Int. J. Heat Mass Transfer. Vol. 10, pp. 755–762. Pergamon Press Ltd. 1967. Printed in Great Britain

ON THE MECHANISM OF DROPWISE CONDENSATION

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(*Received* 27 *July* 1966 *and in revised form* 5 *December* 1966)

Abstract—An extension to a recent theory of heat transfer by dropwise condensation [14] is proposed. The modified theory is compared with recent heat-transfer measurements. Graphs showing the predictions of the theory, for heat fluxes outside the present experimental ranges and for different pressures, are presented.

NOMENCLATURE

a,	defined in equation (2);
g,	local gravitational acceleration;
h_f,	specific enthalpy of saturated liquid;
h_g,	specific enthalpy of saturated vapour;
h_{fg},	$h_g - h_f$;
K_1,	a constant (concerning conduction in a drop);
K_2,	a constant (ratio of base to surface area of a drop);
K_3,	a constant (concerning size of largest adherent drop);
K_4,	a constant (concerning surface coverage by moving drops);
L_0,	$2\sigma v_f/h_{fg}$;
L_3,	$\sqrt{(\sigma/g\rho_{fg})}$;
n,	a constant (concerning drop size distribution);
$\tilde{Q}''$,	average heat flux for surface not covered by moving drops;
$\dot{Q}''_b$,	heat flux through base area of a drop;
$\dot{Q}''$,	average heat flux for surface;
$\dot{Q}''_1$,	$\lambda_f T h_{fg}/2\sigma v_f$;
$\dot{Q}''_2$,	$(h_{fg}^2/v_g)\{(\gamma - 1)/(\gamma + 1)\}\sqrt{(2\pi/RT)}$;
R,	specific gas constant for vapour;
r,	drop radius;
$\check{r}$,	minimum drop radius;
$\hat{r}$,	maximum drop radius;
T,	absolute thermodynamic temperature;
t,	time required for coalescence between two drops;
v_f,	specific volume of saturated liquid;
v_g,	specific volume of saturated vapour;
z,	height of condensing surface.

Greek symbols

$\alpha\,(r/\hat{r})$,	fractional area covered by drops having radii in the interval r, $\hat{r}$;
α,	vapour-to-surface heat-transfer coefficient;
γ,	ratio of isobaric to isochoric specific heat capacity for vapour;
ΔT,	effective vapour-to-surface temperature difference;
ϕ,	fractional area covered by falling drops;
λ_f,	thermal conductivity of condensate;
μ_f,	viscosity of condensate;
ρ_f,	density of condensate;
ρ_g,	density of saturated vapour;
ρ_{fg},	$\rho_f - \rho_g$;
σ,	liquid–vapour interfacial tension.

Conversion factors

$$\frac{\text{MW}}{\text{m}^2} \simeq 0{\cdot}317 \times 10^6 \frac{\text{Btu}}{\text{ft}^2\,\text{h}} \simeq 0{\cdot}860 \times 10^6 \frac{\text{kcal}}{\text{m}^2\,\text{h}}$$

$$\frac{\mathrm{MW}}{\mathrm{m^2\,degC}} \simeq 0{\cdot}176 \times 10^6 \frac{\mathrm{Btu}}{\mathrm{ft^2\,h\,degF}}$$

$$\simeq 0{\cdot}860 \times 10^6 \frac{\mathrm{kcal}}{\mathrm{m^2\,h\,degC}}.$$

1. INTRODUCTION

VARIOUS proposals have been made with regard to the mechanism of dropwise condensation. Certain authors, following Jakob [1], have considered that an important role is played by a thin film or layer of condensate, supposed to form between visible drops [2–8]. Recent experimental and theoretical work [9–12] casts serious doubt on the existence of such films and supports the view of McCormick and Baer [13] that nucleation is an essential feature of dropwise condensation.

A theory of heat transfer during dropwise condensation has recently been put forward [14] which does not invoke the existence of condensate films, and which agrees well with recent experimental measurements [15–21]. In this theory, the heat transfer through a single drop is determined and used, in conjunction with an assumed distribution of drop sizes, to calculate the average heat flux.

The form of distribution adopted is, at best, appropriate only for adhering drops. The main purpose of the present work is to make a correction for the part played by the falling drops.

2. CONDENSATE DISTRIBUTION

Le Fevre and Rose [14] used a distribution of drop sizes:

$$\alpha(r/\hat{r}) = 1 - (r/\hat{r})^{1/n} \qquad (1)$$

where $\alpha(r/\hat{r})$ is the fractional area covered by drops having radii in the interval r, $\hat{r}$; $\hat{r}$ is the "effective maximum" drop radius, and n is a constant.

This distribution fits the broad facts that no area is covered by drops greater than the largest and that if the smallest drops have zero radius then the entire surface is covered. Also, for visible drops, the population is observed to increase with decreasing radius, thus $n > 1$.

In the present work we shall distinguish between two types of drops—the adhering and the falling. While it is possible that equation (1) might represent the distribution of drop sizes from the smallest to the largest adhering drops, it could not include the falling drops. These latter are entirely different in character. Their manner of growth, by sweeping up the adhering drops in their path, is different from that by condensation and random coalescences as in the case of the adhering drops. This is illustrated by the photographs (Fig. 1) showing the appearance of a vertical condensing surface with different condensation rates.

It has been observed [16, 19] that, over a wide range of heat flux, the steam-side heat-transfer coefficient is independent of plate height in the range 1–4 in. Thus for given surface and vapour temperatures the condensation rate should not vary with height in this range. Neither should we expect the maximum size to which adherent drops grow, before falling, to depend on the condensation rate. Thus with increasing heat flux, the number of falling drops should increase, but their average size at a given height should not vary. Thus the number of falling drops should be proportional to the heat flux, while their size, at a given distance from the top of the surface, should be independent of heat flux. We therefore expect that the fraction of area covered by falling drops should vary proportionally with heat flux, in the range 1–4 in from the top of the surface at least.

In order to verify the above and to estimate the constant of proportionality, photographs of a vertical condensing surface (6·2-cm high, 7-cm wide, promoter: dioctadecyl disulphide) were taken at different heat fluxes. Before taking any photographs, the non-condensing gas concentration had been minimized by boiling while "blowing off" steam to atmosphere. The pressure was kept at about 1·05 bar and steam was "blown off" throughout the tests.

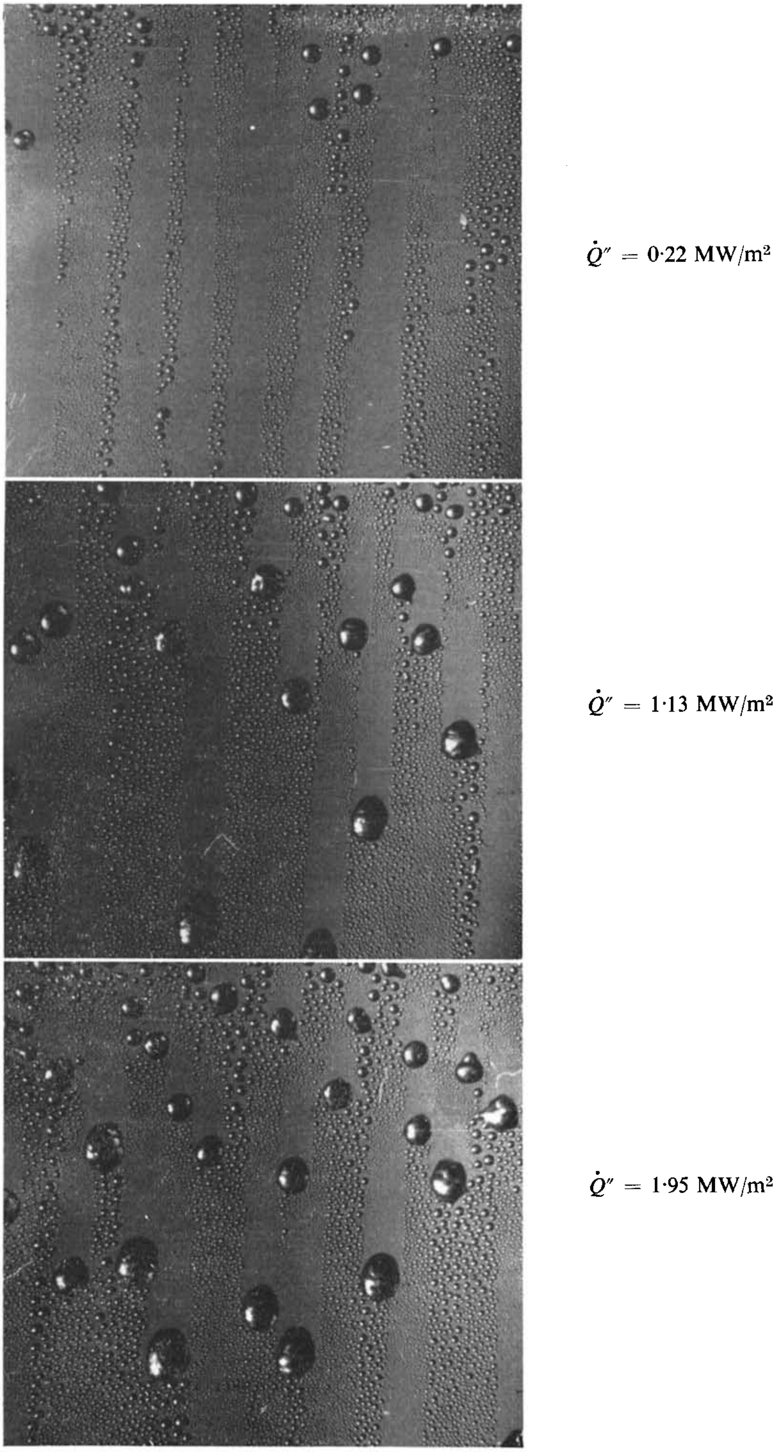

FIG. 1. The appearance of dropwise condensation at different heat fluxes.

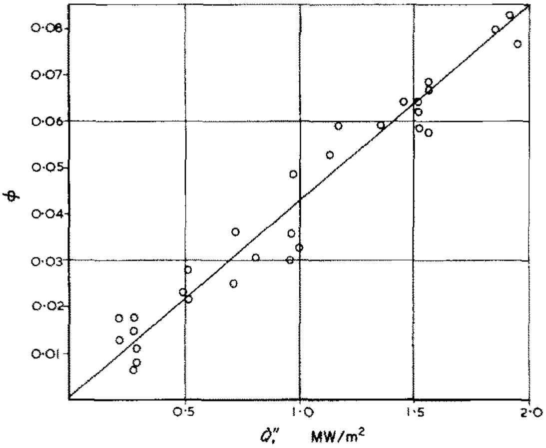

FIG. 2. Variation of fractional area covered by moving drops with heat flux.

Figure 1 shows a selection of the photographs obtained. The fractional areas covered by the falling drops were determined by planimeter from tracings of enlargements of the photographs. The heat flux was found by weighing condensate collected over an observed time interval.

When the fractional area covered by moving drops is plotted against the heat flux it is seen (Fig. 2) that the results are reasonably well represented by a straight line through the origin. Thus we write:

$$\phi = a\dot{Q}'' \tag{2a}$$

where ϕ is the fractional area covered by falling drops, $\dot{Q}''$ is the heat flux, and a is a constant.

In the present case the "least squares" line (assigning all of the error to ϕ) through the origin gives $a = 0{\cdot}043\ \mathrm{m^2/MW}$.

Since falling drops have, initially at least, a significant acceleration, they become progressively more spaced as they move down the condensing surface. Thus it is not apparent whether the area covered by them increases or decreases with plate height, except for surfaces sufficiently tall to enable the drops to attain a limiting velocity. For plates taller than this, the fractional area covered by falling drops would increase with height. The observation [16, 19] that the vapour-side heat-transfer coefficient does not depend on plate height for the range 1–4 in indicates that a does not vary appreciably in this range. The effect of acceleration presumably compensates for the increase in size of the falling drops.

Rather than retaining the empirical factor a, for which we have a value applicable only to the case of steam, at near atmospheric pressure, condensing on a vertical surface of height 62 mm, we suppose that ϕ depends predominantly on: the size at which a drop begins to fall, the gravitational acceleration, the volume condensation rate per area of the surface, the condensate density and the plate height. We then estimate on grounds of dimensional analysis:

$$\phi = f\left\{\frac{\dot{Q}''^2}{h_{fg}^2\rho_f^2\hat{r}g}, \frac{\hat{r}}{z}\right\}.$$

Or, for a given plate height, and having regard to the linear relation between ϕ and $\dot{Q}''$ (Fig. 2)

$$\phi = K_4\frac{\dot{Q}''}{h_{fg}\rho_f\sqrt{(\hat{r}g)}}. \tag{2b}$$

K_4 is dimensionless and a function of $(\hat{r}/z)$. However, for the ranges in which experiments have been carried out, we should not expect significant dependence of K_4 on that height. From the photographs it was estimated that $\hat{r} = 0{\cdot}75$ mm.

Thus, using the value of a given earlier:

$$K_4 = ah_{fg}\rho_f\sqrt{(\hat{r}g)} = 7{\cdot}97.$$

3. EVALUATION OF AVERAGE HEAT FLUX

We shall assume that the adhering drops are distributed according to equation (1) and that no significant heat transfer takes place through the falling drops. In view of the work of Welch and Westwater [3], who found that only a very small fraction of the heat transfer took place through visible drops, this latter assumption seems well justified. Thus we may estimate the average heat flux, $\tilde{Q}''$, through the area not covered by falling drops in the same way that the average heat flux for the whole surface was estimated formerly [14].

Hence we have:

$$\tilde{\dot{Q}}'' = -\frac{1}{\hat{r}}\int_{\check{r}}^{\hat{r}} \dot{Q}_b''\{\alpha'(r/\hat{r})\}\mathrm{d}r \tag{3}$$

and

$$\dot{Q}'' = (1-\phi)\tilde{\dot{Q}}''.$$

Then from (2b)

$$\dot{Q}'' = \tilde{\dot{Q}}''/\{1 + K_4\tilde{\dot{Q}}''/h_{fg}\rho_f\sqrt{(\hat{r}g)}\} \tag{4a}$$

or substituting for $\hat{r}$ from equation (22) of [14] and neglecting ρ_g:

$$\dot{Q}'' = \tilde{\dot{Q}}''/\{1 + K_4\tilde{\dot{Q}}''/K_3{}^{\frac{1}{4}}\, h_{fg}\rho_f^{\frac{3}{4}}\, g^{\frac{1}{4}}\, \sigma^{\frac{1}{4}}\}. \tag{4b}$$

It is interesting to note from equation (4) that regardless of the relation between $\tilde{\dot{Q}}''$ and ΔT, the average heat flux cannot exceed $h_{fg}\rho_f\sqrt{(\hat{r}g)}/K_4$. Thus as ΔT is increased, $\dot{Q}''$ tends to this limiting value, while the heat-transfer coefficient $\dot{Q}''/\Delta T$ passes through a maximum. In the case discussed earlier for which $a = 0{\cdot}043$ m^2/MW, the limiting heat flux would be about 28 MW/m^2, which is well beyond the range of any measurements.

In addition to the above modification of the theory, it has been considered opportune to adopt somewhat different values for the factors K_1, K_2 and K_3. If we assume hemispherical drops the analysis given by Fatica and Katz [22] for conduction through a drop indicates that $K_1 = 0{\cdot}241$. Further, if the promoter layer offers no significant resistance to heat transfer $K_2 = 0{\cdot}5$. Also, adopting the present estimate of $\hat{r}$ (0·75 mm), $K_3 = 0{\cdot}3$.

Having fixed the values of K_1, K_2 and K_3, the series solutions of equation (3) given earlier [14] may be used together with equation (4) to determine the value of n most appropriate to a given set of heat-transfer measurements. Taking the recent experimental data relating to condensation of steam at near-atmospheric pressures on vertical surfaces, promoted with dioctadecyl disulphide [16–21] minimization of the sum of the squares of residuals of $\dot{Q}''$ suggests the following values of n:

Reference	n
Le Fevre and Rose [16, 19]	1·98
Tanner *et al.* [17, 18]	1·93
Citakoglu and Rose [20, 21]	1·97
Above taken together	1·97

Since when $n = 2$, we may express $\tilde{\dot{Q}}''$ and hence $\dot{Q}''$ in terms of simple functions (as opposed to infinite series), this value has been adopted. The values of $\dot{Q}''$ then found, differ negligibly from those obtained by using the above values of n over the respective ranges of heat flux. For $n = 2$ we have:

$$\tilde{\dot{Q}}'' = \left\{\frac{K_1}{K_2{}^3K_3}\frac{L_0}{L_3}\frac{\dot{Q}_2''^3}{\dot{Q}_1''}\right\}^{\frac{1}{2}}\left\{\left(1+\frac{K_2}{K_1}\frac{\dot{Q}_1''}{\dot{Q}_2''}\frac{\Delta T}{T}\right)\arctan\left(\frac{K_2}{K_1}\frac{\dot{Q}_1''}{\dot{Q}_2''}\frac{\Delta T}{T}\right)^{\frac{1}{2}} - \left(\frac{K_2}{K_1}\frac{\dot{Q}_1''}{\dot{Q}_2''}\frac{\Delta T}{T}\right)^{\frac{1}{2}}\right\}. \tag{5}$$

$\dot{Q}''$ may then be found by substitution in equation (4).

In Fig. 3, the theory is compared with the experimental observations. It is seen that for references [16–19] both the original and present versions of the theory are in very good agreement with experiment. In the case of [20, 21] agreement is fair, the original version being marginally better. However, since the difference between the two theoretical curves is less than the uncertainty in the measurements of the steam-to-surface temperature difference at the higher heat fluxes [21], this is not thought to be significant.

No attempt has been made here to explain the differences found between the results ob-

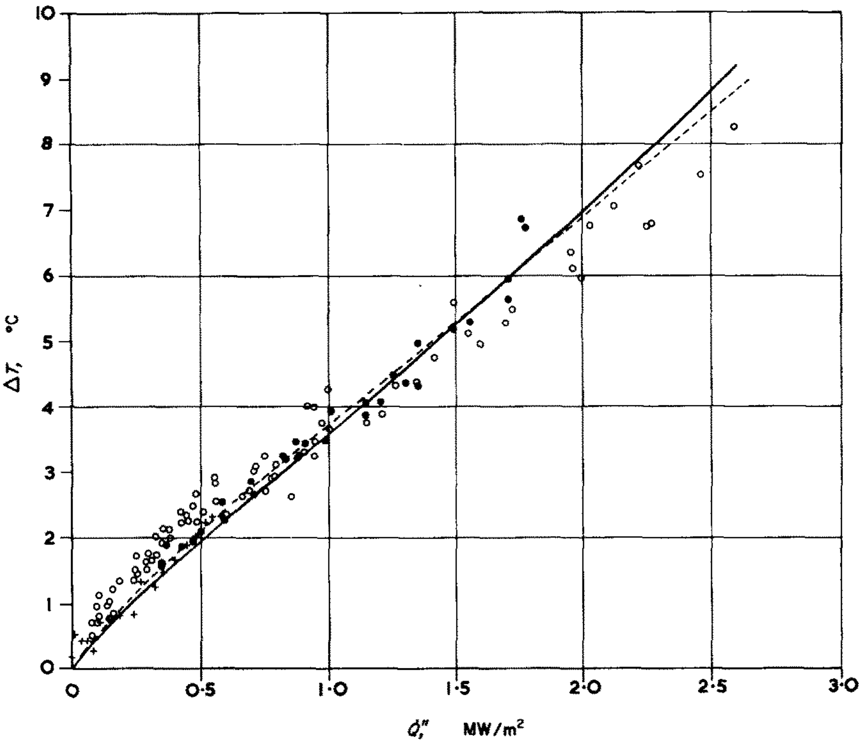

FIG. 3. Comparison between experiment and theory.

———— present
- - - - - - - - - - [14]
○ [20, 21]
+ [17, 18]
● [16, 19]

tained with different promoters [16–19]. It was stated earlier [14] that these may be accommodated by allowing n, K_1, K_2, and K_3 to take slightly different values for different promoters. Le Fevre and Rose [14] tentatively suggested that the promoter layer might offer a significant resistance to heat transfer and thus fixed n, K_1 and K_3 and found that good fits between theory and experiment could be obtained when K_2 was assigned different values for different promoters. In adopting $K_2 = 0{\cdot}5$ here, it is suggested that, in the case of dioctadecyl disulphide, the promoter layer offers negligible resistance.

As may be seen from Fig. 3, for heat fluxes up to about 2 MW/m², the correction for "blanketing" by falling drops makes little difference to the theory as it relates to condensation of steam on vertical surfaces. However, at heat heat fluxes a little beyond this range the effect of the correction becomes increasingly more important as is seen from Fig. 4. It is apparent that for non-vertical surfaces the "blanketing" effect of the falling drops will become significant at lower heat fluxes. This might also be true in the case of vertical surfaces when condensing vapours other than steam.

Figures 5 and 6 show, according to the present version of the theory, the effect of varying the pressure. It is seen that the vapour-side heat-transfer coefficient increases with pressure, the rate of increase being greater at the lower pressures and higher heat fluxes.

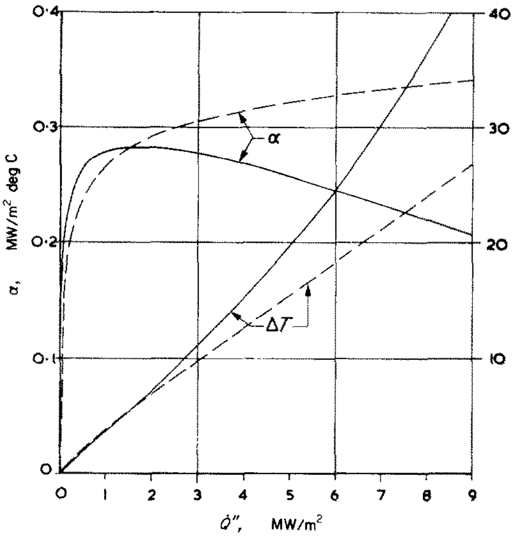

FIG. 4. Comparison between original and present version of the theory.

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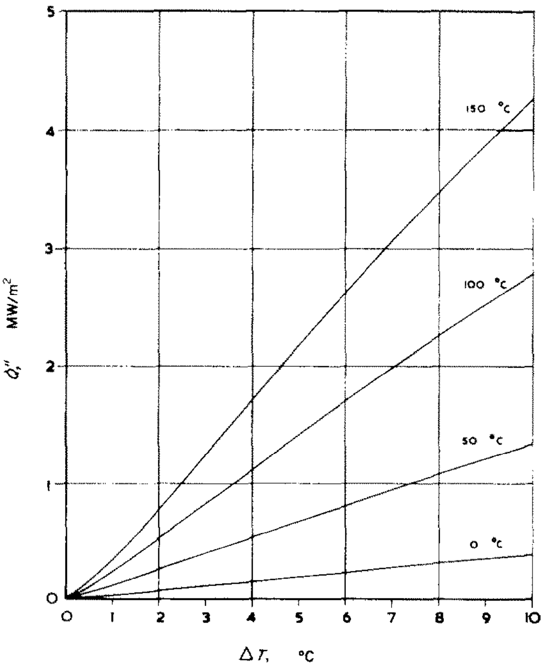

FIG. 5. The relation between heat flux and temperature difference for various pressures (the temperatures shown are the corresponding saturation temperatures).

4. COALESCENCE TIME

When judging the theory in the light of data relating to vapours other than steam, it may be necessary to incorporate new considerations. For instance, the adoption of any form of mean distribution of similarly shaped drops involves the assumption that coalescences take place virtually instantaneously. Such an assumption may be justified in the case of steam, even at the very high heat fluxes which have been obtained (up to about 2·5 MW/m²). There are, however, reports that certain organic vapours [23] (and also steam when using a surface after prolonged intermittent operation without repromoting [20]) will condense in the dropwise mode only at low heat fluxes. This suggests that the time of coalescence may sometimes be significant. It is considered that the change from the dropwise to the filmwise mode of condensation occurs when the time required for drops to coalesce exceeds the time required for their formation.

We would expect the rapidity with which neighbouring drops coalesce on touching, to depend principally on the surface tension, density and viscosity of the condensate as well as the contact angle and size of the drops involved. Thus, if t is the time required for coalescence between drops of radii r_1 and r_2, dimensional analysis suggests a relation of the form:

$$\psi(t\sigma/\mu_f r_1,\ t\sqrt{(\sigma/\rho_f r_1^3)},\ r_2/r_1,\ \textit{contact angle}) = 0.$$

If the viscosity effects overwhelm those of inertia we expect, for drops of given sizes:

$$t \propto r_1 \mu_f/\sigma.$$

Alternatively, if inertia effects dominate:

$$t \propto (r_1^3 \rho_f/\sigma)^{\frac{1}{2}}.$$

In the case of water $\mu_f^2/\rho_f \sigma r_1$ (the square of the ratio of the above "coalescence times") is small. Even for $r_1 = 0{\cdot}002$ μm, about the length of the dioctadecyl disulphide molecule, $\mu_f^2/\rho_f \sigma r_1 < 1$. This suggests that, for water, viscous effects are not important.

It is clear that, since the time required for the

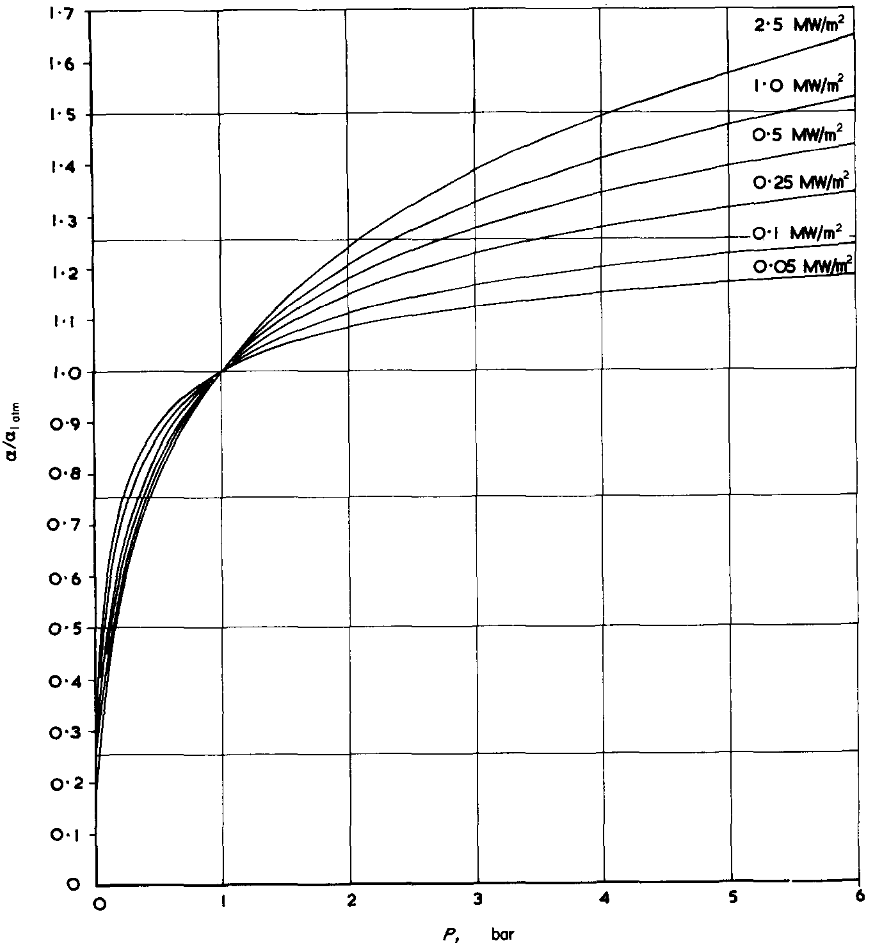

FIG. 6. The effect of pressure variation on the vapour–side heat-transfer coefficient for various heat fluxes.

formation of drops decreases with increasing heat flux while their "coalescence time" remains constant, transition from dropwise to filmwise condensation will ultimately always occur.

In the absence of heat-transfer measurements relating to the region of transition it is not possible to pursue the question further. It is apparent, however, that the theory would not apply for heat fluxes approaching that at which transition occurs.

5. CONCLUSION

The correction for "blanketing" by falling drops leads to an inflexion in the graph of steam-to-surface temperature difference against heat flux, and to a maximum value of the steam-side heat-transfer coefficient.

The effect of the correction becomes marked, for the case of steam condensing on a vertical surface, only at and beyond the highest heat fluxes obtained to date. For non-vertical surfaces the correction will be important at lower heat fluxes. This might also be the case for other condensing fluids even with vertical surfaces.

With increasing heat flux, a value, dependent on the condensing fluid, is reached, at which transition to filmwise condensation occurs. The time required for coalescence between drops is thought to be an important parameter governing this transition.

REFERENCES

1. M. Jakob, Heat transfer in evaporation and condensation, *Mech. Engng* **58**, 729 (1936).
2. E. Baer and J. M. McKelvey, *Heat Transfer in Dropwise Condensation*, ACS Delaware Science Symposium (1958).
3. J. F. Welch and J. W. Westwater, Microscopic study of dropwise condensation, *International Developments in Heat Transfer*, Vol. 2, p. 302. Am. Soc. Mech. Engrs, New York (1961).
4. A. Eucken, Energie und Stoffaustausch an Grenzflächen, *Naturwissenschaften* **25**, 209 (1937).
5. W. Kast, Wärmeübertragung bei Tropfenkondensation, *Chem.-Ingr.-Tech* **35**, 163 (1963).
6. W. Kast, Bedeutung der Keimbildung und der instationären Wärmeübertragung für den Wärmeübergang bei Blasenverdampfung und Tropfenkondensation, *Chemie-Ingr-Tech.* **36**, 933 (1964).
7. R. S. Silver, An approach to a general theory of surface condensers, *Proc. Instn Mech. Engrs* **178**, 339 (1963).
8. S. Sugawara and K. Katsuta, Fundamental study on dropwise condensation, *Proceedings of the Third International Heat Transfer Conference*, Chicago, Vol. 2, p. 354 Am. Instn Chem. Engrs, New York (1966).
9. A. Umur and P. Griffith, Mechanism of dropwise condensation, *J. Heat Transfer* **87** C, 275 (1965).
10. J. L. McCormick and J. W. Westwater, Nucleation sites for dropwise condensation, *Chem. Engng Sci.* **20**, 1021 (1965).
11. J. L. McCormick and J. W. Westwater, Drop dynamics and heat transfer during dropwise condensation of water vapour on a horizontal surface, *Chem. Engng Prog. Symp. Ser.*, No. 64, **62**, 120 (1966).
12. A. C. Peterson and J. W. Westwater, Dropwise condensation of ethylene glycol, *Chem. Engng Prog. Symp. Ser.* No. 64 **62** 135 (1966).
13. J. L. McCormick and E. Baer, On the mechanism of heat transfer in dropwise condensation, *J. Colloid Sci.* **18**, 208 (1963).
14. E. J. Le Fevre and J. W. Rose, A theory of heat transfer by dropwise condensation, *Proceedings of the Third International Heat Transfer Conference*, Chicago, Vol. 2, p. 362 Am. Instn Chem. Engrs, New York (1966).
15. H. Wenzel, Versuche über Tropfenkondensation, *Allg. Wärmetech.* **8**, 53 (1957).
16. E. J. Le Fevre and J. W. Rose, Heat-Transfer measurements during dropwise condensation of steam, *Int. J. Heat Mass Transfer* **7**, 272 (1964).
17. D. W. Tanner, C. J. Potter, D. Pope and D. West, Heat transfer in dropwise condensation—Part I: the effects of heat flux, steam velocity and non-condensable gas concentration, *Int. J. Heat Mass Transfer* **8**, 419 (1965).
18. D. W. Tanner, C. J. Potter, D. Pope and D. West, Heat transfer in dropwise condensation—Part II: surface chemistry, *Int. J. Heat Mass Transfer* **8**, 427 (1965).
19. E. J. Le Fevre and J. W. Rose, An experimental study of heat transfer by dropwise condensation, *Int. J. Heat Mass Transfer* **8**, 1117 (1965).
20. E. Citakoglu, Heat transfer by dropwise condensation, Ph.D. Thesis, London University (1966).
21. E. Citakoglu and J. W. Rose, Some factors influencing the validity of heat-transfer measurements during dropwise condensation, In preparation.
22. N. Fatica and D. L. Katz, Dropwise condensation, *Chem. Engng Prog.* **45**. 661 (1949).
23. R. P. Bobco and A. L. Gosman, Promotion of dropwise condensation of pure organic vapours, Am. Soc. Mech. Engrs Paper No. 57-5-2 (1957).

Résumé—On propose une extension d'une récente théorie du transport de chaleur par condensation en gouttelettes [14]. La théorie modifiée est comparée avec des mesures récentes de transport de chaleur. On présente des courbes donnant les prévisions de la théorie, pour des flux de chaleur en dehors des gammes expérimentales actuelles et pour différentes pressions.

Zusammenfassung—Eine Erweiterung einer neuen Theorie [14] des Wärmeüberganges bei Tropfenkondensation wird vorgeschlagen. Diese modifizierte Theorie wird mit neuen Wärmeübergangsmessungen verglichen. Es werden Diagramme wiedergegeben, welche die Voraussagen dieser Theorie für heute noch unerreichte Wärmestromdichten und verschiedene Drücke Aufzeigen.

Аннотация—Предложено обобщение существующей теории теплообмена при капельной конденсации [14]. Модифицированная теория сравнивается с последними экспериментальными данными по теплообмену. Предсказания теории представлены на графиках для тепловых потоков, лежащих вне возможностей современного эксперимента, и для различных давлений.