*Int. J. Heat Mass Transfer.* Vol. 17, pp. 1241–1251. Pergamon Press 1974. Printed in Great Britain

# DIRECT CONTACT CONDENSATION OF AN IMMISCIBLE VAPOR ON A THIN FILM OF WATER*

A. TAMIR and I. RACHMILEV
Department of Chemical Engineering, University of the Negev, Beer-Sheva, Israel

(*Received* 9 *May* 1973 *and in revised form* 3 *January* 1974)

**Abstract**—Experiments involving direct contact condensation of immiscible organic vapors on water were carried out at atmospheric pressure for the determination of the heat-transfer coefficients of the interface. The system used consisted of a thin film with a spherical geometry on which condensation occurs. The flow is laminar and the atmosphere of the condensing vapor is saturated and stagnant.

Results of the average heat-transfer coefficients of the interface, $\bar{h}_i$, in cal/cm$^2$s°C were: 0·072 for N-pentane, 0·066 for methylenechloride, 0·035 for freon-113 and 0·02 for 1-1 dichloroethane. These results were correlated as a function of the normal boiling point temperature of the vapors by a relationship suggested from nucleation theory.

It was found that the relative magnitude of the surface resistance, $1/\bar{h}_i$, as compared to the total resistance, is appreciable and is in the order of 50 per cent.

## NOMENCLATURE

| | |
|---|---|
| $B$, | dimensionless heat-transfer coefficient of the interface, equation (13); |
| $C_p$, | specific heat; |
| $\bar{C}^*$, | dimensionless average concentration in Fig. 3; |
| $D$, | binary diffusivity in Fig. 3; |
| $g$, | acceleration due to gravity; |
| $h_i$, | heat-transfer coefficient of the interface; |
| $\bar{h}_i$, | mean value of $h_i$ for a particular vapor computed from all experimental results; |
| $\bar{h}$, | overall heat transfer coefficient, defined by equation (15); |
| $k$, | thermal conductivity; |
| $L$, | volumetric flow rate; |
| $M$, | molecular weight; |
| $\overline{Nu}$, | average Nusselt number; equation (16); |
| $N$, | $\lambda_1/C_p(T_\infty - T_0)$; parameter in the solution which takes into account the effect of the mass transfer, Figs. 6–9; |
| $p$, | $k\rho C_p/k_1\rho_1 C_{p1}$, parameter in the solution which takes into account the effect of the mass transfer, Figs. 6–9; |
| $R$, | sphere radius; |
| $\bar{R}$, | universal gas constant, equation (19); |
| $Re$, | Reynolds number defined as $\dfrac{L}{R\nu}$; |
| $S$, | co-ordinate parallel to the liquid surface in the direction of flow; |
| $u_i$, | liquid surface velocity, equation (2); |
| $T$, | local temperature; |
| $T^*$, | dimensionless temperature, equation (6); |
| $X$, | co-ordinate perpendicular to the film surface; |
| $y$, | dimensionless co-ordinate, equation (6); |
| $Z$, | dimensionless co-ordinate, equation (8); |
| $Z_f$, | value of $Z$ at $\theta \cong 176°$, equation (9). |

Greek symbols

| | |
|---|---|
| $\alpha$, | heat diffusivity; |
| $\delta$, | condensate film thickness; |
| $\Delta$, | film thickness (Fig. 1); |
| $\Delta_1$, | film thickness at $\theta = 90°$ (Fig. 1); |
| $\theta$, | angle specifying position on a sphere (Fig. 1); |
| $\rho$, | density; |
| $\nu$, | kinematic viscosity; |
| $\lambda$, | latent heat. |

Subscripts and superscripts

| | |
|---|---|
| 0, | inlet; |
| $c$, | critical; |
| $f$, | outlet, designates that the film made a path on the sphere of about 176°; |
| $i$, | interface; |
| $-$, | average; |
| $\infty$, | bulk of the vapor; saturated conditions; |
| 1, | designates the condensate film. |

---

*Part of the M.Sc. Thesis of I. Rachmilev.

## INTRODUCTION

DIRECT contact heat transfer between immiscible fluids is an effective way of achieving high rates of heat transfer as compared to methods where metallic walls are present in the heat-exchanging system. Among other advantages of this method are the reduction of scale problems and the convenience in separation of the two fluids.

Condensation studies in direct contact were mainly concerned with a single vapor condensing on its own coolant or with a vapor containing a noncondensable gas. There have been, however, very few investigations of direct contact condensation of a single vapor on a coolant in which the condensed vapor is immiscible. This is inspite of its promising use in desalination processes [1–2], for example. In this regard we may mention the work of Goren *et al.* [3] who condensed steam directly on falling drops of Aroclor oil of uniform size, produced by breakup of a cylindrical jet. They concluded that the heat transfer can be predicted reasonably well by using the rigid sphere model for the drops and penetration theory for the jet. In other words, the condensation heat transfer is controlled by the conduction in the drop rather than by surface phenomena. Maa and Hickman, on the other hand, showed recently [2] in very interesting and convincing experiments, that surface phenomena are indeed important. They used a cylindrical jet "condenser" and were able to modify the coolant surface properties with respect to increasing condensation rates of steam by adding small quantities of additives either into the oil or to the condensing vapor. They explained their results according to the theory of heterogeneous nucleation which assumes that the condensate nucleates in the form of tiny droplets on the coolant.

More about the phenomena occurring in direct contact condensation on a coolant of an immiscible vapor may be gathered from non-direct contact condensation experiments of immiscible mixtures on metallic surfaces [4–9]. These experiments were mainly restricted to binary mixtures. Several models were visualized for the condensation of immiscible mixtures. For example, one liquid might form a continuous film on the metallic wall, where the other one rides on the top of it, also in the form of a continuous film. However this laminar two-film model was found incapable of handling the experimental data of Sykes and Marchello [4]. Another condensation model adopted by several investigators [4–9] was as follows: the liquid (usually the organic phase) which wets the metallic wall better condenses in the form of a continuous film; the second liquid (usually water) forms discrete droplets resting on it. Another possibility might be that one liquid occupies part of the condensing surface while the remainder is occupied by the second liquid. It was only recently that Bernhardt *et al.* [9] for the first time obtained visual information concerning this process. They used a high-speed camera and photographed through a microscope the condensation of mixtures of water plus immiscible organic liquids onto a vertical flat surface. Indeed they observed the film-drop behaviour but found that the overall phenomena is even much more complex. On the other hand, they were successful in correlating their data and those of others in a very simple way as compared to previous and more complicated correlations.

The objective of the present investigation was to obtain experimental data for the heat-transfer capability in direct contact condensation on water of various organic vapors. The condensed vapors were immiscible in the water. Our measurements were directed towards the determination of the resistance to heat transfer due to the immiscible condensate. To the best of the present authors' knowledge such data are unavailable. The magnitude of this resistance is important for the design of direct contact units as well as for shedding some light on the condensation phenomena. The experimental apparatus consisted of a thin laminar film which was generated as a flowing water jet spread over the surface of a sphere. On the surface of the film, condensation occurred of a saturated vapor which was immiscible in the water. Applying a heat-transfer model which assumes a constant surface resistance and the temperature measurements made it possible to extract reliable data for the interfacial resistance.

## ANALYSIS OF PHYSICAL MODEL

The physical model consists of a thin film which is generated as liquid spreads over a sphere of radius $R$. The initial temperature of the film is $T_0$ and it is heated by direct contact condensation on its surface by an immiscible vapor at saturation temperature $T_\infty$. Due to the spherical geometry the film thickness, $\Delta$, is continuously decreasing and reaches a minimum value, $\Delta_1$, corresponding to $\theta = 90°$ (Fig. 1). From hereon the thickness increases. Lynn *et al.* [10] who utilized the solution for alminar flow down an inclined plane to approximate the velocity profile in the film, namely,

$$u = u_i(1 - x^2/\Delta^2) \tag{1}$$

obtained the following relationships:

$$\Delta = \Delta_1(\sin\theta)^{-2/3}; \quad u_i = \frac{3L}{4\pi R\Delta_1}(\sin\theta)^{-1/3} \tag{2}$$

where

$$\Delta_1 = \left(\frac{3L\nu}{2\pi Rg}\right)^{1/3} \tag{3}$$

$u_i$ is the free surface velocity, $L$ is the volumetric flow rate and $\nu$ is the kinematic viscosity. In analogous

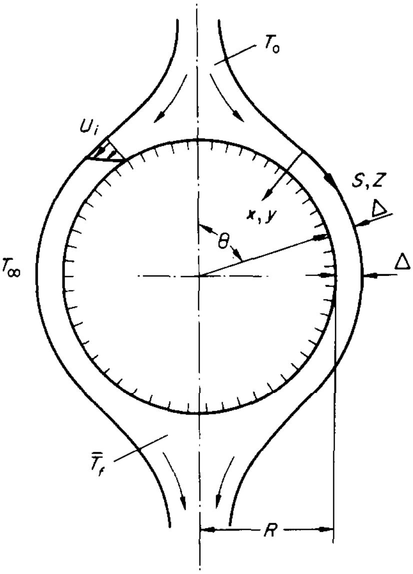

FIG. 1. Physical model and co-ordinates.

way to mass transfer [11], the heat equation in the co-ordinate system depicted in Fig. 1 is given by:

$$u_i\left[1-\left(\frac{x}{\Delta}\right)^2\right]\frac{\partial T}{\partial S}=\alpha\frac{\partial^2 T}{\partial x^2} \tag{4}$$

where $S$ is a distance parallel to the surface in the direction of flow and $x$ is a direction perpendicular to the film surface. In deriving equation (4) it is assumed that $\Delta$ and $x$ may be ignored in comparison to $R$. Practically these conditions hold approximately for $\theta_0 \cong 4°$ up to $\theta_f \cong 176°$. Other simplifying assumptions which may be considered to be reasonably representative of practical conditions prevailing were:

(1) The condensing vapor is saturated and stagnant and hence, the film moves in a medium of constant temperature $T_\infty$.

(2) The resistance to condensation of the immiscible vapor on the coolant is expressed by a coefficient $1/h_i$. $h_i$, the heat-transfer coefficient of the interface, is assumed to be constant.

(3) Effects of interfacial drag and reaction on the film due to vapor initially at rest, on the laminar velocity distribution are ignored.

(4) Axial heat conduction is not being considered here.

(5) The physical properties are assumed constant.

It should be mentioned that part of these assumptions were verified from absorption experiments carried out in our system [12] and by others.

Since $u_i$ and $\Delta$ are functions of $S$ only, a variable $Z$ may be defined:

$$\frac{\mathrm{d}Z}{\mathrm{d}S}=\frac{\alpha}{u_i\Delta^2}. \tag{5}$$

Other dimensionless variables are:

$$T^*=\frac{T_\infty-T}{T_\infty-T_0}; \qquad y=\frac{x}{\Delta}. \tag{6}$$

Hence equation (4) reduces to:

$$(1-y^2)\frac{\partial T^*}{\partial Z}=\frac{\partial^2 T^*}{\partial y^2} \tag{7}$$

where the axial co-ordinate reads

$$Z=\frac{4}{3}\frac{\alpha R^2}{L\Delta_1}\int_{\theta_0}^{\theta}(\sin\theta)^{5/3}\,\mathrm{d}\theta. \tag{8}$$

If $\theta_0 = 0$ and $\theta = \pi$, the above integral is equal to 1·68. It changes only by 0·05 per cent when the limits are 4 and 176°. Hence, after rearrangements, one obtains that the value of $Z = Z_f$ after the film made a path of about 176° over the sphere is:

$$Z_f=89{\cdot}4\left(\frac{R^2\alpha}{L}\right)\left(\frac{R}{Lv}\right)^{1/3} \tag{9}$$

The boundary conditions are:

$$Z=0 \quad : \quad T^*=1 \tag{10}$$

$$y=1 \quad : \quad \frac{\partial T^*}{\partial y}=0 \tag{11}$$

and according to assumption (2),

$$\text{at} \quad y=0 \quad : \quad \frac{\partial T^*}{\partial y}=BT^*(\sin\theta)^{-2/3} \tag{12}$$

where $B$ is a constant dimensionless interfacial heat transfer coefficient defined by

$$B=\frac{h_i\Delta_1}{k}. \tag{13}$$

The "mixing cup" temperature measured in our experiments, is calculated from:

$$\bar{T}^*=\frac{\int_0^1(1-y^2)T^*\,\mathrm{d}y}{\int_0^1(1-y^2)\,\mathrm{d}y}=\frac{3}{2}\int_0^1(1-y^2)T^*\,\mathrm{d}y. \tag{14}$$

In heat-transfer calculations, one is usually interested to obtain information concerning the capability of heat absorption by a given area. For the sphere, it is obtained from

$$L\rho C_p(\bar{T}_f-T_0)=\bar{h}4\pi R^2\frac{(T_\infty-T_0)-(T_\infty-\bar{T}_f)}{\ln[(T_\infty-T_0)/(T_\infty-\bar{T}_f)]} \tag{15}$$

which finally may be represented by the average Nusselt number as

$$\overline{Nu} = \frac{\bar{h}\Delta_1}{k} = -\frac{0{\cdot}56}{Z_f}\ln \overline{T}_f^* \tag{16}$$

where $\bar{h}$ is an overall heat-transfer coefficient.

Equation (7) was elsewhere [11] solved for a limiting case where surface resistance, $1/B$, is ignored, namely $B = \infty$. The general case described here, where surface resistance is finite, was not amenable to an analytical solution and was solved numerically using Crank–Nicolson and the Gauss's elimination methods [13]. The results for the "mixing cup" temperature (which served us for extracting values of $h_i$) vs $Z_f$, with $B$ as parameter, are given in Figs. 6–9 over a wide range of operating conditions. General curves for $\overline{Nu}$ are given in Fig. 2.

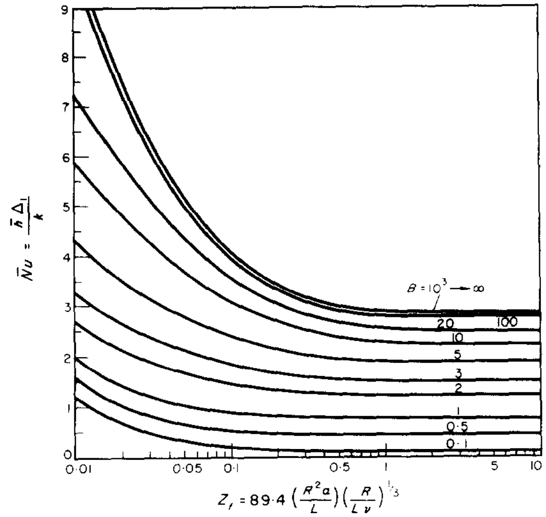

FIG. 2. Average overall Nusselt number.

## EXPERIMENTAL

The following vapors were directly condensed at their normal boiling point on a water film: normal pentane, methylenechloride, Freon-113 and 1-1 dichloroethane. The laminar water film was formed as a liquid jet, issuing from a round orifice, impinges on the top of a sphere and spreads over its surface. Our major aim was to determine the surface resistance to heat transfer, $1/h_i$. This was achieved by measuring $\overline{T}_f^*$ and the quantities required for the determination of $Z_f$. By using the theoretical model results [Figs. 6–9], it was possible to obtain values for $B$ or $h_i$.

Care was taken concerning the following operating conditions:

(1) Laminar flow of the film: it was found that no experimental data were available concerning the transition from laminar to turbulent flow for the sphere model as compared to much information which has been gathered from studies [14] on films having a constant thickness. Therefore, as a preliminary step, the above problem was experimentally studied as detailed in [12]. Briefly, the transition from laminar to turbulent flow was determined from experiments on oxygen absorption in water and it was decided to accept this transition "point" once a deviation from the laminar absorption theory occurred. Figure 3 demonstrates part of the absorption results obtained for a particular sphere out of four which were tested. The major conclusions drawn from this study were:

(a) The transition of the spherical film from laminar to turbulent flow may be characterized by the following critical Reynolds number

$$Re_c = \frac{L/R}{\nu} = 300. \tag{17}$$

(b) No interfacial resistance exists in the absorption of oxygen into water.

(2) Elimination of noncondensable gases: it is well-known that noncondensable gases (usually air) which are present in small quantities in the bulk of the vapor, tend to accumulate at the condensing surface [15] and act as surface resistance. In our study, the elimination of this effect is crucial since we want to detect interfacial resistance which may be caused by other sources. Hence experiments were carried out a few millimeters above atmospheric pressure and with a continuous and small bleed-out of the condensing vapor. Other sources of noncondensable gases were the dissolved air in the water feed and the organic liquid from which vapors were generated. These were eliminated by de-aeration of the water prior to its entering the condensation cell and by boiling the organic liquid before it was supplied to the system.

(3) Stagnant atmosphere of the condensing vapor: in preliminary absorption [12] and heat-transfer experiments it was observed that uncontrolled vapor bleed-out might increase, as expected, the values of $h_i$ since convective motion is set up in the vapor atmosphere. Hence to achieve the above bleed-out, the effect of the heat input required to boil the organic vapor was also tested for optimal operation.

(4) Elimination of heat conduction through the sphere walls was achieved by creating a vacuum inside the sphere.

The essential parts of the experimental system (Fig. 4) were as follows:

*Condensation cell* (*Fig.* 5)

Vapor condensation occurred on the water film inside a glass cell consisting of a cover and a vessel with a heating jacket. The total volume of the cell was about 1·5 l and it was connected to the vapor boiler by a glass tube.

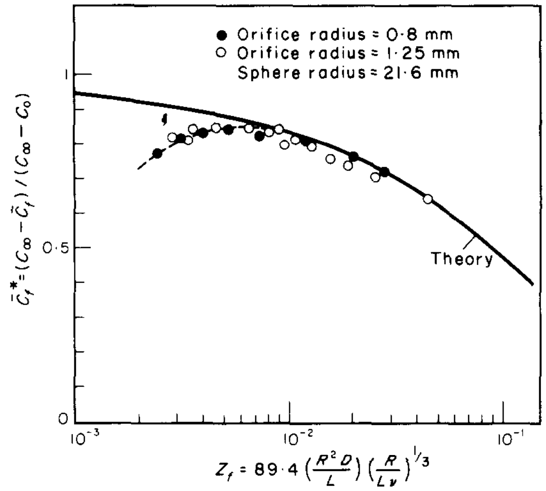

FIG. 3. Oxygen absorption results on a sphere.

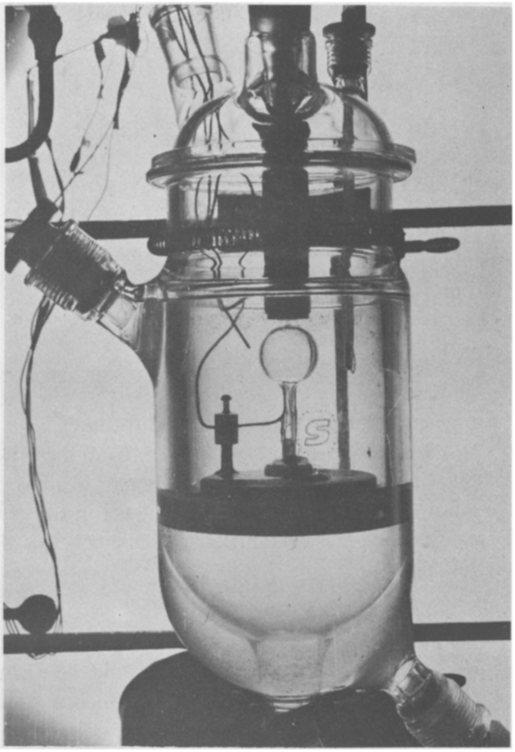

FIG. 5. Condensation cell.

*Orifice holder*

The round orifice with a radius of 0·8 mm was fixed at the bottom of a holder. The holder was placed in one of the cell cover holes and it was possible to regulate its position above the top of the sphere. The distance from the orifice to the water jet impinging point was about 1 mm. The holder was designed to minimize heating in the feed channel leading to the orifice. The feed channel, of 10 mm diameter was drilled in a thick Teflon rod of 25 mm dia. and 200 mm length providing thermal insulation to the flowing water. A tap for temperature measuring was drilled at the minimum practicable distance above the orifice.

*Spheres*

To obtain the spherical film, glass spheres were blown and the air removed. They were mounted on a glass rod of 100 mm length and about 3 mm dia. The radii of the spheres used were 15·5, 17·6 and 21·6 mm. The sphere was placed in a Teflon carrier fixed in a stainless plate mounted in the vessel. Several holes were made in the plate for the water condensate drainage. To reduce condensation over the stainless plate, another pierced Teflon plate was placed.

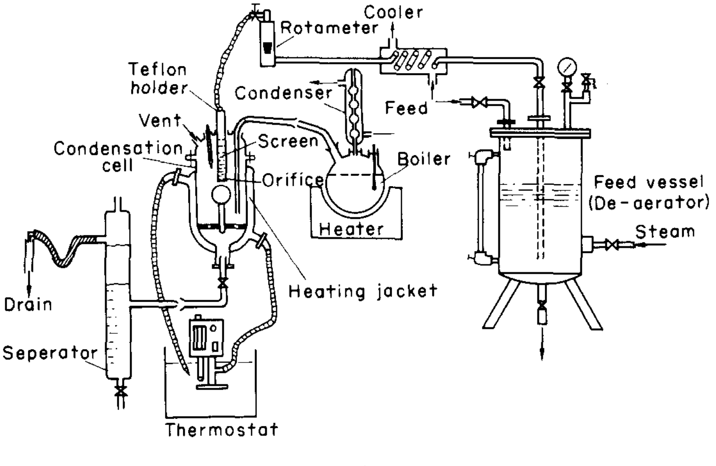

FIG. 4. Experimental system.

Table 1. Range of operating conditions

| Vapor | $T_\infty$ (°C) | $(T_\infty - \bar{T}_f)$ (°C) | $(T_\infty - T_0)$ (°C) | $L$ (cm$^3$/s) | $\Delta_1$ (cm) | $Re = \frac{L}{R\nu}$ |
|---|---|---|---|---|---|---|
| $N$-pentane | 33–35 | 0·9–5 | 3–13 | 1·1–4·6 | 0·014–0·0204 | 82–307 |
| Methylenechloride | 37·5–38 | 2–10 | 7–18 | 1·8–5·4 | 0·0165–0·023 | 146–310 |
| Freon-113 | 45–46 | 6–16 | 15–27 | 1·3–5·4 | 0·0141–0·0216 | 92–292 |
| 1-1 Dichloroethane | 53–55 | 13–29 | 23–35 | 1·3–4·9 | 0·0148–0·0221 | 105–275 |

Total pressure ≅ 740 mm Hg.
Orifice radius to produce the film = 0·8 mm.

*Water–condensate separator*

This was made from glass tube of 70 mm dia. and 750 mm length. The drainage flowed through a glass tube to the middle of the separator. A rubber tube was placed in the upper part of the separator and by regulation of its height, the drainage level in the condensation cell could be adjusted to remain below the stainless plate.

*Boiler*

This was a glass flask 3 l in volume, placed in a heating jacket, the power of which could be varied for optimal operation. A total reflux condenser was fixed in the flask and continuous boiling released the possible dissolved air in the organic liquid. The boiler was connected by a glass tube to the condensing cell for vapor supply.

*Feed vessel*

This was made from stainless steel with a total volume of about 70 l. Steam was bubbled through the water feed which was kept at its boiling point and hence de-aerated. The hot water was injected through the orifice by steam pressure of about 2 atm. Before reaching the orifice, the water was brought to room temperature by the cooler shown in Fig. 4.

*Measurements*

In carrying out runs, direct measurements were taken of the temperature differences, $(T_\infty - T_0)$ and $(T_\infty - \bar{T}_f)$ and hence $\bar{T}_f^*$, by Pyrotenax mineral insulated and pre-assembled copper–constantan thermocouples. Special care was taken to ensure that the thermocouple which measure $\bar{T}_f$ was always covered with the drainage liquid. To achieve this, a groove of 10 mm length was made in the glass rod on which the sphere was mounted. In the groove, the bent thermocouple was fixed and tightened, as shown in Fig. 5, so that its tip measured exactly the temperature after the film made a path over the sphere of about 176°.

The quality of the de-aerated water was determined by measuring the oxygen content in water using the well-known Winkler's method. The quantities were found negligible (less than 0·1 ppm) which indicates the high efficiency of the de-aeration technique adopted here. Additional measurements were the ambient temperature of the vapor in the condensation cell, inlet water temperature and its flow rate. Steady-state conditions were achieved after about one hour.

*Experimental precision and treatment of data*

A single value of $h_i$ from the measured quantities $\bar{T}_f^*$ and $Z_f$ was obtained from the theoretical curves of $\bar{T}_f^*$ vs $Z_f$ with the dimensionless $h_i$, namely $B$, as parameter. An estimate of the relative error $\Delta B/B$, (due to errors employed in the various measurements), was made which showed that for a single measurement, $\Delta B/B = 15$–$20$ per cent. This is not unusual in heat-transfer measurements and therefore a considerable number of measurements were taken and replication of these were made. The physical properties of the liquid film were taken from [16, 17] and evaluated at an average temperature between inlet and outlet to the condensation cell.

## RESULTS AND DISCUSSION

Figures 6–9 contain the results of the direct contact condensation on water of the various vapors. The solid curves represent the theoretical values of the average temperature $\bar{T}_f^*$ plotted vs $Z_f$ with the dimensionless interfacial heat-transfer coefficient, $B$, as a parameter. Each data point is the average of five measurements obtained in one run. Table 1 describes the range of the operating conditions.

As previously noted, values of $h_i$ were deduced from the theoretical curves. In Table 2, detailed results for $N$-pentane obtained in twenty runs are reported to demonstrate typical variations in the values of $h_i$ which were also similar in the other cases.

Table 2. Experimental results of $h_i$ (cal/cm$^2$ s °C) for $N$-pentane

| Runs 1–5 | Runs 6–10 | Runs 11–15 | Runs 16–20 |
|---|---|---|---|
| 0·078 | 0·092 | 0·056 | 0·085 |
| 0·064 | 0·069 | 0·080 | 0·075 |
| 0·060 | 0·076 | 0·070 | 0·055 |
| 0·100 | 0·077 | 0·077 | 0·060 |
| 0·094 | 0·076 | 0·048 | 0·052 |

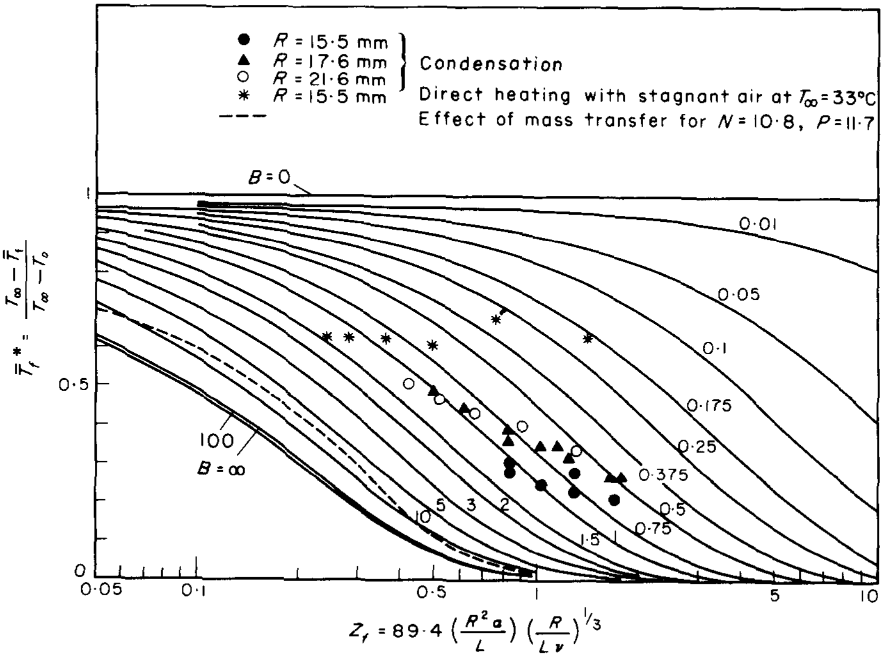

FIG. 6. Condensation of $N$-pentane on water.

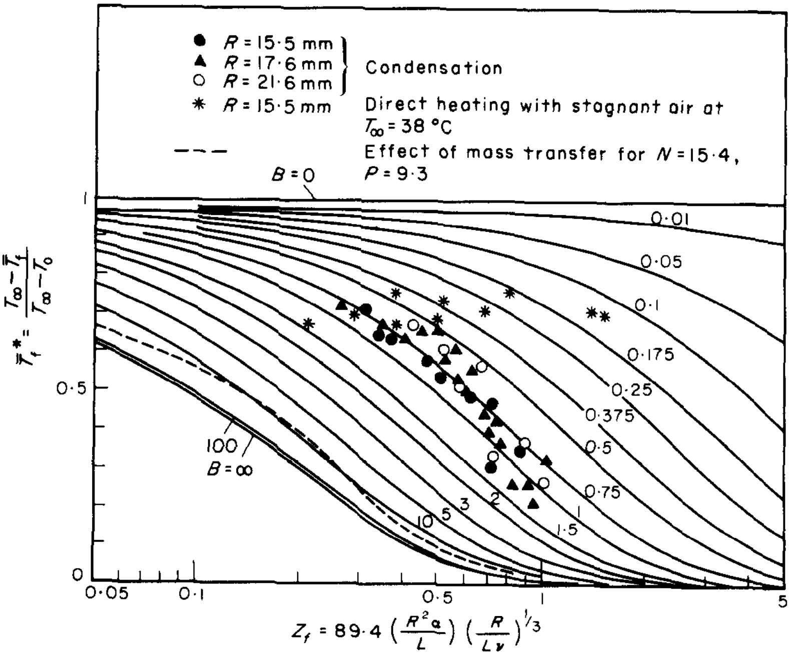

FIG. 7. Condensation of methylene chloride on water.

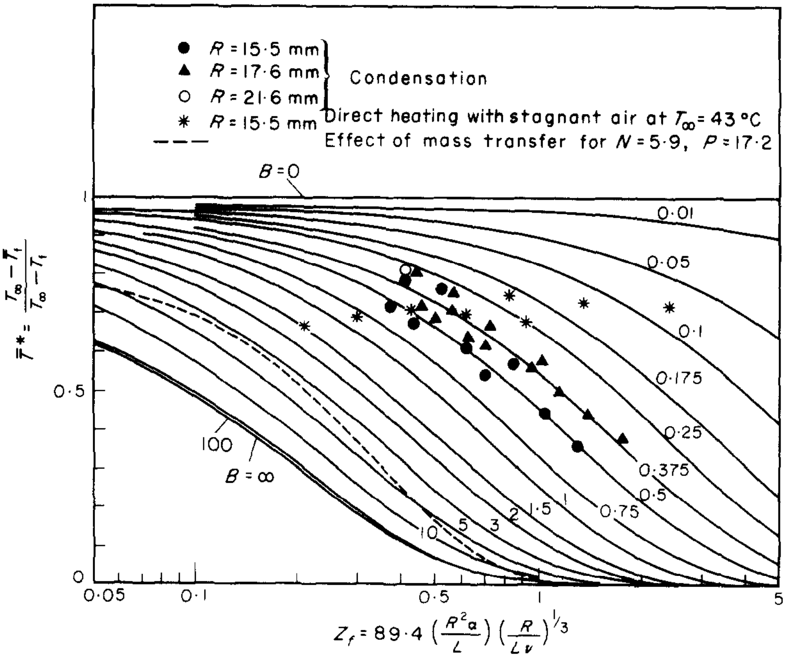

FIG. 8. Condensation of Freon-113 on water.

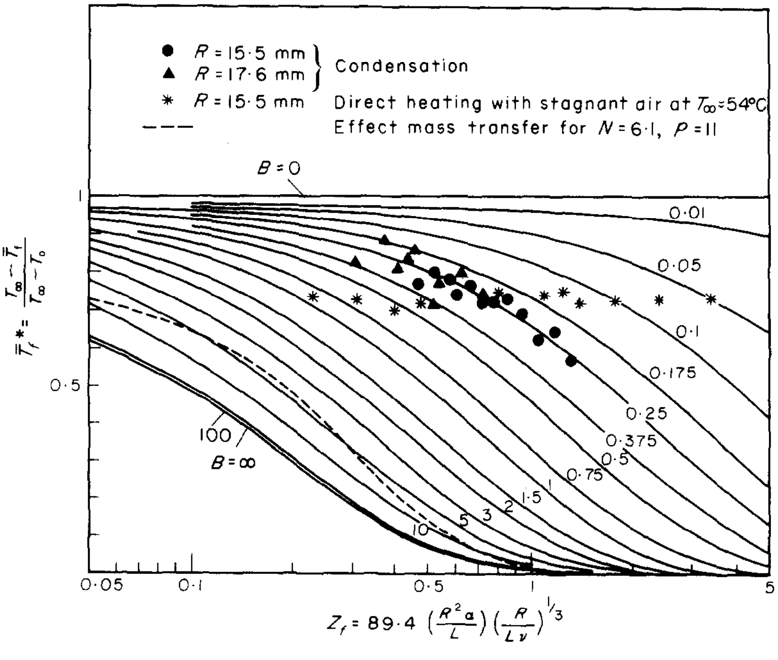

FIG. 9. Condensation of 1-1 dichloroethane on water.

Noting the precision in the measurements, the results were considered to be a randomly scattered replication of the same measurement. Hence, each system was characterized by a single mean value for the interface coefficient of the heat transfer, designated as $\bar{h}_i$. A summary of the final results are reported in Table 3 where the given precision is the confidence range at 0·05 significance level.

Table 3. Main results—mean interface coefficients of heat transfer, $\bar{h}_i$, for the direct contact condensation on water

| Vapor | $T_\infty$ (°C) | $\bar{h}_i$ (cal/cm$^2$ s °C) |
|---|---|---|
| $N$-pentane | 33 | 0·072 ± 0·002 |
| Methylenechloride | 38 | 0·066 ± 0·002 |
| Freon-113 | 45 | 0·035 ± 0·001 |
| 1-1 Dichloroethane | 54 | 0·020 ± 0·001 |

For practical purposes one is usually interested to know the relative magnitude of the surface resistance, $1/\bar{h}_i$, as compared to the total resistance, $1/\bar{h}$, where $\bar{h}$ is defined by equation (15). This may be obtained from the simple series summation formula

$$\frac{1}{\bar{h}} = \frac{1}{\bar{h}_i} + \frac{1}{\bar{h}_{L\infty}} \tag{18}$$

where $\bar{h}_{L\infty}$ is the average heat-transfer coefficient of the liquid film deduced from the case of no interfacial resistance. $\bar{h}_{L\infty}$ is also defined by equation (15) and may be calculated from equation (16) for values of $\bar{T}_f^*$ (Fig. 6) corresponding to $B = \infty$. It should be noted that although equation (18) is an approximate approach, it gives accurate results as compared to the exact solution [19]. Figure 10 contains theoretical curves for the ratio between the interfacial resistance to the total resistance as a function of $Z_f$ with $B$ as parameter. The curves were computed by equation (18) with the aid of Fig. 6. The dotted zone represents our experimental range for the results reported in Figs. 6–9 for all vapors which were tested. It may be seen that the interfacial resistance is appreciable and amounts to 50 per cent of the total resistance. Hence, in the design of direct contact condensation units for immiscible fluids, attention should be paid to the magnitude of the interfacial resistance.

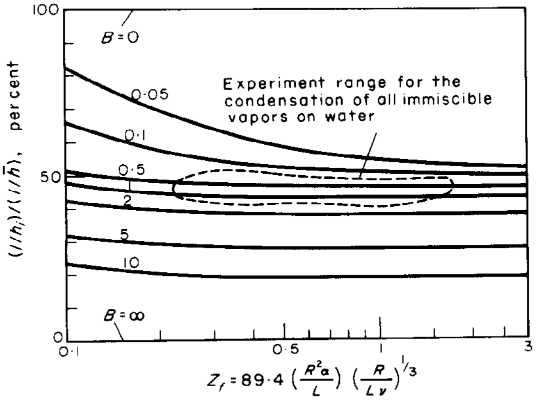

FIG. 10. Percentage of interfacial resistance from the total resistance.

The results obtained so far suggest that the heat-transfer coefficient of the interface, $\bar{h}_i$, is much less than the maximum one predicted from the well-known Hertz–Knudsen model. Following the assumption that all molecules striking the condensing surface indeed condense (namely, the condensation coefficient is unity), the above model yields the following expression for $h_i$ [18]:

$$h_i = \frac{\lambda^2}{v_\infty T_\infty^{3/2}}\left(\frac{M}{2\pi \bar{R}}\right)^{1/2} \tag{19}$$

where $v_\infty$ is the specific volume of the saturated vapor corresponding to $T_\infty$. Calculated values of $h_i$ (cal/cm$^2$ s °C) for the vapors in the order given in Table 3 at their normal boiling point were: 52, 56, 43, 67. As may be seen, our experimental values are about one thousand-fold lower than the maximum attainable quantities. Actually the maximal quantities correspond to the curve $B = \infty$ in Figs. 6–9.

The possibility that the interfacial resistance, $1/\bar{h}_i$, is caused by the additional thickness of the immiscible condensate was checked. This was performed by assuming that the condensate forms a laminar film flowing over the water film while $B = \infty$. The results are given by the dotted lines in Figs. 6–9 which show that the effect of the mass transfer is small especially in the experimental range, and is inadequate to explain the low values of $\bar{h}_i$.

Another possible reason for the low coefficients might be due to a different mechanism by which the spherical water film is heated. Since we deal with dissimilar fluids, it is possible that the major heating of the film is achieved by convection due to direct contact with a hot environment, rather than by condensation (although the atmosphere is saturated). This possibility was tested as follows: the stagnant saturated air atmosphere in the condensation cell was heated and maintained at a temperature identical to the saturation temperature of the various vapors. This was effected by passing water through the heating jacket (Figs. 4 and 5). By covering the entire range of the film flow-rates as in the condensation experiments, the starred points shown in Figs. 6–9 were obtained. It may be observed that in all cases, $\bar{T}_f^*$ remain nearly constant and almost independent of the environment temperature. This was in contrast to the behaviour in the condensation experiments. In addition, values of

$h_i$ increased by about tenfold between extreme values of $Z_f$ as compared to the variations demonstrated in Table 2. It was also found that all air heating data may be well correlated by $B = 0{\cdot}00023\,Re^{1{\cdot}45}$ where the Reynolds number is that of the liquid ( equation 17). It should be noted that the above behavior may be explained as reasonable according to the boundary layer theory where a boundary layer of air is probably formed due to interfacial drag with the liquid film. The above observations indicate, however, that the low values of $\bar{h}_i$ cannot be explained by the above-mentioned phenomena.

According to nucleation theory the rate of critical nuclei formation $r$, is given as [20], $r = v\exp(-\Delta G/kT_\infty)$. $v$ is the frequency factor for nucleation, $\Delta G$ is the free energy change and $k$ is Boltzmann constant. Assuming that the heat-transfer coefficient of the interface is proportional to $r$, and following the approximation made by Sykes and Marchello [4] that the free-energy change may be expected to be proportional to some value of the difference $(T_\infty - \bar{T})$, the results yield that $\bar{h}_i = A\exp(E/T_\infty)$. In other words, $\bar{h}_i$ decreases with the increasing saturation temperature $T_\infty$. Indeed, Fig. 11, shows such behavior where experimental values of $\bar{h}_i$ for each vapor (Table 3) were reasonably correlated by:

$$\bar{h}_i = 0{\cdot}225 \times 10^{-10}\exp\left(\frac{6735}{T_\infty}\right). \qquad (20)$$

$T_\infty$ is the absolute temperature in °K and $\bar{h}_i$ is in cal/cm$^2$ s °C. Supporting evidence to the nucleation mechanism in direct contact condensation of a single vapor on a coolant may also be found in the study of Maa and Hickman [2] who investigated the condensation of steam on a cylindrical jet of paraffinic oil. It is also of interest to mention that for such a chemical system, heat-transfer coefficients of the interface were in the order of 0·01–0·04, which are similar to ours. Goren [3], on the other hand, found that condensation heat transfer of steam over an oil drop is controlled by conduction through the drop rather than by nucleation phenomena.

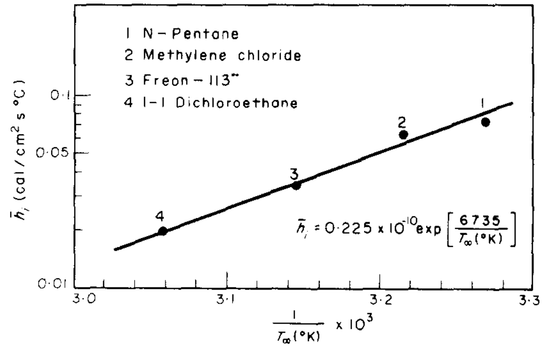

FIG. 11. Correlation of the heat-transfer coefficients of the interface.

The major conclusions which are suggested from our study are:

(1) When vapors of $N$-pentane, methylenechloride, freon-113 and 1-1 dichloroethane are directly condensed on water in which they are immiscible, the heat-transfer coefficients of the interface (Table 3) are remarkably low. Their correlation (equation 20) gives some evidence that nucleation phenomena are involved in this process.

(2) The magnitude of the interfacial resistance, $1/\bar{h}_i$ was found equal to the average resistance, $1/\bar{h}_{L\infty}$, offered by the water film on which condensation took place. Hence it is recommended to consider its effect in the design of direct contact condensation units for disimilar fluids.

*Acknowledgement*—The authors wish to thank Professor S. Sideman for his advice and Professor Y. Taitel for reviewing the manuscript.

## REFERENCES

1. K. S. Spiegler, *Principles of Desalination.* Academic Press, London (1966).
2. J. R. Maa and K. Hickman, Direct contact condensation of steam on a modified oil coolant, *Desalination* **10**, 95–111 (1972).
3. S. L. Goren and G. R. Wilke, Sea water evaporation by immiscible liquid heat transfer, Sea Water Conversion Laboratory Report No. 67-1, Water Resources Center Desalination Report No. 14, 31–39 (1967).
4. J. A. Sykes and J. M. Marchello, Condensation of immiscible liquids on horizontal tube, *I/EC Process Des. Dev.* **9**, 63–71 (1970).
5. J. Stepanek and G. Standart, Heat transfer during the condensation of mixtures of vapors of immiscible liquids, *Colln Czech. Chem. Commun.* **23**, 995–1011 (1958).
6. M. Tobias and A. E. Stoppel, Condensation of vapors of water and immiscible organic liquids, *Ind. Engng Chem.* **46**, 1450–1455 (1954).
7. W. W. Akers and M. M. Turner, Condensation of vapors of immiscible liquids, *A.I.Ch.E. Jl* **5**, 587–589 (1962).
8. E. Barnea and J. Mizrahi, Heat transfer coefficient in the condensation of a hydrocarbon–steam mixture, *Trans. Instn Chem. Engrs* **50**, 286–288 (1972).
9. S. H. Bernhardt, J. J. Sheridan and J. W. Westwater, Condensation of immiscible mixtures, *A.I.Ch.E. Symp. Ser.* **68**, 21–37 (1972).
10. S. Lynn, J. R. Straatemeier and H. Kramers, Absorption studies in the light of the penetration theory, *Chem. Engng Sci.* **4**, 63–67 (1955).
11. W. E. Olbrich and J. D. Wild, Diffusion from the free surface into a liquid film in laminar flow over defined shapes, *Chem. Engng Sci.* **24**, 25–32 (1969).
12. A. Tamir and I. Rachmilev, Experimental study on turbulence onset in flows of thin films over single spheres, *Chem. Engng Sci.* **28**, 1931–1934 (1973).
13. G. D. Smith, *Numerical Solution of Partial Differential Equations.* Oxford University Press, London (1965).

14. G. D. Fulford, The flow of liquids in thin films, *Adv. Chem. Engng* **5**, 151–236 (1964).
15. A. Tamir and Y. Taitel, Improving condensation rates by interfacial suction and forced convection in the presence of noncondensable gases, *Israel J. Technol.* **9**, 69–81 (1971).
16. J. H. Perry, *Chemical Engineer's Handbook*, 4th Edn. McGraw-Hill, New York (1963).
17. W. H. McAdams, *Heat Transmission*, 3rd Edn. McGraw-Hill, New York (1954).
18. A. Tamir and D. Hasson, Evaporation and condensation coefficient of water, *Chem. Engng Jl* **2**, 200–211 (1971).
19. A. Tamir and Y. Taitel, Diffusion to flow down an incline with surface resistance, *Chem. Engng Sci.* **26**, 799–808 (1971).
20. A. S. Michaels, *Nucleation Phenomena*. American Chemical Society, Washington D.C. (1966).

## CONDENSATION D'UNE VAPEUR NON MISCIBLE PAR CONTACT DIRECT AVEC UN FILM D'EAU

**Résumé**—On a conduit des expériences sur la condensation de vapeurs organiques non miscibles par contact avec l'eau, à la pression atmosphérique, pour déterminer les coefficients de transfert thermique à l'interface. Le système utilisé est celui du film mince à géométrie sphérique, sur lequel se fait la condensation. L'écoulement est laminaire et la vapeur est saturée et stagnante.

Les résultats sur les coefficients moyens de transfert à l'interface $\bar{h}_i$, sont les suivants en cal/cm$^2$.s.K: 0,072 pour le N-pentane, 0,066 pour le chlorure de méthyle, 0,035 pour le fréon 113 et 0,02 pour le 1-1 dichloréthane. Ces résultats sont reliés à la température d'ébullition normale des vapeurs par une relation suggérée par la théorie de la nucléation.

On trouve que la grandeur relative de la résistance surfacique $1/\bar{h}_i$, comparée à la résistance totale, est appréciable et de l'ordre de 50 pour cent.

## KONDENSATION EINES NICHTMISCHBAREN DAMPFES IM DIREKTEN KONTAKT MIT EINEM DÜNNEN WASSERFILM

**Zusammenfassung**—Zur Bestimmung der Wärmeübergangskoeffizienten an der Phasengrenzfläche wurden bei Atmosphärendruck Experimente ausgeführt, in denen die Kondensation von nicht mischbaren organischen Dämpfen in direktem Kontakt mit Wasser untersucht wird. Das benutzte System besteht aus einem dünnen Film mit sphärischer Geometrie, auf dem kondensiert wird. Die Strömung ist laminar und der Zustand des kondensierenden Dampfes gesättigt und gleichbleibend.

Ergebnisse für den mittleren Wärmeübergangskoeffizienten an der Phasengrenzfläche $\bar{h}_i$ in cal/(cm$^2$ s °C) waren: 0,072 für *n*-Pentan, 0,066 für Methylchlorid, 0,035 für Freon 113 und 0,02 für 1–1 Dichloräthan. Die Ergebnisse wurden als Funktion der normalen Siedepunktstemperatur dargestellt mittels einer Beziehung, die von der Kernbildungstheorie abgeleitet wird.

Es wurde gefunden, daß die relative Größe des Oberflächenwiderstandes $1/\bar{h}_i$ im Vergleich mit dem Gesamtwiderstand beträchtlich ist und die Größenordnung von 50% hat.

## НЕПОСРЕДСТВЕННАЯ КОНТАКТНАЯ КОНДЕНСАЦИЯ НЕСМЕШИВАЮЩЕГОСЯ ПАРА НА ПОВЕРХНОСТИ ТОНКОЙ ПЛЕНКИ ВОДЫ

**Аннотация** — Для определения коэффициентов теплообмена на поверхности раздела экспериментально исследовалась непосредственная контактная конденсация несмешивающихся органических паров на поверхности воды. Используемая система состояла из тонкой сферической пленки, на поверхности которой происходит конденсация. Поток является ламинарным; атмосфера конденсирующегося пара насыщена и неподвижна.

Получены следующие значения средних коэффициентов теплообмена $\bar{h}_i$ на поверхности раздела, измеренные в кал/см$^2$ сек °C: 0,072 для *N*-пентана; 0,066 для хлористого метила; 0,035 для Фреона-113 и 0,02 для дихлорэтана. На основе зависимости, используемой в теории образования ядер, результаты обобщаются в виде функции нормальной температуры в точке кипения.

Найдено, что по сравнению с суммарным сопротивлением относительная величина поверхностного сопротивления $1/\bar{h}_i$ является существенной и имеет порядок 50%.