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PARAMETRIC STUDY OF BOILING HEAT TRANSFER IN POROUS MEDIA

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ABSTRACT

Detailed numerical modeling and parametric variation studies were conducted on boiling heat transfer processes in porous deposits with emphasis on applications associated with light water nuclear power reactor systems. The processes of boiling heat transfer in the porous corrosion deposits typically involve phase changes in finite volumetric regions in the porous media. The study examined such processes in two porous media configurations, without chimneys (homogeneous porous structures) and with chimneys (heterogeneous porous structures).

A 1-D model and a 2-D model were developed to simulate two-phase flows with phase changes, without dry-out, inside the porous media for both structural configurations. For closure of the governing equations, an empirical correlation of the evaporation rate for phase changes inside the porous media was introduced. In addition, numerical algorithms were developed to solve the coupled nonlinear equations of mass, momentum, energy, capillary pressure, and evaporation rate.

The distributions of temperature, thermodynamic saturation, liquid pressure, vapor pressure, liquid velocity, and vapor velocity were predicted. Furthermore, the effects of heat flux, system pressure, porosity, particle diameter, chimney population density, chimney radius, and crud thickness on the all superheat, critical heat flux, and minimum saturation were examined. The predictions were found to be in good agreement with the available experimental results.

The principle conclusions drawn from the study were:

1. Optimum porosity, particle diameter, chimney population density, and the chimney radius exist to yield the lowest wall superheat.
2. The critical heat flux (CHF) of a porous coating surface with/without chimneys increases with increasing porosity, particle diameter, chimney population density and chimney radius within a broad range of parameter variation.
3. The wall superheat of a homogeneous porous coating surface is, in general, higher than that of a surface covered by porous media with chimneys (such as naturally formed corrosion deposits on boiling surfaces) except for the 2-D systems with high chimney population density and/or with larger chimney radius.
4. The wall superheat of a homogeneously coated surface is lower than that of a clean surface except for systems with high porosity and/or with thick porous layers.
5. The wall superheat of a surface covered by porous media with chimneys is lower than that of a clean surface except for the 2-D systems with large chimney population density, chimney radius, porosity, and/or thickness.
6. The CHF of a homogeneously coated surface is lower than that of a surface covered by a porous media with chimneys and higher than that of a clean surface, except for thick porous coatings.
7. Due to the presence of the chimneys (with certain chimney population density and chimney radius), the reduction of the wall superheat may not be significant (i.e., can be less than 1 °C), but the CHF may be increased significantly. Thus, it is suggested that porous media with chimneys (with certain chimney population densities and chimney radii) give better thermal performance under boiling conditions than similarly structured homogeneous porous media without chimneys.
8. Over the range of porous layer parameters examined, an increase of system pressure causes the wall superheat of a porous coating surface with/without chimneys to decrease and the CHF of the porous media to increase.
9. Over the range of parameters examined, an increase of crud thickness causes the wall superheat of a porous coating surface with/without chimneys to increase and the CHF of the porous media to decrease.

The paper includes selected results from these observations which pertain closely to LWR applications.

NOMENCLATURE

C_e = evaporation rate coefficient (Eq. 6)

C_{pl}, C_{pv} = liquid, vapor specific heat

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d_p = particle diameter

g = gravitational acceleration

h_e = interfacial heat transfer coefficient

h_{fg} = latent heat

k = molecular permeability

k_r = molecular relative permeability

m = turbulent permeability

m_r = turbulent relative permeability

P_c = capillary pressure

P_l , P_v = liquid, vapor pressure

S = liquid saturation

T = temperature

V_l , V_v = liquid, vapor velocity vector

Greek

ε = porosity

μ_l , μ_v = liquid, vapor dynamic viscosity

ρ_l , ρ_v = liquid, vapor density

τ = evaporation rate

Subscripts

l = liquid

v = vapor

ρ = particle

INTRODUCTION

The topic of boiling heat transfer in porous media has drawn broad attention in many areas,such as post-accident thermal analyses of boiling in nuclear reactor core debris, high flux electronic cooling, geothermal reservoir performance prediction, thermally enhanced oil recovery, nuclear waste disposal, drying of paper and food products, porous insulation moisture transport, among others. Of particular interest to this study is the boiling process in the porous corrosion deposits on heat transfer surfaces in water-cooled nuclear power reactors.

The study of boiling heat transfer in porous corrosion deposits deals with two-phase flows with phase change in heterogeneous porous media. It also requires a basic understanding of boiling heat transfer in homogeneous porous media. Therefore, both topics involving boiling heat transfer in porous media were studied in this work. The first is boiling heat transfer in homogeneous porous media heated from one end and cooled at the other end, which considers one-dimensional two-phase flows with phase change in porous media. The second is the boiling heat transfer in the porous deposits, which involves two-dimensional two-phase flows with phase change in porous media.

The structure of corrosion deposits in boiling systems has been extensively investigated (Macbeth, et al., 1971; Kawaguchi, et al., 1983; Thomazet, et al., 1985; Solomon, et al., 1976; Rooth, et al., 1971). It is found that these corrosion deposits are porous material layers with densely populated chimneys as shown in Fig 1. The existence of chimneys in the porous deposit appears to result from the deposit forming while boiling occurs. Cooking rice in a rice cooker is a vivid animation of this deposit forming process. With evaporation near the heating surface, these chimneys serve as venting channels which expel vapor to the bulk coolant. From the literature cited at the beginning of this paragraph, the typical dimensional characteristics of a porous deposit and selected for evaluation in this study, are: 25 μm thick layer, 2.5 μm chimney diameter, 3×10^9 chimneys/m^2 , 0.6 porosity, and 0.5 μm particle diameter.

The boiling process in porous media differs significantly from that over a clean surface. First, for the majority of the porous material layers capillary effects are important whereas the gravitational effects are typically not important. Moreover, the effective phase change area in a porous coating is much larger than that of a clean surface since, without dry-out, the vapor is formed not only at the heating surface but also at internal surfaces of the porous media where the local wall temperature sufficiently exceeds the saturation temperature of the local liquid to cause vapor generation. Due to the availability of a larger phase change area boiling inside a porous medium can yield more effective heat transfer than that on a clean plane surface. Finally, a quasi-isothermal liquid and vapor two-phase zone adjacent to the heating surface exists in the porous layer, in the absence of dry-out (Bau, 1980; Su, 1981; Bau, et al., 1982; Udell, 1983; Udell, 1985). In this two-phase zone, the liquid is drawn to the heating surface under the combined action of capillary pressure and gravity, and the vapor generated moves to the cold end. Between this counter-current two-phase zone and the cooling surface where the temperature is less than the saturation temperature, there is a conduction dominated liquid zone. The length of the two-phase zone increases with increasing heat flux.

Theoretical studies of steady one-dimensional two-phase flows inside thick homogeneous porous media have been carried out by Bau and Torrance (1982) and Udell (1983, 1985). In their models, it was assumed that the two-phase zone was isothermal and the phase changes only occurred at the bounding surfaces of the two-phase zone. To the authors' knowledge, there is no reported theoretical study which considers phase changes inside the two-phase zone and predicts the wall superheat for pool boiling inside a homogeneous porous layer.

Theoretical modeling of chimney-type deposits is reported in the literature (Stratton, et al., 1969; MacBeth, 1971; Cohen, 1974; Eckert, et al., 1983; Pan, et al., 1985). However, these previous studies dealt with single phase flow in the porous layer and some of them assumed that evaporation could only occur at the chimney walls.

Figure 1. Structure of porous deposits (Thomazet *et al.*, 1985).

In addition, the effects of parameters, such as system pressure, thickness of the porous coating, layer porosity, particle diameter, chimney population density, and chimney diameter, on the critical heat flux (CHF) and wall superheat have been studied by several researchers. The results given by different researchers are, however, sometimes contradictory. Furthermore, the reported correlations of wall superheat and imposed heat flux vary significantly. To date, there is no commonly accepted universal correlation for pool boiling heat transfer inside homogeneous porous media.

In order to investigate the parametric effects, this study developed both a 1-D model and a 2-D model to describe two-phase flows with phase change inside the homogeneous porous media heated from one end and cooled at the other end, without and with the presence of chimneys. The models were validated quantitatively with available experimental results. The boiling curves were predicted and the effects of system pressure, porosity, particle diameter, chimney population density, chimney radius, and crud thickness on the wall superheat, CHF, and minimum saturation were examined. The predictions were found to be qualitatively in good agreement with the available experimental measurements.

MODEL DEVELOPMENT

The steady-state mass and linear momentum conservation equations for the continuous vapor and liquid phases in two-phase flows with phase changes inside porous media can be expressed as (Shi, 1994)

$$\nabla\cdot(\rho_l V_l) = -\tau(T) \qquad (1)$$

$$\nabla\cdot(\rho_v V_v) = \tau(T) \qquad (2)$$

$$\nabla P_l = -\left(\frac{\mu_l}{k k_{rl}} + \frac{\rho_l}{m m_{rl}}\left|V_l\right|\right)V_l + \rho_l g \qquad (3)$$

$$\nabla P_v = -\left(\frac{\mu_v}{k k_{rv}} + \frac{\rho_v}{m m_{rv}}\left|V_v\right|\right)V_v + \rho_v g \qquad (4)$$

where the molecular permeability k and turbulent permeability m are modeled by the Kozeny-Carman relation (Scheidegger, 1974); the molecular relative permeabilities, k_{rl} and k_{rv}, are evaluated using the well accepted correlation by Fatt and Kickoff (Fatt, et al., 1959); the turbulent relative permeabilities m_{rl} and m_{rv} are assumed to be equal to k_{rl} and k_{rv}, respectively; and the capillary pressure P_c, defined as the pressure difference between the vapor and liquid, is evaluated by Leverett's formula (1941). The quantities P_c, k_{rl}, k_{rv}, m_{rl}, and m_{rv} are all functions of the liquid saturation, S.

The energy equation for vapor and liquid two-phase flows inside porous media can be derived as (Shi, 1994)

$$\nabla\cdot\left(\rho_v c_{pv} T V_v + \rho_l c_{pl} T V_l - \lambda\nabla T\right) = 0 \qquad (5)$$

where λ is the effective thermal conductivity of the porous media and is expressed as $(1-\varepsilon)\lambda_p + S\,\varepsilon\lambda_l$.

By considering evaporation from the liquid film on the particle surface, the evaporation rate, $\tau(T)$, in the mass balance equations is modeled as[17]

$$\tau(T) = \frac{C_e \dfrac{6S(1-\varepsilon)}{d_p} h_e (T - T_{sat})}{h_{fg}} \qquad (6)$$

where h_e is the evaporation heat transfer coefficient from the kinetic theory of interfacial mass transfer (Pan, 1986), $6(1-\varepsilon)/d_p$ represents the total particle surface area per unit volume, and C_e is an empirical constant and was determined to be 0.002 (Shi, 1994) in this study.

With appropriate boundary conditions (Shi, 1994), the above coupled nonlinear equations were solved numerically for both 1-D homogeneous porous media and 2-D porous deposits with chimneys.

RESULTS AND DISCUSSION OF PARAMETRIC ANALYSES

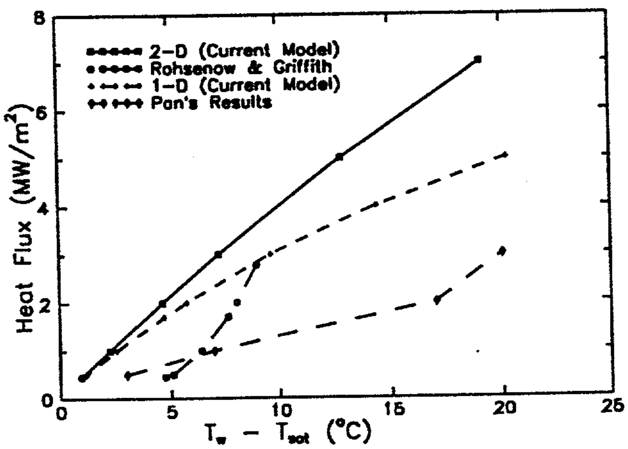
Our discussion here is focused on the results from parametric analyses. The quantitative validation of the modeling and the discussion of the predicted distributions of temperature, saturation, velocities, and pressure can be found elsewhere (Shi, 1994).

The system pressure and the heat flux imposed on the corrosive heating surface differ from one boiling system to another. In addition, characteristic parameters of the corrosion deposits, such as porosity, particle diameter, chimney population density, chimney radius, and crud thickness, vary over a wide range due to different formation processes, and these differences affect the boiling heat transfer. In this section, effects of these parameters on the wall superheat, defined as T_w-T_{sat}, and the critical heat flux (CHF) are examined. It should be mentioned that the following parametric analyses were conducted by changing only one parameter at a time.

In addition, the parametric study may provide general guidelines for the optimum design of electronic cooling systems where thin porous layers with/without chimneys are proposed to be coated on the heating surface to enhance heat removal under boiling conditions. However, the detailed study of boiling heat transfer in an electronic cooling system is beyond the scope of this presentation.

Effect of Heat Flux

The boiling curves predicted from the current models, under a system pressure of 155 bars, are presented in Fig. 2. The results from Rohsenow's correlation (Rosenow, 1952) for pool boiling over a clean surface and by Pan (1986) are also shown in Fig. 2 for comparison. It should be mentioned that the wall superheats presented from Pan's work (1986) are the maximum wall superheats predicted at the base surface of the plane of symmetric between chimney locations. Thus, the wall superheats cited from Pan in this section refer to these maximum values. However, the wall superheats from the current models represent the radially-averaged values over the area surrounding a chimney.



P_o = 155 bar; δ = 25 μm; ε = 0.6; d_p = 0.5 μm;
r_o = 2.5 μm; N_v = 3800 chimneys/mm^2

Figure 2. Boiling performance curves.

According to Zuber's correlation (Zuber, 1958), the critical heat flux for pool boiling over a clean surface under a system pressure of 1.55 10^7 Pa is 2.77 MW/m^2. Hence, the boiling curve for a clean surface, referred to Rohsenow in Fig.2, is plotted below this CHF point. The Rohsenow's correlation used in this study is given by

$$T_w - T_{sat} = c_{sf}\frac{h_{fg}}{c_{pl}}\left[\frac{q}{\mu_l h_{fg}}\left(\frac{\sigma}{g(\rho_l-\rho_v)}\right)^{1/2}\right]^{0.33}\left(\frac{c_{pl}\mu_l}{k_l}\right)^{1.7} \qquad (7)$$

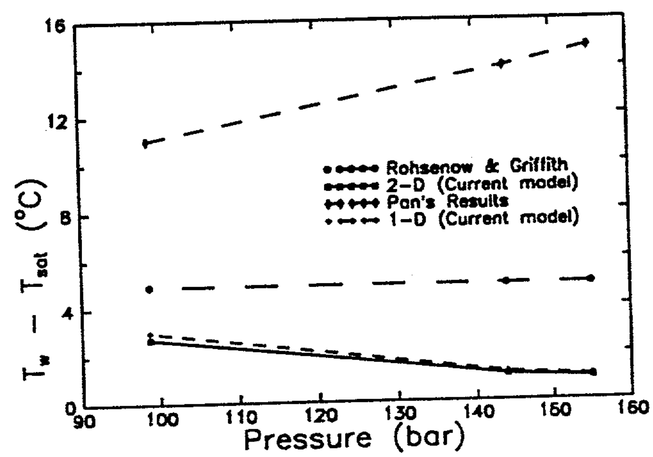
where the coefficient c_{sf} depends on surface-fluid combinations and a c_{sf} of 0.013 is used for this application.

Figure 2 shows that the wall superheat increases with increasing heat flux. For heat fluxes below the CHF point, the wall superheats from the current models are lower than those from Rohsenow's correlation. This suggests that surfaces coated with the selected porous media characteristics have better boiling heat transfer capability than clean surfaces. This suggestion is also supported by the higher critical heat flux (i.e., higher than 2.77 MW/m^2) predicted for the coated surface and observed in Fig. 2. In addition, the wall superheats from the 1-D model are higher than those from the 2-D model.

Effect of System Pressure

The effect of system pressure on wall superheat is illustrated in Fig. 3. The results from Pan's model with a higher heat flux and from Rohsenow's correlation with the same heat flux are also included in Fig. 3 for comparison.

Figure 3 shows that the wall superheat from Pan's model increases with increasing system pressure, whereas the wall superheat from the current models decreases with increasing system pressure. The latter trend is supported by both experimental evidences (Nakayama, et al., 1982; Czikk, et al., 1981) and an other model prediction (El-Wakil, 1978). This trend is also true for pool boiling over a clean surface, as reported by Rohsenow (1952).



Parameters in the 1-D and 2-D models and Rohsenow's model:
q = 0.45 MW/m^2; δ = 25 μm; ε = 0.6;
d_p = 0.5 μm; r_o = 2.5 μm; r_l = 10.3 μm; N_v = 3000 chimneys/mm^2
Parameters in Pan's model:
q = 1.7 MW/m^2; δ = 25 μm; = 0.6;
d_p = 0.5 μm; r_o = 2.5 μm; N_v = 2600 - 3500 chimneys/mm^2

Figure 3. Effect of system pressure on wall superheat.

Figure 3 also indicates that, under the same pressure, the predicted wall superheats from the 1-D model are higher than those from the 2-D model but lower than those from Rohsenow's correlation. This signifies that the wall superheat for the homogeneous porous coating system is higher than that for the porous coating system with chimneys but lower than that for a clean surface system.

A much lower heat flux than that in Pan's model is used in the current research. This is because the heat flux in Pan's study exceeds the CHF in the current model. Therefore, the comparison between the current models and Pan's model on the effect of system pressure is qualitative rather than quantitative.

In addition, both current 1-D and 2-D models suggest that the CHF, along with the minimum saturation, increases with increasing system pressure as shown in Fig. 4. This finding is consistent with the effect of system pressure on wall superheat. Both imply that under higher system pressure the porous coating surface has better thermal performance. However, this tendency shown in Fig. 4 is contrary to that predicted by Zuber (1958) for boiling over a clean surface and by Pan's (1986) and by MacBeth's (1971) models. Figure 4 also shows that the difference between the minimum saturations for the one-D and the two-D systems is reduced with increasing system pressure.

The trends from the current models, shown in Figs. 3 and 4, consistently suggest that a higher system pressure makes it easier for the liquid to access the heating surface and for the vapor to escape from the porous media. In other words, high system pressure enhances boiling heat transfer performance of porous layers.

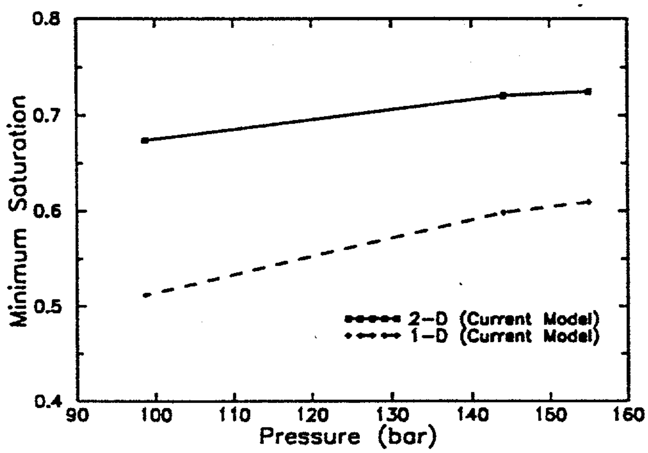


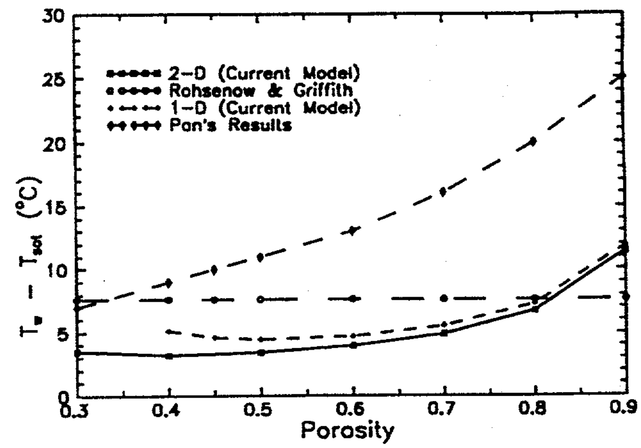
Figure 4. Effect of system pressure on minimum saturation.

Effect of Porosity

The wall superheats predicted by both 1-D and 2-D models are plotted against the porosity in Fig. 5. The results from Pan's two-dimensional model (1986) and from Rohsenow's correlation (1952) for boiling over a clean surface, under the same conditions, are also presented in Fig. 5 for comparison.

While Pan's results suggest that the wall superheat increases monotonically with increasing porosity for porosities of 0.3 to 0.9, both current 1-D and 2-D models show that the wall superheat decreases with increasing porosity at first and then increases. The transition porosity, where the wall superheat reaches the lowest value, in the 2-D case is about 0.4 which is smaller than that of about 0.5 in the 1-D homogeneous case.

The experimental investigations by Zhao and Zhang (1988) revealed that, at low heat flux, porous layers with low porosities yield better boiling heat transfer performance, whereas, at high heat flux, porous layers with high porosities do better. Physically, this could be rationalized because the high heat flux requires a high rate of liquid evaporation. But phase change in layers with small pores is limited by insufficient liquid supply. On the other hand, layers with large pores make it more convenient for large amounts of water to penetrate so that the water loss from evaporation can be compensated for in time, which supports a high phase change rate. This agreement and the trends in Fig. 5 suggest that there is an optimum porosity for the lowest wall superheat at a given heat flux for fixed pore size.



P_o = 155 bar; q = 1.7 MW/m^2; δ = 25 μm; ε = 0.6;
d_p = 0.5 μm; r_o = 2.5 μm; r_l = 10.3 μm; N_v = 3000 chimneys/mm^2

Figure 5. Effect of porosity on wall superheats.

Considering that the predicted CHF of the 2-D system is larger than that of the 1-D system, the ratio of the imposed heat flux to the critical heat flux for the 2-D system is lower than that for the 1-D system. Thus, according to Zhao and Zhang's experimental results, the optimum porosity for the 2-D system must be lower than that for the 1-D system. Consistently, the current results from Fig. 5 show that the optimum porosity, which yields the lowest wall superheat, for the 2-D system is about 0.4 and for the 1-D system is about 0.5, which is higher than that for the 2-D system as expected.

The existence of an optimum porosity can also be explained by two competing effects associated with increasing porosity. First, an increase of porosity leads to smaller internal phase change surfaces per unit volume, which tends to cause an increase of the wall superheats. On the other hand, with increasing porosity, the interconnecting void space increases. As a result, both liquid and vapor move more easily, which causes a decrease in the wall superheats. Hence, in the 1-D case, for porosities larger than 0.5, the first effect is dominant, whereas, with smaller porosities, the latter effect is dominant. Similarly, in the 2-D cases, the first effect is significant for porosities larger than 0.4, whereas the latter effect becomes dominant with smaller porosities.

Figure 5 also shows that the wall superheats from the current models are lower than those from Rohsenow's correlation for boiling over a clean surface except for loosely packed porous coatings (i.e. porosity > 0.8). The wall superheats in the 1-D case are slightly higher than those in the 2-D case.

The minimum saturation level increases with increasing porosity in both 1-D and 2-D model predictions as shown in Fig. 6. This can be understood because the larger porosity provides more interconnecting void spaces which allow liquid to flow more easily towards the heating surface and the vapor to escape more easily from the porous media. The trend in Fig. 6 also indicates that the CHF must increase with increasing porosity. In addition, the increase of porosity also narrows the difference between the minimum saturations predicted by the current two models.

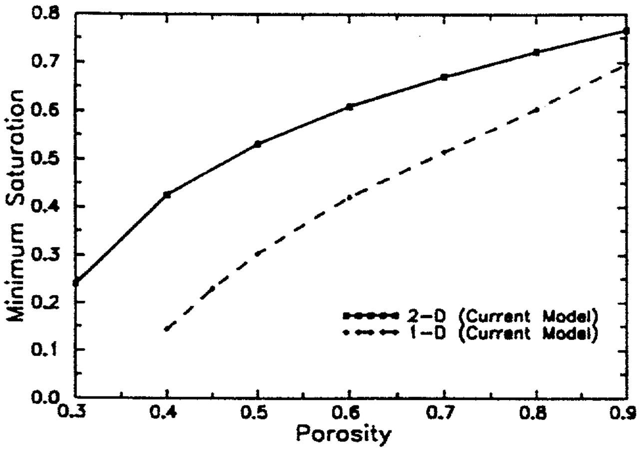


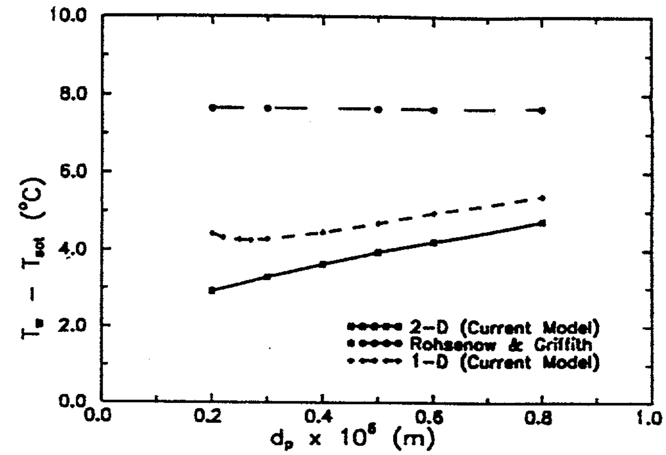
Figure 6. Effect of porosity on minimum saturation.

Effect of Particle Diameter

Figure 7 shows the relationship of the wall superheat and the particle diameter (i.e. characteristic length scale of the porous media) for the current two models. The wall superheat predictions by Rohsenow (1952) for boiling over a clean surface under the same condition are also presented in Fig. 7.

Different from the trend in the 2-D case, where the wall superheat increases monotonically with the particle diameter, the wall superheat of the homogeneous 1-D system decreases as the particle diameter increases from 0.2 to 0.3 μm. However, further increasing of the particle diameter results in an increase of the wall superheat. Thus, the optimum particle diameter for the 1-D system is about 0.3 μm. According to the argument in the effects of porosity from Zhao & Zhang's experiments (1988), the optimum particle diameter for the 2-D system should be lower than 0.3 μm. However, the heat flux under the current consideration is higher than the CHF for the 2-D systems with a particle diameter lower than 0.2 μm. Hence, the lower limit in particle diameter in this study is 0.2 μm. Although in the 2-D system no decrease in wall superheat is observed with increasing particle diameter, the 2-D results are not contrary to Zhao and Zhang's findings. The optimum particle diameter for the 2-D system is 0.2 μm. The experiments by Fujii (1984) at atmosphere pressure also showed an effect of particle diameter on the wall superheat similar to that from the current 1-D model. In addition, the existence of an optimum particle diameter can also be explained by two opposing effects caused by increasing particle diameter in a similar way to the explanation of porosity. It can be deduced from Fig. 7 that, with a sufficiently large particle diameter, the wall superheat of the porous coating surface might be higher than that of a clean surface under pool boiling conditions.

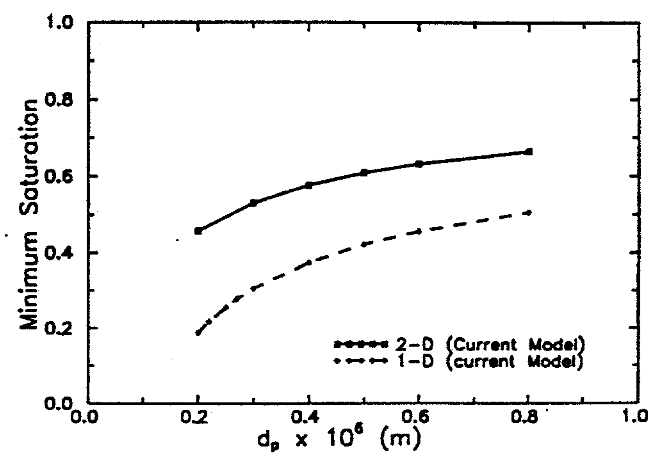
As shown in Fig. 7 and within the range of this study, the wall superheats for boiling over porously coated surfaces with/without chimneys are lower than those over a clean surface. Also, the wall superheats of porously coated surfaces with chimneys are lower than those of homogeneously coated surfaces.



$P_o = 155$ bar; $q = 1.7$ MW/m^2; $\delta = 25$ μm; $\varepsilon = 0.6$;
$r_o = 2.5$ μm; $N_v = 3000$ chimneys/mm^2

Figure 7. Effect of particle diameter on wall superheats.

The minimum saturation is plotted against particle diameter in Fig. 8. Similar to the effect of porosity on wall superheat, the minimum saturation obtained by both 1-D and 2-D models increases monotonically with increasing particle diameter, which indicates that the CHF increases with increasing particle diameter. It is also found that the difference between the two predictions and the gradient of the wall superheat become smaller with increasing particle diameter.

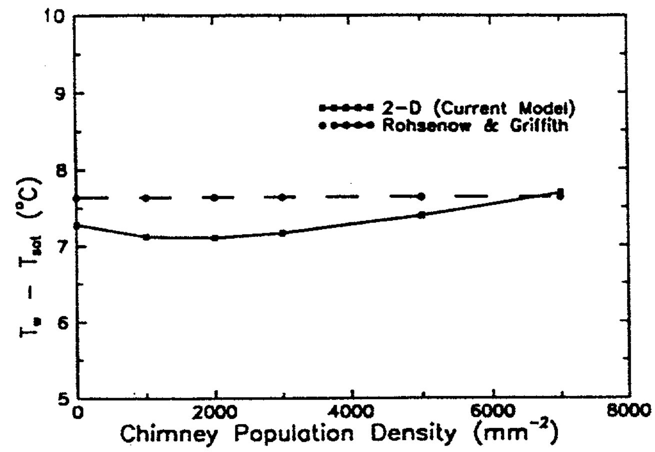


$P_o = 155$ bar; $q = 1.7$ MW/m^2; $\delta = 25$ μm; $\varepsilon = 0.6$;
$r_o = 2.5$ μm; $N_v = 3000$ chimneys/mm^2

Figure 8. Effect of particle diameter on minimum saturation

Effect of Chimney Population Density

The predicted relations of wall superheat versus the chimney population density are presented in Fig. 9. Different from Pan's results (1986) where the wall superheat decreases monotonically with increasing chimney population density, the current model suggests that there is an optimum chimney density (about 2×10^9 chimneys/m^2) with which the lowest wall superheat occurs for the selected porous layer characteristics.



P_o = 155 bar; q = 1.7 MW/m^2; δ = 25 μm; ε = 0.8;
d_p = 0.5 μm; r_o = 2.5 μm

Figure 9. Effect of chimney population density on wall superheat.

Physically, increasing chimney population has two conflicting effects. First, since the chimney can provide an easier passage for vapor to escape from the porous media, the efficiency of heat removal from the heating surface increases with increasing chimney population density, which causes a decrease in wall temperature. On the other hand, the effective heat flux imposed to the porous region excluding the chimney bottoms increases with increasing chimney population, which results in an increase in the wall temperature. As shown in Fig. 9, for chimney population densities less than the optimum value, the latter effect is dominant and causes an increase of the wall superheat with decreasing of chimney population density. In contrast, for chimney population densities greater than the optimum value, the second effect is dominant and gives rise to an increase in wall superheat with increasing chimney population density.

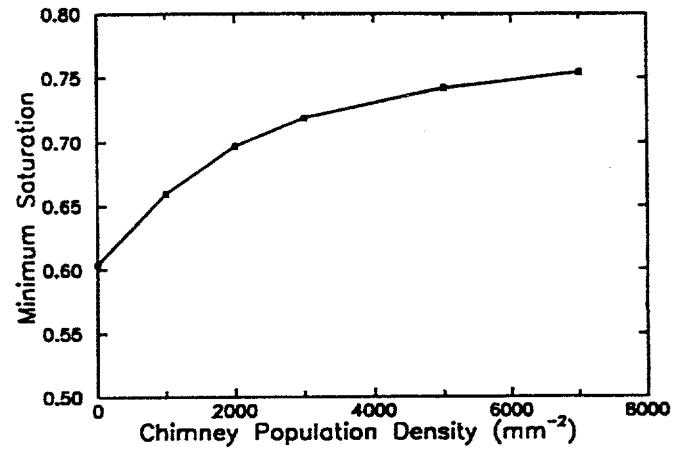
It is also found for those specific conditions, with a chimney density higher than about 4×10^9 chimneys/m^2, that the wall superheat becomes higher than that of a homogeneous porous coating surface without chimneys, and with a chimney density higher than 7×10^9 chimneys/m^2, the wall superheat exceeds that (from Rohsenow's correlation) of a clean surface.

The minimum liquid saturation, at the base of the symmetric surface between two nearby chimneys, is found to increase monotonically with increasing chimney population density as shown in Fig. 10. This indicates that the CHF increases with increasing chimney population density.

Effect of Chimney Radius

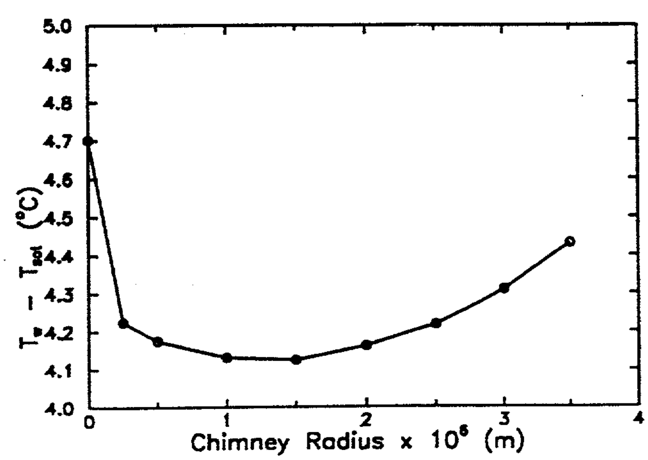
The correlation between the wall superheat and the chimney radius is illustrated in Fig. 11. The figure shows that there is an optimum chimney radius, about 1.5 μm, for which the lowest wall superheat is predicted.

Similar to the effect of the chimney population density on the wall superheats, an increase of the chimney radius can also lead to two opposing effects. The larger chimney radius may provide an easier channel for vapor to escape from the porous media, which gives a reduction of the wall superheat. On the other hand, the increase of the chimney radius enhances the effective heat flux on the porous layer of each region associated with a chimney, which causes an increase of the wall superheat. This may explain the trend shown in Fig. 11.



P_o = 155 bar; q = 1.7 MW/m^2; δ = 25 μm; ε = 0.8;
d_p = 0.5 μm; r_o = 2.5 μm

Figure 10. Effect of chimney population density on minimum saturation.

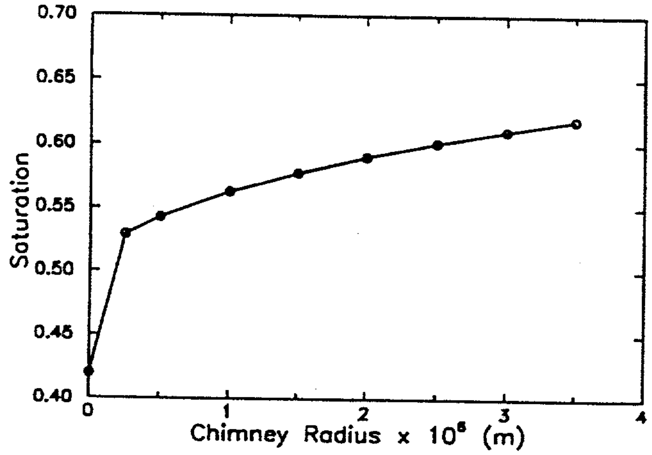


P_o = 155 bar; q = 1.7 MW/m2; δ = 25 μm; ε = 0.6;
d_p = 0.5 μm; N_v = 3000 chimneys/mm^2

Figure 11. Effect of chimney radius on wall superheat.

In addition, Figure 11 may suggest that the wall superheat of a 2-D system with a sufficiently large chimney radius can be higher than that (about 4.7 °C) of a 1-D system without chimneys. With an even larger chimney radius, the wall superheat of the 2-D system can exceed that (about 7.7 °C) of a clean surface under boiling conditions.

Minimum saturations are also plotted against the chimney radius as shown in Fig. 12. It is found that the minimum saturation increases monotonically with increasing chimney radius. This finding implies that the CHF increases with increasing chimney radius.



P_o = 155 bar; q = 1.7 MW/m^2; δ = 25 μm; ε = 0.6;
d_p = 0.5 μm; N_v = 3000 chimneys/mm^2

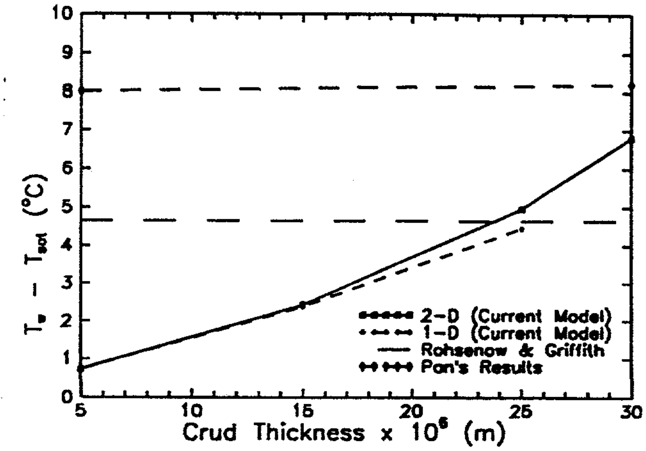
Figure 12. Effect of chimney radius on minimum saturation.

Effect of Crud Thickness

The effect of crud thickness on the wall superheat is illustrated in Fig. 13 for the test case. It shows that the wall superheat increases with increasing crud thickness. This tendency agrees with Pan's predictions and is also supported by experimental observations (Lu, et al., 1987; Liu, et al., 1985). When the crud thickness is less than about 25 μm, the wall superheat for boiling over a surface with porous media is smaller than that for boiling over a clean surface. However, when the crud thickness is larger than about 25 μm, the wall superheat for boiling over a coated surface is higher than that for boiling over a clean surface. In Fig. 13, the heat flux and the chimney population density are smaller than those in Pan's model because the use of the same conditions as in Pan's model would give rise to dry-out conditions within the current range of crud thicknesses for the 1-D and 2-D models.

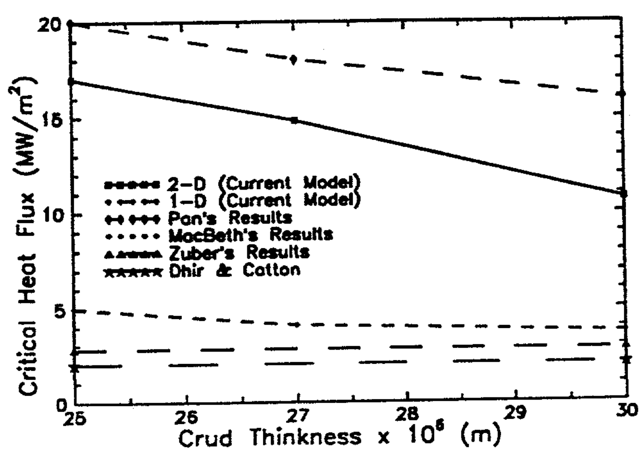
It is worth noting that the wall superheats predicted by the current models are much more sensitive to crud thickness than those predicted by Pan. This is believed to be due to the heat flux in the current model being very close to the predicted critical heat fluxes of surfaces with crud thicknesses higher than 25 μm. Also of interest is that, from Fig. 13, the wall superheats of the 1-D system without chimneys are slightly lower than those of the 2-D system with chimneys. According to conclusions from the above two sections, this negative effect of the chimneys on the wall superheats implies that the chimney density and chimney radius may be too large to enhance the boiling heat transfer under the investigated conditions. With smaller chimney diameters or chimney densities, the results from the 2-D model showed that the wall superheat is reduced.

Figure 14 shows the effect of the crud thickness on CHF. The CHF predicted by the current models is lower than those from Pan's and MacBeth's models but higher than those by either Zuber's (1958) or Dhir and Catton's (1977) correlations. The CHF decreases with increasing crud thickness, which agrees with others' predictions [21,27] and is confirmed by experimental findings[30,31]. Figure 14 also shows that the CHF for the 1-D system is lower than that for the 2-D system even, though the wall superheat for the 1-D system is predicted to be lower than that of the 2-D system shown in Fig. 13. It should be noted that Zuber's correlation is for boiling over a clean surface whereas Dhir & Catton 's correlation[32] is for boiling in inductively heated particulate beds.



Parameters in Pan's model:
P_o= 69 bar; q = 1.5 MW/m^2; ε= 0.7;
d_p = 0.5 μm; r_o = 2.5 μm; N_v = 5000 chimneys/mm^2
Parameters in other's models:
P_o= 69 bar; q = 0.3 MW/m^2; ε = 0.6;
d_p = 0.5 μm; r_o = 2.5 μm; N_v = 2000 chimneys/mm^2

Figure 13. Effect of crud thickness on wall superheat.



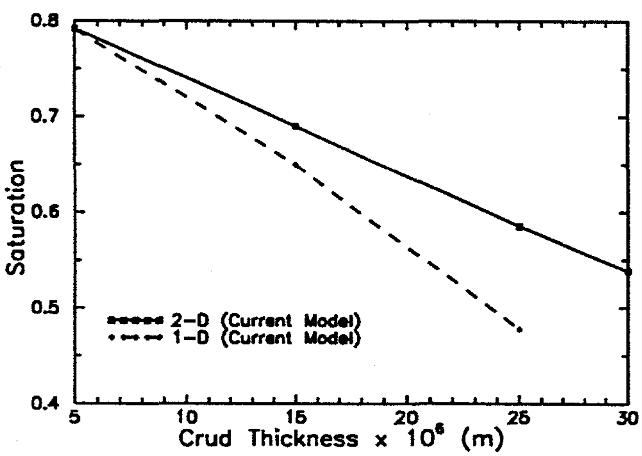
P_o= 155 bar; = 0.6; d_p = 0.5 μm;
r_o = 2.5 μm; N_v = 5000 chimneys/mm^2

Figure 14. Effect of crud thickness on CHF.

Also of interest is the comparison of the current results with the findings of Bau and Torrance (1982) and with Udell (1985). Their studies suggested that the two-phase zone length in their two-zone or three-zone region decreases with increasing heat flux. This also agrees with the trends shown in Fig. 14 through equating the crud thickness to the two-phase zone length and equating the critical heat flux to their heat flux.

The effect of crud thickness on the minimum saturation shown in Fig. 15 also suggests that the critical heat flux decreases with increasing crud thickness since the minimum saturation predicted increases as the crud thickness increases. It is worth noting that minimum saturations predicted from the 1-D model are lower than those predicted from the 2-D model despite the wall superheats from the 2-D model being slightly higher than those from the 1-D model. This is consistent with the CHF predictions. In addition, it can be seen for conditions of increasing crud thickness that the minimum saturation in the 1-D case becomes much more lower than that in the 2-D case.

These many observed trends and sensitivities of the thermal performance of the 1-D and 2-D models of thin porous layers to changes in layer characteristics have been presented. Good consistancy of results of the models is obtained with those of other models and with experimental observations. The results are also consistant with physical and intuitive understanding, providing further insights into the parametric influences.



$P_o = 69$ bar, $q = 0.3$ MW/m^2; $\varepsilon = 0.6$;
$d_p = 0.5$ μm; $r_o = 2.5$ μm; $N_v = 2000$ chimneys/mm^2

Figure 15. Effect of crud thickness on minimum saturation.

SUMMARY AND CONCLUSIONS

A study to more insightfully assess the parametric effects on boiling heat transfer in porous layers was conducted. Models without and with chimneys, homogeneous and heterogeneous porous layers, respectively, were developed. Ample use of published information on both modeling and experimental observations and data was made in developing the models and in assessing the validity of the model's predictions.

In the current parametric study, effects of heat flux, system pressure, porosity, particle diameter, chimney population density, chimney radius, and crud thickness on the wall superheat, CHF, and minimum saturation were rendered. In general, model predictions are in good agreement with the available experimental results.

Several conclusions can be drawn from this study:

1. Optimum porosity, particle diameter, chimney population density, and chimney radius exist to yield the lowest wall superheat and, thus, best thermal performance.
2. The CHF of a porous coating surface with/without chimneys increases with increasing porosity, particle diameter, chimney population density and chimney radius, within a broad range of parameter variation.
3. The wall superheat of a homogeneous porous coating surface is, in general, higher than that of a surface covered by porous media with chimneys (such as naturally formed corrosion deposits in boiling systems) except for the 2-D systems with high chimney population density and/or with larger chimney radius.
4. The wall superheat of a homogeneous coating surface is lower than that of a clean surface except for systems with high porosity and/or with thick porous layers.
5. The wall superheat of a surface covered by porous media with chimneys is lower than that of a clean surface except for the 2-D systems with large chimney population density, chimney radius, porosity, and/or thickness.
6. The CHF of a homogeneous coating surface is lower than that of a surface covered by porous media with chimneys (such as corrosion products) and higher than that of a clean surface, except for thick porous coatings.
7. Due to the presence of chimneys (with certain chimney population density and chimney radius), the reduction of the wall superheat may not be significant (i.e., can be less than 1 °C), but the CHF may be increased significantly. Thus, it is suggested that porous media with chimneys (with certain chimney population density and chimney radius) give better thermal performance under boiling conditions than homogeneous porous media without chimneys.
8. Over the range of parameters examined, with an increase of system pressure, the wall superheat of a porous coating surface with/without chimneys decreases and the CHF of the porous media increases.
9. Over the range of parameters examined, with an increase of crud thickness, the wall superheat of a porous coating surface with/without chimneys increases and the CHF of the porous media decreases.

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