Dropwise condensation theory and experiment: a review

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Abstract: The paper reviews progress in dropwise condensation research from 1930 to the present. Particular attention is given to heat transfer measurements, theory, transition and effects of surface material. Although it has been known since the 1930s that heat transfer coefficients for dropwise condensation of steam are much higher than those for film condensation, there were, until the 1960s, wide discrepancies between the results of different investigators. Subsequently, more accurate measurements have shown good consistency and the mechanism and theory of the dropwise condensation have become better understood. There has been considerable controversy over the magnitude of the so-called 'constriction resistance' and the effect of the surface thermal conductivity on the heat transfer coefficient. The balance of evidence suggests that this is only significant at very low heat fluxes and for very small condensing surfaces. Measurements have also been made with sufficiently high cooling intensities to cover the range of dropwise to filmwise condensation transition. The detailed mechanism of transition is still not fully understood. Perhaps most importantly, the practical problem of promoting lasting dropwise condensation under industrial conditions remains to be solved despite considerable effort spanning more than 70 years.

Keywords: condensation, dropwise, heat transfer measurements, theory

NOTATION

$A(r)\,\mathrm{d}r$	fraction of the surface area covered by drops with a base radius in the range r to $r + \mathrm{d}r$
f	fraction of the surface area covered by drops with a base radius greater than r
g	specific force of gravity
h_{fg}	specific enthalpy of evaporation
k	thermal conductivity of the condensate
K_1	dimensionless constant in equation (2)
K_2	promoter-dependent dimensionless constant in equation (2)
K_3	constant (see equation (8))
$N(r)\,\mathrm{d}r$	number of drops per area with a base radius in the range r to $r + \mathrm{d}r$
q	mean heat flux for the surface
q_{b}	heat flux through the base of a drop
r	radius of a drop
$\hat{r}$	radius of the largest drop
$\check{r}$	radius of the smallest drop
R	specific ideal-gas constant
t	Celsius temperature
T	thermodynamic (absolute) temperature
T_{sat}	saturation temperature
T_{v}	vapour temperature
v_{f}	specific volume of the saturated liquid
α	heat transfer coefficient
γ	ratio of the principal specific heat capacities
ΔT	vapour–surface temperature difference
ρ	density of the condensate
ρ_{f}	density of the saturated liquid
ρ_{g}	density of the saturated vapour
σ	surface tension

1 INTRODUCTION

Dropwise condensation may occur when condensation takes place on a surface that is not wetted by the condensate. Surface heat transfer coefficients for dropwise condensation are much higher than for the filmwise mode that occurs when the surface is wetted. Dropwise condensation was first recognized by Schmidt *et al.* (1930), and much interest was stimulated by their report that heat transfer coefficients were between 5 and 7 times those found with film condensation. Over the years there have been a few demonstrations of successful applications on an industrial scale. Tanzola and Wiedman (1954) and Kajanne (1957) reported significant increases in rates of paper drying through promotion of

dropwise condensation of steam used in the drying process. Brunt and Minkin (1958) obtained an increase in output of an experimental sea water evaporator of around 50 per cent while Eibling and Hyatt (1958) found increases in the range 13–33 per cent. Poll *et al.* (1967) reported improvements of 10–15 per cent in output of a triple-effect evaporator associated with a power station. Garatt (1958), on the other hand, found no improvement in output of a large saline water evaporator after injecting a dropwise promoter into the supply steam over the course of a year, although a pilot plant study had indicated that an improvement of around 30 per cent should have been obtainable. An attempt by the CEGB (1960) to promote dropwise condensation in a condenser at Bradford power station gave no improvement in performance.

The key problem has been, and is, to devise reliable means of promoting long-lived dropwise condensation under industrial conditions. Clean metal surfaces are normally wetted and the mode of condensation is filmwise, where the condensate forms as continuous film on the surface. An exception to this is the case of condensation of a metal; mercury condenses in the dropwise mode on stainless steel [Ivanovskii *et al.* (1967), Kollera and Grigull (1971), Necmi and Rose (1977) and Niknejad and Rose (1978)]. It has been shown that metal surfaces such as gold, silver or chromium, which had earlier been thought to give dropwise condensation of steam, owe their promoting properties to surface impurities, and, when the surface is chemically clean, film condensation is obtained (Erb and Thelen (1966), Erb (1973), Wilkins *et al.* (1973) and Woodruff and Westwater (1979)). The monolayer promoters, such as dioctadecyl disulphide, which are effective on copper or copper-containing surfaces, have proved very useful for laboratory heat transfer measurements with lifetimes of tens of hours or more for condensation of steam. They have not, however, shown sufficient durability under industrial conditions, possibly owing to their removal by oxidation of the surface. Layers of hydrophobic polymers such as polytetrafluorethylene (PTFE, 'Teflon') give excellent dropwise condensation but have only been found to be sufficiently durable when the thickness of the low-conductivity polymer layer is so large (around 20–30 μm) as to offset the advantage of dropwise condensation.

By around 1990 it seemed unlikely that any satisfactory method of promoting dropwise condensation under industrial conditions would be developed. Interest was then revived by reports of a new, and apparently successful, method for promoting dropwise condensation ('dynamic mixing magnetron sputtering/ion plating') developed and used by Zhao and coworkers (Zhao *et al.* (1990), Zhao *et al.* (1991a), Zhao *et al.* (1991b), Zhao and Lin (1991), Zhao and Xu (1993), Zhao and Burnside (1994), Zhao *et al.* (1994), Burnside *et al.* (1996), Wang *et al.* (1992) and Zhao and Wang (1993)). However, follow-up reports of successful industrial application seem not, as yet, to have materialized. An outline history of dropwise condensation research is given in Table 1.

Table 1 Outline history of dropwise condensation research

1930–1960	Identification of dropwise mode by Schmidt *et al.* (1930). Measurements show $\alpha_{dropwise} \gg \alpha_{filmwise}$. Identification/development of successful promoters for laboratory measurements. Many experimental heat transfer investigations for steam—widely scattered data showing different trends.
1960–1980	Error in earlier measurements attributed to the presence of air in steam. More refined experiments. Good agreement between measurements in different countries. Development of theory.
1980–1990	Diminishing activity due to the failure to find an industrially reliable promoter.
1990–2000	Reports of a successful promoter and its use in a power plant condenser (China). Renewed interest.
2000–	Follow-up on Chinese reports awaited.

2 HEAT TRANSFER MEASUREMENTS

As noted above, it is possible to promote dropwise condensation satisfactorily under clean laboratory conditions. Fatty acids have often been used as promoters as well as specially synthesised long hydrocarbon chain promoters (see Blackman *et al.* (1957)) such as dioctadecyl disulphide. Unfortunately, many conflicting experimental results have been published over the years, and this has hindered progress towards understanding of the mechanism and development and acceptance of theory. Figure 1 summarizes most of the data reported since 1930 for dropwise condensation of steam at atmospheric pressure. The choice of vapour–surface temperature difference as ordinate and heat flux as abscissa is deliberate. Most writers prefer to represent data of this sort by plotting heat transfer coefficient, considered to be the quantity of interest, against temperature difference, regarded as the 'driving force' for heat transfer. In dropwise condensation the vapour–surface temperature difference is very small and in many cases its magnitude may be comparable with the uncertainty in its measurement so that misleading inferences can be drawn when plotting a quantity involving its reciprocal. Also, in dropwise condensation the coolant-side resistance dominates and the heat flux is usually adjusted by varying the coolant flowrate, with temperature difference the dependent measured quantity.

There is little doubt that the correct results lie in the shaded region with lower temperature differences. Error in the other data is primarily due to the presence in the steam of air as well as, in some cases, to insufficient accuracy in the measurement of the surface temperature. The extreme sensitivity of the dropwise condensation heat transfer coefficient to even minute quantities (a few ppm) of air in the steam has been demonstrated in several investigations. In the presence of a non-condensing gas, the observed vapour–surface temperature difference includes that occurring in the vapour–gas boundary layer; the latter depends not only on
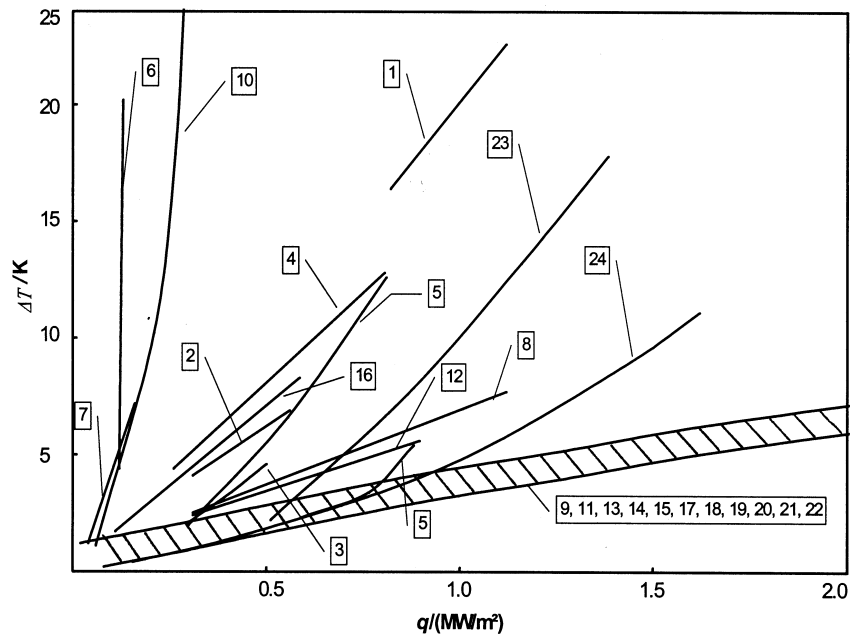
Fig. 1 Heat transfer measurements for dropwise condensation of steam at near-atmospheric pressure: 1, Schmidt *et al.* (1930); 2, Nagle *et al.* (1935); 3, Gnam (1937); 4, Fitzpatrick *et al.* (1939); 5, Shea and Krase (1940); 6, Fatica and Katz (1949); 7, Kirschbaum *et al.* (1951); 8, Hampson and Ozisik (1952); 9, Wenzel (1957); 10, Welch and Westwater (1961); 11, Le Fevre and Rose (1964); 12, Kast (1963); 13, Le Fevre and Rose (1965); 14, Tanner *et al.* (1965a); 15, Citakoglu (1966); 16, Griffith and Lee (1967); 17, Citakoglu and Rose (1968); 18, Graham (1969); 19, Wilmshurst and Rose (1970); 20, Tanasawa and Ochiai (1973); 21, Aksan and Rose (1973); 22, Stylianou and Rose (1980); 23, Ma *et al.* (1994); 24, Leipertz and Koch (1998)

the amount of gas present in the supply vapour but also on the geometry of the apparatus, venting arrangements and the condensation rate. This explains the diversity of values and trends shown in Fig. 1. For dropwise condensation, accurate measurements, free from error due to the presence of non-condensing gas, show that the heat transfer coefficient *increases* with vapour–surface temperature difference or heat flux, as shown by the curvature of the shaded data band in Fig. 1. One of the early investigations to show this clearly (see Fig. 2) is that of Wenzel (1957). This behaviour

is due to the increase in numbers of active nucleation sites with increase in vapour–surface temperature difference as discussed below.

Measurements, in general agreement with each other, have been made by several investigators (Tanner *et al.* (1968), Graham (1969), Wilmshurst and Rose (1970), Stylianou and Rose (1980), Tsuruta and Tanaka (1983) and Hatamiya and Tanaka (1986)) for dropwise condensation of steam at pressures ranging from atmospheric down to around 1 kPa. A typical dataset is shown in Fig. 3. These data show that the heat transfer coefficient decreases with decreasing pressure. This is due to interphase matter transfer resistance which plays an important role in the mechanism of dropwise condensation, as discussed below.

The data for dropwise condensation of steam that are considered to be accurate are quite well represented, for pressures near and below atmospheric, by the empirical equation

$$q/(kW/m^2) = t^{0.8}\left[5\left(\frac{\Delta T}{K}\right) + 0.3\left(\frac{\Delta T}{K}\right)^2\right] \quad (1)$$

where *t* is Celsius temperature.

Dropwise condensation is obtained most readily with high surface tension fluids. For this reason, and because of its industrial importance, most experiments have been done with steam. However, some measurements have been made
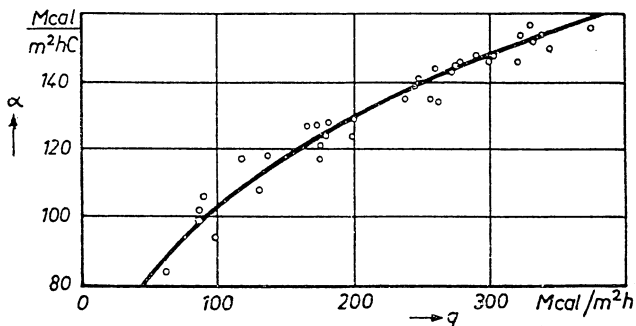

Fig. 2 Dependence of heat transfer coefficient on heat flux for dropwise condensation of steam on copper at near-atmospheric pressure; promoter oleic acid (Wenzel (1957))
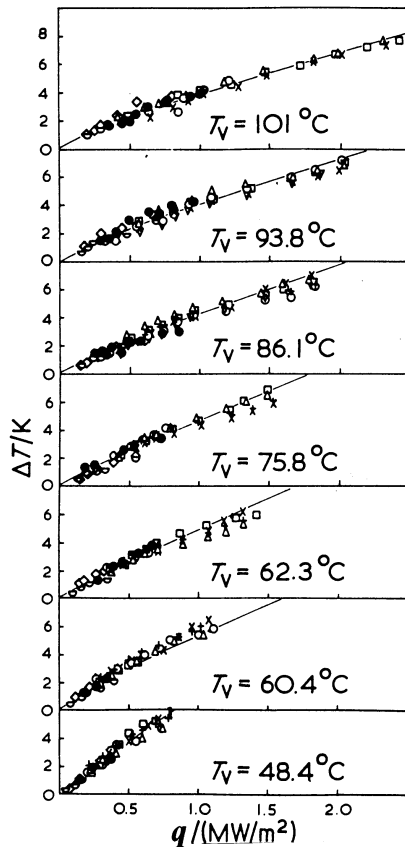
Fig. 3 Dropwise condensation of steam on a dioctadecyl disulphide-promoted copper surface at various pressures. Closed circles denote data of Wilmshurst and Rose (1970); other points are from runs on eight different days by Stylianou and Rose (1980); the lines are from theory (Le Fevre and Rose (1966)). Figure from Stylianou and Rose (1980)

using relatively high surface tension organic fluids. Wilmshurst and Rose (1974) condensed ethylene glycol, aniline and nitrobenzene on a PTFE-coated surface. Stylianou and Rose (1983) used ethylene glycol and Utaka *et al.* (1987, 1994) condensed propylene glycol, ethylene glycol and glycerol on a copper surface using a monolayer promoter. As in the case of steam, the heat transfer coefficient was found to be significantly larger than for film condensation.

Heat transfer measurements have also been made for dropwise condensation of mercury (Misra and Bonilla (1956), Gelman (1958), Ivanovskii *et al.* (1967), Kollera and Grigull (1971), Necmi and Rose (1977) and Niknejad and Rose (1978)). The older data of Misra and Bonilla and Gelman show significantly lower heat transfer coefficients and probably involve substantial error caused by the presence of non-condensing gas. The more recent data are in broad agreement and indicate heat transfer coefficients similar to those found with film condensation of mercury. This is due to the high conductivity of the condensate, giving correspondingly high coefficients in film condensation.

3 MECHANISM AND THEORY

The appearance of ideal dropwise condensation of steam is shown in Fig. 4. Falling drops originate from a region within a few millimetres of the top of the surface and fall very quickly (with acceleration apparently little short of that of gravity) after commencing to move. By collecting the drops in their path, the moving drops grow and thus sweep diverging tracks, causing lower regions on the surface to be swept more frequently which ensures that only drops near the top of the surface can reach the size at which gravity exceeds the net surface tension restraining force. Stationary drops grow both by condensation on their surface and by coalescence with neighbours. Micro-cine studies, notably those of Westwater and coworkers (McCormick and Westwater (1965, 1966) and Peterson and Westwater (1966)) have removed uncertainties regarding the origin of the smallest or 'primary' drops. These form at nucleation sites on bare surface exposed by coalescence between larger drops or sweeping by falling drops, and are typically around 1000 times smaller than nucleation bubbles in boiling. Nucleation site densities are typically in the range 10^7–10^9 sites/mm^2. The ratio of the largest drop radius to the smallest is around 10^6, and several thousand coalescences typically occur during the formation of a large drop with a diameter of around 1 mm.

A theory proposed by Le Fevre and Rose (1966) has subsequently been found to be in good agreement with the data for steam over wide ranges of heat flux and vapour pressure and with data for other fluids (see, for instance,
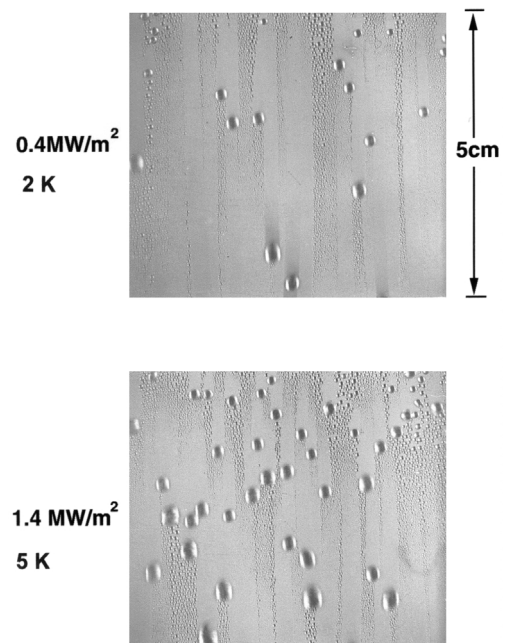

Fig. 4 Appearance of dropwise condensation on a monolayer-promoted copper surface at atmospheric pressure for two coolant flowrates. The corresponding heat fluxes and vapour–surface temperature differences are indicated (from Rose (1964))

Rose (1988) and Rose *et al.* (1999)). The essence of the approach is to combine an expression for the heat transfer through a drop of given size with an expression for the *average* distribution of drop sizes and to integrate over all drop sizes to obtain, for a given vapour–surface temperature difference, the heat flux for the surface. The crucial factors in the expression for heat transfer through a drop are conduction, surface curvature and interface temperature drop (matter transfer resistance). A related approach by Tanaka (1975) includes detail of the coalescence process.

In the theory of Le Fevre and Rose (1966) the heat flux through the base of a hemispherical (a good approximation for water) drop with radius r is given by

$$q_b = \frac{\Delta T - \frac{(2\sigma T_{sat})}{(r\rho_f h_{fg})}}{K_1 \frac{r}{k} + K_2\left(\frac{0.627}{0.664}\right)\frac{T_{sat}}{(h_{fg}^2\rho_g)}\frac{(\gamma+1)}{(\gamma-1)}\left[\frac{(RT_{sat})}{(2\pi)}\right]^{1/2}} \quad (2)$$

(note that equation (2) has been slightly modified by introduction of the factor 0.627/0.664 (see Rose (1998)) from that originally used by Le Fevre and Rose (1966)).

The second term in the numerator on the right-hand side of equation (2) is the amount by which the vapour must be subcooled in order to condense on the convex liquid surface. The first term in the denominator is the conduction resistance, and the second term accounts for interphase matter transfer as well as possible promoter layer resistance (included in K_2).

For the distribution of drop sizes, Le Fevre and Rose (1966) used

$$f = 1 - \left(\frac{r}{\hat{r}}\right)^{1/3} \quad (3)$$

where f is the fraction of surface area covered by drops having a base radius greater than r. Equation (3) indicates that no area is covered by drops larger than the largest, and that, as the drop radius approaches zero, the fraction of area covered by all larger drops approaches unity.

Equation (3) may be written as

$$A(r)\,dr = \frac{1}{3}\left(\frac{r}{\hat{r}}\right)^{-1/3}\frac{dr}{\hat{r}} \quad (4)$$

or

$$N(r)\,dr = \frac{1}{3\pi r^2}\left(\frac{r}{\hat{r}}\right)^{-1/3}\frac{dr}{\hat{r}} \quad (5)$$

Equations (2) and (4) were combined and the average heat flux for the surface (in terms of the temperature difference) was obtained by integration over all drop radii:

$$q = \frac{1}{3\hat{r}^{1/3}}$$
$$\times \int_{\check{r}}^{\hat{r}}\left\{\frac{T - \frac{(2\sigma T_{sat})}{(r\rho_f h_{fg})}}{K_1\frac{r}{k} + K_2\left(\frac{0.627}{0.664}\right)\frac{T_{sat}}{(h_{fg}^2\rho_g)}\frac{(\gamma+1)}{(\gamma-1)}\left[\frac{(RT_{sat})}{(2\pi)}\right]^{1/2}}\right\}$$
$$\times r^{-2/3}\,dr \quad (6)$$

where the minimum drop radius is taken to be that of the smallest viable (thermodynamically) drop

$$\check{r} = \frac{2\sigma v_f}{h_{fg}\Delta T} \quad (7)$$

and the maximum, on the basis of dimensional analysis, is taken as

$$\hat{r} = K_3\left[\frac{\sigma}{(\rho g)}\right]^{1/2} \quad (8)$$

The theory was originally criticized for the fact that it contained four apparently adjustable constants, two (K_1 and K_2) in equation (2), one in the expression for the drop size distribution [the index 1/3 in equation (3)] and one in equation (8) (K_3), which were chosen to give the best fit to the available reliable data for steam at atmospheric pressure. However, all of the constants were known to be close to unity, and three could have been estimated independently. The fourth, that in the distribution function, was subsequently verified, together with the form of equation (3), by measurements (Graham (1969) and Tanasawa and Ochiai (1973)), computer simulations (Glicksman and Hunt (1972)) and theory (Rose and Glicksman (1973)) (see Fig. 5). The final result for the dependence of heat flux on temperature difference may be expressed in, albeit lengthy, closed form. To facilitate comparison with experiment, a detailed graph giving the theoretical prediction for dropwise condensation of steam at different pressures is given in Fig. 6.

The theory is compared with subsequent experiments for dropwise condensation of steam at various pressures in Fig. 3, where it is seen that the dependence of temperature difference on heat flux and pressure is accurately predicted. As outlined by Rose (1988), the theory is also in fair agreement with data for ethylene glycol and mercury (with some adjustment of the constants anticipated as a result of different contact angles for the different fluids) and predicts very well the dependence of heat transfer coefficient on maximum drop size (see Fig. 7).

4 DROPWISE TO FILMWISE CONDENSATION TRANSITION

Several experimental studies have been made concerning transition from dropwise to filmwise condensation, analogous to that from nucleate to film boiling. Measurements with very intensive cooling using liquid nitrogen have been made for steam by Takeyama and Shimuzu (1974) and
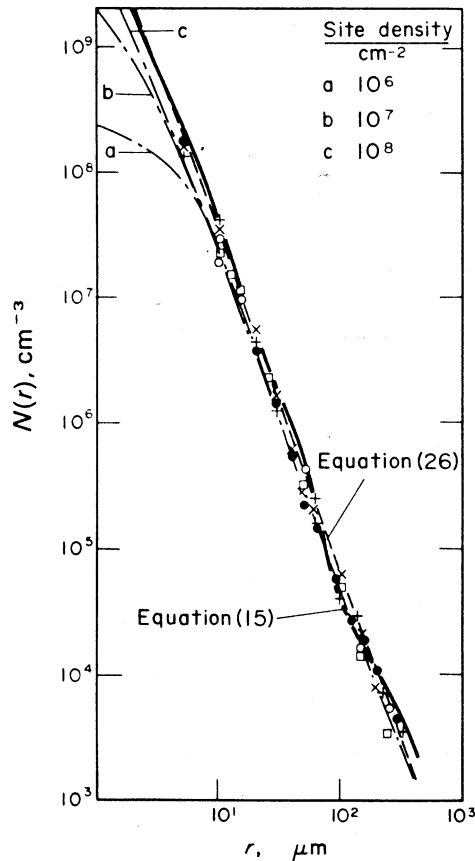
Fig. 5 Distribution of drop sizes (Rose and Glicksman (1973)). Equation (26) denotes the distribution used by Le Fevre and Rose (1966); equation (15) denotes the theoretical result of Rose and Glicksman (1973); lines a, b and c denote simulations of Glicksman and Hunt (1972); points denote measurements of Graham (1969) and Tanasawa and Ochiai (1973)

Tanasawa and Utaka (1983). The peak heat flux for steam at atmospheric pressure exceeded $10\,\text{MW/m}^2$ when the vapour–surface temperature difference was around $20\,\text{K}$. Investigations of transition using other fluids, where the peak heat flux is lower and less intensive cooling is required, are easier to perform.

Wilmshurst and Rose (1974) and Wilmshurst (1979) have reported measurements for condensation of aniline and nitrobenzene on a PTFE-coated copper surface. Figure 8 shows the appearance of the condensing surface for a fixed vapour temperature during condensation of aniline as the cooling intensity was increased. When the cooling intensity was reduced, the mode of condensation reverted to dropwise. The appearance of the transition regime for the organic fluids was similar to that reported by Takeyama and Shimuzu (1974) and Tanasawa and Utaka (1983) for steam—the larger drops became irregular in shape and eventually formed into patches of film with no apparent preferred location on the surface. For aniline the peak heat flux ranged from about $150\,\text{kW/m}^2$ at the lowest pressure used

($T_{\text{sat}} = 59\,^\circ\text{C}$) to about $400\,\text{kW/m}^2$ at the highest pressure ($T_{\text{sat}} = 151\,^\circ\text{C}$); for nitrobenzene the peak heat flux ranged from about $100\,\text{kW/m}^2$ at the lowest pressure used ($T_{\text{sat}} = 62\,^\circ\text{C}$) to about $250\,\text{kW/m}^2$ at the highest pressure ($T_{\text{sat}} = 113\,^\circ\text{C}$).

Stylianou (1980) and Stylianou and Rose (1983) examined transition for condensation of ethylene glycol on a copper surface promoted with oleic acid and on a PTFE-coated copper plate. In the case of the oleic acid-promoted surface, patches of film formed on the lower part of the surface in some cases (but not always), but these were evidently not formed by the running together of falling drops or streaks. Figure 9 shows the appearance of the condensing surface at three vapour temperatures as the cooling intensity was increased. On the PTFE surface the falling drops grew into streaks at higher condensation rates and there was evidence that transition was accompanied with agglomeration of these streaks. For both surfaces, the mode reverted to ideal dropwise as the cooling intensity was reduced. Complete transition to film condensation was not achieved with the PTFE-coated surface, but the peak heat flux was about the same for the two surfaces and ranged from about $500\,\text{kW/m}^2$ at the lowest pressure used ($T_{\text{sat}} = 113\,^\circ\text{C}$) to about $900\,\text{kW/m}^2$ at the highest pressure used ($T_{\text{sat}} = 167\,^\circ\text{C}$).

Utaka and coworkers have published several papers relating to transition for organic condensing fluids. Utaka *et al.* (1986, 1988) report and discuss data for condensation of propylene glycol, ethylene glycol and glycerol on a small circular (8 mm diameter) copper surface promoted with trilauryl trithiophosphate. In these tests, significant vapour cross-flow velocity was used so that the condensate drops or streaks moved almost horizontally across the surface. Rather higher maximum heat fluxes than those found by Stylianou and Rose (1983) are reported. For propylene glycol the maximum heat flux varied from about $300\,\text{kW/m}^2$ at the lowest pressure ($T_{\text{sat}} = 363\,\text{K}$) to around $600\,\text{kW/m}^2$ at the highest pressure ($T_{\text{sat}} = 400\,\text{K}$); for ethylene glycol the approximate maximum heat fluxes were $800\,\text{kW/m}^2$ at $T_{\text{sat}} = 368\,\text{K}$ and $2000\,\text{kW/m}^2$ at $T_{\text{sat}} = 428\,\text{K}$; for glycerol the corresponding approximate maxima were $500\,\text{kW/m}^2$ at $T_{\text{sat}} = 411\,\text{K}$ and $1000\,\text{kW/m}^2$ at $T_{\text{sat}} = 423\,\text{K}$. The results taken as a whole suggested broadly that the peak heat flux decreased with increase in liquid viscosity. The appearance of the condensing surface as revealed by photographs [Utaka *et al.* (1988)] suggests that, as the cooling intensity increased, the moving drops became streaks which ran together to form a continuous film further along the plate. The appearance is quite different to that found in the above investigations where the surfaces were larger and the drainage by gravity.

The photographic evidence, showing apparently different behaviour for different surfaces, promoters and mode of drainage, raises questions regarding the transition mechanism. Stylianou and Rose (1983) considered three possible mechanisms of transition. The first was that film condensation could result from total coverage or flooding by moving/falling drops at some distance along/down the condensing
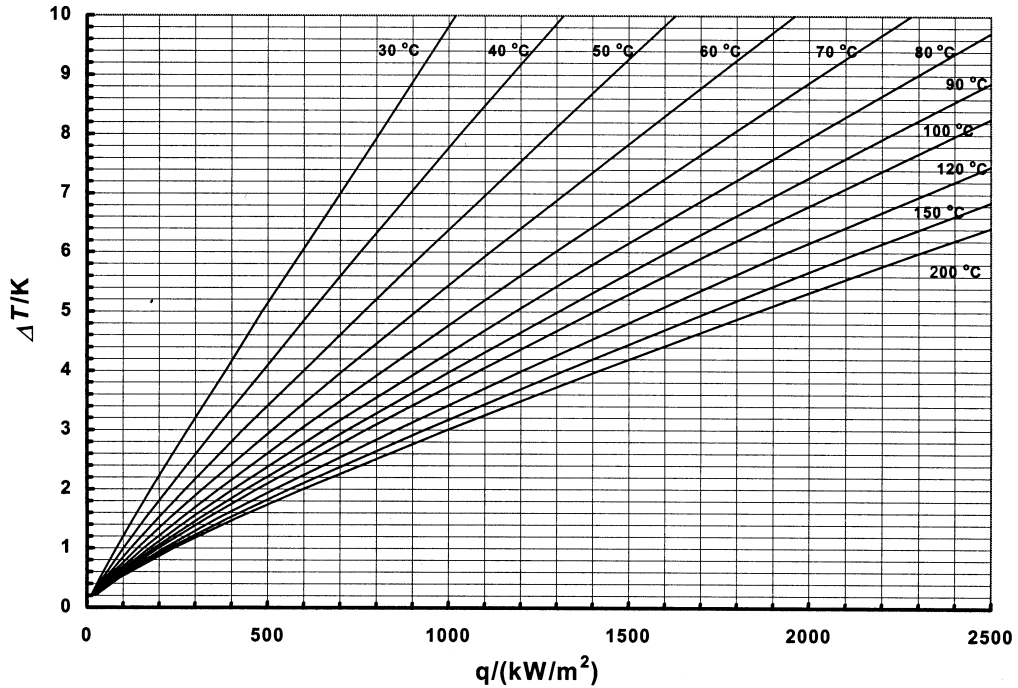
Fig. 6 Theoretical results of Le Fevre and Rose (1966) for dropwise condensation of saturated steam at different vapour temperatures for a monolayer promoter (Rose (2000))

surface. This was discarded in the light of the visual/photographic evidence. The second, raised earlier by Rose (1967) and Wilmshurst and Rose (1974), was the suggestion that the time taken for drops of given size to coalesce, governed by surface tension, inertia and viscosity, should be

constant, while the growth rate of drops would increase with increase in condensation rate; at sufficiently high condensation rates, area would be covered more quickly than it was made available, resulting in transition. The third possible mechanism was that, with increasing vapour–surface
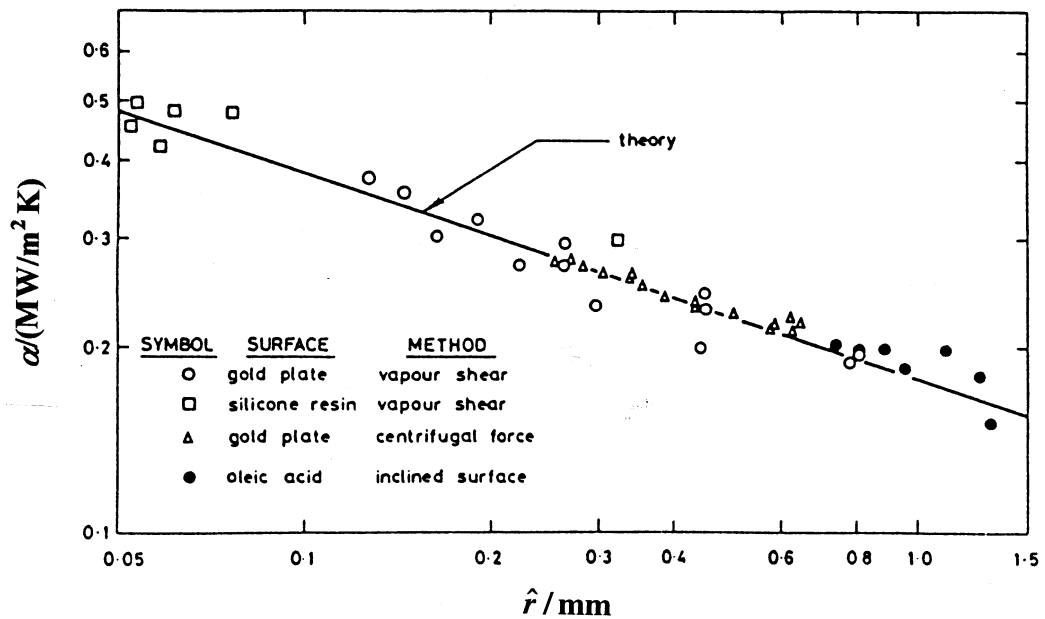

Fig. 7 Dependence of heat transfer coefficient on maximum drop size. Comparison of the theory of Le Fevre and Rose (1966) with measurements of Tanasawa (1974) (Rose (1988))
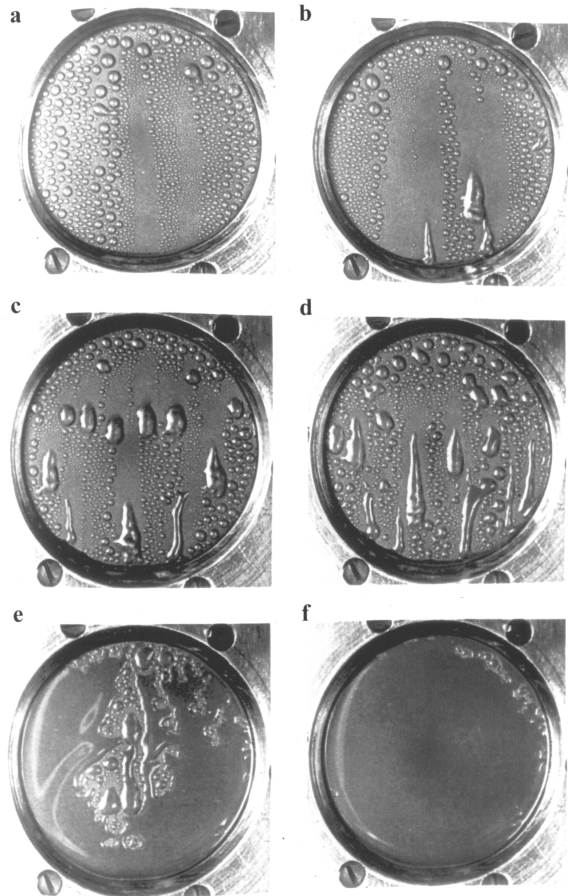
Fig. 8 Condensation of aniline on a PTFE-coated copper surface (a) using air as the coolant, (b) to (f) using water as the coolant with the flowrate increasing from (b) to (f). Vapour temperature 113 °C; diameter of the condensing surface 38 mm (Wilmshurst (1979))

Fig. 9 Condensation of ethylene glycol on an oleic acid-promoted copper surface at three vapour temperatures. The coolant flowrate increases from left to right (Stylianou (1980))

temperature difference, the number of active nucleation sites increases to the extent where, in some locations, the active sites are effectively in contact with neighbours, giving rise to a wetted patch.

The last of the above-mentioned might explain the difference in behaviour of PTFE and monolayer-promoted surfaces. These surfaces would be expected to have different distributions of nucleation sites, the PTFE surface perhaps having fewer of the smallest sites. The tails formed behind moving drops, particularly evident in the case of the work of Utaka *et al.* (1986, 1988), might be explained by the fact that the surface temperature beneath a large moving drop is lower than elsewhere on the surface and the active nucleation site density on the relatively cold surface immediately behind them correspondingly higher. This could lead to locally 'wetted' conditions at the rear of the moving drop and hence the tail. The effect of viscosity could be explained by a coalescence time effect between neighbouring primary drops.
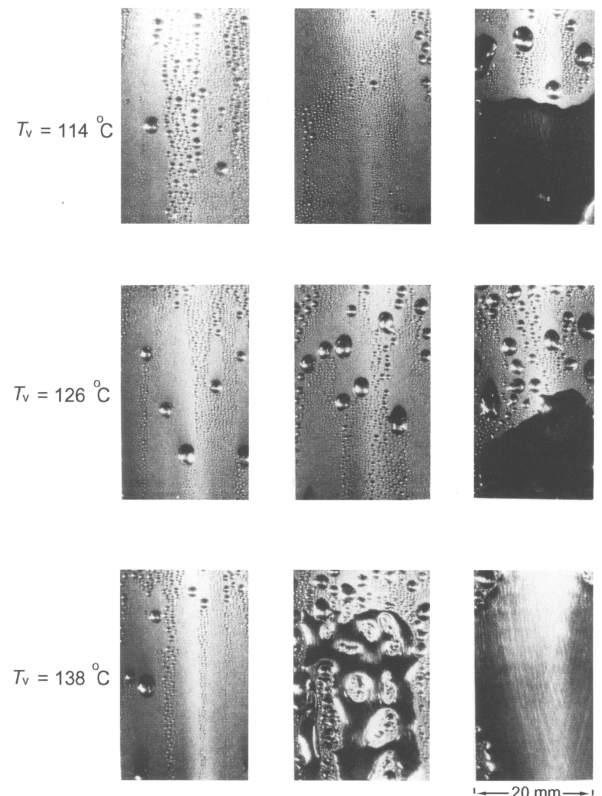
5 EFFECT OF SURFACE THERMAL CONDUCTIVITY—'CONSTRICTION RESISTANCE'

There has been continuing controversy about whether or not the thermal conductivity of the condensing surface plays a significant role in determining the effective heat transfer coefficient in dropwise condensation. Results of several investigations (Griffith and Lee (1967), Tanner *et al.* (1965a), Wilkins and Bromley (1973), Hannemann and Mikic (1976b) and Tsuruta *et al.* (1991)) have been interpreted as indicating a strong effect of the thermal conductivity of the substrate on dropwise condensation heat transfer. Theoretical models (Hannemann and Mikic (1976a), Tsuruta and Tanaka (1991) and Tsuruta (1993)) have also indicated additional thermal resistance dependent on surface thermal conductivity, owing to constriction of heat flux lines around larger drops. Other experimental data (Graham (1969), Aksan and Rose (1973), Rose (1978a, 1978b), Wilmshurst and Rose (1970), Wilmshurst (1979), Stylianou (1980) and Stylianou and Rose (1980)) show this effect to be negligible. With the exception of the measurements of Tsuruta *et al.* (1991), those data which indicate a significant 'constriction effect' have been discussed in detail

by Stylianou and Rose (1980) and the findings attributed to experimental error or promoter ineffectiveness on the surfaces used. The data of Tsuruta *et al.* (1991) and Tsuruta (1993) are discussed below.

In those investigations that showed negligible constriction resistance, the heat flux and temperature gradient were found from accurately determined temperature distributions in the condenser plate as opposed to surface thermometry. The materials tested included phosphor bronze, steel and polytetrafluorethylene (PTFE, Teflon). In the case of PTFE, thin layers of known thickness (Graham (1969), 1.5 μm; Wilmshurst (1979), 10 ± 1 μm; Stylianou (1980), 13.7± 0.4 μm) were used, and the temperature at the copper–PTFE interface was found by extrapolation. Samples of temperature distributions in a PTFE-coated copper condensing plate (Stylianou, 1980) are shown in Fig. 10. The temperature drop across the PTFE layer was found by one-dimensional conduction to obtain the temperature of the PTFE surface.

Figure 11 shows plots of vapour–surface temperature difference obtained from the measurements of Stylianou and Rose (1980) for copper, bronze and PTFE surfaces. The lines in Fig. 11 are from the theory of Le Fevre and Rose (1966) and, as may be seen from Fig. 3, accurately represent the data for a copper surface. The thermal conductivities of the phosphor bronze and PTFE are approximately 7 and 1500 times lower than that of copper. Although there might appear to be evidence of slightly higher temperature differences for bronze at the highest and lowest pressures, the data for PTFE and bronze surfaces are otherwise closely in line with those for copper. Similar conclusions may be drawn from the works of Graham (1969), Aksan and Rose (1973),
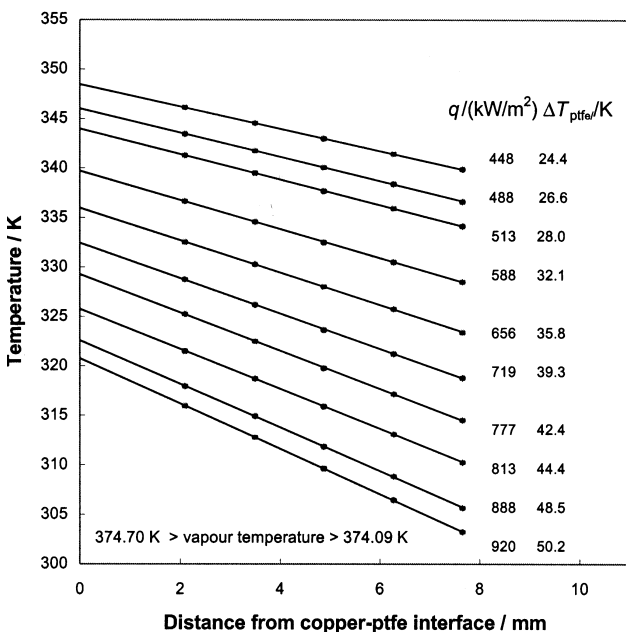
Rose (1978a, 1978b), Wilmshurst and Rose (1970) and Wilmshurst (1979).

The more recent experimental investigation of Tsuruta *et al.* (1991), which showed clear evidence of constriction resistance, merits special consideration. In this work,
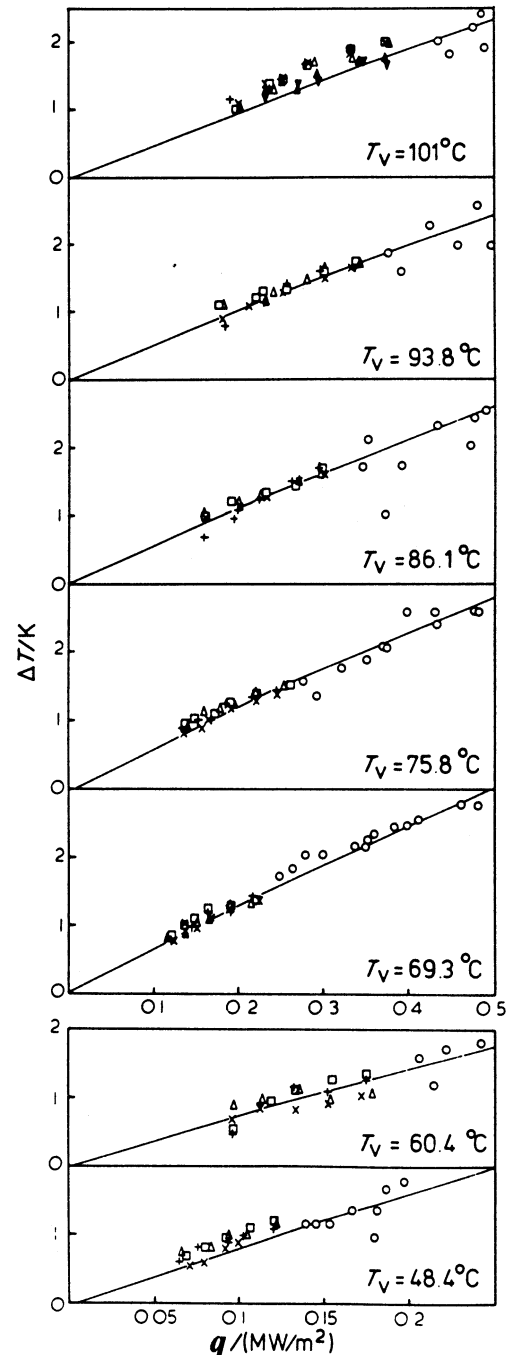
Fig. 11 Dropwise condensation of steam at various pressures on dioctadecyl disulphide-promoted bronze and PTFE-coated copper surfaces (data of Stylianou and Rose (l980)); the open circles denote the PTFE data, the lines are from theory (Le Fevre and Rose (1966)). Figure from Stylianou and Rose (1980)
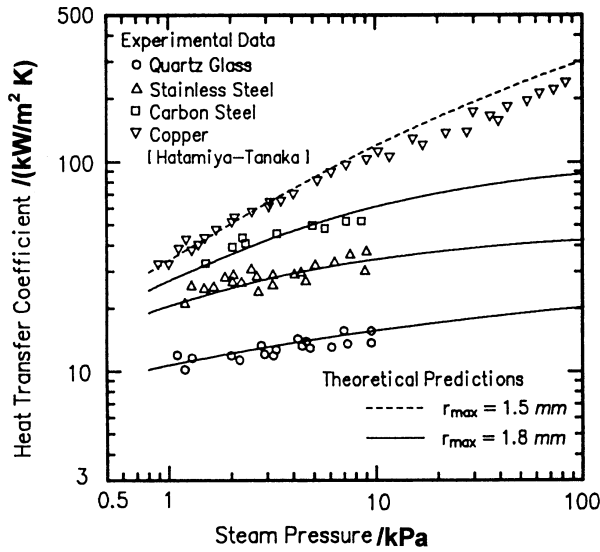
Fig. 10 Measured temperature distributions in a PTFE-coated copper condensing plate during dropwise condensation of steam (Stylianou (1980))

Fig. 12 Dependence of heat transfer coefficient on surface material (Tsuruta *et al.* (1991))

Fig. 13 Surface temperature measurements for dropwise condensation on glass (Tsuruta *et al.* (1991))

measurements were made using surface thermometry. Condensation took place on a thin layer of Si_3N_4 promoted with oleic acid. Figure 12 shows the results for carbon steel ($k = 51.6$ W/m K), stainless steel ($k = 16.0$ W/m K) and quartz glass ($k = 1.3$ W/m K), together with earlier data for copper of Hatamiya and Tanaka (1987). The fact that heat transfer coefficients are plotted against pressure suggests that a relatively small range of heat flux was used, for which the dependence of heat transfer coefficient on heat flux or temperature difference was small and the heat transfer coefficient apparently constant. The data appear to be in quite good agreement with the theory of Tsuruta and Tanaka (1991), although it may be noted that different values of large drop radius were used in the model for copper and the lower-conductivity materials. At a pressure of 10 kPa the heat transfer coefficient found for quartz glass, having a thermal conductivity less than that of copper by a factor of approximately 300, is seen to be less than that for copper by a factor of almost 10.

Examples of surface temperature–time traces of Tsuruta *et al.* (1991) for the quartz glass surface for three heat fluxes are shown in Fig. 13. The mean vapour–surface temperature differences exceed those expected for a copper surface under the same conditions by around 0.9 K at the lowest heat flux and around 2.2 K at the highest heat flux. Although these differences are smaller than the temperature fluctuations, it appears that the mean values of the surface temperature are sufficiently well established and an effect of surface thermal conductivity is demonstrated. It may be noted, however, that the factor of almost 10 ratio between heat transfer coefficients for copper and glass indicated in Fig. 12 is due to a difference in ΔT of only 1.3 K.

Figure 14 illustrates the basis of the model of Tsuruta *et al.* (1991). The condensing surface is envisaged as typified
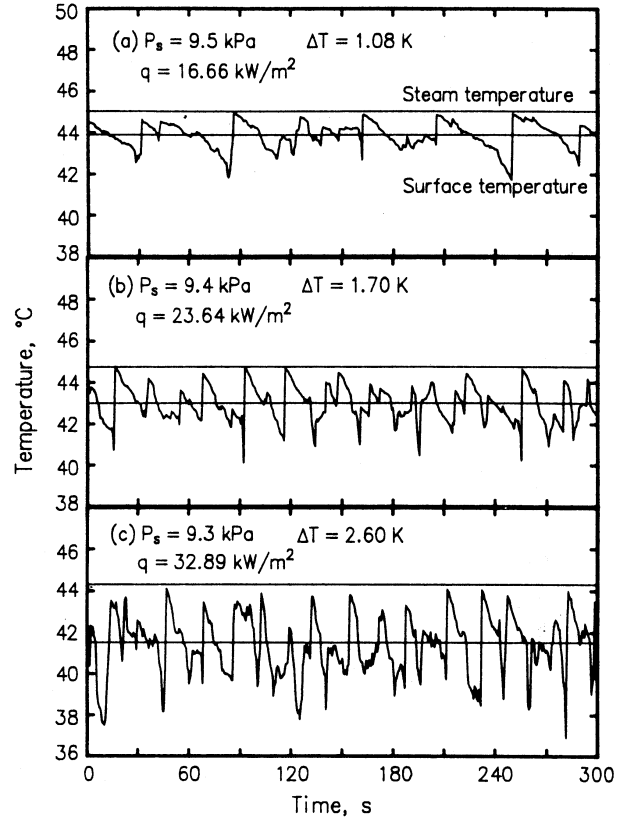
by a large stationary drop surrounded by a region with smaller drops bounded by an adiabatic cylinder (in the condensing plate) centred on the large drop. Tsuruta (1993) extended the model of Tsuruta *et al.* (1991) to the case of a thin low-conductivity polymer layer on a metal
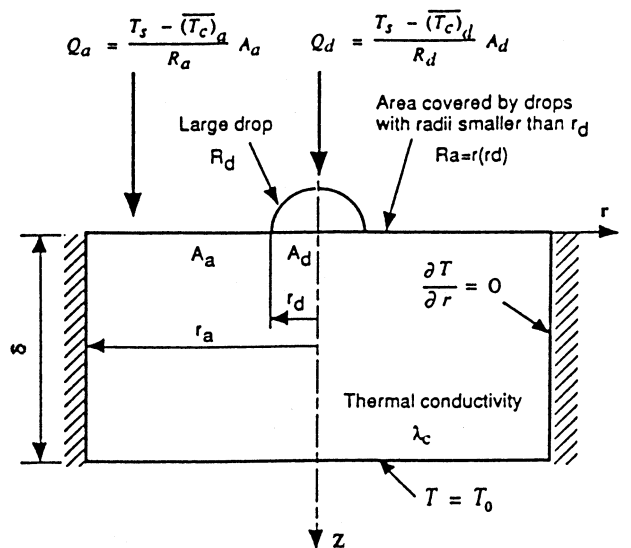

Fig. 14 Basis of the 'constriction resistance' model of Tsuruta and Tanaka (1991) (Tsuruta *et al.* (1991))
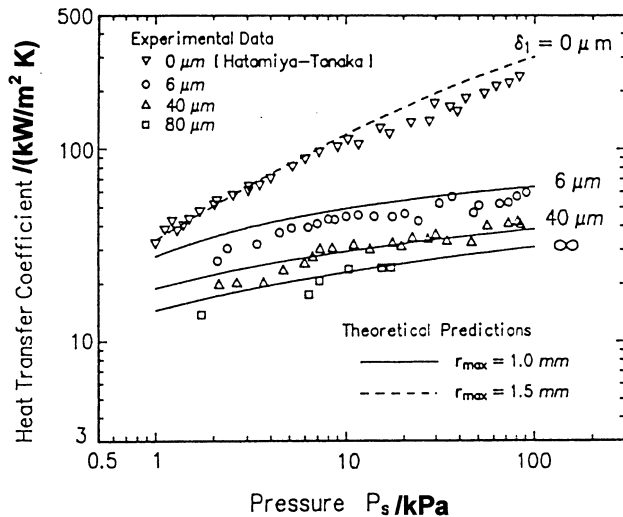
Fig. 15 Comparison of measurements and predictions for a polymer-coated surface (Tsuruta (1993))

Fig. 16 Dropwise condensation on a PTFE-coated surface. Measurements of Stylianou (1980). Open and closed points denote tests on different days. The error bands are due to uncertainty in the PFTE layer thickness (Rose (2000))

surface. Figure 15 shows results of calculations and experiments for a fluorethylene propylene (FEP, $k = 0.251$ W/m K) layer. Again, the calculated results are apparently in good accord with the measurements. Rather surprisingly, however, the predicted heat transfer coefficient for an infinitely thick FEP layer is *higher* than that for quartz glass (see Fig. 12), while the thermal conductivity of the polymer is around 5 times *lower*. It is evident that the model is very sensitive to the value chosen for the radius of the largest stationary drops (1.8 mm was used for the quartz glass calculations, whereas a value of 1 mm was used for the FEP).

Figure 16 shows in more detail the data of Stylianou (1980) for condensation of steam at near-atmospheric pressure on a PTFE-coated surface. The error bars denote uncertainty in the value of ΔT owing to uncertainty in the PTFE layer thickness (the estimated error in the temperature of the PTFE–copper interface was around 0.1 K). It is evident that these data are in conformity with the earlier measurements for copper. At this pressure the calculation of Tsuruta (1993) indicates a heat transfer coefficient of around 50 kW/m² K for the PTFE layer thickness used in the experiments. The line representing this value is also shown in Fig. 16 and clearly is not in agreement with the measurements.

The conditions under which the measurements of Tsuruta *et al.* (1991) and Tsuruta (1993) were made differ in two important ways from those of Aksan and Rose (1973), Rose (1978), Wilmshurst and Rose (1970), Wilmshurst (1979), Stylianou (1980), Stylianou and Rose (l980) and Stylianou and Rose (1983). For the latter group of investigations, where no constriction effect was found, the heat fluxes and the heights of the condensing surfaces were much larger than for those of Tsuruta *et al.* (1991) and Tsuruta (1993). High heat fluxes and tall surfaces have important implications for constriction resistance. The frequency with which a
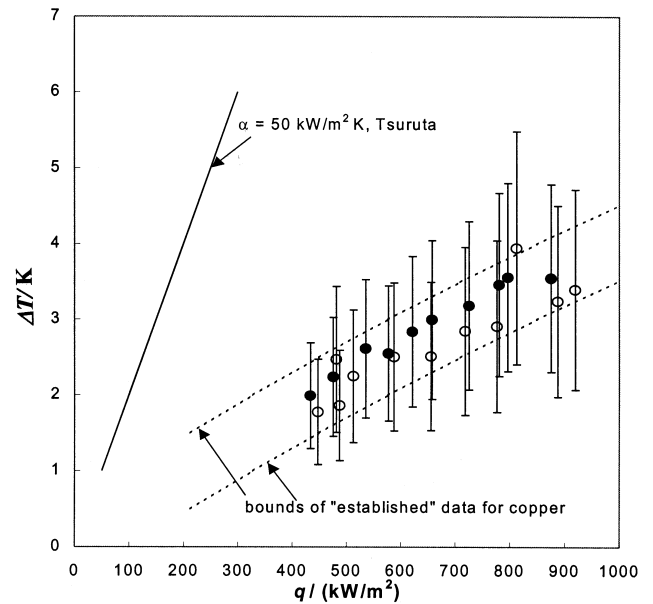
location on the surface is swept by moving drops increases with both condensation rate and with distance down the surface (falling drops sweep diverging tracks so that lower locations are swept more frequently). Rose (1978a, 1978b) suggested that constriction resistance may not be seen at higher heat fluxes owing to the very rapid coalescences and sweeping which have a smoothing effect on the surface temperature. Only at low heat fluxes, and near the top of the condensing surface, where the larger drops are relatively stationary, would surface temperature non-uniformity have a significant effect.

In the model of Tsuruta and Tanaka (1991) and Tsuruta (1993), the magnitude of the mean effective radius of the largest stationary drops plays a critical role as noted. It is not clear (see the photographs in Fig. 4) which radius should be used as *the effective radius of large stationary drops* in the model. It would seem evident that, except in the case of a very short condensing surface, the departing drop radius is not typical, and any 'average' largest drop radius would be much smaller. Regarding comparisons between constriction resistance theory and measurements, it is probably more accurate to say, in view of the strong dependence of the theory on large drop radius, that good agreement with the measurements (for short surfaces at low heat fluxes) can be found when choosing *reasonable* values for this radius, although, in the present author's experience, a maximum radius of drops of around 0.5 mm is more typical for the top of a well-promoted surface, with much smaller values further down.

On the basis of the available evidence, it would seem that 'constriction resistance' effects, leading to effective addi-

tional vapour-side temperature drops of around 1 or 2 K for very low-conductivity surfaces, may be found for heat fluxes perhaps less than about $100 \, \text{kW/m}^2$ and for condensing surface heights less than around 10 mm. For heat fluxes in excess of about $250 \, \text{kW/m}^2$ and for taller surfaces, constriction effects are negligible. Accurate measurements, using a low-conductivity material and covering a wide range of heat flux, are needed to establish these limits more precisely.

6   CONCLUDING REMARKS

Many of the earlier, and some of the more recent, heat transfer measurements for dropwise condensation are vitiated by the presence of air in the vapour. The heat transfer coefficient for dropwise condensation is not constant (heat flux is not proportional to vapour–surface temperature difference). Even when effects of air have been eliminated, misleading conclusions have been reached by comparing heat transfer coefficients measured at widely different heat fluxes, for example, when comparing data at different pressures or for condensation on different surface materials.

Sufficient accurate experimental data are now available to validate theory. The theory of Le Fevre and Rose (1966) is in satisfactory agreement with these data and would thus appear to incorporate the important mechanisms. Effect of thermal conductivity of the substrate is not incorporated in this theory. On the available evidence this effect would only seem to be important for very short condensing surfaces and for low heat fluxes. Accurate measurements covering sufficient ranges of surface thermal conductivity, plate height and heat flux, and preferably using the same experimental technique for all materials, are needed to establish the conditions under which constriction resistance might be significant.

The phenomenon of transition to film condensation is still imperfectly understood. The important mechanisms would appear to be associated with increased proximity of active nucleation sites with increasing vapour–surface temperature difference and with the speed at which small or primary drops coalesce as compared with their growth rate.

Finally, and most importantly from a practical view point, a reliable means of promoting dropwise condensation under industrial conditions has yet to be convincingly demonstrated.

Other relatively recent reviews, which give somewhat different viewpoints and emphasis, have been given by Tanasawa (1991), Rose (1997), Rose *et al.* (1999) and Rose (2000).

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