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Heat transfer characteristics of the closed tube aerosyphon

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Abstract—The paper deals with heat transfer in the closed tube aerosyphon in which the upper section is cooled, the lower section is heated, and buoyancy forces are largely attributable to air bubbles introduced at the base of the device. Consistent with previous work, experimental data are used to demonstrate that substantial increases in heat transfer rates arise from increased advection and turbulence produced by the buoyant streaming. An attempt is made to describe the behaviour of the system in terms of bubble hydrodynamics. Predicted trends based on the behaviour of single bubbles are in general agreement with observations thus suggesting a simple physical basis for a complex convection system. This flow model is consistent with the observed effects of aerator geometry and aeration rate, and may be used in establishing levels of confidence for extrapolation of an empirical correlation above or below the range of the variables measured.

1. INTRODUCTION

THE TUBULAR thermosyphon, in which circulation is essentially confined to a single tube, has been studied extensively in recent decades. The device appears in two different but interrelated forms: the open system, in which only one end of the tube is open to a reservoir of the filling fluid; and the closed system, in which neither end is open, the filling fluid being completely contained within the device. The pioneer analysis of Lighthill [1], who considered single phase conditions in a vertical tube, has served as a point of departure for many related studies, notably the work of Martin [2] and Japikse and Winter [3] for the open system, and Bayley and Lock [4] and Japikse *et al.* [5] for the closed system.

When boiling takes place in the lower (heated) section of the tube and condensation occurs in the upper (cooled) section, heat transfer rates may be increased dramatically. Although no physico-mathematical model for this system has yet appeared, a considerable body of experimental data has been produced. Representative studies include those of Larkin [6], Andros [7] and Groß and Hahne [8]. Not surprisingly, the comparatively large heat transfer coefficients associated with pool boiling and film condensation are reflected in the performance of the tubular thermosyphon. There is, however, a lack of circulatory control with the result that, under normal circumstances, the refluent material introduces an additional thermal impedance high enough to prevent the phase change heat transfer coefficients for plane surfaces from being attained.

The aerosyphon is an alternative system in which buoyancy is augmented in the liquid filling by introducing a stream of gas bubbles the rate and effect of which may be imposed externally. In either open or closed form, the aerosyphon is not unlike the single phase thermosyphon, at least in terms of general circulation. Flow control provides an advantage over the natural phase change system but the absence of a latent heat effect is a disadvantage. Previous work [9] has demonstrated that heat transfer coefficients in the aerosyphon are comparable with those in the phase change thermosyphon.

There are many potential applications of the aerosyphon but current interest is mainly centred on northern development. In particular, the device would be effective in maintaining permafrost in a frozen condition and is thus a valuable tool in foundation design. It has also been proposed as a means of creating a thin wall of ice in flowing water [9] thereby providing some control of spring run off.

The exploratory work of Lock and Maezawa has established the device as a viable alternative but is limited in its scope: it presents only a restricted range of experimental data, and makes no attempt to interpret the data in terms of bubble behaviour. The present work is intended to confirm and extend the early experimental data, and to offer a plausible explanation of heat transfer performance based on our current knowledge of bubble physics as it may be applied to the complex convection system within the aerosyphon tube.

The experiments reported here have been limited to the closed tube system but it is intended that their interpretation will also provide equal understanding of the open system which may be considered as one of the two main elements of the closed system. In general, the results reveal the effect of air flow rate on overall heat transfer rate. In particular, they examine the role of bubble dynamics: in emergence of bubbles

NOMENCLATURE

A, B	coefficients
d	tube diameter [m]
g	gravitational acceleration [$m\,s^{-2}$]
h	average heat transfer coefficient [$W\,m^{-2}\,°C^{-1}$]
k	thermal conductivity [$W\,m^{-1}\,°C^{-1}$]
L	tube length [m]
m, n, p	exponents
N	number of orifices
Nu	Nusselt number
$\dot{Q}$	heat supply rate [W]
r	orifice radius [m]
R	bubble radius [m]
Ra	Rayleigh number
Re	Reynolds number
T	absolute temperature [K]
U	velocity [$m\,s^{-1}$]
$\dot{V}$	air flow rate [$m^3\,s^{-1}$].

Greek symbols

α	void fraction
β	coefficient of thermal expansion [K^{-1}]
δ	distance from tube wall to bubble ring [m]
ν	kinematic viscosity [$m^2\,s^{-1}$]
ρ	density [$kg\,m^{-3}$]
σ	surface tension [$N\,m^{-1}$].

Subscripts

a	air
b	bubble
C	cold section
d	tube diameter
H	heated section
l	liquid
m	methanol
o	orifice
r	orifice radius
∞	terminal.

from the aerator orifices; in advective augmentation in the upper and lower sections; and in the mutual interaction of bubbles.

2. EXPERIMENTAL CONSIDERATIONS

2.1. *The rig*

The experimental rig, shown schematically in Fig. 1, was essentially the same as that used in a previous investigation of the single phase behaviour of the closed-tube thermosyphon [10]. It consisted of a long vertical steel pipe, 102 mm i.d., the top section of which was cooled by a water jacket while the bottom was heated by electric-resistive metal tape wrapped over a thin layer of electric insulation (of negligible thermal resistance). In the studies reported here, the heated and cooled sections were both 1.0 m in length, thus fixing the length–diameter ratio at about 10 : 1.

The two sections were separated by an 18 mm thick ring of insulating material ($k = 0.15$ W m^{-1} K^{-1}), thus producing a thermal boundary condition at the junction which was considered to be a satisfactory compromise between field conditions (in which the insulating piece would be absent) and laboratory conditions (which, ideally, approach a step discontinuity in temperature). Field conditions usually offer insufficient control for a rigorous test of physical models whereas idealized laboratory conditions may produce data which are difficult to apply in the field. The compromise used here was successful in escaping

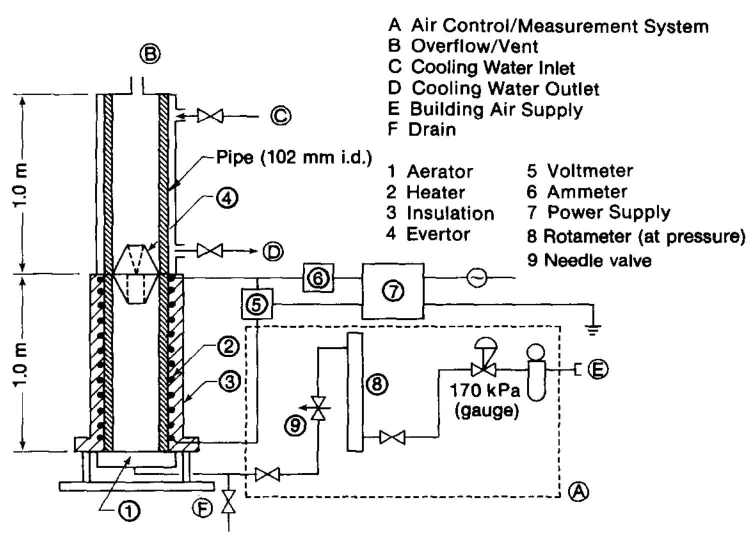

Fig. 1. Schematic of experimental rig.

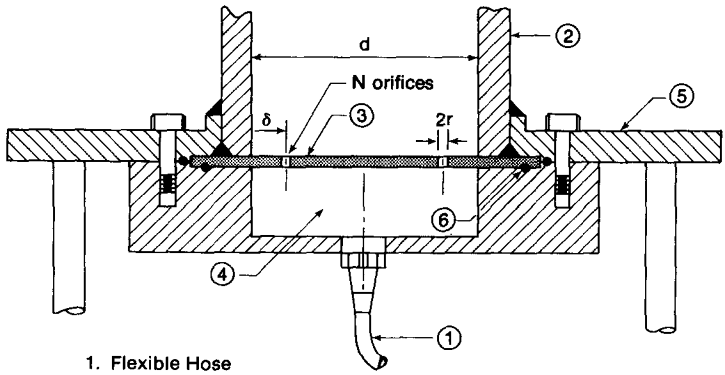

FIG. 2. Diagrammatic of aerator.

these limitations for single phase conditions [11] and therefore offered some promise for the two-phase experiments.

Also positioned in the junction region was an evertor. This device has been used on other occasions [4,9] and has the effect of everting the flow in both sections simultaneously: the ascending annulus of the lower section becomes the ascending core of the upper section, and vice versa. Within the operating limits of the aerosyphon, the evertor helps maintain circulatory control by creating a figure-of-eight pathway around which the liquid travels while the air simply moves through the system in a shallow S-shaped trajectory which is almost vertical. A wire mesh bulb was fitted to the base of the evertor.

Air was bubbled into the system through a ring of small orifices located in the base of the lower section. Details of the aerator are shown in Fig. 2. The air supply was regulated and monitored before it entered a settling chamber from which its only exit was by passage through a ring of equally-spaced, uniform-diameter holes which communicated with the liquid-filled aerosyphon cavity above. Interchangeable orifice plates enabled a variety of aerator geometries to be studied with ease. As indicated in Fig. 2, the aerator was sealed with strategically located O rings. The system was drained through the orifices and a diversion in the air supply system (see Fig. 1).

2.2. *Instrumentation*

Electrical power was supplied from the building mains to the lower section of the rig and was controlled with a 1.4 kV A Variac transformer. The gross electrical power thus supplied was calculated from the product of current and voltage measured using a Fluke 8062A multimeter. As mentioned above, heat was taken out of the system through water circulating in the cooling jacket of the upper section. Since the system was operated in the steady state mode, the net heat supply rate to the lower section was equal to the net heat removal rate from the upper section, but no attempt was made to measure the latter. Prior to beginning the tests proper, the system was filled with small chips of insulating material and calibrated in order to determine the radial heat leakage through the thermal insulation wrapped around the heating tape of the lower section. During this calibration run, in which heat loss was plotted against the difference between T_H and the ambient temperature, the cooling water temperature and flow rate were held close to the values used in the experiments proper. Typically, the loss was of the order of 10% of the power supplied.

Temperatures were measured with thermocouples located at various points throughout the rig [10]. Principal locations included regular intervals along the tube wall in both sections, and in the air, nearby in the laboratory and at the entry to the aerator. Output from the thermocouples was fed into a Fluke 2175A digital thermometer for the main (steady) data; system response data required the use of a strip chart recorder, in this case a single channel being sufficient to follow the change in overall temperature difference as a function of time.

Air was supplied from the building system via a regulator which enabled the pressure, and hence the flow rate, to be controlled. Precise setting of the flow rate was effected with a needle valve and bypass. Volumetric flow rates in the range $0.97 \times 10^{-6}\ \mathrm{m^3\ s^{-1}} < \dot{V} < 0.94 \times 10^{-4}\ \mathrm{m^3\ s^{-1}}$ were measured using a set of rotameters. Typically, the change in temperature of the air while passing through the rig was less than 1°C. Its effect on the heat balance was negligible. The main elements of the air control and measurement system are shown schematically in Fig. 1.

2.3. *Procedural observations*

Each test run in the main series followed the same procedure. Given the thermal calibration data, the

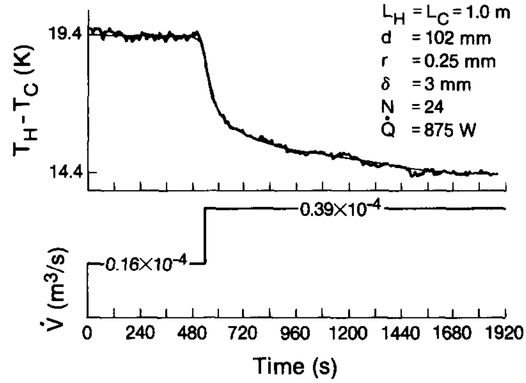

FIG. 3. Transient response of aerosyphon.

tube was fully charged with liquid and inspected for leakage. The cap was then placed on the top of the upper section and, with the cooling water flowing, the heat supply turned on and adjusted to some predetermined value. The air supply was then turned on and the flow rate set at a low value. Thermocouple output was then monitored at regular intervals until, for all practical purposes, a steady state had been reached. Experience showed that a period of about 6 h was required to limit errors attributable to transience and drift. Power supply, air supply rate and temperatures were then recorded. After adjusting the air flow rate to a slightly higher value the monitoring and recording process was repeated.

A supplementary series of transient tests was conducted in order to measure the response time of the system. Each of these consisted of setting the level of the overall temperature difference ($T_H - T_C$) on a strip chart recorder and noting the subsequent variation when the air flow rate was suddenly set to a new value. A typical response curve is shown in Fig. 3 for which it should be noted that the air adjustment, which took only a few seconds, may be regarded as a step function.

Once the above procedures had been tried out in exploratory runs, the subsequent production of raw data proceeded systematically and essentially without difficulty. However, two observations are worth making. First, the thermal boundary condition at the junction is bound to lead to a departure from isothermal walls, thus requiring an average wall temperature for use with the heat transfer data. This procedure was shown to be quite satisfactory [11] for single phase data where the overall temperature difference and the corresponding curvature in the temperature profile were greater than the values observed with the two-phase data generated here. Accordingly, the procedure was again adopted.

The second observation concerns the working fluid. This was chosen because its thermal properties are essentially constant over the temperature range envisaged. However, methanol proved to be troublesome: it attacked and pitted the tube wall and orifice plate. Most important, once the overflow at the top of the system had adjusted to a new air flow rate, the exiting air continued to carry out methanol in vapour and droplet form thus slowly reducing the effective cooled length of the upper section. Inspection of the final liquid level was therefore carried out after each test. Data which failed to pass this inspection have been excluded from the presentation and discussion below.

Table 1. Test schedule (0.97×10^{-6} m^3 s^{-1} < $\dot{V}$ < 0.94×10^{-4} m^3 s^{-1}, $L_H = L_C = 1.0$ m, $d = 102$ mm, $\dot{Q} = 875$ W)

Test	δ/d	r/d	N
1	0.250	0.0098	24
2	0.125	0.0098	24
3	0.032	0.0098	24
4	0.032	0.0054	24
5	0.032	0.0024	24
6	0.032	0.0054	48
7	0.032	0.0054	60
8†	0.250	0.0098	24

† $\dot{Q}$ = 42–850 W.

2.4. *Schedule of tests*

Given the main purpose of the experiments, the aerosyphon geometry was kept fixed with $L_H/d = 10$ and $L_H/L_C = 1.0$. The parameters used in the main test series are listed in Table 1. The schedule details the systematic study of aerator geometry as defined by the orifice ring diameter, the number of orifices and the orifice diameter. Also listed is a supplementary run to check the effect of power level.

Not listed in Table 1 are the transient test conditions. Transient response was not investigated systematically over a wide range of parameters and will therefore be treated as supplementary information which is of secondary importance.

3. DISCUSSION OF RESULTS

3.1. *Hydrodynamic characteristics*

Heat transfer rates in the single phase closed tube thermosyphon have been available for over two decades [4,5]. The more recent data of Lock and Simpson [11] confirm and extend previous work and reveal that the Nusselt number may be represented by a power law function of the Rayleigh number, length–diameter ratio and heated–cooled length ratio. For $L_H/d = 10$ and $L_H/L_C = 1.0$ it has been shown that $10^{1.25} < Nu_d < 10^{1.8}$ for $76 < Ra_d < 92$ if the fluid is methanol. This corresponds to a range of 34 W m^{-2} K^{-1} $< h <$ 132 W m^{-2} K^{-1} when 3 K $< T_H - T_C <$ 40 K.

For the same apparatus operating under the same temperature difference, the effect of aeration, to a rough approximation, may be estimated by replacing the thermal buoyancy $g\beta(T_H - T_C)$ with the mixture buoyancy $g\alpha(\rho_m - \rho_a)/\rho_m$, where α is the void fraction. This implies that the buoyancy would be multiplied by a factor of the order of 10–10^2 for an air–methanol system operating in the bubbly flow regime. If the effect of aeration is manifest principally in the advective velocity, with a corresponding enhancement of turbulence, normalization and dimensional analysis suggest that the exponent of the Rayleigh number for single phase conditions should reappear as the exponent of the modified Rayleigh number applicable to bubbly flow. The exponent has a value of approximately 0.3, thus suggesting that aeration would increase the heat transfer coefficient by about 2–4 times if circulating control is maintained. The results discussed below generally reveal increases of this magnitude and therefore suggest that a useful point of departure in attempting to model the aerated heat transfer system would be the idea of turbulence enhancement. For bubbles which do not interact, or interact weakly, the turbulence field may be attributed to wake and trajectory effects which gradually increase as bubble size increases [12]. Bubble size, which is related directly to the bubble velocity, is strongly influenced by the diameter of the orifice at which the bubble forms. For a given orifice size, the effect of increasing the air flow rate is evidently to increase the number of bubbles per unit volume, thereby increasing the void fraction. Air flow rate thus controls the mixture buoyancy which exerts a strong influence on the mixture velocity; this in turn influences the intensity of turbulence.

For isolated bubbles moving in a shallow, quiescent layer of liquid, the relations between orifice diameter, bubble diameter and terminal velocity are simple and well established [12–14], as is the relation between bubble diameter and air flow rate. For such conditions, it is reasonable to expect a monotonic relation between air flow rate and both terminal velocity and void fraction. This implies a monotonic relation between air flow rate and mixture velocity, but since the latter directly determines the advective flux it is evident that the heat transfer rate should be a monotonically increasing function of air flow rate.

This expectation would appear to be consistent with intuition. The particular relationships mentioned above are usually power laws for the bubbly flow regime and thus the form $h \propto \dot{V}^p$ is an obvious starting point in the search for a heat transfer correlation. The closed tube aerosyphon, however, is a more complex system in which other factors and effects come into play. The bubbles rise, not in a quiescent liquid, but in a non-uniform velocity field adjacent to a solid and they will not attain their terminal velocity immediately. After travelling some distance they will begin to interact with each other, to coalesce, and, in the upper section, may form bubbles of size comparable with the tube diameter. Such effects are more pronounced in longer tubes. The aerator itself will exert an influence through the size, number and positioning of the orifices. Finally, the air flow rate may eventually reach values beyond the bubbly flow regime.

3.2. *The effect of air flow rate*

The dependence of heat transfer rate on air flow rate is clearly shown in Figs. 4–6, and again in Figs.

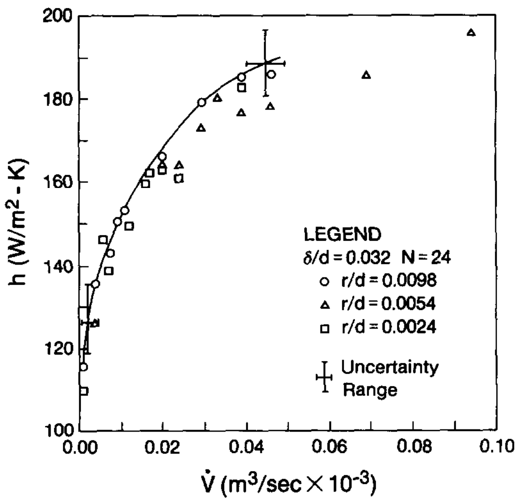

FIG. 4. Effect of orifice diameter on heat transfer coefficient.

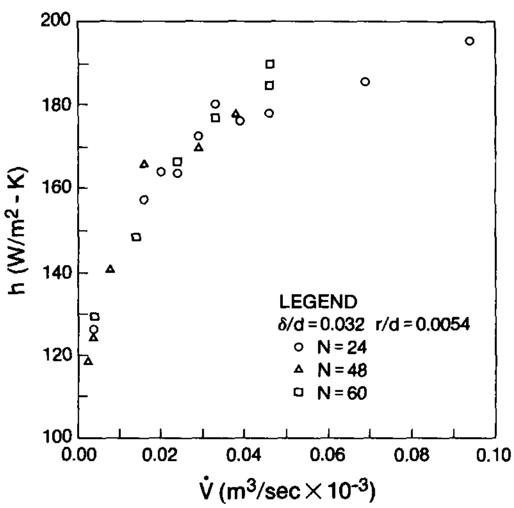

FIG. 5. Effect of number of orifices on heat transfer coefficient.

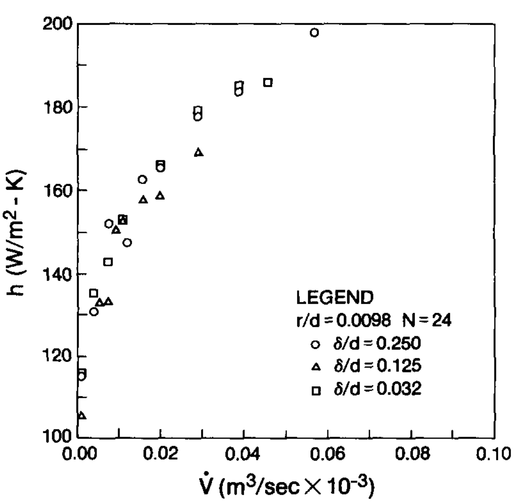

FIG. 6. Effect of orifice ring diameter on heat transfer coefficient.

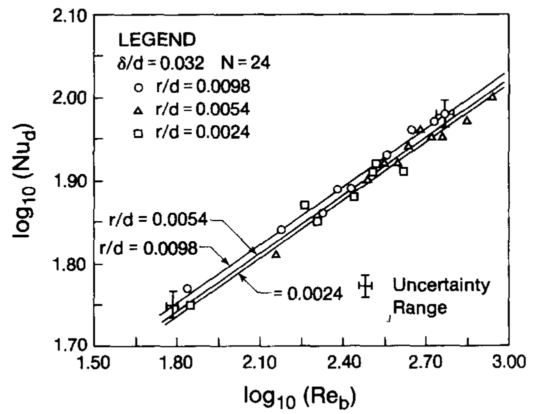

FIG. 7. Effect of orifice diameter ratio on non-dimensional heat transfer rate.

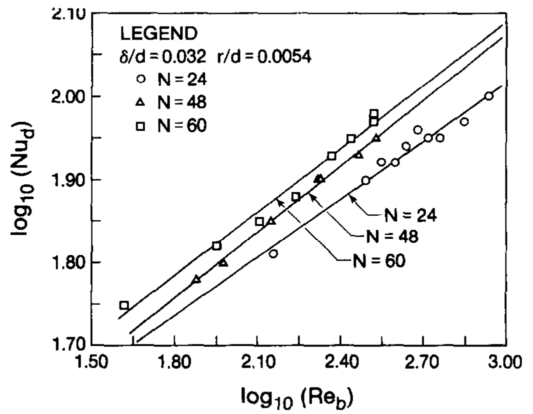

FIG. 8. Effect of number of orifices on non-dimensional heat transfer rate.

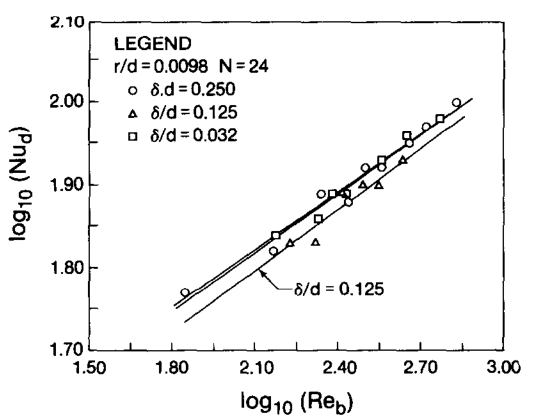

FIG. 9. Effect of orifice ring diameter ratio on non-dimensional heat transfer rate.

7–9. As expected, the introduction of bubbles into the rising lower annulus and upper core has the effect of increasing the mean mixture velocities and enhancing the vigour of turbulence. This effect increases rapidly when the aeration rate is small, thus indicating the substantial changes in mean buoyancy which are possible even with a few dispersed bubbles. As the aeration rate is increased the marginal effect on buoyancy decreases with increasing void fraction, as would be expected, although there is not enough evidence presented to draw conclusions applicable to very high void fractions.

In addition to the role of buoyancy, it appears that the effect of the evertor is to maintain circulatory control, at least to the extent of minimizing interaction between the annular and core regions. In the lower section, the natural tendency of the bubbles to move radically inwards was partly offset by the inverted fine mesh bulb extending downward from the base of the evertor. This did not prevent counter-current mixing entirely, at least not for this particular tube length, but it did facilitate the downward passage of air-free liquid from the upper annulus while denying access to upward-moving bubbles which have strayed across into the lower core. Obviously, no such measure is necessary on the top of the evertor.

The visual observations of Lock and Maezawa [9] suggest that the h–$\dot{V}$ results obtained are in the bubbly flow regime in which both mixture advection and turbulence intensity increase with air flow rate. These appear to be the principal cause of an increased heat transfer rate. In the lower section, it is to be expected that the thickness of the annular 'boundary layer' would tend to decrease with an increase in buoyancy (aeration rate) but the oscillatory behaviour of faster bubbles, together with their natural migration away from the tube wall, produces a countervailing effect which appears to dominate [9]. Bubble motion generates highly turbulent mixing adjacent to the lower tube wall and thus promotes a high radial heat flux, compatible with the mixture advective velocity.

In the upper section, such an argument does not apply because the descending annulus is essentially bubble free. However, the descending annulus is undoubtedly highly turbulent and does have a higher thermal conductivity than the bubble mixture; these features offer some compensation. In addition, the core of the upper section is populated with bubbles the diameters of which continue to increase with height, and the continuing interaction and coalescence of which may well generate higher mixture velocities than in the lower section core. If so, the downward annular liquid velocities would receive a further augmentation.

Clearly, the total heat flux must be the same in both sections but their heat transfer coefficients may be different. Some indication of their relative magnitudes may be obtained from Fig. 5 of ref. [9]. Limited though the data are, this figure displays heat transfer results for a closed aerosyphon with and without an evertor and reveals that the non-everted flow result is about 30% less than the everted value. In the absence of an evertor the upper section does not have a focused aerated core and may therefore be expected to exhibit a significantly higher impedance than the upper section of an everted flow. The observed result could not occur if the upper impedance was significantly less than the lower impedance, thus requiring that the latter be less than or equal to the former when the flow is everted. Only a detailed examination of the two temperature fields would uncover the precise details.

3.3. *The effect of aerator geometry*

Figures 4 and 7 illustrate the effect of aerator orifice diameter on heat transfer. As noted earlier, the diameter of the orifice is directly related to the diameter of the bubbles released. Bubble diameter in turn is directly related to terminal velocity which, in general terms, controls turbulence and advection. It is therefore to be expected that an increase in orifice diameter would improve the rate of heat transfer if all other conditions remained identical. Figures 4 and 7 do indeed reveal this expected tendency but the results deserve closer analysis.

For a given total air flow rate, an increased orifice diameter implies no change in the flow rate for any given orifice but since the orifice area is increased the mean air velocity will be reduced in the same proportion. This change is reflected in a decreased bubble frequency and a slight decrease in bubble diameter. It appears from Figs. 4 and 7 that these two effects are smaller than the main effect of orifice diameter. In the range of conditions studied, the net effect of increasing orifice size appears to be an increase in the scale and intensity of turbulence and perhaps an increase in mixture velocity. The simple, quasisteady expression relating bubble and orifice diameters may not apply exactly under these modified conditions but it does suggest that R will be proportional to r raised to a small power, thus implying an effect which will become increasingly smaller as the orifice diameter becomes increasingly larger. Figures 4 and 7 are consistent with this tentative hypothesis, and may in fact illustrate conditions in which the marginal effect has become small enough that it is virtually offset by the minor adverse effects mentioned above; behaviour may be asymptotic. As the error bars indicate, the effect is clouded by experimental error.

The effect of the number of orifices in the aerator is shown in Figs. 5 and 8 which reveal, as anticipated, that increasing the circumferential bubble density improves the heat transfer rate. With only a few orifices, the rising bubble streams will be separated by regions in which the mean advective velocity and turbulence level will be smaller. Increasing the number of orifices has the effect of reducing these intermediate regions, but the resulting improvement becomes smaller as the number of orifices becomes larger. It is to be expected that the improvement would depend upon the mutual reinforcement of adjacent bubble streams, and would therefore diminish when the lateral influence of the streams overlaps substantially.

This decrease in marginal improvement is partly offset by another effect. For a given air flow rate, increasing the number of orifices obviously decreases the rate for each orifice, and therefore the bubble frequency and bubble size are reduced correspondingly. As in Figs. 4 and 7, this small secondary effect may become increasingly important, relatively, and may ultimately produce an asymptotic or extremal relationship, although no evidence of this is apparent in Figs. 5 and 8.

The effect of the third aerator parameter, the orifice ring diameter, is shown in Figs. 6 and 9 from which it appears that the effect is not monotone. The explanation of this fact may be quite complex and certainly involves several factors about which little is known. The slight reduction noted for the intermediate value of δ/d is not regarded as an extremum, given the accuracy of the data. To a first approximation, the results imply that orifice ring diameter has no effect on heat transfer rate.

It was noted earlier that the proximity of the wall may exert an influence on bubble dynamics. This is especially true for the lower section in which the bubbles populate a narrow region of relatively high velocity gradient. Within this region, the effect of a bubble moving with a given vertical velocity is likely to be greatest adjacent to the wall in a narrow film within which the flow is either laminar or only intermittently turbulent. The effect of the bubble is to disturb and mix this relatively quiescent film and thus increase the local heat transfer rate. On the other hand, the wall exerts a damping effect on the bubble's primary and secondary motion and thereby reduces its impact on the flow field.

Changing the orifice ring diameter evidently brings two opposing effects into play. The two effects produce no net change in heat transfer rate, although the picture is complicated somewhat by several other factors. The tendency for bubbles to migrate radially inwards, for example, tends to keep the bubbles out of the laminar film region; this tendency may be largely independent of orifice ring diameter. Natural oscillation and spiralling in the bubble trajectory results in a radial span of influence essentially defined by the width of the rising annulus. Depending upon the air flow rate, and the corresponding bubble radii and terminal velocities, this radial span of influence is not likely to be altered by orifice ring diameter, at least for the range of conditions studied here. The picture may change for air flowing very slowly through a ring of very small orifices. Similarly, it is conceivable that longer tubes may exhibit greater interference between the annulus and the core, at least in the lower section, in which case the orifice ring diameter may become crucially important in maintaining circulatory control.

3.4. *Empirical correlation*

From the above discussion it appears that analysis and interpretation of the heat transfer data may be couched in terms of bubble behaviour. In particular, variations in system behaviour closely follow expectations based upon the hydrodynamic characteristics of single bubbles suitably modified to reflect the complexity of an aerosyphon. Such explanations are

Table 2. Empirical test data

Test	$Nu_d = A\,Re_b^m$ log A	m	log B	m	$Nu_d = B\,Re_o^m$ B_o	n_1	n_2	n_3
1	1.32	0.237	1.69	0.131	62.73		0	
2	1.28	0.246	1.67	0.136	59.60		0	
3	1.33	0.233	1.70	0.129	63.18		0	0.16
4	1.32	0.234	1.64	0.133	60.58	0.15		0.16
5	1.32	0.234	1.61	0.129	63.57			0.16
6	1.28	0.265	1.66	0.144	59.94	0.15		
7	1.33	0.255	1.69	0.140	61.10	0.15		
	$m = 0.24 \pm 0.01$			$m = 0.14 \pm 0.005$, $B_o = 61.53 \pm 1.5$, $B = B_o\,(N)^{n_1}\,(\delta/d)^{n_2}\,(r/d)^{n_3}$				

$Nu_d = hd/k_m$; $h = \dot{Q}/\pi d L_H (T_H - T_C)$.

speculative at this stage but they are consistent with ref. [9] and preliminary observations in a comprehensive visual study to be published in due course. Only a detailed study of bubble dynamics above intense line and core sources would test the hypothesis completely. The flow model outlined above is adequate for present needs.

As noted, the principal effect of aeration is advective augmentation. If the principal cause is bubble motion then it follows that the heat transfer rate should correlate with the quantities which describe bubble motion. Central in these descriptors is the bubble Reynolds number Re_b which not only contains the bubble size and velocity but determines the local flow regime around the bubble, its trajectory and wake in particular. Figures 7–9 show the data of Figs. 4–6 respectively, replotted in terms of the tube Nusselt number and bubble Reynolds number. It is immediately apparent that our expectation is realized and that the data correlate well in the form

$$Nu_d = A\,Re_b^m \tag{1}$$

where $Re_b = 2U_\infty R/\nu$ is calculated according to ref. [13] and $Nu_d = hd/k_m$. Table 2 summarizes the findings. The data are seen to lie largely in the range $10^2 < Re_b < 10^3$ which suggests that the flow was in the upper reaches of the bubbly flow regime [13]. For $Re_b < 10^2$, the single bubble basis of the flow model would become increasingly accurate and thus establishes confidence in extrapolation down into this region. For $Re_b > 10^3$, the model would become increasingly inaccurate, especially when bubble interaction and higher void ratios in the core of the upper section begin to generate large slugs of air.

The above correlation supports the flow model suggested and thus sheds light on the physics of the system but it suffers from the obvious practical disadvantage that the bubble Reynolds number is not ordinarily known in a given application. Earlier discussion attempted to outline the connection between the bubble radius, velocity (hence Re_b) and void fraction, on the one hand, and orifice radius, together with air flow rate, on the other. A correlation of the form given by equation (1) therefore suggests a more practical correlation in the form

$$Nu_d = B\,Re_o^m \tag{2}$$

in which $B(d, \delta, r, N)$ is determined from the aerator geometry, $Re_o = 2U_o r/\nu_a$ and $U_o = \dot{V}/\pi r^2 N$.

Each of the aerator geometrical effects discussed earlier may be incorporated separately into the coefficient B by making the assumption

$$B = B_o N^{n_1} \left(\frac{\delta}{d}\right)^{n_2} \left(\frac{r}{d}\right)^{n_3}. \tag{3}$$

Although the data are limited, it was found that $m = 0.14 \pm 0.005$, $B_o = 61.5 \pm 1.5$, $n_1 = 0.15 \pm 0.01$, $n_3 = 0.16 \pm 0.02$ and $n_2 = 0 \pm 0.04$. A plot of Nu_d/B vs Re_o is shown in Fig. 10. The remarks regarding extrapolation given above apply equally to this curve. The exponents in equation (3) fit the data but do not have a theoretical basis; however, they are consistent with the physical model of bubble-driven advective augmentation.

Finally, it is worthwhile examining the water data of Lock and Maezawa in the light of the above methanol data. Fluid properties, like geometrical properties, are not expected to alter the exponent of the Reynolds number but they may alter its coefficient in equations (1) and (2). The magnitude of the change is difficult to predict for such a complex system but a qualitative estimate may be found, at any given flow rate, from

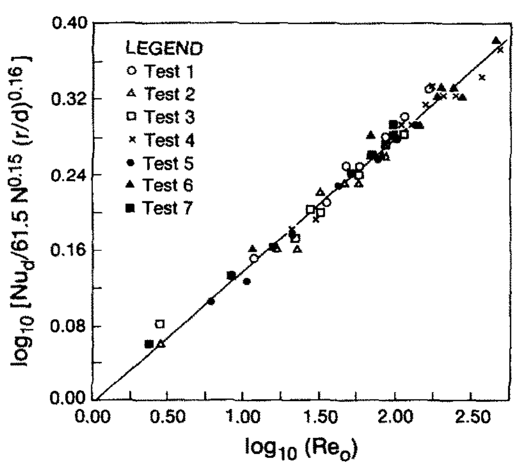

FIG. 10. Empirical correlation.

alterations in the Reynolds number itself. For a single bubble forming quasi-steadily, the radius is proportional to $[\sigma/(\rho_l-\rho_a)]^{1/3}$; altering this produces a corresponding change in terminal velocity. The momentum diffusivities of the two fluids are also different.

The overall change in Re_b corresponding to a change from methanol to water is found to be much less than an order of magnitude, and not enough to explain the difference between the present data and those of Lock and Maezawa. However, the latter data are very limited, and conditions were not identical in the two experiments. More important, the use of 'pure' thermophysical properties is only a crude approximation; it is known, for example, that the purity of the liquid may make an enormous difference in the flow regime [13] and may thus cause a considerable alteration in heat transfer rate. Since no attempt was made to control the cleanliness of either of the two systems, it would appear that quantitative reconciliation of these particular data sets is not possible. In general, the type and amount of each impurity would have to be specified; otherwise system performance may only be predicted within upper and lower bounds [13] which may, however, be close enough for many field purposes.

3.5. *Transience*

The curve shown in Fig. 3 was used to examine the nature of the system response to a step change in the air flow rate. A plot of the slope of this curve vs the temperature difference was found to be linear, thus identifying a first-order system. The time constant thus measured was 76 s, consistent with rough estimates from a number of other tests, and compatible with the 6 h period used to ensure 'steady state' conditions.

The hydrodynamic response of the system overall is clearly much more rapid than the thermal response. The latter, however, is governed not only by the thermal capacity of the fluids but by the surrounding materials also. The thermal insulation around the lower section in particular was very slow to respond. The intrinsic thermal response of the aerosyphon is evidently much closer to the hydrodynamic response.

4. CONCLUSIONS

In general terms, the experimental results presented here reveal that the aerosyphon exhibits heat transfer rates several times greater than those for the corresponding thermosyphon operating under single phase conditions. This fact leads to the belief that the overall effect of aeration is one of augmentation manifest through increases in the advective velocities and the level of turbulence. Consistent with the earlier findings of Lock and Maezawa, the use of an evertor in the region where the heated and cooled sections meet appears to have maintained circulatory control and thus limited the thermal impedance caused by refluence.

Despite the complexity of the system, the data suggest that its behaviour is largely governed by the hydrodynamic characteristics of individual bubbles. Although no internal temperature measurements were taken, the evidence suggests that heat transfer coefficients in the heated section were of the same order as in the cooled section. In the lower section, augmentation adjacent to the wall has been attributed to bubble motion combined with an increase in void fraction. The descending core is largely, if not entirely, liquid. In the upper section, the additional buoyancy in the aerated ascending core is evidently the cause of high velocities and turbulence in the descending annulus which may have been penetrated by some wayward bubbles.

Using the established, direct relations between bubble motion and orifice size and the attended relation between void fraction and aeration rate, it has been possible to reconcile the data with expectations based on the behaviour of single bubbles. In particular, the effect of aerator geometry, as defined by the orifice ring radius, orifice diameter and number of orifices was found to be generally consistent with predicted trends. Since the effect is largely limited to the lower section, it was not expected to be large. For practical convenience, the geometrical trends have been represented by simple power law relationships but it is believed that they are actually more complex; extrapolation beyond the range of aerator geometries studied is not recommended.

The general effect of aeration on heat transfer rate is adequately represented by a power law relation in which the exponent is independent of aerator geometry. In terms of the bubble Reynolds number, the results lend support to the basic flow model proposed and reveal that the system was operating at the upper end of the bubbly flow regime. When plotted in terms of the orifice Reynolds number, the Nusselt number again correlated well when the effect of aerator geometry was factored out empirically. It is suggested that extrapolation down to lower Reynolds numbers should present no difficulty whereas extrapolation above the tested range should be treated with caution. The effectiveness of the device now appears to be well established but its performance under other conditions, different fluids and geometries in particular, cannot be ascertained without further work.

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CARACTERISTIQUES DU TRANSFERT DE CHALEUR PAR AEROSIPHON DANS UN TUBE FERME

Résumé—On étudie le transfert thermique par aérosiphon dans un tube fermé avec extrémité supérieure froide, l'extrémité basse étant chauffée, les forces d'Archimède étant largement attribuables à des bulles d'air introduites à la base du système. Cohérentes avec un travail antérieur, des données expérimentales sont utilisées pour démontrer que des accroissements substantiels de transfert de chaleur sont dus à l'augmentation de l'advection et de la turbulence produites par l'entrainement provoqué. Un essai est fait pour décrire le comportement du système en termes d'hydrodynamique de bulles. Des configurations prédites et basées sur le comportement de bulles uniques sont en accord général avec les observations, ce qui suggère une base physique simple pour un système complexe de convection. Ce modèle d'écoulement est cohérent avec les effets observés de géométrie d'aérateur et il peut être utilisé dans l'établissement des degrés de confiance pour l'extrapolation d'une formule empirique au-dessus ou au-dessous du domaine des variables mesurées.

EIGENSCHAFTEN DES WÄRMEÜBERGANGS IN EINEM GESCHLOSSENEN AEROSYPHON

Zusammenfassung—Die Veröffentlichung befaßt sich mit dem Wärmeübergang in einem geschlossenen Aerosyphon. Der obere Teil wird gekühlt, der untere Teil wird beheizt, und die Auftriebskräfte sind weitgehend den Luftblasen zuzuschreiben, die am unteren Ende der Apparatur eingeblasen werden. Übereinstimmend mit vorausgegangenen Arbeiten zeigen die experimentellen Daten, daß die übertragene Wärmemenge durch Advektion und Turbulenz infolge der Auftriebsströmung wesentlich zunimmt. Es wird ein Versuch unternommen, das Systemverhalten mit Gleichungen der Blasen-Hydromechanik zu beschreiben. Vorhergesagte Tendenzen, basierend auf dem Verhalten von Einzelblasen, stimmen im allgemeinen mit den Beobachtungen überein. Es wird damit eine einfache physikalische Basis für ein umfassendes konvektives System vorgeschlagen. Dieses Strömungsmodell gibt die beobachteten Einflüsse der Geometrie und der Stärke der Belüftung wieder. Es kann benutzt werden zum Errichten von Vertrauensgrenzen für die Extrapolation einer empirischen Beziehung außerhalb des durch Messungen abgesicherten Variablenbereichs.

ХАРАКТЕРИСТИКИ ТЕПЛООБМЕНА В ЗАМКНУТОМ ТРУБЧАТОМ АЭРОСИФОНЕ

Аннотация—Рассматривается теплообмен в замкнутом трубчатом термосифоне, охлаждаемом сверху и нагреваемом снизу. Действие подъемных сил обязано, в основном, пузырькам воздуха, вводимым в основание установки. Как и в предыдущей работе, с помощью экспериментальных данных показано, что существенное увеличение интенсивности теплообмена происходит за счет увеличения адвекции и турбулентности, вызванной восходящими струями. Предпринята попытка описать поведение системы с помощью гидродинамики пузырьков. Результаты расчетов, основанных на поведении единичных пузырьков, в основном согласуются с экспериментальными данными, что позволяет предложить простую физическую модель для сложной конвективной системы. Модель течения адекватно учитывает влияние конфигурации аэратора и скорости аэрации и может использоваться для установления пределов достоверности экстраполяции эмпирической зависимости за пределами диапазона измеренных значений переменных.