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International Journal of Heat and Mass Transfer 49 (2006) 3180–3189

International Journal of HEAT and MASS TRANSFER

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Effect of inundation for condensation of steam on smooth and enhanced condenser tubes

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Received 13 November 2005

Abstract

The paper presents new measurements on the effect of inundation during condensation of steam in tubes banks. Most of the data relate to wire-wrapped enhanced tubes but measurements are also reported for low-finned and smooth tubes. The technique of artificial inundation has been used where liquid is supplied above a single horizontal test condenser tube to simulate condensate draining from higher tubes. Inundation rates have been used to simulate a column of up to almost 30 tubes. The surface temperature of the condenser tube was measured at four locations around the tube using buried thermocouples. The heat transfer and hence condensation rate was determined from the mass flow rate and temperature rise from coolant. The temperature and flow rate of the simulated inundation was carefully controlled. All tests were carried out at atmospheric pressure with constant vapour downflow approach velocity and constant coolant flow rate. For the given coolant and vapour flow rates and temperatures (same for all tests), and in the absence of inundation, the vapour-side heat-transfer coefficient for the finned tube was around four times that of the smooth tube while the heat-transfer coefficient for the wire-wrapped tubes was independent of winding pitch and around 30% higher than for the smooth tube. For inundation conditions the smooth tube data are in line with the widely used Kern equation relating the heat-transfer coefficient to the depth of a tube in the bank. The heat-transfer coefficient for the finned tube was virtually unaffected by inundation up to the maximum used which was equivalent to a depth of about 20 finned tubes in a bank. At this depth level the heat-transfer coefficient for the finned tube was around six times that of the smooth tube. For the wire-wrapped tubes the deterioration in performance with increasing inundation was least for the smallest winding pitch used for which the heat-transfer coefficient fell by around 9% at an equivalent depth in a bank of 25 tubes. At this depth level the heat-transfer coefficient for the wire-wrapped tube was almost twice that of the smooth tube.

Keywords: Condensation; Inundation; Wire-wrapped tube; Low-finned tube; Enhanced heat transfer; Condenser; Steam

1. Introduction

For a bank of horizontal condenser tubes *inundation* is a term used to describe the effect of condensate falling from higher tubes in the bank on the performance of a given row. The condensate film thickness around lower tubes increases due to inundation and consequently the heat-transfer coefficient decreases. The condensate as it leaves a given tube may be in the form of droplets, columns or a continuous sheet depending on the inundation flow rate and the vertical separation of the tubes. It is difficult to isolate the effect of inundation in a tube bank from other factors, namely decrease of vapour shear stress on the condensate film and increase in non-condensing gas concentration as the condensation process proceeds. The problem has been studied by many investigators.

Laboratory experiments with even small tube banks require high vapour generation capacity. The more convenient approach using artificial inundation supplied above a single condenser tube or above the uppermost tube of a small column of horizontal tubes has been widely used. In measurements of this kind it is important to set and control the inundation supply temperature correctly,

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0017-9310/$ - see front matter
doi:10.1016/j.ijheatmasstransfer.2006.02.003

Nomenclature

B	dimensionless constant in Eq. (11)
d	outside diameter of the tube
d_w	wire diameter
d_{ct}	pitch circle diameter for location of thermocouples (Fig. 4)
F	$\frac{\mu d h_{fg} g}{u_\infty^2 k \Delta T}$ indicating the relative importance of gravity and vapor velocity, see Eq. (10)
g	specific force of gravity
h_{fg}	specific latent heat of evaporation
k	conductivity of condensate
N	number of tubes in a tube bank or column
Nu	Nusselt number, $\frac{q}{\Delta T}\frac{d}{k}$
p	pitch of winding
q	heat flux based on smooth tube area
q_0	heat flux for tube with no inundation
q_1	heat flux of the uppermost tube in a tube bank
q_N	heat flux for Nth tube in a vertical column
$q_{m,N}$	mean heat flux for N tubes in a vertical column
Re	"two-phase Reynolds number", $u_\infty \rho d/\mu$
T_v	vapor temperature
T_s	saturation temperature
T_w	local outside tube wall temperature
T_{wo}	mean outside tube wall temperature
T^*	inundation temperature, see Eqs. (9) and (A.1)
ΔT	vapour-to-surface temperature difference
u_∞	vapour approach velocity
z	dimensionless film thickness, defined in Eq. (2)

Greek symbols

ρ	density of condensate
ρ_v	density of vapor
μ	dynamic viscosity of condensate
σ	surface tension
α_0	heat-transfer coefficient for a tube with no inundation
α_1	heat-transfer coefficient for a tube or the uppermost tube in a tube bank given by Eq. (3)
α_N	heat-transfer coefficient for Nth tube in a tube bank
$\alpha_{m,N}$	mean heat-transfer coefficient for N tubes in a tube bank

particularly when using enhanced tubes. The correct inundation supply temperature is that at which inundation would impinge on a tube in an actual bank. The correspondence between the rate at which simulated inundation is supplied and equivalent depth in a tube bank also needs careful consideration.

Many investigations have been made using smooth condenser tubes and data for condensation on small banks of low-finned tubes have become available more recently. While there is appreciable variation among results it is clear that, with low-finned and wire-wrapped tubes, the enhanced heat-transfer coefficients are degraded less under inundation than are smooth tubes. The present experimental investigation formed part of a study of condensation on wire-wrapped tubes, see Murase et al. [1], and was prompted by a report, in a literature survey by Marto [2,3] citing the work of Brower [4] for condensation of steam, that heat-transfer coefficients for wire-wrapped tubes, while being significantly smaller than for low-finned tubes, were less degraded by inundation (see Fig. 1).

The problem of inundation was first analysed theoretically for smooth tubes by Nusselt [5] who considered an in-line column of horizontal tubes each with the same and uniform surface temperature. For the case of laminar condensate flow, and neglecting inertia and convection in the condensate film, whose thickness was assumed much smaller than the tube radius, Nusselt obtained the dependence of film thickness on angle around the tube:

$$z = \frac{2}{\sin^{4/3}\theta}\left\{\int_0^\theta \sin^{1/3}\theta \, d\theta + C\right\} \tag{1}$$

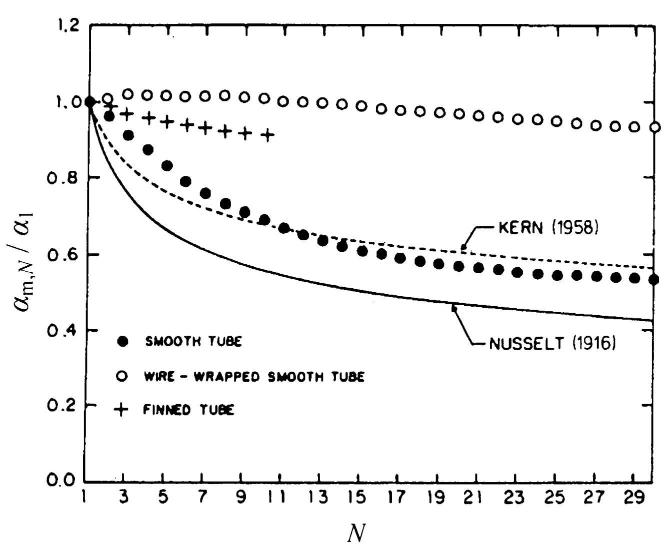

Fig. 1. Dependence of normalised heat-transfer coefficient on tube depth in a column, from Marto [2,3].

where

$$z = \left(\frac{\rho(\rho-\rho_v) g h_{fg}}{\mu d k \Delta T}\right)\delta^4 \tag{2}$$

For the top tube in a bank or column the constant C has to be zero for finite film thickness at the top. This leads to the well-known result for a single tube or top tube of a bank

$$\alpha = 0.728\left\{\frac{\rho(\rho-\rho_v) g h_{fg} k^3}{\mu d \Delta T}\right\}^{1/4} \tag{3}$$

For lower tubes inundation is effectively treated as emanating from an (infinitely high) plane surface source along the top generator of the tube (where the tangential velocity is zero) with total flow rate to one side set equal to half the inundation flow rate from above. This gives the values of C for the lower tubes and leads to the results

$$\alpha_{m,N}/\alpha_1 = N^{-1/4} \tag{4}$$

where $\alpha_{m,N}$ is the average heat-transfer coefficient for a column of N tubes given by

$$\alpha_{m,N} = \frac{1}{N}\sum_{i=1}^{N}\alpha_i = \frac{1}{N\Delta T}\sum_{i=1}^{N} q_i \tag{5}$$

when ΔT is the same for all tubes. α_1 is the heat-transfer coefficient for the top tube in the column given by Eq. (3), and

$$\alpha_N/\alpha_1 = N^{3/4} - (N-1)^{3/4} \tag{6}$$

where α_N is the heat-transfer coefficient for the Nth tube from the top in the column.

It may be noted in passing that the calculated profile of the film at the top of the lower tube does not blend with that of the falling liquid from the upper tube and the assumption of a thin condensate film is violated near the top of lower tubes as well as near the bottom of all tubes.

Except for the top tube in a bank, experimental data (for which ΔT is not the same for all tubes) have generally indicated the Nusselt [5] model to be over conservative, i.e., the detrimental effect of condensate inundation on the performance of lower tubes is significantly exaggerated by Eqs. (4) and (6). A commonly used empirical correlation due to Kern [6,7] uses a modified index in the Nusselt result thus:

$$\alpha_{m,N}/\alpha_1 = N^{-1/6} \tag{7}$$

and

$$\alpha_N/\alpha_1 = N^{5/6} - (N-1)^{5/6} \tag{8}$$

which has been found to represent the data better. In practice Eqs. (4), (6)–(8) cannot be strictly applicable since the temperature difference across the condensate film is not the same for all tubes.

As noted above, comparisons with data may also be affected by non-condensing gas and vapour shear stress. In addition, for a tube cooled by a liquid flowing internally (the usual case) with inside heat-transfer coefficient uniform around the perimeter, the outer surface temperature varies significantly around a condenser tube, see Memory and Rose [8].

Several investigators have found that the effect of condensate inundation is much less severe for low-finned tubes (e.g., Webb and Murawski [9] for R-11, Honda et al. [10–14] for R-113, R-123, R-134a and R-407c) and wire-wrapped tubes (e.g., Rifert et al. [15] for ammonia, Marto and Wanniarachchi [16] for steam). This has been explained by Honda et al. [17] by the fact that the fins or wires suppress lateral spreading of condensate along the tube (see Fig. 2) so that the parts of the tube between columns of falling condensate are not affected and behave like the top tube. Chen [18] and Kutateladze et al. [19] drew attention to the fact that the heat transfer could be significantly affected by condensation on the film between tubes. Leicy [20] noted the importance of inundation temperature control in measurements using simulated inundation, particularly in the case of enhanced tubes.

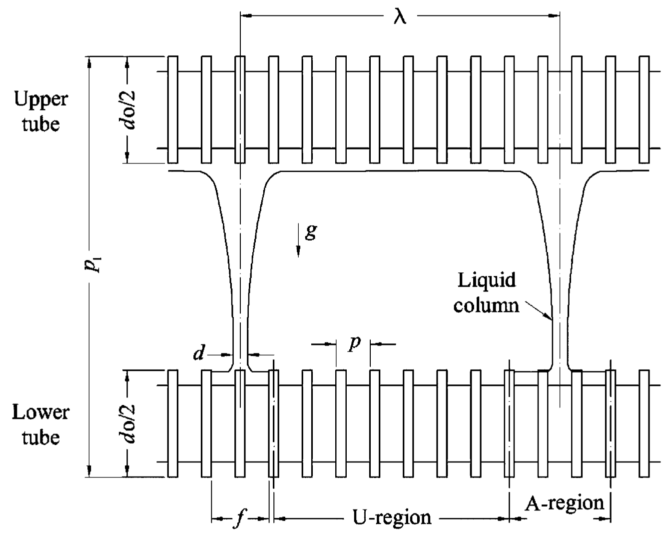

Fig. 2. Model of inundation in column of tubes of Honda et al. [17].

2. Present investigation

The apparatus is shown schematically in Fig. 3 and is the same as that used by Murase et al. [1] modified for inundation tests. Vapour, generated in an electrically heated, stainless steel boiler, was directed vertically downward over the horizontal, water-cooled, copper test condenser tube. Excess vapour passed to auxiliary condensers from which the condensate returned to the boiler by gravity. The cooling water temperature rise was measured using a 10-junction thermopile. Care was taken to ensure adequate mixing and isothermal immersion of the leads in the vicinity of the junctions. A small predetermined correction for the dissipative temperature rise of the cooling water in the tube and mixing boxes was incorporated in the calculation of the heat-transfer rate from the coolant flow rate and temperature rise. The surface temperature of the tube was measured by four thermocouples embedded in the tube wall as shown in Fig. 4. All thermocouples were calibrated in a high precision constant temperature bath against a platinum resistance thermometer, accurate to 0.005 K. The coolant flow rate was measured using a float-type flow meter with accuracy better than 2%. The vapour velocity at approach to the test section was found from the measured power input to the boiler (together with the condensate return temperature) with a small, predetermined correction for the heat loss

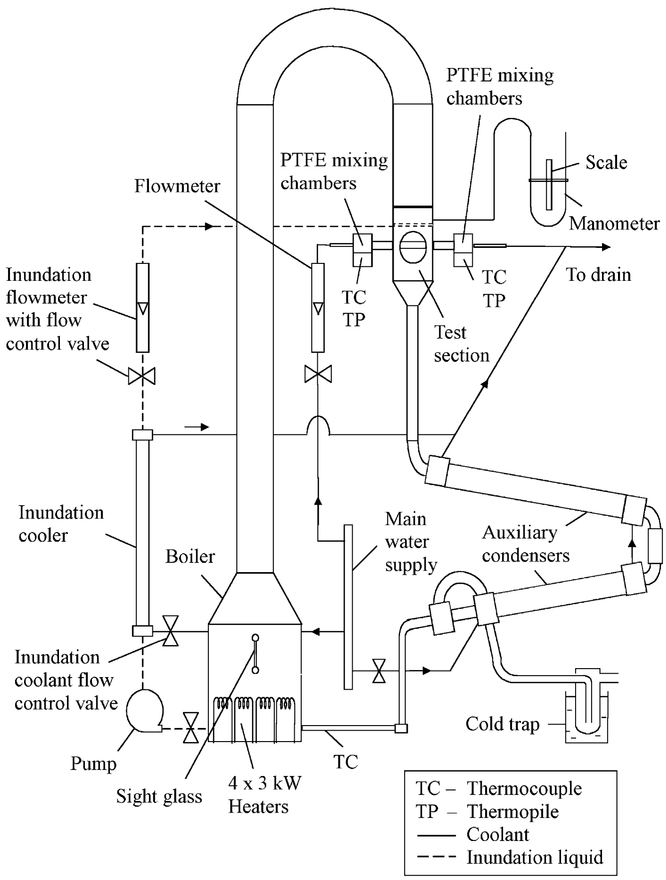

Fig. 3. Schematic of apparatus.

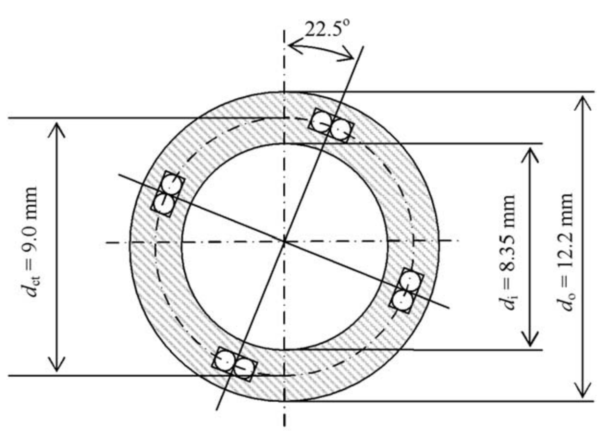

Fig. 4. Location of thermocouples in test tube wall.

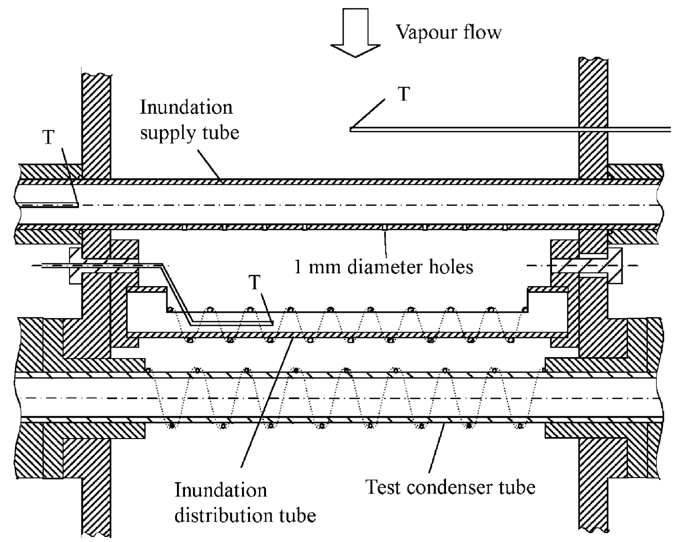

Fig. 5. Detail of test section. T denotes position of thermocouple junction.

from the well-insulated apparatus between boiler and test section, see Lee and Rose [21].

To simulate inundation, water from the boiler was pumped to the inundation supply tube via a tube-in-tube cooler and flow meter (see Fig. 3). Referring to Fig. 5, the test section consisted of three vertically in-line horizontal tubes. Water from the inundation cooler entered the inundation supply tube from both ends, passed through holes at the bottom of the inundation supply tube into the wire-wrapped distribution tube. The upper part of the inundation distribution tube was removed to collect liquid from the supply tube which overflowed onto the test condenser tube. The distribution tube was fitted with a thermocouple as shown. The inundation liquid and condensate were returned to the boiler by gravity.

Preliminary tests to establish uniform inundation flow were first conducted without condensation. Inundation flow rates up to 1.5 l/min were tested. Trial and error was used to establish optimum diameter and spacing of holes in the bottom of the inundation supply tube and the wire pitch for the inundation distribution tube to achieve uniform inundation. Uniformity of flow was strongly dependent on the inclination of the inundation distribution tube and provision was made to adjust this during operation. Uniform flow distribution was established in a range of flow rates up to 0.8 l/min. For the present tube and test conditions this is equivalent to the condensate flow from a column of about 30 smooth tubes and 20 low-finned tubes. Details of the apparatus are given by Murase. [22]. Fig. 6 shows photographs of the inundation flow.

Heat-transfer measurements were made using smooth, wire-wrapped and low-finned tubes. The smooth and wire-wrapped tubes with outside diameter 12.2 mm were the same as used in the single-tube investigation [1]. For the wire-wrapped tubes a wire diameter of 1.6 mm with winding pitches of 4, 8, and 16 mm (see Fig. 7) were used for comparison with the data of Brower [4]. The low-finned tube had outside diameter at fin root of 12.7 mm and fin thickness, fin height and interfin space of 0.5, 1.59, and 1.5 mm, respectively. This was the best performing tube (in the absence of inundation) in the work of Briggs et al. [23]. Accuracy estimation and tube cleaning procedure to ensure film condensation and precautions to avoid non-condensing gas effects were the same as those described by Murase et al. [1].

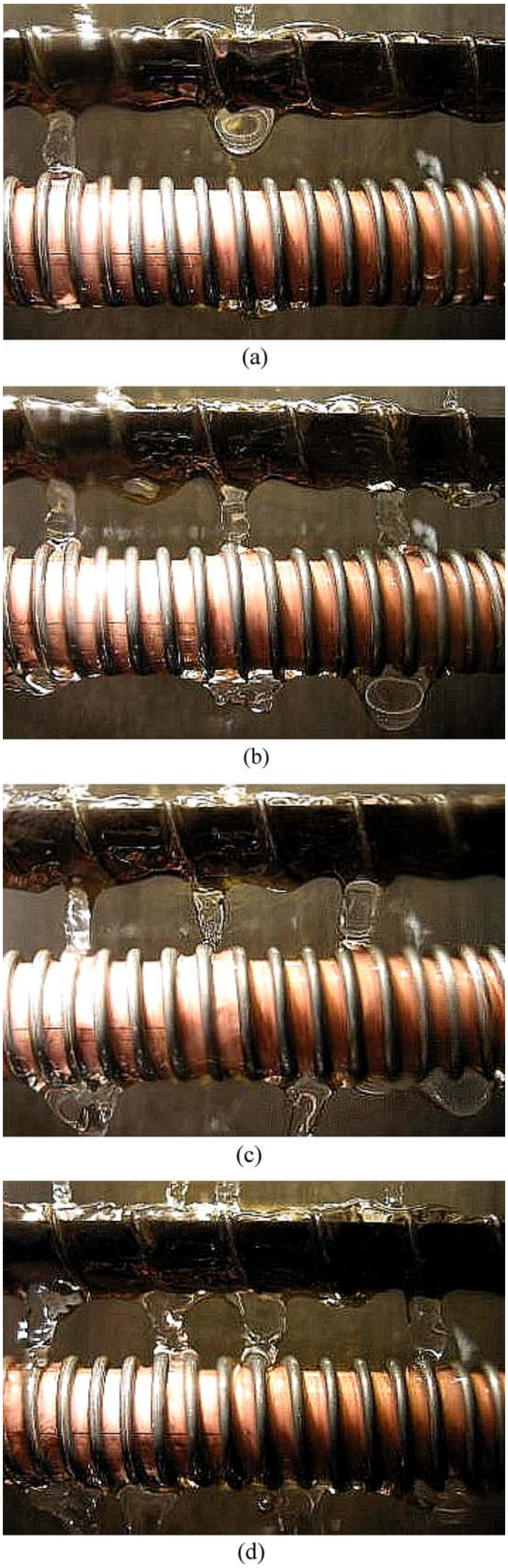

Fig. 6. Photographs of simulated inundation (wire diameter 1.6 mm, wire pitch 4 mm). (a) Inundation rate 0.1 l/min; (b) inundation rate 0.3 l/min; (c) inundation rate 0.5 l/min; (d) inundation rate 0.8 l/min.

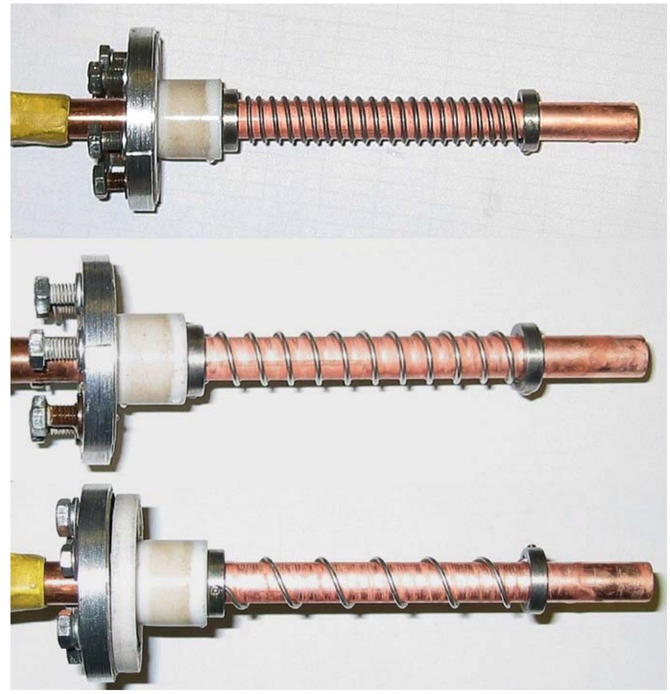

Fig. 7. Photographs of wire-wrapped tubes tested.

All tests were conducted at a little above atmospheric pressure with a fixed coolant flow rate to the condenser tube of 2 l/min. The inundation flow rate was increased in steps up to 0.8 l/min. For each inundation flow rate the inundation temperature, measured in the inundation distribution tube as shown in Fig. 5, was adjusted using the inundation cooler, to the desired value determined as described below. This involved trial and error adjustment of the coolant flow rate to the inundation cooler. Typically adjustment and attainment of the new steady condition took around 10 min.

Except for a few additional data for the finned tube taken at the end of the investigation the inundation temperature was adjusted to the mean condensate temperature at which condensate would leave a tube according to the Nusselt [5] theory

$$T_{\mathrm{i}}^{*} = \frac{5}{8}T_{\mathrm{v}} + \frac{3}{8}T_{\mathrm{wo}} \tag{9}$$

where the wall temperature T_{wo} was taken as the arithmetic mean of the four measured values. The vapour and wall temperatures substituted into Eq. (9) were values measured at the previous lower inundation rate.

All experiments were done with a vapour approach velocity to the test tube of approximately 0.56 m/s. The maximum proportion of steam condensed on the test condenser tube (for the cases with zero inundation) were approximately 14% for the smooth tube, 15% for the wire-wrapped tubes and 20% for the finned tube. The coolant inlet temperature was always around 10 °C. The coolant temperature rise varied between about 2 and 10 K (generally higher than about 5 K) and was measured with an accuracy of around 0.05 K by the 10-junction thermopile.

3. Results

Preliminary tests were first conducted without inundation over a range of coolant flow rates. The results are shown in Fig. 8. It is seen that the smooth tube data are a little above the Nusselt theory and in close agreement with the forced convection condensation equation

$$Nu\widetilde{Re}^{-1/2} = \frac{0.9 + 0.728F^{1/2}}{(1 + 3.44F^{1/2} + F)^{1/4}} \tag{10}$$

Eq. (10), from Rose [24], takes account of vapour shear stress on the condensate film and is a very close fit to numerical solutions based on the theory of Shekriladze and Gomelauri [25]. The data for the wire-wrapped tube are somewhat higher and virtually the same for the three winding pitches used. The results for the low-finned tubes are substantially higher and somewhat higher than predicted by the model of Rose [26].

In all cases, as may be seen, the data are well fitted by

$$q = B\left\{\frac{\rho(\rho - \rho_v)gh_{fg}k^3}{\mu d}\right\}^{1/4} \Delta T^{3/4} \tag{11}$$

as found in earlier investigations, see Murase et al. [1].

Fig. 9 shows the variation of heat flux with simulated inundation rate. The vapour and coolant conditions were essentially the same for all cases. Over the range of inundation rates used, the heat flux fell by approximately 25% for the smooth tube, and by 20%, 13%, and 9% for the wire-wrapped tubes with pitch of winding, 16, 8, and 4 mm, respectively. The fall in heat flux with inundation over the range used was negligible for the low-finned tube. Fig. 10 shows the same data plotted on a heat-transfer coefficient basis. The trends are broadly the same as in the case of heat flux. The high heat-transfer coefficients for the finned tube are due to the much smaller values of the vapour-to-surface temperature difference in this case.

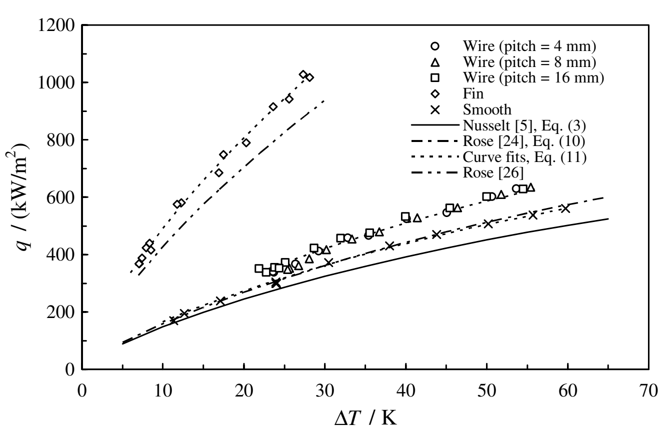

Fig. 8. Dependence of heat flux on vapour-to-surface temperature difference without inundation.

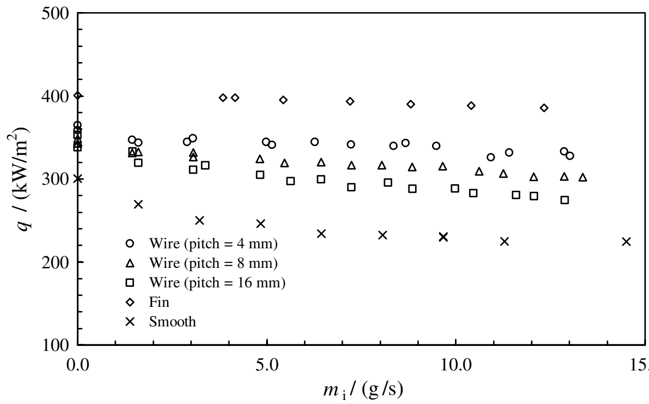

Fig. 9. Dependence of heat flux on inundation rate.

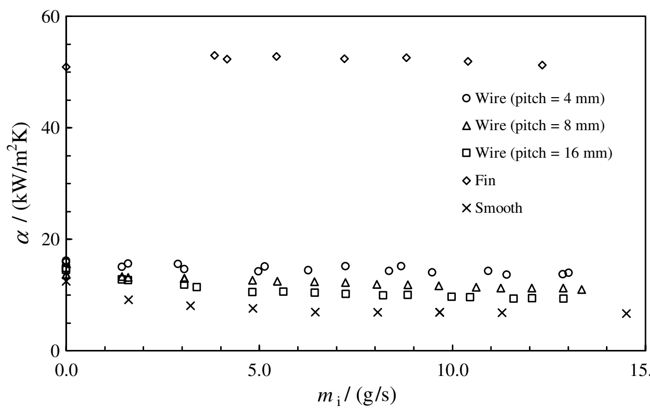

Fig. 10. Dependence of heat-transfer coefficient on inundation rate.

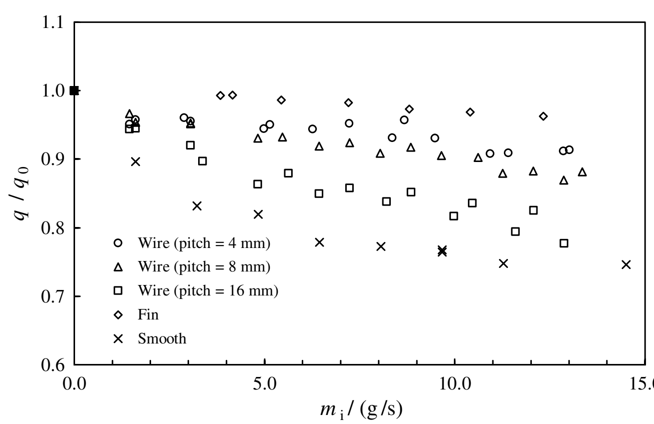

Fig. 11. Dependence of normalised heat flux on inundation rate.

The same results are plotted on a normalised basis in Figs. 11 and 12. It is seen that deterioration of heat transfer with increasing inundation is least for the finned tube in contrast with the data shown in Fig. 1. For the wire-wrapped tube the performance under inundation improves as the winding pitch decreases. For the largest winding pitch, 16 mm, the performance falls more steeply with increasing inundation rate but is still significantly better than the smooth tube in this respect. It is noteworthy that

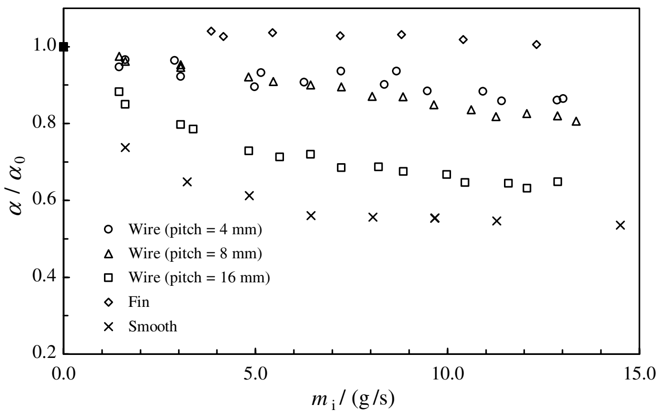

Fig. 12. Dependence of normalised heat-transfer coefficient on inundation rate.

in the case of the finned tube the pitch, 2 mm, was significantly less than the pitch of the smallest used for the wire-wrapped tube, 4 mm. It is possible that performance of a more closely wound wire-wrapped tube under inundation might have been better. The smallest possible pitch of winding for a wire-wrapped tube is the wire diameter.

It may be noted that in the literature on inundation only normalised heat-transfer coefficients are usually given. Plotting only normalised values disguises the fact that the values without inundation might be affected by presence in the vapour of non-condensing gas. In cases where diffusion resistance in the vapour is appreciable the normalised heat-transfer coefficient would appear much less affected by inundation.

3.1. Equivalent depth in tube bank

We seek the effective row number from the top of a column which corresponds to the inundation rates used in the present experiments. Clearly the inundation flow rates used do not correspond directly to any specific depth (tube number from the top) in a column of vertical tubes. However, it may be seen from Fig. 13 that, for the present data, inundation and condensation rates are smoothly related in all cases so that corresponding values for rates intermediate between those used in the tests can readily be found.

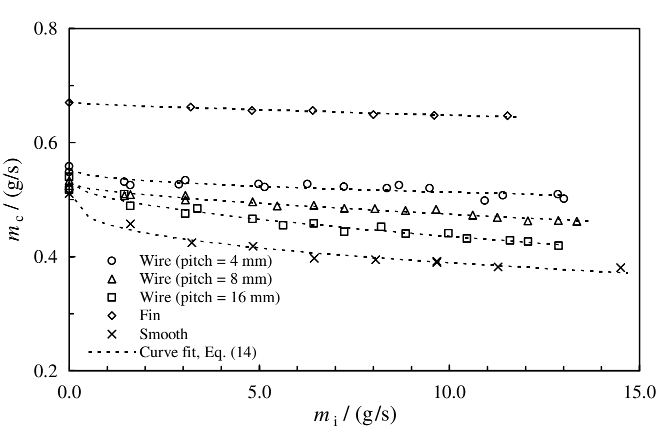

Fig. 13. Dependence of condensation rate on inundation rate.

For a column of tubes the condensation rate on tube N from the top is given by

$$m_{c,N} = Q_N/h_{fg} \tag{12}$$

The inundation rates are given by

$$m_{i,1} = 0 \quad \text{top tube} \tag{13a}$$

$$m_{i,2} = m_{c,1} \quad \text{2nd tube down} \tag{13b}$$

$$m_{i,3} = m_{c,2} + m_{c,1} \quad \text{3rd tube down, etc.} \tag{13c}$$

so that inundation and condensation rates are related for a column of tubes. For the simulation experiments, the condensation rate corresponding to tube 2 in a column may be read off from Fig. 13 at inundation rate equal to the (measured) condensation rate, Q_1/h_{fg}, for zero inundation. Then the condensation rate corresponding to row 3 in a column is read from Fig. 13 at an inundation rate equal to the sum of the condensation rates for rows 1 and 2 and so on up to the maximum inundation rate used in the simulation measurements. The process described above and illustrated in Fig. 14 was facilitated by fitting the data in Fig. 13 by

$$m_c = a_1 + a_2 m_i^{a_3} \tag{14}$$

where a_1, a_2, and a_3 are constants. An iterative process was used for the non-linear constant a_3 in which for each value of a_3 the corresponding values of the linear constants a_1 and a_2 are immediately found by linear regression. As can be seen, excellent fits were obtained in all cases.

Fig. 15 shows the results for smooth, finned and wire-wrapped tubes on an effective row number basis determined as described above. The same general conclusions

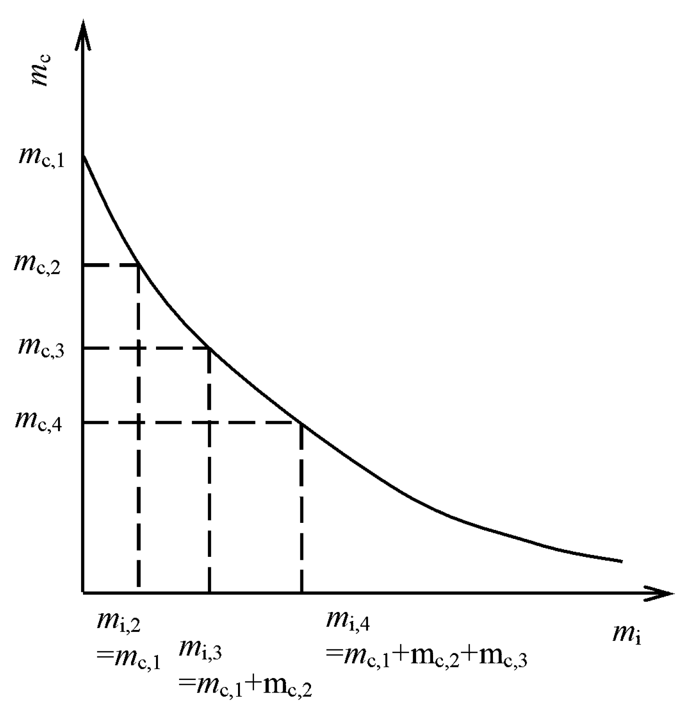

Fig. 14. Calculation of effective depth in column of tubes.

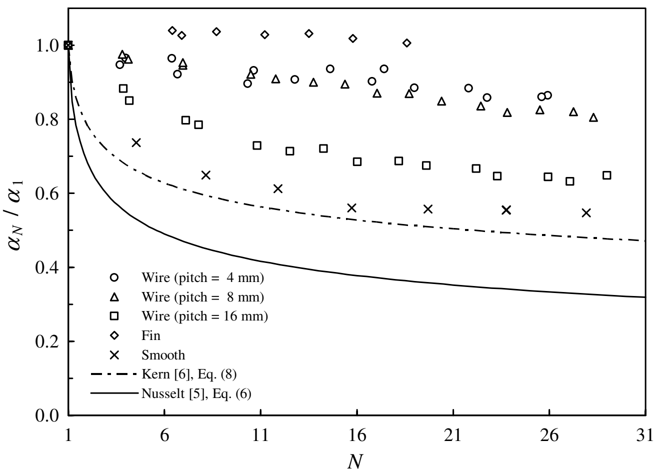

Fig. 15. Dependence of normalised heat-transfer coefficient on row number in simulated column of enhanced tubes.

as those discussed on the basis of inundation flow rate can be drawn. It is seen that for the same maximum inundation rate fewer rows are simulated for the tubes with higher heat-transfer coefficients and hence higher condensation rates. The smooth tube inundation data are in quite good agreement with Kern, Eq. (8).

4. Conclusions

Earlier measurements for condensation of steam on single tubes without enhancement have been confirmed. The best enhancement was that found for the low-finned tube. Note that the enhancement ratio of 4 reported here for the finned tube is the ratio of the heat-transfer coefficient for the finned tube divided by that for the smooth tube *at the same coolant flow rate* and would depend on the coolant flow rate used. Values of enhancement ratio reported earlier by Briggs et al. [27] for a finned tube having the same dimensions are enhancement ratios are 3.01 for *the same vapour-side temperature difference*, and 4.36 *for the same heat flux*. The measured enhancement ratio for the wire-wrapped tube was only around 1.3 and almost the same for the three winding pitches tested.

Under inundation the finned tube performed best, with the heat-transfer coefficient virtually unaffected up to the highest inundation rate used, equivalent to a depth of around 20 tubes in a bank. For the wire-wrapped tubes the inundation performance increased with decreasing winding pitch. At the highest inundation rate, equivalent in this case to around 28 tubes, the heat-transfer coefficient for the tube with the smallest winding pitch was around 50% higher than that for the smooth tube.

Appendix A. Further consideration of inundation temperature in simulation tests

In the present work, as noted, the inundation supply temperature was set using Eq. (9) based on the Nusselt theory for an isothermal, smooth tube. In reality, as found in the present tests (see Fig. A.1) and many earlier investigations of condensation on horizontal tubes, the surface temperature around the tube varies significantly and has an approximately cosine distribution. Moreover, there is no reason why the Nusselt theory should give the correct mean temperature of condensate draining from the bottom of an *enhanced* tube.

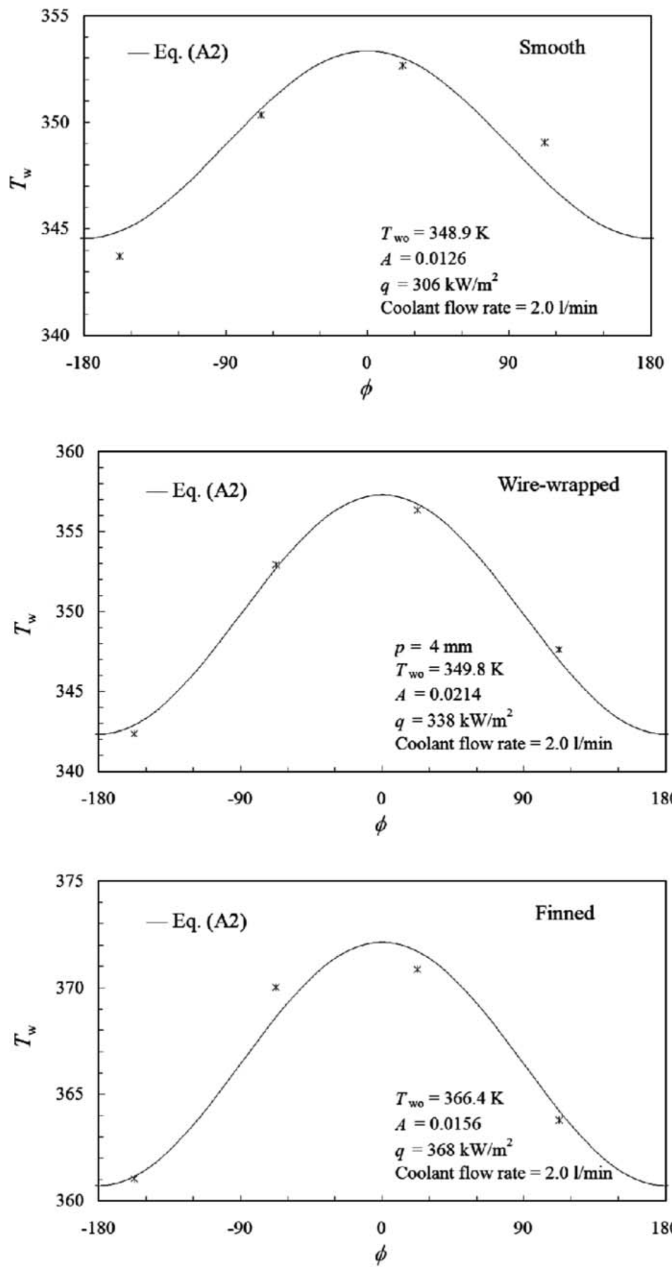

Fig. A.1. Specimen surface temperature distributions around the tube for condensation of steam on smooth, wire-wrapped and finned tubes and comparison with curve fits Eq. (A.2).

To assess the sensitivity to inundation supply temperature further tests were done using the finned tube (most sensitive to inundation supply temperature) with the inundation supply temperature set according to Eq. (9) but with

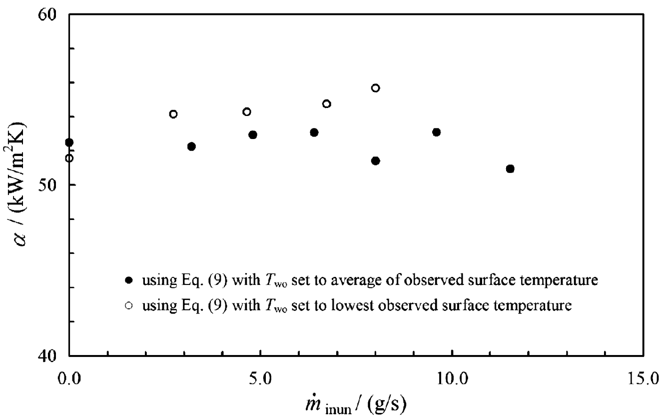

Fig. A.2. Effect of inundation supply temperature for condensation of steam on low-finned tube (vapour approach velocity 0.56 m/s, coolant flow rate 2.0 l/min).

T_{wo} set equal to the temperature indicated by the lowest thermocouple on the tube surface (157.5° from the top of the tube, see Fig. 4) rather than the mean of the four surface temperatures. It is seen (Fig. A.2) that the measured heat-transfer coefficients are somewhat lower when using the lower inundation temperature and show a marginal decrease with increasing inundation rate suggesting that the inundation temperatures used earlier may have been a little too high. The smooth and wire-wrapped tube results would be less sensitive to inundation supply temperature.

For future tests it is recommended that the inundation temperature be set to

$$T_i^* = \frac{5}{8}T_v + \frac{3}{8}T_{wo}(1 - A) \tag{A.1}$$

where A is the constant in the curve fit to the measured tube wall temperature distribution.

$$T_w = T_{wo}(1 + A\cos\phi) \tag{A.2}$$

Eq. (A.1) is derived in the same way as Eq. (9) using the cosine surface temperature distribution, Eq. (A.2), rather than uniform wall temperature. This procedure should at least be correct for smooth tubes and is probably not a bad approximation for finned and wire-wrapped tubes.

The suggested experimental procedure is then as follows:

1. Measure surface temperatures for tube with no inundation.
2. Fit temperature distribution by Eq. (A.2) to determine constant A used in Eq. (A.1).
3. Determine inundation temperature for next test using Eq. (A.1).
4. From the measured heat flux calculate drainage rate from the tube.
5. Set the inundation temperature to that calculated at step 3.
6. Set inundation rate equal to drainage rate calculated at step 4.
7. Conduct set of measurements corresponding to the second tube in the column.
8. Repeat steps 1–5.
9. Set inundation rate equal to previous value plus current condensation rate and obtain results for row 3 and so on.

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