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ENHANCEMENT OF BOILING ON A SMALL DIAMETER TUBE DUE TO BUBBLES FROM BELOW

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ABSTRACT

An experimental investigation was carried out using distilled water at nominal atmospheric pressure boiling outside horizontal tubes of diameter 3.0mm arranged one above the other. This diameter is of industrial importance in compact systems and has the interesting characteristics of being similar to the bubble size. The heat flux on the lower tube is kept constant while that of the upper tube is varied. It is shown that at low to moderate heat fluxes there tends to be an enhancement due to the influence of bubbles from the lower tube on the upper tube, but at high heat flux this effect is negated as the density of the bubbles increases. A superposition model has been proposed and is successful in explaining the enhancement and, at high heat flux the restriction to the heat transfer caused by bubbles from the lower tube.

INTRODUCTION

Boiling heat transfer finds a wide application in process industries in areas such as reboilers, evaporators, condensers and tubular heat exchangers due to its ability to transfer a large amount of energy at low temperature differences.

There is a move towards more compact equipment using smaller diameter tubes but the existing research and consequent correlations apply to large tubes of more than 6 mm diameter. This study extends the research to small tubes and attempts to evaluate the influence of bubbles on tubes when the bubble is of similar size or larger than the tube.

There has been considerable research in the area of boiling heat transfer on the outside of tube bundles and various correlations have been developed to describe the phenomenon that occurs in these geometries. Correlations such as those developed by Mostinski's (1963) and Cooper (1984) are all independent of tube diameter. An attempt to model the pool boiling heat transfer on tubes, which included the influence of the tube diameter, was made by Cornwell et al (1982) using a large data base. They concluded that the diameter has some small effect on the heat transfer coefficient for larger diameter tubes.

Recently a study by Kew et al (2002) concluded that pool boiling correlations used for larger tubes can be used with reasonable accuracy to predict the boiling heat transfer coefficient on tubes of the order 1-6mm.

Boiling on tube bundles has been studied extensively by investigators such as Palen et al (1972), Cornwell et al (1986; 1990; 1990) , Fujita et al (1987; 1997), Jensen (1988) and Chan et al (1987) , it was found that there is an increase in the heat transfer coefficient in the upper tubes due to rising bubbles from tubes below sometime termed as the bundle effect. Fujita (1990) developed a model based on micro layer evaporation to explain this bundle effect. Sliding bubbles have long been investigated as the mechanism by which this enhancement takes place. Cornwell and Schuller (1982) showed that boiling on tube columns and also on a single tube was characterized by sliding bubbles that slid around the periphery. Studies by Kenning et al (2000) have shown that these sliding bubbles do have a strong effect on the heat transfer. Cornwell et al (1992) were able to show experimentally the contribution using bubble boiling in a column of tubes.

Studies on twin tube arrangement have been carried out by Wall et al (1978), Chan (1987), Cornwell (1994), Hahne (1983; 1991) and more recently by Kumar (2002), but these investigations all considered using tubes with diameter greater than 6 mm. All these investigations confirmed the earlier assertion that there is an increase of heat transfer coefficient between upper and lower tube. Gorenflo et al (2002) notes that sliding bubbles generated on the lower surface of a tube influence both the convective and evaporative heat transfer, the effect being most pronounced at an intermediate level of bubble generation.

More recently the group at Heriot-Watt University Cornwell et al (2000) investigated the boiling mechanisms outside a column of horizontal wires and concluded that there is a small increase in the heat transfer coefficient in bubbly flow above that of nucleate boiling for the upper tubes, but vapour blanketing mitigated this effect at higher heat fluxes. To the authors' knowledge there is no published data on the effect of the lower tube on the upper tube for small tubes and wires for diameters below 3 mm. This work is pertinent to the growing interest in the area of process intensification by reducing conventional equipment.

EXPERIMENTAL APPARATUS AND PROCEDURE

The apparatus used in this experiment, shown in Fig. 1, consists of an aluminum block of dimensions 180 high by 120 wide mm connected to a condenser. In the front section there is a glass window for visualization studies. The tubes are made of stainless steel of outside diameter 3.0 mm and inside diameter 2.4 mm with a heating length of 85 mm which is connected to a terminal block whiles the centre distance between the tubes is 7mm.

A 500W DC power supply is connected independently to each of the two test tubes with a voltage tapping connected to a digital voltmeter whiles the other two tubes are dummy tubes used to keep the instrumented tubes in horizontal position. A single Type K thermocouple in each tube was used to measure the internal temperature of the tubes. There are also two type K thermocouples to measure the saturation temperature of the liquid. Electrical heaters are embedded at the back of the test section to keep the liquid at the saturation temperature.

The experiment was carried out by filling the test section with the liquid to a level of about 120 mm from the base of the test section. The heaters at the back of the test section were switched on to heat the liquid to its saturation temperature and the liquid was allowed to boil continuously for about 20 minutes to allow condensable gases to escape before experimental readings are taken.

The power supply to the tubes was switched on and the current and voltage across the tubes was measured after a steady state had been achieved (2-3minutes). The experiment was carried out by increasing the heat flux in predetermined steps and then at decreasing the heat flux through the same steps.

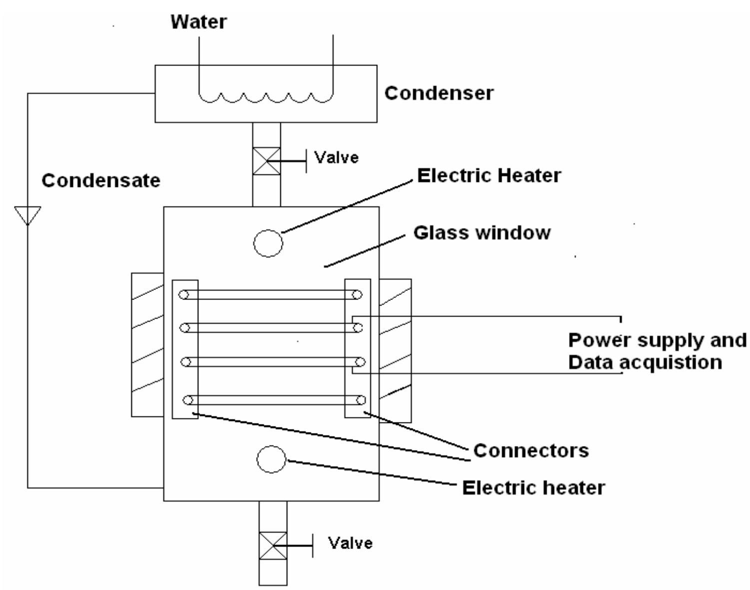

Fig.1: Schematic diagram of the experimental set-up

The heat flux on the upper tube was varied from 7-192 kW/m^2 while the heat flux on the lower tube was held at a series of different values between 0-192 kW/m^2.

Readings from the thermocouples are estimated as accurate to within 0.1^{o}C of the measured temperature and that of the digital voltmeter and the power supply are within ±3 % accurate. The relative error for the heat transfer coefficient was found to be 10%. Random error would be expected to less than this; systematic errors influence the absolute values of heat transfer coefficient calculated, but would be expected to have little effect on the enhancement.

The heat flux q was found from

$$q = \frac{VI}{\pi d_o l} \tag{1}$$

and the heat transfer coefficient is evaluated from

$$\alpha = \frac{q}{\left(T_{wall} - T_{sat}\right)} \qquad (2)$$

where T_{wall} is the surface temperature of the tube and T_{sat} is the saturation temperature of the fluid. The surface temperature is evaluated by considering the heat conduction through a cylinder at the appropriate heat flux. The results obtained with distilled water as a working fluid are shown in Figs. 2 and 3.

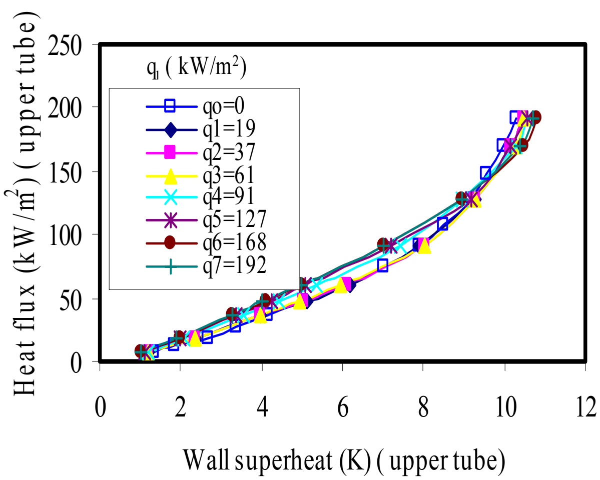

Fig 2: Variation heat flux with wall superheat with distilled water as the working fluid

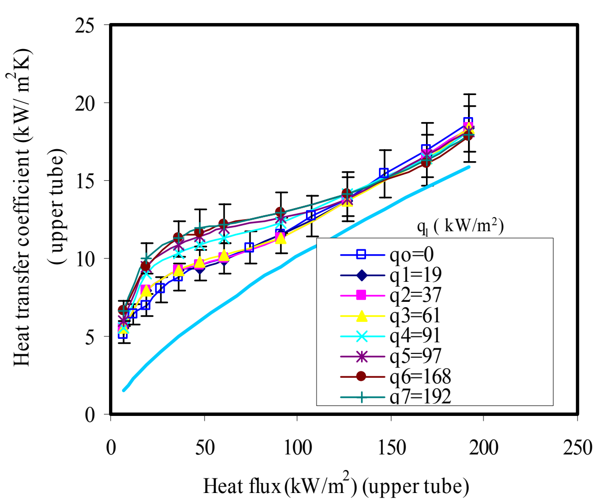

Fig 3: Variation of heat transfer coefficient with heat flux with distilled water as working fluid.

SUPERIMPOSITION MODEL FOR BOILING OF THE UPPER TUBE.

Experimental results for the two tubes indicate that the heat transfer coefficient of the upper tube is enhanced by the translating bubbles up to a particular value equating the point where the heat flux (q) yields a total enveloping of the tube. It is suggested that this is due to the effect of bubbles from the lower tube passing over the upper tube. The model, shown schematically in Fig. 4 assumes that translating bubbles envelop a proportion, p, of the length of the tube and heat is transferred by nucleate boiling from a fraction ($1-p$) of the tube. The factor p may be applied instantaneously to a length of the tube, or represent the proportion of the time during which a single point is traversed by translating bubbles. The resulting heat transfer coefficient may thus be determined from Eq. (3),

$$\alpha_1 = p\alpha_{tf} + (1-p)\alpha_{nb} \quad (3)$$

where α_1 is the mean total heat transfer coefficient for the upper tube, α_{nb} is the nucleate boiling heat transfer coefficient at the appropriate heat flux, and α_{tf} is the heat transfer coefficient through the thin film laid down by a translating bubble originating from the lower tube. Under boiling conditions without the lower tube the total heat transfer coefficient is given by the nucleate boiling alone;

$$\alpha_2 = \alpha_{nb} \quad (4)$$

The value of α_{tf} may be more than α_{nb} (as immediately after a passage of the translating bubble while a layer of the liquid is evaporating under the bubble) or less than α_{nb} (as later in the passage when the liquid layer has evaporated away or when the intensity of passing bubbles precludes sufficient liquid reaching the tubes, typically at high heat flux). The enhancement, $\Delta\alpha$, of boiling on the upper tube over that for pure nucleate boiling is given by

$$\Delta\alpha = p(\alpha_{tf} - \alpha_{nb}) \quad (5)$$

The experimental analysis allows the estimation of p by the following reasoning. Fig. 5 shows the enhancement to be zero at a heat flux of 125kW/m^2 at all values of lower tube heat flux. Since observation indicates that 'p' is not zero, then at this point $\alpha(tf) = \alpha(nb)$ and from reference to Fig 3 for lower tube q=0 yields approximately $\alpha(tf)$ =14 kW/m^{2}K.

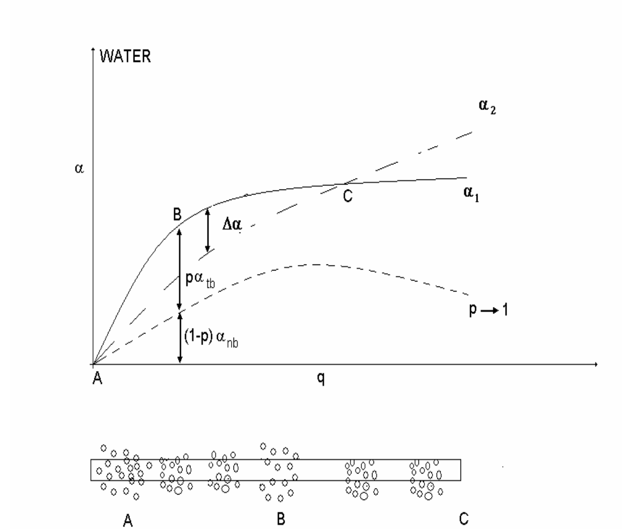

Fig. 4: Diagram showing the superimposition model for boiling of the upper tube

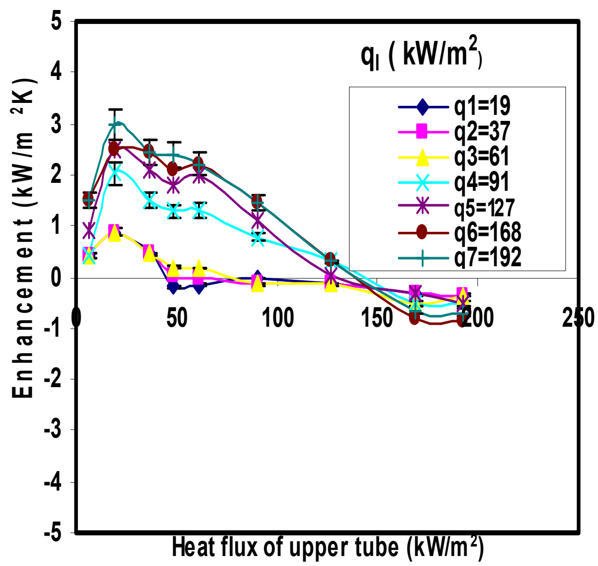

Fig 5: Variation of enhancement against heat flux of upper tube for distilled water

This value may also be estimated by theoretical consideration of the thin film under the bubbles for this part of the tube. It is assumed that the heat transfer due to translating bubbles on the same tube is essentially through a thin film rather like that under sliding bubbles on a large tube. Appendix 1 shows the analysis of conduction heat flow through such a thin film thickness δ is given as

$$\alpha_{tf} = \frac{k}{\delta} \quad (6)$$

where

$$\delta^2 = \delta_i^{\,2} - (\frac{2\Delta T k_f}{\rho_f h_{fg}})t \qquad (7)$$

From substitution of reasonable observed values from experimental and also from previous work on sliding bubbles, it is evident that for small diameter tubes $\delta \approx \delta_i$. That is the thin film suffers very little thickness change due to evaporation during the short passage time (approximately a microsecond) of the enveloping bubble. Hydrodynamic analysis of the initial thickness of layer under a bubble on a surface in water by Addlesee and Kew (2002) and thermal analysis by Kenning (2000) have established this to be about 50µm under boiling conditions at 1 atmosphere. Substitutions in Eq. (6) yields very approximate a similar value of 14 kW/m^2 K for *α(tf)* under these conditions. This corresponds reasonably well with the value estimated experimentally and gives comfort in using this experimental constant value of *α (tf)* for the estimation of *p*. The values of *p* may now be found for each experimental point by substituting measured values of α_1, *α(nb)* and *α(tf)* into Eq. (5). A plot of p against heat flux of upper tube is shown in Fig. 6.

As q_{upper} approaches 125kW/m^2, the level at which *α(nb) = α(tf), p* becomes indeterminate. At higher heat fluxes vigorous nucleate boiling tends to displace translating bubbles and the model, as presented here, is not valid. Further work is necessary to determine the influence of induced turbulence and bulk fluid movement on the heat transfer from the upper tube.

A tentative correlation has been attempted with the limited data obtained to date. Values of *p* for the range $10<q_{upper}<100$kW/m^2 have been correlated against q_{lower}. The form of the correlation, given in Eq. (8), and shown on Fig. 7, was chosen to meet the requirement that $q = 0 \Rightarrow p = 0$ and $q = \infty \Rightarrow p = 1$.

$$p = 1 - e^{(4*10^{-6}q)} \qquad (8)$$

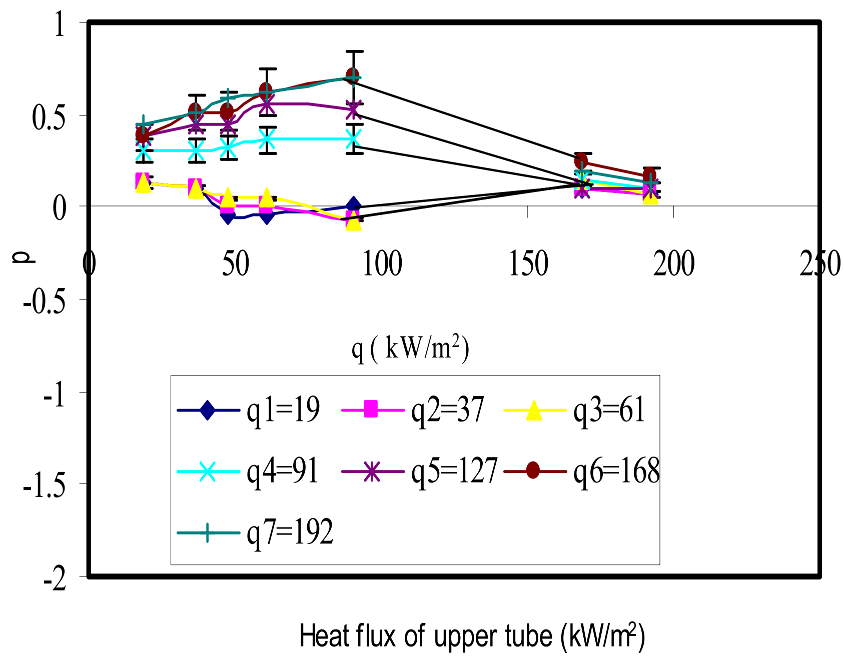

Fig. 6: A plot of p vs. q (upper tube) using experimental values

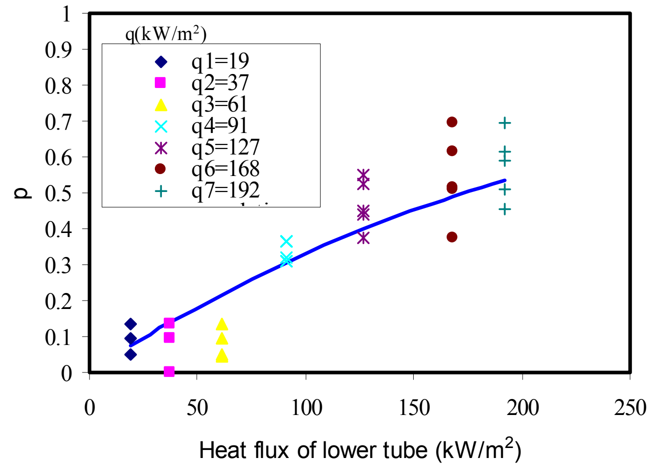

Fig. 7: A plot of p against heat flux of lower tube

The predicted enhancement due to translating bubbles was then calculated using Eq. (4) and Eq. (7) with $\alpha(nb)$taken as that for an isolated tube and a value of $14kW/m^2K$ used for $\alpha(tf)$.. A plot showing the enhancement predicted by the model against heat flux of the upper tube is shown in Fig. 8**.**

This shows reasonable agreement as expected, given the limited data employed in developing and testing the correlation. However, it demonstrates that the concepts underlying the model are valid and that the influence of lower tubes on heat transfer from tubes above them may be explained in terms of translating bubbles. Further investigation is required to determine the apparently low enhancement when the upper tube heat flux is below $37kW/m^2$.

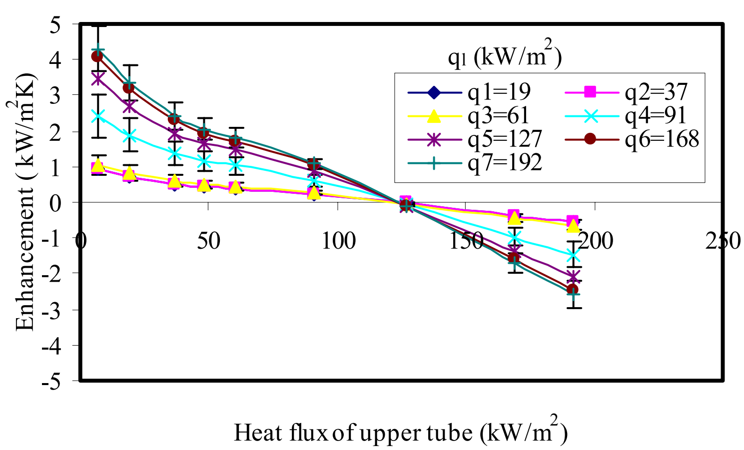

Fig. 8: Correlated plot of enhancement against heat flux for upper tube

CONCLUSIONS

An experimental investigation has been carried out during which the boiling heat transfer coefficient from a 3mm tube was measured at various heat fluxes while the heat input to a similar tube situated below the test tube was varied. The study has led to the following conclusions:

1. If the heat flux from the upper tube is held constant at a low to moderate value, the presence of a heated tube below leads to an increase in heat transfer coefficient for the upper tube.

2. At higher values of upper tube heat flux, the increase in heat transfer coefficient with lower tube heat flux is less marked, and may be zero or negative above a certain threshold.

3. A new superposition model combining nucleate boiling with heat transfer to translating bubbles has been developed which is consistent with the experimental observations at moderate heat fluxes.

4. Future work will include examination of photographic and video recordings and comparison of observed values of p with those inferred from equation 3.

Work is continuing to further develop the model by studying heat transfer from tubes in the range 1-3mm diameter and with different fluids. The model will also be tested on tube bundles.

APPENDIX 1

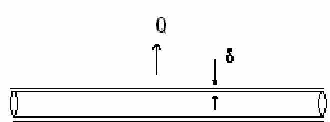

Fig A0: Heat conduction through a thin film

For heat flow through the film,

$$Q = -\frac{dm_l}{dt} h_{fg} = -\rho_f A \frac{d\delta}{dt} h_{fg} \qquad \text{(A1)}$$

Also

$$Q = q_{tf} A \qquad \text{(A2)}$$

Equating Eq. (A1) and Eq. (A2) we have

$$d\delta = -\frac{q_{tf}}{\rho_f h_{fg}} dt \qquad \text{(A3)}$$

The heat transfer through the film is by linear conduction such that

$$q_{tf} = -k_f \frac{\Delta T}{\delta} \qquad \text{(A4)}$$

Substituting equation Eq. (A3) into Eq. (A4) we have

$$\int_{\delta_i}^{\delta} \delta d\delta = \int_0^t \frac{k_f \Delta T}{\rho_f h_{fg}} dt \qquad \text{(A5)}$$

this gives

$$\delta^2 = \delta_i^{\,2} - \left(\frac{2\Delta T k_f}{\rho_f h_{fg}} \right) t \qquad \text{(A6)}$$

where δ_i is the initial thickness of film.

NOMENCLATURE

d_o	outside diameter of tube (m)
I	electrical current (A)
l	length of heated tube (m)
q	heat flux (kW/m^2)
T	temperature (K)
V	voltage (V)
p	proportion of tube covered with vapour

Greek Symbols

α	heat transfer coefficient(kW/m^2K)
Δ	change in heat transfer coefficient (kW/m^2K)

Subscripts

sat	saturation condition
wall	wall temperature of tube
nb	nucleate boiling
tf	thin film
upper	upper tube
lower	lower tube

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