ment of a Solid/Liquid/Fluid Contact Line," *J. Colloids and Interface Science*, Vol. 35, pp. 85–101.

Israelachvili, J. N., 1985, *Intermolecular and Surface Forces: With Applications to Colloidal and Biological Systems*, Academic Press, New York.

Lopez, J., Miller, C. A., and Ruckenstein, E., 1976, "Spreading Kinetics of Liquid Drops on Solids," *J. of Colloids and Interface Science*, Vol. 56, pp. 460–468.

Miller, C. A., and Ruckenstein, E., 1974, "The Origin of Flow During Wetting of Solids," *J. of Colloids and Interface Science*, Vol. 48, pp. 368–373.

Truong, J. G., 1987, "Ellipsometric and Interferometric Studies of Thin Films Wetting on Isothermal and Nonisothermal Surfaces," Ph.D. Thesis, Rensselaer Polytechnic Institute, Troy, NY.

Truong, J. G., and Wayner, P. C., Jr., 1987, "Effects of Capillary and van der Waals Dispersion Force on the Equilibrium Profile of a Wetting Liquid: Theory and Experiment," *J. Chem. Phys.*, Vol. 87, pp. 4180–4188.

Wayner, P. C., Jr., Kao, Y. K., and LaCroix, L. V., 1976, "The Interline Heat Transfer Coefficient of an Evaporating Wetting Film," *Int. J. Heat Mass Transfer*, Vol. 19, pp. 487–492.

Wayner, P. C., Jr., 1978, "The Effect of the London–van der Waals Dispersion Force on Interline Heat Transfer," ASME JOURNAL OF HEAT TRANSFER, Vol. 100, pp. 155–159.

An Upper Bound for the Critical Boiling Heat Flux

W. R. Gambill[1] and J. H. Lienhard[2]

Nomenclature

$\bar{c}$ = mean molecular velocity
h_{fg} = latent heat of vaporization
J = flux of molecules
k = Boltzmann constant (gas constant per molecular mass)
m = mass of a molecule
n = number density of molecules
p, p_c, p_r = pressure, critical pressure, reduced pressure
q_{max} = peak or "burnout" heat flux
$q_{max,max}$ = $\rho_g h_{fg} \sqrt{RT/2\pi}$ (theoretical upper bound for q_{max})
R = ideal gas constant on a unit mass basis
T_{sat} = saturation temperature
T_w = wall temperature of a heating (or cooling) surface
ΔT = $T_w - T_{sat}$
ρ_g = density of saturated vapor
ϕ = $q_{max}/q_{max,max}$

Introduction

The great value of boiling and condensation in process heat transfer is that they yield the highest known heat transfer coefficients. People who have to transfer a great deal of energy rapidly, under fairly low driving temperature differences, usually turn to these processes. We therefore constantly face the question, "What is the upper limit on these heat fluxes—how far can they be driven?"

Several investigators have taken particular pains to push the peak boiling heat flux q_{max} to its limits. For example, Gambill

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and Greene (1958) used inlet tangential-slot swirl-flow generators with water at pressures up to 7 MPa and at axial velocities up to 30 m/s. They obtained burnout heat fluxes as high as 172.8 MW/m². When this study was later extended to swirl flows induced by internal twisted tapes, Gambill et al. (1961) obtained q_{max} up to 117.8 MW/m² in water, and Gambill and Bundy (1963) obtained q_{max} up to 28.4 MW/m² in ethylene glycol.

Ornatskii and Vinyarskii (1965) have observed q_{max} in water flowing axially in small (0.5 mm i.d.) tubes at speeds up to 90 or 100 m/s, and attained a q_{max} of 224.5 MW/m².

Japanese investigators (see, e.g., Monde and Katto (1978) and Katto and Shimizu (1979)) have measured q_{max} in a different kind of system—a liquid jet impinging perpendicularly upon a heated disc. The set of experiments using water jets at atmospheric pressure yielded several values of q_{max} in excess of 10 MW/m². Monde and Katto's highest value was 18.26 MW/m². (Katto and Shimizu also did experiments using Freon-113 jets and Freon-12 jets under a wide range of elevated pressures, but burnout heat fluxes in these cases were not extremely high.)

The existing data strongly indicate that we should hold the hope of reaching far higher heat fluxes than we presently reach. Our present aim is to set a theoretical and/or practical limit on attainable heat fluxes, so we might henceforth aspire to reach that limit in practical design.

The Upper Limit for the Limiting Heat Flux

The highest heat flux that can conceivably be achieved in a phase-transition process—which we designate as $q_{max, max}$—was given by Schrage (1953), who made reference to antecedents of the idea extending back into the 19th century. Tien and Lienhard (1976, 1979) independently made this calculation in a simple approximate way, in reference to boiling burnout, in 1970. They noted that, if one could contrive to collect every vapor molecule that leaves a liquid-vapor interface without permitting any vapor molecules to return to the liquid, then

$$q_{max,max} = (mJ)h_{fg} \qquad (1)$$

where m is the mass of a molecule and J is the flux of molecules in one direction. The kinetic theory gives $J = n\bar{c}/4$, where n is the number density of vapor molecules and $\bar{c}$ is the average molecular speed. They took the vapor leaving the liquid as having roughly the average speed of a Maxwellian gas, $\bar{c} = \sqrt{8RT/\pi}$. (This will be a reasonable approximation, even at pressures rather close to the critical pressure, because the average energy of molecules is little affected by the forces that give rise to nonideality in a gas.) The use of this $\bar{c}$ in the preceding equation gives

$$J \approx n\sqrt{kT/2\pi m} \qquad (2)$$

We substitute equation (2) into (1) and obtain

$$q_{max,max} \equiv \rho_g h_{fg} \sqrt{RT/2\pi} \qquad (3)$$

Schrage's more complete calculation took account of the distortion of the Maxwellian distribution when molecules are prevented from returning to the surface. He indicated that this would give rise to a hard-to-evaluate constant on the right side of equation (3), but that this constant would remain on the order of unity. Consequently the right side of equation (3) is not an exact upper bound, but rather a group that *characterizes* the upper limit correctly and gives its correct order of magnitude.

The arguments underlying this result would apply equally well in reverse—for vapor molecules condensing on an inter-

face. Thus equation (3) provides an approximate upper bound for either boiling or condensation. (In this connection we do not include an accommodation coefficent in equation (3) because the preponderance of evidence (see, e.g., Mills, 1965) suggests that molecular accommodation is imperfect only when an interface is impure, and our interest is restricted to the kind of rapidly renewed interfaces that one encounters in intense nucleate boiling or condensation.)

Values of $q_{max,max}$ are plotted against reduced pressure, p/p_c, for some typical liquids in Fig. 1 to illustrate the astonishing heat fluxes that might seem to be attainable.

There are several reasons that this limit will be unattainable in practice. The most serious restriction is that many vapor molecules will inevitably be returned to the interface by molecular collisions. We can only hope to slow the return flow of vapor molecules, not to eliminate it.

Another problem lies in the premise that all the heat ultimately passes through a liquid–vapor interface. The problem in designing a process is to get the heat to flow through the liquid, up to an interface, and away from the interface on the vapor side. That is what the various investigators reviewed in the preceding section contrived to do with the help of swirl inserts, jets, high subcooling, high velocities, elevated pressure, and so forth. If a given strategy permits some of the heat to find a less efficient path to the liquid bulk, without involving phase change at a liquid–vapor interface, it will undoubtedly yield a lower heat flux.

Comparison of the Upper Bound With Data

Boiling Burnout. Table 1 and Fig. 2 display all of the highest experimental burnout heat fluxes we could locate, over a full range of pressures. We represent these data in the dimensionless form suggested by equation (3), as a function of reduced pressure

$$\phi \equiv \frac{q_{max}}{\rho_g h_{fg} \sqrt{RT/2\pi}} = f(p_r) \qquad (4)$$

The data for the pressure range $0.004 < p_r < 0.10$ appear to be consistently bounded at $\phi = 0.1$. However, at higher pressures, the highest existing values of ϕ fall off roughly as $(p_r)^{-2}$. We see two possible reasons for this decrease:

As q approaches $q_{max,max}$ in any configuration in the lower pressure range, there will be a great deal of local boiling and of

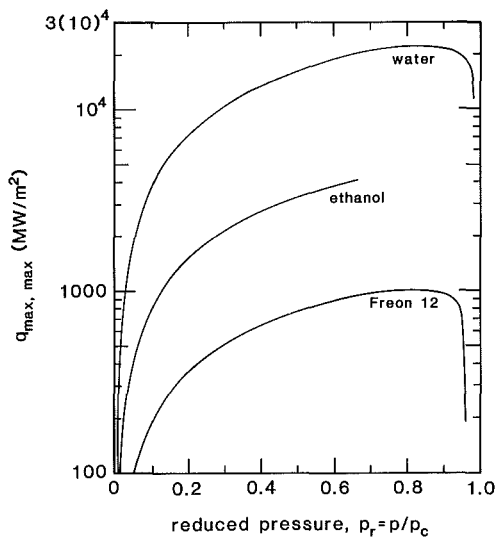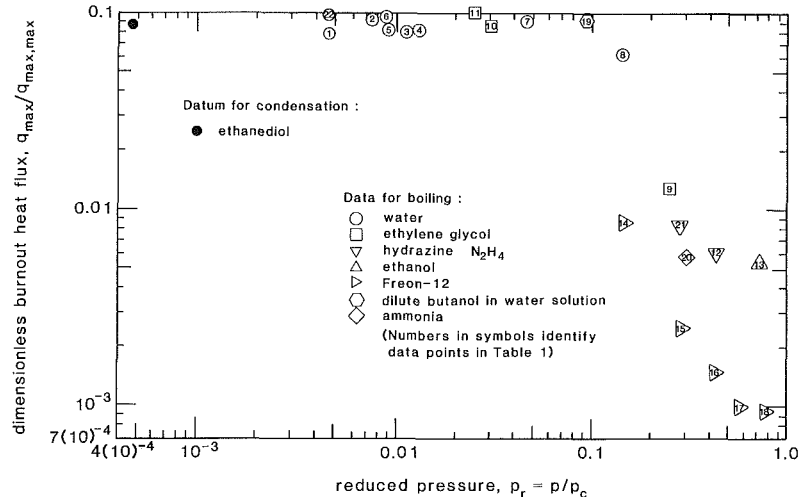

Fig. 1 Pressure dependence of $q_{max,max}$

Fig. 2 Pressure dependence of the highest measured burnout heat fluxes

Table 1 List of extreme values of q_{max}

No.	Boiled liquid	$p_r = p/p_c$	ϕ	Source and configuration	
1	water	0.0046	0.0815	Monde (1978)	jet impinging on a disk
2	water	0.0074	0.0955	Gambill (1958)	axial tube flow
3	water	0.0111	0.0809	" "	inlet generated swirl
4	water	0.0130	0.0813	" "	" " "
5	water	0.0090	0.0840	" (1961)	twisted tape generated swirl
6	water	0.0091	0.0978	" "	" " " "
7	water	0.0459	0.0907	Ornatskii (1965a)	axial flow, uniform q
8	water	0.1423	0.0625	" "	axial flow, nonuniform q
9	ethylene glycol	0.2497	0.0128	Gambill (1963)	twisted-tape swirl
10	ethylene glycol	0.0298	0.0863	" "	" " "
11	ethylene glycol	0.0243	0.1033	" "	" " "
12	hydrazine	0.4269	0.0060	Hines (1959)	axial tube flow
13	ethanol	0.7137	0.0054	Kutateladze (1965)	" " "
14	Freon-12	0.146	0.0087	Katto (1979)	jet impinging on a disk
15	Freon-12	0.282	0.0025	" "	" " " " " "
16	Freon-12	0.430	0.0015	" "	" " " " " "
17	Freon-12	0.571	0.0010	" "	" " " " " "
18	Freon-12	0.678	0.00095	" "	" " " " " "
19	water/3 percent butanol	0.0965	0.0961	Ornatskii (1965b)	axial flow, nonuniform q
20	ammonia	0.3056	0.0058	Noel (1961b)	axial tube flow
21	hydrazine	0.2791	0.0084	Noel (1961a)	" " "
22	water	0.0047	0.0916	Weatherhead (1955)	crossflow over a cylinder

interfacial structure near the surface. Not much of the heat *can* flow straight into the liquid without passing through an interface. At the higher pressures, however, $\rho_g h_{fg}$ increases, so less vapor volume is generated at a given heat flux. It is very probable that more of the heat finds a less efficient, direct convection path, at higher pressures.

Consider the Katto–Shimizu Freon data, for example. The Japanese visual observations (all made at low pressure), and observations of a cold-air-in-water analogy in support of Lienhard and Eichhorn's (1979) study of the jet-disk system, show that, at low pressure, burnout occurs at the outer edge of the disc. But even just before burnout, portions of the very thin sheet spreading over the disk are still intact at the outer edge of the disk. As the pressure increases and less vapor is generated, more and more of the sheet remains intact and more of the heat has to escape by direct conduction/convection through these portions. Thus, increased access to alternate flow paths is one reason that ϕ drops off with increased pressure.

The second reason becomes evident when we consider Fig. 1, where we read $q_{max,max}$ values for water in excess of 20,000 MW/m^2 at high pressures. It would require a temperature drop of about 4700 K to drive one-tenth of this heat flux through a 1-mm thickness of pure silver. At higher pressures it therefore becomes impossible to supply a heater surface with these heat fluxes (at steady state) by any known means. Gambill et al. (1960, 1961), for example, encountered the wall ΔT limit in a few of their early swirl-flow tests at $q > 94$ MW/m^2. Using 304 stainless steel tubes with a 0.89 mm wall thickness at these flux levels, they suffered tube failures initiated by melting of the external wall.

It is important to note that it was not the wall ΔT limitation that caused the high-pressure Katto–Shimizu measurements to be so low. The wall ΔT limitation and the inefficient-path limitation are independent of one another, with the wall ΔT limitation arising when $q_{max,\,max}$ becomes very high. The Katto–Shimizu data become limited by an inefficient heat transfer path long before the wall ΔT limitation is reached. Increasingly inventive strategies for configuring the convective flow are needed to beat the inefficient-path limitation as p_r is increased.

Effusion-Limited Burnout. The first motivation for looking at $q_{max,max}$ has been that of setting target levels for the heat transfer to which we may aspire in the design of boiling processes. From a practical standpoint, that limit appears to be about 0.1 $q_{max,max}$.

However, another very important use for the limit is immediately evident. If burnout consistently reaches one tenth of the limit, then burnout is occurring because molecular effusion fails to provide sufficient cooling. Molecular effusion is a hitherto unobserved burnout mechanism.

A very recent study of burnout is subcooled boiling by Elkassabgi and Lienhard (1988) shows that, at very high levels of liquid subcooling, the pool boiling q_{max} for horizontal cylinders becomes independent of subcooling. With reference to a prepublication version of this note, they show that highly subcooled burnout occurs in each of four different fluids at 0.01 $q_{max,\,max}$. That means that, even at fairly low heat fluxes, the breakdown in the boiling process occurs because molecular effusion fails to keep up with the heat supply. (We have subsequently discovered the same kind of a breakdown—at higher heat fluxes—during subcooled *flow* boiling over cylinders.)

Condensation. Condensation, like boiling, can occur in a "film" mode or a "nucleate" (more commonly called "dropwise") mode. Indeed, one can quite reasonably anticipate an approximate reflection of the boiling curve in the negative ΔT, negative heat transfer, portion of the boiling curve—a reflec-
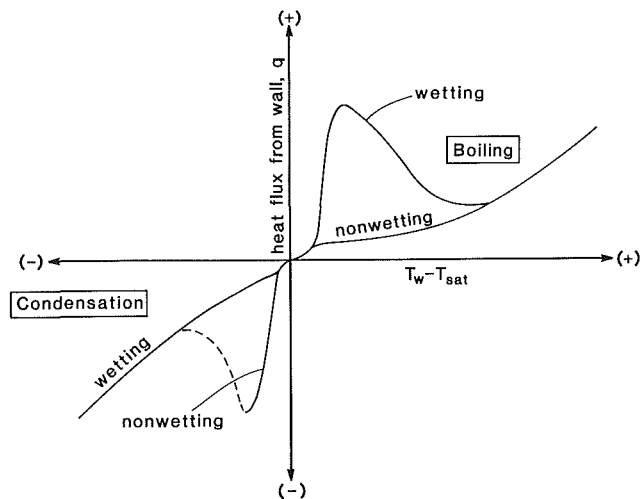

Fig. 3 Boiling and condensation curves viewed as approximate reflections of one another

tion that could rightly be called a "condensation" curve[2] (see Fig. 3).

A possible objection to such an analogy is that *either* dropwise or film condensation can exist at a given low ΔT. However, it is known that the same thing is true in pool boiling. Experiments conducted in completely nonwetting liquids have revealed that film boiling can be established in what would normally be the nucleate boiling range of temperature differences (see, e.g., Winterton, 1983). Thus either the boiling or condensation portion of the curve can exhibit hysteresis in the film or nucleate modes (Witte and Lienhard, 1982).

One would then look for a peak flux in condensation, analogous to that which occurs in boiling. Few investigators of dropwise condensation have pushed the heat flux to a maximum. Stylianou and Rose (1983) recently reviewed the history of extreme values of dropwise condensation measurements. They also developed an apparatus in which they subjected ethanediol (ethylene glycol) vapor to walls that could be brought to temperatures more than 80 K below saturation, at subatmospheric pressures. Their excellent photographs show clear evidence of hydrodynamic transitions from dropwise to filmwise condensation at maximum heat fluxes that range from 0.56 to 0.97 MW/m^2.

None of these measurements were obtained with serious augmentation; yet they reach values of ϕ that range from 0.09 at the lowest pressure of 3800 Pa, down to 0.02 at the highest pressure of 35,600 Pa. The clear implication is that very-low-pressure dropwise condensation also reaches the apparent practical limit of $\phi = 0.1$. Stylianou and Rose include some earlier data for aniline and nitrobenzene that they indicate are too crude to use quantitatively, but which verify these trends. (The aniline data range from $\phi \cong 0.14$ at low pressure down to 0.01 at higher pressure.) They report extreme condensing heat flux values reported by Tanasawa and Utaka (1979) for water at one atmosphere that yield ϕ values as high as 0.054. We include the limiting ethanediol value in Fig. 2.

Conclusions

1 The upper bound for heat flux by phase change, $q_{max,max}$, is given approximately by equation (3).

2 Boiling burnout can occur as the result of the failure of molecular effusion to keep up with a heat supply.

3 The practical limit to heat transfer by phase change is one

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[2]This view was suggested to the second author by Prof. J. T. Kimbrell, Washington State University, ca. 1963.

tenth of $q_{\mathrm{max,max}}$, at least at pressures less than about one tenth of the critical pressure.

4 At higher pressures, $q_{\mathrm{max,max}}$ is such a large number that, for many liquids, we are not capable of supplying it.

5 If means exist for supplying $q_{\mathrm{max,max}}$ at higher pressures, it then becomes increasingly difficult to capitalize on the enormous magnitude of $q_{\mathrm{max,max}}$ because it is hard to design processes that prevent heat from finding non-phase-change flow paths.

6 The present ideas appear to apply to the limiting heat flux in dropwise condensation as well as to q_{max} in nucleate boiling. The few available observations of q_{max} in dropwise condensation appear to define the $\phi = 0.1$ limit, *without serious augmentation*, at low pressures.

References

Elkassabgi, Y., and Lienhard, J. H., 1988, "Influences of Subcooling on Burnout of Horizontal Cylindrical Heaters," ASME JOURNAL OF HEAT TRANSFER, Vol. 110, No. 2, pp. 479–486.

Gambill, W. R., and Greene, N. D., 1958, "Boiling Burnout With Water in Vortex Flow," *Chem. Engr. Prog.*, Vol. 54, No. 10, pp. 68–76.

Gambill, W. R., Bundy, R. D., and Wansbrough, R. W., 1960, "Heat Transfer, Burnout, and Pressure Drop for Water in Swirl Flow Through Tubes With Internal Twisted Tapes," Oak Ridge Nat'l. Lab. Rpt. ORNL-2911, Appendix 2, pp. 95–99.

Gambill, W. R., Bundy, R. D., and Wansbrough, R. W., 1961, "Heat Transfer, Burnout, and Pressure Drop for Water in Swirl Flow Through Tubes With Internal Twisted Tapes," *Chem. Engr. Prog. Symp. Series*, Vol. 57, No. 32, pp. 127–137.

Gambill, W. R., and Bundy, R. D., 1963, "High-Flux Heat-Transfer Characteristics of Pure Ethylene Glycol in Axial and Swirl Flow," *AIChE Journal*, Vol. 9, No. 1, pp. 55–59.

Hines, W. S., 1959, "Forced Convection and Peak Nucleate Boiling Heat Transfer Characteristics for Hydrazine Flowing Turbulently in a Round Tube at Pressures to 1000 psia," Rocketdyne Rept. R-2059.

Katto, Y., and Shimizu, M., 1979, "Upper Limit of the CHF in the Forced Convection Boiling on a Heated Disc With a Small Impinging Jet," ASME JOURNAL OF HEAT TRANSFER, Vol. 101, No. 2, pp. 265–269.

Kutateladze, S. S., 1965, "Criteria of Stability in Two-Phase Flow," *Symposium on Two-Phase Flow*, Vol. 1, Univ. of Exeter, United Kingdom, pp. 83–92.

Lienhard, J. H., and Eichhorn, R., 1979, "On Predicting Boiling Burnout for Heaters Cooled by Liquid Jets," *Int. J. Heat Mass Transfer*, Vol. 22, pp. 774–776.

Mills, A. F., 1965, "The Condensation of Steam at Low Pressures," Tech. Rpt. on NSF GP-2520, Series No. 6, Issue No. 39, Space Sciences Lab., Univ. of Calif. Berkeley.

Monde, M., and Katto, Y., 1978, "Burnout in High Heat-Flux Boiling System With an Impinging Jet," *Int. J. Heat Mass Transfer*, Vol. 21, pp. 295–305.

Noel, M. B., 1961a, "Experimental Investigation of Heat Transfer Characteristics of Hydrazine and a Mixture of 90 percent Hydrazine and 10 percent Ethylenediamine," JPL Technical Rept. No. 32-109.

Noel, M. B., 1961b, "Experimental Investigation of the Forced-Convection and Nucleate-Boiling Heat-Transfer Characteristics of Liquid Ammonia," JPL Technical Rept. No. 32-125.

Ornatskii, A. P., and Vinyarskii, L. S., 1965a, "Heat Transfer Crisis in a Forced Flow of Underheated [Subcooled] Water in Small-Bore Tubes," *High Temp.*, Vol. 3, No. 3, pp. 400–406.

Ornatskii, A. P., and Vinyarskii, L. S., 1965b, "Critical Heat Transfer in the Forced Motion of Underheated Water-Alcohol Mixtures in Tubes of Diameter 0.5 mm," *High Temp.*, Vol. 3, No. 6, pp. 881–882.

Schrage, R. W., 1953, *A Theoretical Study of Interphase Mass Transfer*, Columbia University Press, New York, Chap. II.

Stylianou, S. A., and Rose, J. W., 1983, "Drop-to-Filmwise Condensation Transition: Heat Transfer Measurements for Ethanediol," *Int. J. Heat Mass Transfer*, Vol. 26, No. 5, pp. 747–760.

Tanasawa, I., and Utaka, Y., 1979, "Measurement of Condensation Curves for Dropwise Condensation Heat Transfer," *Proc. 18th Nat'l. Heat Transfer Conf.*, ASME, p. 63.

Tien, C. L., and Lienhard, J. H., 1976, *Solutions Manual to Accompany Statistical Thermodynamics*, revised printing, Hemisphere, Washington, p. 13.

Tien, C. L., and Lienhard, J. H., 1979, *Statistical Thermodynamics*, revised printing, Hemisphere, Washington, p. 55, Problem 2.21.

Weatherhead, R. T., 1955, "Boiling Burnout Heat Flux for Wires in Water Crossflow at Atmospheric Pressure," Report NDA-9 (Boiling Burnout Progress No. 5).

Winterton, R. H. S., 1983, "Comments on 'On the Existence of Two 'Transition' Boiling Curves'," *Int. J. Heat Mass Transfer*, Vol. 26, No. 7, pp. 1103–1104.

Witte, L. C., and Lienhard, J. H., 1982, "On the Existence of Two 'Transition' Boiling Curves," *Int. J. Heat Mass Transfer*, Vol. 25, No. 6, pp. 771–779.

A New Interpolation Formula for Forced-Convection Condensation on a Horizontal Surface

J. W. Rose[1]

Nomenclature

$$G = \left(\frac{k\Delta T}{\mu h_{fg}}\right)\left(\frac{\rho\mu}{\rho_v\mu_v}\right)^{1/2}$$

h_{fg} = specific enthalpy of evaporation
k = thermal conductivity of condensate
Nu = local Nusselt number = $qx/k\Delta T$
Pr = Prandtl number of condensate
q = heat flux
$\tilde{\mathrm{Re}}$ = local "two-phase" Reynolds number = $U_\infty \rho x/\mu$
U_∞ = free-stream vapor velocity
x = distance from leading edge of surface
ΔT = temperature difference across condensate film
μ = viscosity of condensate
μ_v = viscosity of vapor
ρ = density of condensate
ρ_v = density of vapor

The problem of forced-convection condensation, with (saturated) vapor flow parallel to a horizontal isothermal plane surface, has been considered in detail by Koh (1962). Solutions were obtained on the basis of the uniform-property boundary-layer equations for both condensate and vapor. It was shown that the surface heat transfer coefficient was expressible in terms of a relationship between four dimensionless parameters, thus

$$\mathrm{Nu}\,\tilde{\mathrm{Re}}^{-1/2} = \psi_1\left\{\frac{k\Delta T}{\mu h_{fg}}, \ \mathrm{Pr}, \ \left(\frac{\rho\mu}{\rho_v\mu_v}\right)^{1/2}\right\} \qquad (1)$$

or alternatively

$$\mathrm{Nu}\,\tilde{\mathrm{Re}}^{-1/2} = \Phi_1\left\{G, \ \mathrm{Pr}, \ \left(\frac{\rho\mu}{\rho_v\mu_v}\right)^{1/2}\right\} \qquad (2)$$

where

$$G = \frac{k\Delta T}{\mu h_{fg}}\left(\frac{\rho\mu}{\rho_v\mu_v}\right)^{1/2} \qquad (3)$$

However, in the range of practical interest, the Prandtl number dependence was weak, leading to

$$\mathrm{Nu}\,\tilde{\mathrm{Re}}^{-1/2} = \psi_2\left\{\frac{k\Delta T}{\mu h_{fg}}, \ \left(\frac{\rho\mu}{\rho_v\mu_v}\right)^{1/2}\right\} \qquad (4)$$

or

$$\mathrm{Nu}\,\tilde{\mathrm{Re}}^{-1/2} = \Phi_2\left\{G, \ \left(\frac{\rho\mu}{\rho_v\mu_v}\right)^{1/2}\right\} \qquad (5)$$

Equations (4) and (5) relate to the case where the condensate film is treated on the basis of the Nusselt approximations, i.e., neglecting the inertia and convection terms in the conservation equations. The presence of $(\rho\mu/\rho_v\mu_v)^{1/2}$ arises from the conditions of continuity of velocity and shear stress at the vapor-liquid interface. The separate dependence on this parameter disappears in the limiting cases of zero and infinite condensation rate. In the former case Cess (1960) had earlier shown that

$$\mathrm{Nu}\,\tilde{\mathrm{Re}}^{-1/2} = 0.436G^{-1/3} \qquad (6)$$

valid for $G \rightarrow 0$, while Shekriladze and Gomelauri (1966) had obtained, for infinite condensation rate,

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