Int. J. Heat Mass Transfer. Vol. 18, pp. 875–878. Pergamon Press 1975. Printed in Great Britain

FILM BOILING FROM A PARTLY SUBMERGED SPHERE

E. MARSCHALL and L. C. FARRAR
Department of Mechanical Engineering, University of California,
Santa Barbara, California 93106, U.S.A.

(*Received* 11 *March* 1974 *and in revised form* 13 *August* 1974)

Abstract—An experimental study of film boiling from a partly submerged sphere is carried out. The experimentally obtained results are compared with theoretically predicted data. At high temperature differences, the experimental Nusselt numbers are larger than the predicted Nusselt numbers.

NOMENCLATURE

c_p, specific heat of vapor;
c_{pl}, specific heat of liquid;
g, gravitational acceleration;
h, heat-transfer coefficient;
h_{fg}, latent heat of vaporization;
k, thermal conductivity;
Nu, Nusselt number;
P, total pressure;
P_0, atmospheric pressure;
q_{rad}, radiant heat transfer;
r, radial coordinate;
R_0, radius of sphere;
T, temperature;
T_s, saturation temperature;
T_l, temperature of liquid;
T_w, sphere temperature;
v_r, radial velocity component;
v_θ, theta velocity component.

Greek symbols

δ, gap thickness;
ε_l, emissivity of liquid;
ε_w, emissivity of sphere;
λ, modified latent heat;
ν, kinematic viscosity of vapor;
ν_l, kinematic viscosity of liquid;
ρ, vapor density;
ρ_l, liquid density;
σ_r, Stefan–Boltzmann constant.

INTRODUCTION

A MAJOR problem in the evaporation of salt solutions is the formation of scale on heating surfaces. This problem is especially severe when it is required to completely evaporate the solution, since, in this case, the usually employed methods for scale prevention fail. Therefore, a new technique for a complete, scale-free evaporation of aqueous salt solutions is under study. The basic idea of this technique is to use film boiling in the evaporating process. Heating surfaces are maintained at temperatures high enough to ensure a stable vapor film between the heating surface and the solution. If the vapor–liquid interface remains undisturbed or is only slightly disturbed, evaporating liquid and heating surfaces are effectively separated and scale formation on the heating surface is prevented. As a possible heater configuration for this process, partly submerged spheres have been investigated. In the following, heat-transfer experiments are described, and experimental results are compared with a simplified theory on film evaporation. These experiments have been carried out with distilled water as fluid.

EXPERIMENTAL APPARATUS

A schematic of the film boiling apparatus is presented in Fig. 1. Distilled water at a constant flowrate leaves the constant head tank (1), passes through a metering valve (2) and a calibrated flowmeter (3), and enters a glass pipe (4). At the top of the glass pipe, an inconel sphere (5) is located which is supported by a ceramic tube (6). The inconel sphere is heated by a 2·5 kW induction heater operating at about 400 KHz. Inconel 600 has been chosen because of its good corrosion resistance at high temperatures. The temperature of the sphere is measured with a shielded thermocouple which is inserted through the ceramic pipe, so that the thermocouple junction is placed in the center of the sphere. Since the surface of the sphere is heated by induction, the thermocouple measures the average surface temperature. The water level relative to the sphere is controlled by flowrate, sphere diameter, and sphere temperature. However, water flowrates are adjusted to values small enough to ensure complete evaporation of the liquid at steady-state operation. The water level can be read from a graduated glass pipe (7).

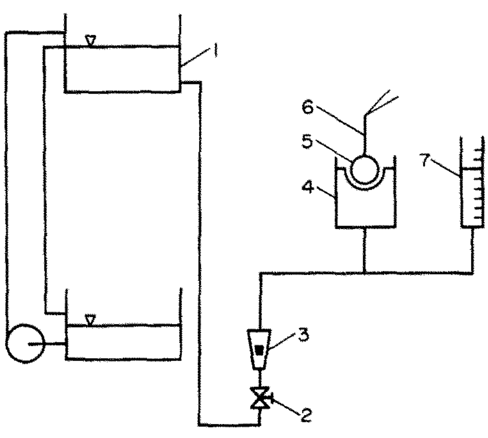

FIG. 1. Experimental apparatus.

HEAT-TRANSFER ANALYSIS

Film boiling from fully submerged spheres has frequently been studied [1–4]. A quite successful analysis of this physical situation has been carried out by Hendricks and Baumeister [2]. After some modifications, their approach has been adopted to analyze film boiling heat transfer from partially submerged spheres. The analytical model and some nomenclature is shown in Fig. 2. If the vapor flow in the gap between

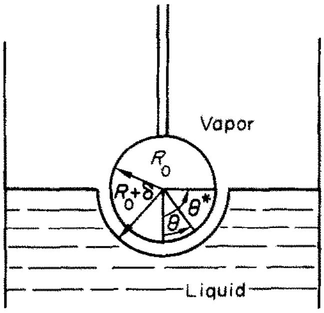

FIG. 2. Film boiling model.

liquid and sphere is assumed to be laminar and at a steady state, and if it is further assumed that inertia and body force terms can be neglected, the governing equations for conservation of mass and momentum are:

continuity:

$$0 = \frac{1}{r^2}\frac{\partial}{\partial r}(r^2 v_r) + \frac{1}{r\sin\theta}\frac{\partial}{\partial\theta}(v_\theta \sin\theta) \tag{1}$$

momentum:

$$0 = -\frac{1}{\rho}\frac{\partial P}{\partial r} + \nu\left[\nabla^2 v_r - \frac{2}{r^2}v_r - \frac{2}{r^2}\frac{\partial v_\theta}{\partial\theta} - \frac{2}{r^2}v_\theta\cot\theta\right] \tag{2}$$

and

$$0 = -\frac{1}{r\rho}\frac{\partial P}{\partial\theta} + \nu\left[\nabla^2 v_\theta + \frac{2}{r^2}\frac{\partial v_r}{\partial\theta} - \frac{v_\theta}{r^2\sin^2\theta}\right]. \tag{3}$$

The vapor participation in radiant energy exchange is assumed to be small. Thus, the convective terms in the energy equation can be neglected if the latent heat of vaporization is replaced by an enthalpy difference, which contains the latent heat. With this the energy equation may be reduced to

$$\frac{k}{r^2}\frac{\partial}{\partial r}\left(r^2\frac{\partial T}{\partial r}\right) = 0. \tag{4}$$

To solve equations (1)–(4), appropriate boundary conditions have to be specified. For an angle $\theta < \theta^*$, these conditions are:

$$\text{at} \quad r = R_0: \quad v_r = 0, \quad v_\theta = 0, \quad T = T_w. \tag{5}$$

At the liquid–vapor interface, the following conditions are considered:

(a) non-slip conditions

$$\text{at} \quad r = R_0+\delta: \quad v_r = v_r(R_0+\delta), \quad v_\theta = v_l \approx 0, \quad T = T_s$$

or (b) slip conditions

$$\text{at} \quad r = R_0+\delta: \quad v_r = v_r(R_0+\delta), \quad \frac{\partial v_\theta}{\partial r} = 0, \quad T = T_s. \tag{6}$$

Two additional conditions are:

$$\text{at} \quad \left.\begin{matrix} r = R_0+\delta \\ \theta = \theta^* \end{matrix}\right\} P = P_0 \tag{7}$$

and

$$\text{at} \quad \left.\begin{matrix} R_0 \leqslant r \leqslant R_0+\delta \\ \theta = 0 \end{matrix}\right\} v_\theta = 0. \tag{8}$$

Furthermore, it is assumed that δ is constant for $0 \leqslant \theta \leqslant \theta^*$. As pointed out in [5], this simplification has little effect on the values of the average heat transfer coefficients. Since δ and v_r are unknown at this point, two additional mathematical constraints have to be specified. These constraints can be derived from an energy and a pressure balance.

The energy conducted and radiated to the liquid–vapor interface is equal to the energy conducted from the interface into the liquid and the energy used to evaporate the liquid. Thus, for $r = R_0+\delta$ and $0 \leqslant \theta \leqslant \theta^*$

$$-\left(k\frac{\partial T}{\partial r}\right)_l - \rho h_{fg} v_r = -k\frac{\partial T}{\partial r} + q_{\text{rad}}. \tag{9}$$

Introducing a modified latent heat,

$$h = h_{fg} + a.c_p(T_w - T_s),$$

to account for superheat and vapor inertia forces and assuming the emissivity ε_l of the liquid to be essentially unity, equation (9) yields

$$-\rho v_r(h_{fg} + c_{pl}[T_s - T_l + ac_p(T_w - T_s)])_{r=R_0+\delta} = -\left(k\frac{\partial T}{\partial r}\right)_{r=R_0+\delta} + \varepsilon_w\sigma_r\frac{R_0^2}{(R_0+\delta)^2}(T_w^4 - T_s^4) \tag{10}$$

where a is a number of the order 0·5.

As pointed out in [2], a static equilibrium requires that the average pressure $\bar{P}$ in the vapor at the liquid interface resulting from the weight of the supported liquid, surface tension, and atmospheric pressure balance the average pressure due to flow. This condition written as a force balance over the entire area from $\theta = 0$ to $\theta = \theta^*$ gives

$$\int_0^{2\pi}\int_0^{\theta^*} P_{(R_0+\delta)}(R_0+\delta)^2\sin\theta\,d\theta\,d\phi = \int_0^{2\pi}\int_0^{\theta^*}\bar{P}(R_0+\delta)^2\sin\delta\,d\theta\,d\Phi. \tag{11}$$

The total pressure P in the vapor gap is found from the solution of the momentum equation; $\bar{P}$ is calculated from

$$\bar{P} = \frac{P\,dA}{\int dA} \tag{12}$$

where

$$P = P_0 + (R_0+\delta)(\rho_l - \rho)(\cos\theta - \cos\theta^*)g. \tag{13}$$

In equation (13), P_0 is the atmospheric pressure, $(R_0+\delta)(\rho_l-\rho)*(\cos - \cos\theta^*)g$ represents the liquid head, the influence of the surface tension has been neglected, since it is small compared with the liquid head. At $\theta = \theta^*$, equations (13) reduces to $P = P_0$.

A lengthy solution of equations (1)–(4) with boundary conditions (5)–(7), and (8), (10), and (11), which follows essentially the approach used by Hendricks and Baumeister, leads to

$$Nu = \frac{h}{k}R_0 = 1+\frac{R_0}{\delta}+\frac{R_0}{k}\varepsilon_w\sigma_r\frac{(T_w^4-T_s^4)}{T_w-T_s}. \quad (14)$$

The gap thickness δ is found for non-slip conditions from

$$\left(\frac{\delta}{R_0}\right)^4 = -\left(\frac{2\varepsilon_w\sigma_r(T_w^4-T_s^4)}{\rho\mu\lambda c}+\frac{2k(T_w-T_s)}{R_0\rho\mu\lambda c}\right)\frac{\delta}{R_0} - \frac{2k(T_w-T_s)}{R_0\rho\mu\lambda c} \quad (15)$$

and for slip conditions from

$$\left(\frac{\delta}{R_0}\right)^4 = \left[-\left(\frac{2\varepsilon_w\sigma_r(T_w^4-T_s^4)}{\rho u\lambda c}+\frac{2k(T_w-T_s)}{R_0\rho u\lambda c}\right)\frac{\delta}{R_0} - \frac{2k(T_w-T_s)}{R_0\rho u\lambda c}\right]\frac{1}{(1+3\phi)} \quad (16)$$

where

$$C = \frac{\dfrac{1-\cos\theta^*}{u^2}\left[\tfrac{1}{2}R_0\phi(\rho_l-\rho)g(1-\cos\theta^*)\right]}{12\left[\ln\cos^2\dfrac{\theta^*}{2}+\sin^2\dfrac{\theta^*}{2}\right]} \quad (17)$$

and

$$\lambda = [h_{fg}+c_{pl}(T_s-T_l)+ac_p(T_w-T_s)] \quad (18)$$

$$u = \frac{v}{R_0}$$

$$\phi = \frac{R_0+\delta}{R_0}.$$

It should be noted that the heat-transfer coefficients defined by equation (14) represent heat transfer by conduction, convection, and radiation. The vapor flowrate is determined from

$$F = 2\pi(1-\cos\theta^*)R^2h(T_w-T_s)/\lambda. \quad (19)$$

RESULTS AND CONCLUSIONS

Experiments were carried out with an 18·75 mm (3/4 in) dia. inconel 600 sphere. Flowrates were always adjusted so that the sphere was half submerged ($\theta^* = \pi/2$). The difference between sphere temperature and saturation temperature $(T_w - T_s)$ was varied between 300 and 900°C. A comparison between predicted and experimentally obtained Nusselt numbers is shown in Fig. 3. The solid lines represent the Nusselt numbers as they are calculated with equation (14) for non-slip and slip conditions. Data points for the experimental values of the Nusselt number were obtained from the measured flowrates and temperature differences with help of equation (19). Errors due to heat losses to ambient conditions and due to radiation heat transfer from the upper part of the sphere to the water level were estimated to be in the order of 2 per cent.

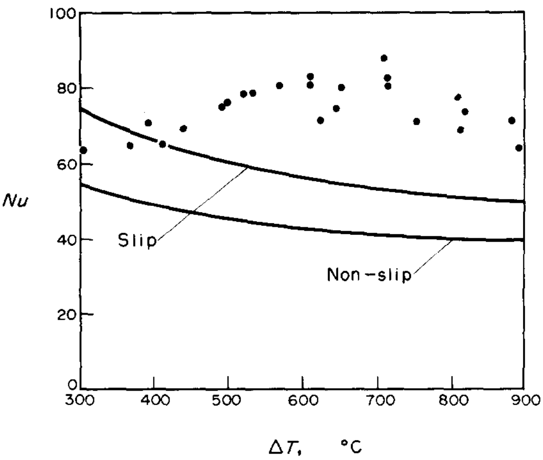

FIG. 3. Nusselt numbers vs temperature difference $(T_w - T_s)$.

At small temperature differences in the order of 300–400°C, the experimental Nusselt numbers are between those predicted for slip and non-slip conditions as one might expect. At higher temperature differences, the experimental Nusselt numbers are larger than the predicted Nusselt numbers. From visual observations it appears that with increasing temperature differences the vapor–liquid interface is more and more disturbed. Consequently, the assumption of laminar vapor flow is increasingly violated. At temperature differences above 600°C, large, almost spherical vapor cavities are formed at the vapor–liquid interface. These cavities move constantly. In addition, bubbles are formed within the liquid near the liquid–vapor interface. Although these phenomena violate the basic assumptions of the analytical model, equation (14), along with equations (15) and (16), allows a conservative prediction of film boiling heat transfer from a half submerged sphere.

Acknowledgement—Acknowledgement is made to the Research Committee and the Computer Center of the University of California, Santa Barbara, for support of this research in the form of computer time.

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EBULLITION EN FILM SUR UNE SPHERE PARTIELLEMENT SUBMERGEE

Résumé—Une étude expérimentale de l'ébullition en film sur une sphère partiellement submergée est effectuée. Les résultats obtenus expérimentalement sont comparés aux prévisions théoriques. Pour des différences de température élevées, les nombres de Nusselt expérimentaux sont plus grands que les nombres de Nusselt calculés.

FILMSIEDEN AN EINER TEILWEISE EINGETAUCHTEN KUGEL

Zusammenfassung—Die bei einer experimentellen Untersuchung des Filmsiedens an einer teilweise eingetauchten Kugel gewonnenen Ergebnisse werden mit den theoretisch vorausgesagten Werten verglichen. Bei hohen Temperaturdifferenzen sind die experimentell ermittelten Nusselt-Zahlen größer als die vorausgesagten.

ПЛЕНОЧНОЕ КИПЕНИЕ НА ЧАСТИЧНО ПОГРУЖЕННОЙ СФЕРЕ

Аннотация — Проведено экспериментальное исследование пленочного кипения на частично погруженной сфере. Экспериментальные результаты сравнивались с теоретически рассчитанными данными. Найдено, что при больших температурных перепадах экспериментальные числа Нуссельта больше расчетных.