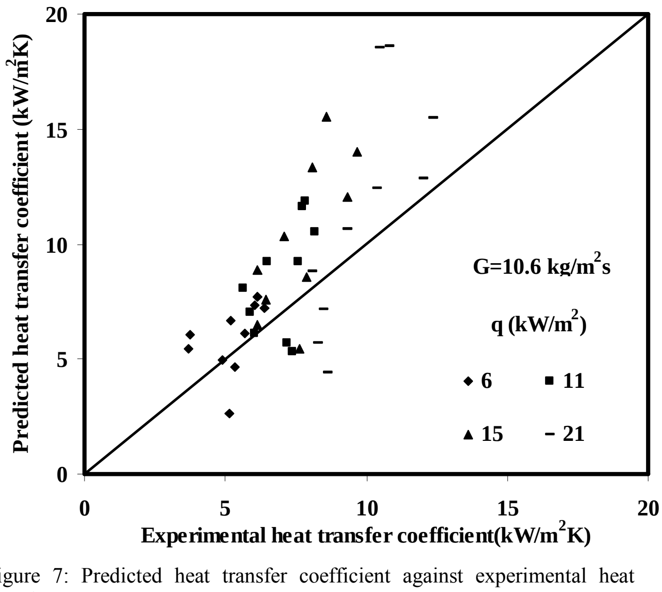

Figure 7: Predicted heat transfer coefficient against experimental heat transfer

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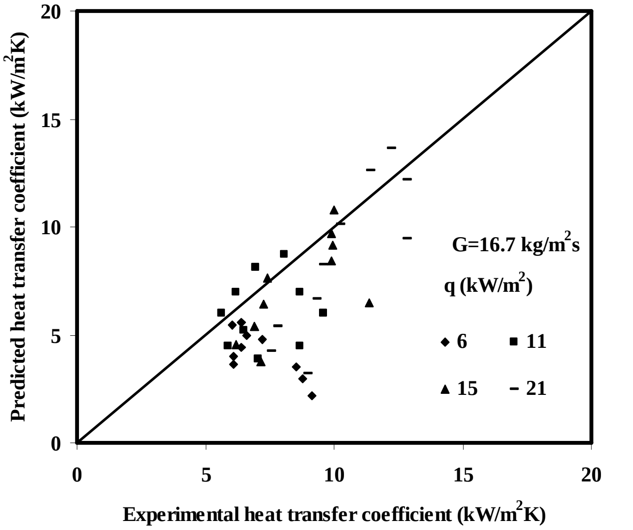

Figure 8: Predicted heat transfer coefficient against experimental heat transfer

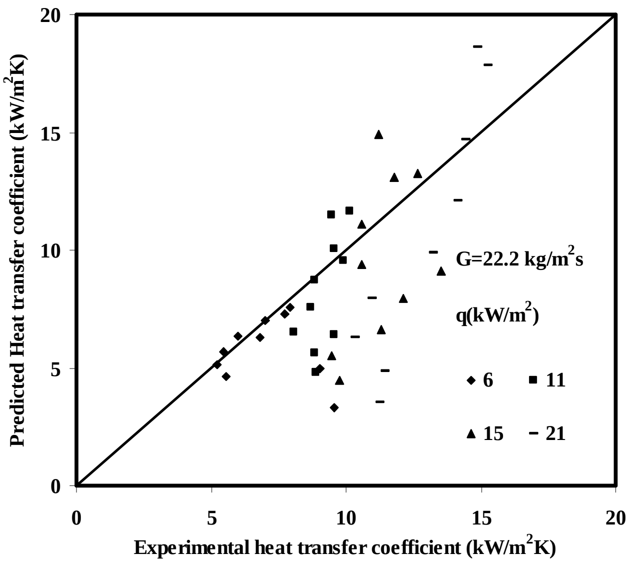

Figure 9: Predicted heat transfer coefficient against experimental heat transfer

V. Conclusions

There has been substantial study on boiling on the outside of tubes and tube bundles of diameter more than 6mm with various correlations and mechanism of heat transfer. Recent studies by the authors show that the heat transfer coefficient in a compact tube bundle is pronounced when diameter ranges is below 6mm. This pronounced increase in heat transfer coefficient is due the effect of confined bubbles and the effect of mass flux is negligible.

Future work would involve the use of computational fluid dynamics packages in comparing experimental results with that of simulation work, an analytical approach to determine the initial film layer under a passing bubble within a tube bundle.

Nomenclature

B_o	Bond number
Co	confinement number
d	tube diameter(m)
f	frequency (Hz)
g	acceleration due to gravity (m/s^2)
h	heat transfer coefficient (kW/m^2)
Nu	Nusselt number
t	time (s)
q	heat flux (kW/m^2)
Pr	Prandtl number
Re	Reynolds number
U	pair velocity (m/s)
x	quality

Greek symbols

ρ	density (kg/m^3)
μ	absolute viscosity (N/m^{2}s)
σ	surface tension (N/m)

Subscripts

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