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Measurement of condensation heat transfer in a thermosyphon

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Abstract—A novel technique is described for studying vapour condensation in a two-phase reflux thermosyphon. The technique involves accurate measurement of the thickness of the condensed liquid film using a rotating needle contact method. For laminar flow of this liquid ($Re_L < 10$), this information can be used to determine the spatial dependence of heat flux in the condenser. The validity of the method is demonstrated by the agreement between the results it gives and those predicted for a calculable condenser system, and by the direct measurement of the temperature difference across the known thermal impedance of the condenser wall. The technique is being used to study condensation in the presence of non-condensable gas.

INTRODUCTION

IN THE two-phase closed reflux thermosyphon, the condenser plays an important role in the process of energy transfer. Experimental studies of heat transfer rate in the condenser have adopted several different approaches that have usually involved measurement of heat transfer rates for the condenser as a whole. In order to understand the condensation process in more detail, it is desirable to obtain data on the spatial dependence within the condenser of the condensing vapour.

Information about the distribution of heat and mass transfer rate in the condenser has been obtained in several recent studies, particularly related to the condenser of a gas-loaded reflux thermosyphon. This work involved measurement and modelling of the vapour phase. Peterson and Tien [1] used a wet-bulb/dry-bulb method to measure concentration profiles around the gas–vapour interface. They also developed theoretical approaches to this heat transfer problem. Parfentiyev [2] measured the temperature gradient within the vapour near a wicked condenser surface and calculated the local heat flux under the assumption that the vapour in this region is saturated. Kobayashi and Matsumoto [3] obtained two-dimensional temperature and density information, and the mole fraction of non-condensable gas in the interfacial region using a laser holographic interferometer.

In this paper, we describe a technique which we have developed to study the spatial dependence of heat and mass transfer in a vertical condenser. The method relies on measurements of the condensed liquid film rather than of the vapour phase as with previous studies. Accurate measurements of thickness, coupled with a knowledge of the behaviour of the falling liquid film, permit detailed information to be obtained of the spatial dependence of condensing vapour.

Techniques for measuring the thickness of liquid films have been most extensively developed for the study of two-phase flow [4]. Such techniques include conductance and capacitance, the needle contact method, light absorption, photographic and shadowgraph methods, etc. However, so far no work has been reported on the application of such methods to closed reflux thermosyphons, presumably because of the difficult geometry and environment: enclosure and vacuum.

Among the measurement methods mentioned above, one simple but practical technique is the needle contact method. However, there have been few reports of the use of this method for measuring the film thickness on the inside surface of a tubing condenser. Ueda *et al.* [5, 6] used this method for measuring the thickness of condensate film during downward vapour flow inside a vertical tube. The probes were fixed at three positions along the condenser, and the needles were inserted into the condenser through the wall and moved backwards and forwards by a micrometer. The film thickness was therefore measured at three points and detailed information was not obtained on the spatial dependence of condensing vapour.

In the present work, an improved needle contact probe is described, with which the thickness of the condensed liquid film can be measured at any point within the condenser of a thermosyphon. The spatial distribution of vapour condensing on the surface of the condenser is determined using a simple model of downward laminar flow for the liquid film. The validity of the technique is demonstrated by applying it to a condenser system the properties of which can be calculated quite accurately. The results obtained by the method are also confirmed by direct measurement of the temperature difference across the wall of the condenser.

THEORETICAL CONSIDERATIONS

In this section the principle of the method for determining spatial distribution of condensing vapour is

NOMENCLATURE

d	distance between the axis of rotation of the probe and the tube centre [m]	Greek symbols	
$\dot{F}$	mass flux [kg s^{-1} m^{-2}]	α	angle defined in equation (10)
$\dot{F}^*$	dimensionless mass flux	α_h	heat transfer coefficient [W m^{-2} °C^{-1}]
g	gravitational acceleration [m s^{-2}]	β	angle measured by experiments
k	thermal conductivity [W m^{-1} °C^{-1}]	δ	liquid film thickness [m]
l	vertical length of condenser [m]	γ	latent heat [J kg^{-1}]
$\dot{m}$	mass flow rate [kg s^{-1}]	μ	dynamic viscosity [kg m^{-1} s^{-1}]
$\dot{Q}$	heat flux [W m^{-2}]	ρ	density [kg m^{-3}].
R	radius of condenser [m]	Subscripts	
Re	Reynolds number	h	heat
r	radius of rotation of probe tip [m]	L	condensate liquid
T	temperature [°C]	s	saturated
t	wall thickness [m]	total	total
x	distance along the condenser [m].	w	wall.

first discussed. The accuracy of the method, and the limitations on its applicability are also analysed. Finally, a condenser configuration is described which has heat transfer properties that can be calculated quite accurately, and which can be used to demonstrate the validity of the method.

The method for measuring spatial distribution of heat flux relies on an accurate determination of the thickness of the falling film of condensed liquid over the entire condenser surface. In its simplest configuration, the method is applicable to a vertical cylindrical condenser of inner radius R in which the condensed liquid completely wets the surface of the condenser. For laminar flow of such liquid, it is easy to show that the thickness, δ, of this liquid film at any point x measured from the top of the condenser is related to the mass flux per unit width of liquid flowing past that point, $\dot{m}(x)$, by

$$\dot{m}(x) = 2\pi R\rho_L^2 g\delta^3(x)/3\mu_L. \tag{1}$$

In this equation, ρ_L is the density of the liquid, g the acceleration due to gravity, and μ_L the viscosity of the liquid. This relationship ignores buoyancy forces on the condensed liquid film due to the vapour and neglects nonuniformities which may exist in the thickness of the liquid film due to instabilities induced by vapour shear or motion.

If $\dot{F}(x)$ is the mass flux of vapour condensing per unit area, we can write, for a vertical condenser

$$\frac{d\dot{m}(x)}{dx} = 2\pi R\dot{F}(x). \tag{2}$$

Therefore

$$\dot{F}(x) = \frac{\rho_L^2 g\delta^2(x)}{\mu_L}\frac{d\delta(x)}{dx}. \tag{3}$$

Equation (3) gives the relationship between the thickness of the condensed liquid film and the mass flux of condensing vapour. The existence of the derivative in this equation implies that very accurate thickness data are necessary in order to achieve reasonable accuracy in estimating mass flux. The procedure for processing the thickness data is therefore of importance in the application of the method.

The validity of the technique described here was demonstrated by performing measurements in a condenser, the properties of which could be calculated accurately. In his seminal work on the condensation process, Nusselt [7] derived the heat transfer coefficient for vapour condensing on an isothermal surface. In the simplest form of this calculation, it is assumed that the vapour is isothermal, and the only significant thermal impedance for heat transfer is that of the condensed liquid film. This film is assumed to move under gravity, with laminar flow, and to have a smooth ripple-free surface as discussed above. This simple theory predicts a film thickness $\delta(x)$ in a vertical condenser given by

$$\delta(x) = \left[\frac{4k_L(T_s - T_w)\mu_L x}{\rho_L^2 g\gamma}\right]^{1/4} \tag{4}$$

where γ is the latent heat of condensation of the working fluid, T_s the temperature of the saturated vapour, T_w the temperature of the condenser surface, and k_L the thermal conductivity of condensed liquid. The local heat transfer coefficient $\alpha_h(x)$ is given by

$$\alpha_h(x) = \left[\frac{k_L^3 g\rho_L^2\gamma}{4\mu_L(T_s - T_w)x}\right]^{1/4}. \tag{5}$$

In the derivation of these results, it is assumed that the only significant heat flow is in a direction normal to the liquid film. Sensible heat transport associated with the motion of the liquid itself is ignored.

The theory has been progressively developed and refined over the years. Reviews of work on condensation [8–10] did not report any experimental measurements of the spatial dependence of heat flux under conditions which are the same as that described by Nusselt. The problems involved in undertaking such experiments arise from two major sources: the difficulties in achieving an isothermal boundary condition at the interface between the condenser wall and the condensed liquid; and the influence of instabilities in the condensed liquid. The isothermal boundary condition is difficult to achieve because thermal impedances associated with condensing vapour are very small. It is therefore necessary to make other impedances involved in the heat transfer even smaller, particularly that between the outer condenser surface and the cooling liquid flowing past it. Thus it is quite difficult to build a condenser in which the condensation process behaves according to the simple Nusselt model. Such a model, therefore, does not serve as a useful basis for designing a condenser in which the spatial dependence of heat flux can be calculated accurately.

In order to overcome this difficulty, we have built a condenser in which a large thermal impedance exists between the condensing vapour and rapidly moving, turbulent cooling water held at constant temperature. This thermal impedance is chosen to be large compared with that of the condensed liquid film, and of that between the outer surface of the condenser wall and the cooling water. Under these conditions, provided that the temperature in the vapour is uniform (a condition that can be experimentally verified), the thermal impedance between condensing vapour and cooling water is nearly constant and the heat flux through the wall of the condenser will be independent of position. In other words, $\dot{F}(x)$ in equation (2) is constant, so integrating this equation gives

$$\dot{m}(x) = 2\pi Rx\dot{F}. \tag{6}$$

The total mass flow rate at the bottom of the condenser, $\dot{m}_{\text{total}}$, can be obtained directly from equation (6)

$$\dot{m}_{\text{total}} = 2\pi R\dot{F}l \tag{7}$$

where l is the vertical length of the condenser.

Dividing equations (6) and (7), and inserting into equation (1), we obtain

$$\delta^3(x) = 3\dot{m}_{\text{total}}\mu_L x/2\pi R\rho_L^2 gl. \tag{8}$$

Thus

$$\delta^3(x) = 3\dot{Q}_{\text{total}}\mu_L x/2\pi R\rho_L^2 gl\gamma \tag{9}$$

where $\dot{Q}_{\text{total}}$ is the total heat flux in the condenser.

The existence of instabilities in the condensed liquid film will not significantly affect the heat transfer rate in such a condenser. However, accurate measurements of the thickness of the film will only be possible if the condenser is operated in a domain where waves and instabilities do not occur. The experimental realization of such a condenser is quite simple to achieve, and is discussed below.

THE ROTATING NEEDLE CONTACT METHOD

The needle contact method for measuring the thickness of liquid films [11] has been widely used in the study of condensation heat transfer. The principle of the method is very simple: the point of a needle is brought up to the surface of the film and when the needle contacts the surface, the position of the point of the needle is recorded relative to the position of the surface over which the liquid is flowing. Despite its widespread use in many situations, this method has not previously been used to measure the condensate film thickness in the condenser of a two-phase closed reflux thermosyphon.

This paper describes a needle contact technique for measuring the thickness of a condensed liquid film anywhere within the relatively inaccessible interior of a cylindrical condenser. The probe configuration and system geometry are shown in Figs. 1 and 2, respectively. The probe is rotated around an axis which is

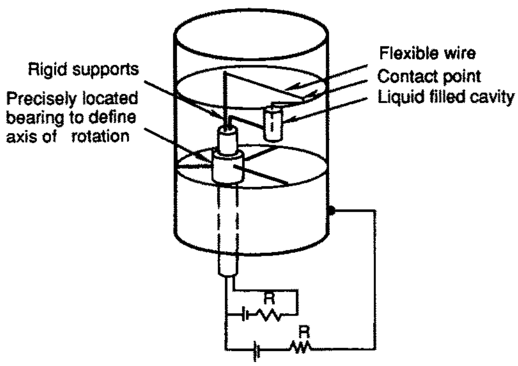

FIG. 1. The configuration of the rotating probe measuring system.

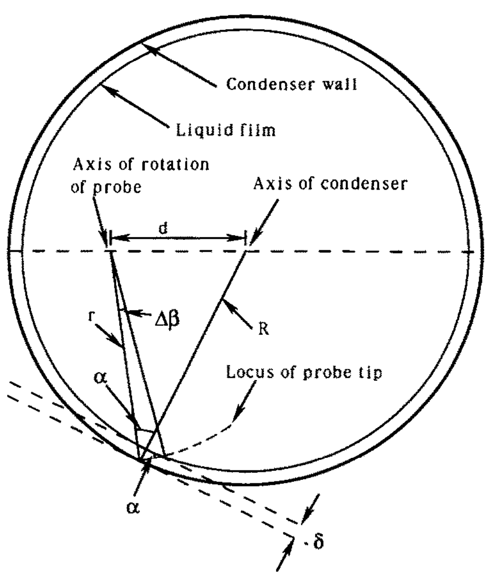

FIG. 2. The geometrical relationships of the rotating probe.

parallel to the axis of the condenser and is located nonconcentrically with respect to it. The probe enters the evacuated thermosyphon through a lower rotating seal and can be raised vertically to any desired position in the condenser.

The relationship between the angular position of the probe and the thickness of the condensed liquid film is given by

$$\cos \alpha = \frac{R^2 + r^2 - d^2}{2Rr} \tag{10}$$

$$\delta = r\Delta\beta \sin \alpha. \tag{11}$$

In these equations the angles α and $\Delta\beta$ are defined in Fig. 2, and d is the distance between the axis of rotation of the probe and the tube axis, r the radius of rotation of the probe tip and R the inner radius of the tube. In our present system, $R = 11.5$ mm, $d \approx 2.5$ mm, and $r = 12$ mm. The angular position of the probe can be measured to an accuracy of approximately $\pm 5 \times 10^{-5}$ rad by a precision turning table. The accuracy for controlling the table is about $\pm 1.5 \times 10^{-4}$ rad using manual adjustment.

With the rotating probe, a measurement can be made at any position on the wall of the condenser. It is necessary to determine the distance between the probe and the surface of the wall at arbitrary points. Therefore, two positions must be detected: one at which the probe first touches the condensed liquid film, and the second where the probe touches the inside surface of the condenser. The position of the point of contact of the probe and the liquid surface can be determined by measuring the electrical resistance between the probe and a distant conductor inside the condenser and below the probe. Experimentally, the angular position of this point of contact is found to be reproducible to within about $\pm 1.5 \times 10^{-4}$ rad. For an electrically conducting condenser wall it is an easy matter to determine the point of contact of the probe with the wall of the condenser. This is considerably more difficult, however, for a condenser which is electrically insulating, such as a glass tube. It is possible to determine the point of contact between the probe and an insulating condenser wall using a second contact which moves relative to the probe when the probe touches the condenser wall.

This arrangement is also shown in Fig. 1. The slight displacement caused when the probe contacts the wall causes the second probe to move relative to its surroundings. A consequent change in the electrical current through a pool of water around this second probe is measured. Using a measurement on a conducting surface it is found that the accuracy of detecting the point of contact of the probe with the wall is also about $\pm 3 \times 10^{-4}$ rad. The relative error (measurement error/real film thickness) depends on the geometry of the probe (see equations (10) and (11)). Reduction of d may increase the sensitivity of the measurement, but also introduces progressively larger errors due to uncertainties in d (see equation (10)). In the present work, d is made large enough that the contribution of its variations to the accuracy of measurement are unimportant. In such a system described above, the total error of measurement of a liquid film, 0.05 mm thick, is about $\pm 2\%$. Considerable improvements in the accuracy of the measuring system will be possible with further development.

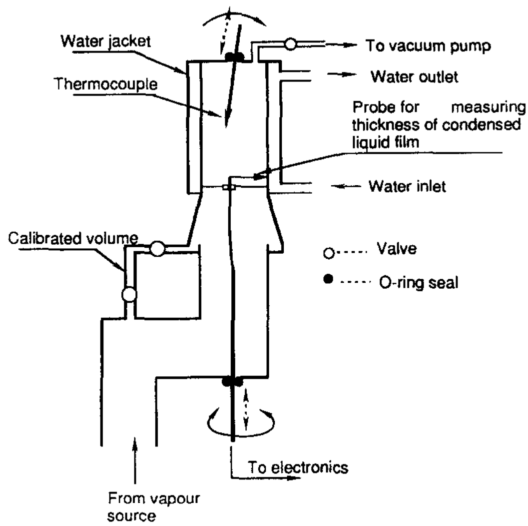

FIG. 3. Schematic diagram of apparatus.

APPARATUS

A schematic diagram of the apparatus used to simulate the thermosyphon is shown in Fig. 3. Vapour is supplied from a boiler which is connected by flexible tubing to a condenser consisting of a vertical glass tube surrounded by a water jacket. The system can be evacuated through a valve at the top of the condenser, and measurements can be made with either pure vapour or with non-condensable gas present. The condensing vapour wets the surface of the condenser and returns to the boiler under gravity. The returning liquid can be separated from the incoming vapour stream using an enlarging tapered pipe, and then channelled to a side measuring tube. In this way, the flux of vapour entering the condenser can be determined accurately.

In order to satisfy the condition of uniform condensation within the condenser (equation (6)), a Pyrex glass tube is used as the condenser, and water as the working fluid. The geometry, and relevant thermal parameters of the system components are shown in Table 1. Because the liquid film is very thin (less than 0.1 mm at several hundred watts input), the glass represents the dominant thermal impedance between the vapour and the constant temperature water jacket. For the system described here, the thermal impedance of the glass is 15 times greater than that of the liquid film at its thickest point. We therefore expect that the

Table 1. System parameters in calculable condenser

($T = 20°C$)	Condenser wall	Liquid film
Material	Glass	Water
Thickness (mm)	1.5	0.05
Thermal conductivity ($W\ m^{-1}\ °C^{-1}$)	1.0	0.56
Thermal impedance ($m^2\ °C\ W^{-1}$)	0.0015	0.0001

variations in heat flux to the condenser wall will be small. Figure 4 shows the calculated spatial dependence of heat flux on distance from the top of the condenser. These results are obtained using published values for physical properties of glass and water, and system dimensions as given in Table 1. The power density of 7.6 $kW\ m^{-2}$ corresponds to a power of 126 W in this system. As expected, the departures from uniformity of the heat flux are less than $\pm 3\%$.

In this method, it is essential that the surface of the condenser be completely wet by the condensing vapour. In previous work [12], contamination was considered as an important factor influencing the type of liquid film attained. Droplet condensation was observed to occur, even though the surface and other parts in the system had been cleaned carefully. Such droplets were observed on the glass condenser wall at the mixing region in the system with non-condensable gas [1]. In the present system, complete wetting was achieved by thoroughly cleaning the internal surface of the glass condenser and subjecting it to a mild acid etching process prior to assembly of the system. As a result, no droplets occurred whether the system was operating under pure vapour condensation or with non-condensable gas.

As mentioned above, one of the important factors which may influence the measurement is the occurrence of instabilities in the condensed liquid film. This phenomenon generally results from two causes: shear force at the liquid–vapour interface if the vapour is not stationary, and a gravity induced wave on the surface of the liquid.

Experiments by Binnie [13] indicate that gravity induced surface waves exist down to Reynolds numbers as low as 18.6 ($Re_L = 4\dot{m}/2\pi R\mu_L$). Calculations

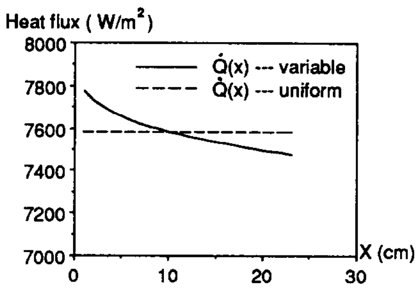

FIG. 4. Effect of thermal impedance of condensed liquid film on the spatial dependence of heat flux, $\dot{Q}(x)$.

[14] confirmed this result. In the present work, the Reynolds number in the liquid is less than 10, indicating that waves should not occur. This is supported, both by the fact that waves were not observed through the transparent condenser, and by the reproducibility of measurements of the position of the liquid surface.

In the present system the average velocity of the vapour at the entrance to the condenser is about 1.8 $m\ s^{-1}$ for 150 W input. At these low velocities, the shear force between vapour and liquid is negligible compared with the weight of the liquid, confirming an assumption made in deducing equation (1) of zero velocity gradient in the liquid at the liquid–vapour interface.

The temperature within the volume of the condenser is measured with a thermocouple (chromel–alumel, 0.076 mm diameter) inserted through a vacuum tight O-ring feedthrough at the upper end of the condenser, which is shown in Fig. 3. The thermocouple is located within a narrow (2 mm diameter) stainless steel tube which can slide through the seal and rotate around the sealing point. With this method, the temperature within virtually the entire volume of the condenser can be measured with a single thermocouple.

The measurement of temperature within the condenser permits an independent estimate to be made of the heat flux at the condensing surface. The rate of flow of heat per unit area, $\dot{Q}(x)$, through the wall of the condenser is calculated simply from the temperature difference ΔT across it

$$\dot{Q}(x) = K_w \Delta T / t \quad (12)$$

where t is the thickness of the condenser wall, and K_w the thermal conductivity of the wall material. Equation (12) is only applicable when the thermal impedance of the wall is large compared with that of the liquid film so that the temperature drop across the liquid is small. This is the situation in the present experiment. According to the assumptions previously stated, the mass flux can be determined from the heat flux by

$$\dot{F}(x) = \dot{Q}(x)/\gamma. \quad (13)$$

It will be appreciated that measurements of temperature of the liquid film provides a very direct method of estimating the spatial dependence of heat flux. This method is only useful, however, in condensers for which the thermal impedance of the wall is large compared with that of the condensed liquid

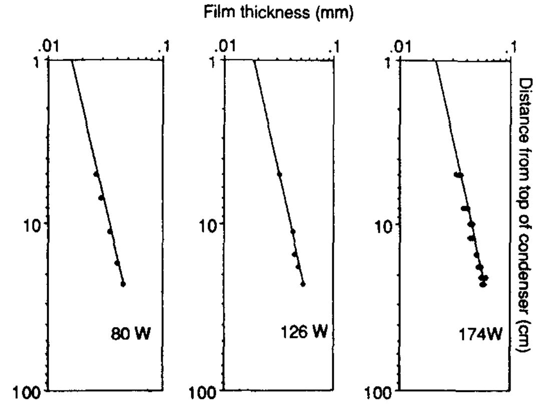

FIG. 5. Experimental data (points) and calculated results (lines) of liquid film thickness under the condition of pure vapour condensation. The lines have slope of 1/3 as given in equations (8) and (9), and the values of points are determined by actual measurements of the geometry of the probe (equations (10) and (11)).

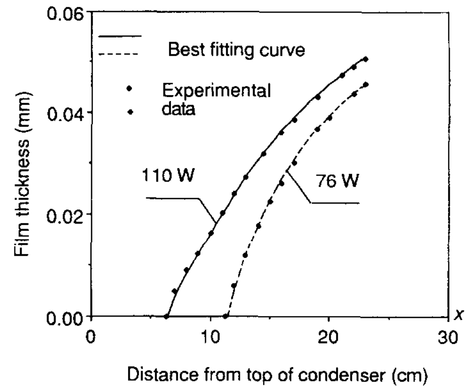

FIG. 6. Experimental data and the best fitting curve of liquid film thickness under the condition of condensation with non-condensable gas.

film, as is the case here. For condensers which are likely to be used in practical heat transfer systems, such a condition would not apply and measurements of liquid film thickness would be a much more satisfactory method of obtaining estimates of the distribution of heat flux in the condenser.

RESULTS

Condensation of pure vapor

Measurements were made in a reflux thermosyphon without non-condensable gas in order to verify that the rotating needle probe does indeed give correct results for film thickness in a calculable system. Experimental results for a cylindrical glass condenser with water as the working fluid are shown in Fig. 5. The functional dependence and absolute value predicted by equations (6) and (8) are indeed observed, as indicated by the agreement shown between the experimental data and the results of calculations. The results are therefore consistent with a uniform heat flux over the vertical walls of the condenser. Further demonstration of the uniformity of the heat flux is obtained from the observation that the inner surface of the condenser is isothermal to within $\pm 0.1^\circ$C. Such results verify that the contacting method reproduces predicted results in a system which is calculable. It is with some confidence therefore, that this method can be applied to a condenser containing non-condensable gas.

Condensation with non-condensable gas

Figure 6 shows measurements of thickness of the condensed liquid film in a reflux thermosyphon containing a fixed quantity of air as non-condensable gas [15]. The interface where vapour is mixed with gas is located towards the top of the condenser, and there is a significant temperature drop across the interface. The measurements were made at power levels of 76 and 110 W. The absence of a liquid film above the vapour–gas interface (detected by temperature measurement with the thermocouple shown in Fig. 3) clearly indicates that condensation is limited to the lower part of the condenser. More quantitatively, the data of Fig. 6 can be used to derive the spatial dependence of condensing mass flux, using equation (3). Direct application of equation (3) to the data does not yield particularly useful results, however, since very small errors in measurements of film thickness δ result in large variations in its derivative. A more fruitful approach is to choose a smoothly varying analytic function to describe the heat flux. This function should be of a form which incorporates the physical constraints of the experiment, specifically, constant heat flux well below the interface and zero heat flux well above the interface. This analytic function can be chosen with a few adjustable parameters which are determined by a best fit with the experimental data through application of equation (3).

The simplest function consistent with the above constraints requires two adjustable parameters and is of the form

$$\dot{F}^*(x) = \dot{F}(x)/\dot{F} = [\tanh\{A(x-x_0)\}+1]/2. \quad (14)$$

This function varies symmetrically about x_0 from 1 for $x \gg x_0$ to 0 for $x \ll x_0$. x_0 is thus the centre of the interface and A^{-1} provides a measure of the width of the interface. It turns out that the experimental data cannot be fitted satisfactorily by such a symmetric function and it is necessary to include some asymmetry into the form for the condensing heat flux. This requires three adjustable parameters. One such function (by no means the only one) which includes asymmetry is

$$\dot{F}^*(x) = \dot{F}(x)/\dot{F} = [\tanh\{A(x-x_0)\}+B]/[1+B]. \quad (15)$$

It is necessary to apply the constraint that values of $\dot{F}^*$ are defined to be zero where the functional form

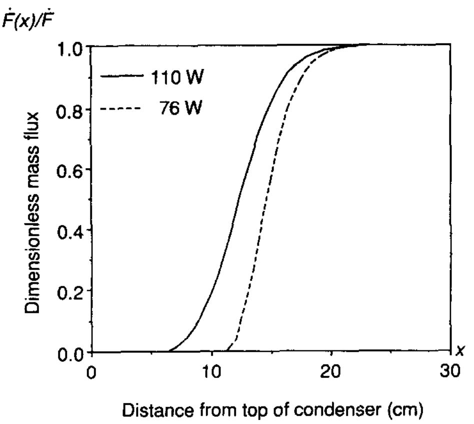

FIG. 7. Mass flux density deduced from the best fitting curve of liquid film thickness in Fig. 6.

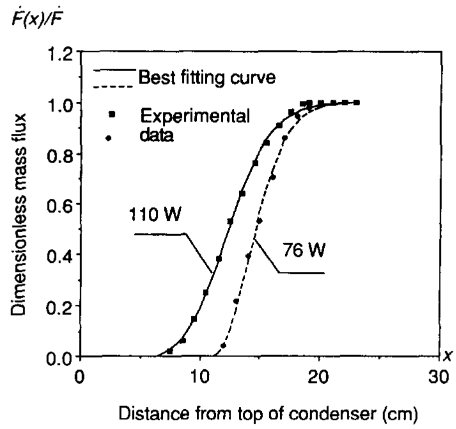

FIG. 8. Comparison of results of mass flux data obtained by the film thickness method (lines) and by direct measurements of temperature difference across the condenser wall (points).

assumes negative values. The third parameter, B, essentially shifts the origin of the $\dot{F}^*$ axis. Values of B slightly less than 1 effectively eliminate the long tail of the heat flux above the interface.

It was found that the functional form of equation (15) could be used to derive values of heat flux which in turn yielded a calculated spatial dependence of film thickness that was an extremely good fit to the experimental data for film thickness. Examples of the best fit for the condensing mass flux are shown in Fig. 7, and the agreement of the derived film thickness with the experimental data is shown in Fig. 6. It is clear that there is little more to be gained from trying alternative functional forms for the mass flux since the agreement is within the experimental uncertainty of the data. It is possible that with more accurate data it may be necessary to introduce other functional forms, or even more adjustable parameters in order to get an adequate fit. For the present, however, this approach yields considerable information about the spatial dependence of condensing heat flux.

In the condenser design discussion here, the spatial dependence of heat flux can be independently determined from a measurement of the temperature difference across the glass wall, ΔT (equation (12)). Results of such measurements are shown in Fig. 8 and compared with the distributions of heat flux calculated from measurements of film thickness. The good agreement observed provides further confirmation of the validity of the method discussed here.

Compared with previous work done by Ueda *et al.* [5, 6], the present method can give significantly more information on the condensation process, since this method can provide information on the spatial dependence of condensing vapour.

In the method discussed here it has been assumed that the surface of the liquid is smooth and stable. This usually occurs only for fairly low levels of condensing heat flux. It may be possible to extend the technique to situations in which the falling liquid film develops waves and unstabilities by making measurements of the surface profile of the liquid. Such measurements, involving needle contact techniques, have been described by Collier and Hewitt [4].

CONCLUSION

The needle contact method of measuring the thickness of liquid film has been developed for determining the spatial dependence of heat flux within the condenser of a vertical two-phase reflux thermosyphon. The method involves measurement of the thickness of the condensed liquid film. The validity of the method has been demonstrated by measurements within a condenser in which the spatial dependence of heat flux is known.

The method is being applied to the study of condensation phenomena in thermosyphons containing non-condensable gas. Work is in progress on the effect of the gas species, and inclination of the thermosyphon. Instabilities at the gas–vapour interface are also under study. The method is of particular applicability in practical condensers where the thermal impedance of the condenser wall is not large compared with that of the condensed liquid film.

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MESURE DU TRANSFERT THERMIQUE AVEC CONDENSATION DANS UN THERMOSIPHON

Résumé—On décrit une nouvelle technique pour étudier la condensation de vapeur dans un thermosiphon de reflux diphasique. Le technique permet une mesure précise de l'épaisseur du film liquide condensé en utilisant une méthode de contact à aiguille tournante. Pour l'écoulement laminaire de ce liquide ($Re_L < 10$), cette information peut être utilisée pour déterminer la dépendance spatiale du flux thermique dans le condenseur. La validité de la méthode est démontrée par l'accord entre les résultats obtenus et ceux prédits pour un système condenseur calculable et par la mesure directe de la différence de température à travers l'impédance thermique connue de la paroi du condenseur. La technique est utilisée pour étudier la condensation en présence de gaz incondensable.

MESSUNGEN ZUM WÄRMEÜBERGANG BEI KONDENSATION IN EINEM THERMOSYPHON

Zusammenfassung—Es wird eine neuartige Technik zur Untersuchung der Kondensation in einem Zwei-Phasen-Rückstrom-Thermosyphon vorgestellt. Die Technik beruht auf einer genauen Messung der Filmdicke über den Kontakt einer Nadel. Für laminare Filmströmung ($Re_L < 10$) kann diese Information dazu benutzt werden, die räumliche Verteilung des Wärmestroms im Kondensator zu bestimmen. Die Anwendbarkeit der Methode wird durch Vergleich der Versuchsergebnisse mit solchen aus einem berechenbaren Kondensatorsystem gezeigt. Zusätzlich wird die Temperaturdifferenz quer zur Kondensatorwand, deren Wärmeleitwiderstand bekannt ist, direkt gemessen. Die beschriebene Technik wird nun dazu benutzt, die Kondensation in Gegenwart von Inertgasen zu untersuchen.

ОПРЕДЕЛЕНИЕ ТЕПЛОПЕРЕНОСА ПРИ КОНДЕНСАЦИИ В ТЕРМОСИФОНЕ

Аннотация—Описывается новая методика исследования конденсации пара в двухфазном термосифоне. Методика включает точное измерение толщины пленки конденсированной жидкости методом контакта с вращающейся иглой. При ламинарном течении жидкости ($Re_L < 10$) эта информация может использоваться для определения пространственной зависимости теплового потока в конденсаторе. Применимость метода иллюстрируется согласием между полученными на его основе данными и результатами для вычисляемой системы конденсаторов, а также непосредственным измерением разности температур вследствие термического импеданса стенки конденсатора. Предложенная методика используется для исследования конденсации при наличии неконденсирующегося газа.