Analysis of film condensation within inclined thin porous-layer coated surfaces

K. J. Renken and M. Aboye
Department of Mechanical Engineering, University of Wisconsin–Milwaukee, Milwaukee, WI, USA

Film-condensation enhancement by a porous/fluid composite system is investigated numerically. The numerical simulations detail laminar film condensation from an inclined thin porous-layer coated plate that is immersed in pure saturated steam at atmospheric pressure. A mathematical model is formulated, and the numerical results are compared with Nusselt's theory and preliminary experimental data. The dependence of the average heat transfer coefficient (represented by the Nusselt number and the average heat flux) on the surface subcooling, the gravity field, and the thin porous-layer coating characteristics (thickness, effective thermal conductivity model, porous structure, etc.) is documented.

Keywords: porous coating; film condensation; inclined surface; numerical model

Introduction

The process of film condensation in a porous medium has been receiving increased attention in the phase-change area of the porous-media literature. Recent interest in enhanced condensation by thermal engineering industries has sparked the demand for in-depth understanding of this phenomenon. Condensation within and over thin porous layers is of great interest in heat pipes where porous wicks are used to transport the condensate from the condensing section to the evaporator and in surface modifications for the purpose of heat transfer enhancement. Some specific examples of applications that solicit higher efficiencies and more compact sizes are thermal insulations, direct-contact heat exchangers, condensers, dehumidifiers, cooling electronic devices, etc. A general review of published works on condensation in porous media is provided by Kaviany (1991) and Nield and Bejan (1992).

The majority of studies using thin porous-layer coated surfaces to augment thermal communication have been devoted to evaporation. Relevant theoretical and experimental studies that have addressed very thin porous-layer boiling include those of Afgan *et al.* (1985), Avedisian and Koplik (1982), Avedisian *et al.* (1984), Bergles and Chyu (1982), Cheng and Verma (1981), Ito *et al.* (1982), Kajikawa *et al.* (1983), Konev and Mitrovic (1986), Konev *et al.* (1987), Kovalev *et al.* (1987), Marto and Lepere (1982), Nakayama *et al.* (1980a, 1980b, 1982), Styrikovich *et al.* (1987), and Webb (1981, 1983).

Two common conclusions drawn by all these authors are that the employment of a porous surface structure (1) produces high heat transfer coefficients with small temperature differences when compared to uncoated surfaces and (2) is therefore a very attractive and viable heat transfer enhancement alternative.

Address reprint requests to Professor Renken at the Department of Mechanical Engineering, University of Wisconsin-Milwaukee, P.O. Box 784, Milwaukee, WI 53201, USA.

Received 25 January 1992; accepted 9 June 1992

While a number of studies have focused on the evaporation issue, very little attention has been paid to the problem of condensation in a thin porous-layer coated surface; a remarkably few published results pertaining to this problem exist. Some of the more interesting related investigations include the works of Woodruff and Westwater (1979, 1981), who discovered that gold surfaces promoted dropwise condensation (DWC) of steam at atmospheric pressure; Carnavos (1980), who tested augmented tubes for overall R-11 condensing performance; Rifert *et al.* (1984), Marto (1986), and Marto *et al.* (1987), who reported that wire-finned tubes enhanced the heat transfer coefficients in condensation by approximately 60 to 100 percent as compared with smooth tubes; Yau *et al.* (1984), who studied the effects of drainage strips for horizontal finned condenser tubes; Shekarriz and Plumb (1986), who theoretically demonstrated that capillary porous fins can have a significant effect on filmwise condensation rates; and others, who have used organic coatings (Holden *et al.*, 1987) and scratched rough surfaces (Izumi *et al.*, 1989) to promote dropwise and filmwise condensation. More relevant to the present study are the works of Renken *et al.* (1989) and Renken and Aboye (1992), who presented results showing that a conductive porous coating may yield a considerable heat transfer enhancement during laminar film condensation.

The present investigation summarizes the subsequent results of a numerical study of condensation enhancement by a porous/fluid composite system. The composite system is composed of a relatively thin, highly conductive and permeable porous coating that is attached to an inclined impermeable surface. The analysis is based on the usual approximations for thin film condensation and on the Darcy–Brinkman model, which is used to describe the flow and heat transfer processes in the porous region. Matching conditions are utilized at the interface of the porous/fluid layer in the solution of the condensation velocity and temperature distributions. A comparison of the results of the present study with Nusselt's correlation (Nusselt 1916) and to experimental data (Renken and Aboye 1992) demonstrates the possibilities that exist for condensation enhancement with the employment of a thin porous metallic-layer coated surface system.

Analysis

Mathematical formulation

Figure 1 shows a schematic of the problem of interest, namely, film condensation within a thin porous-layer coated surface. The coordinate system is also defined in Figure 1, where x and y are Cartesian coordinates in the horizontal and vertical directions, respectively, with the positive y-axis pointing toward the porous medium and the positive x-axis pointing along the length of the plate. The origin of the coordinate system is at the top end of the plate. In more detail, we have a condensing surface of length L with a constant cold temperature of T_w that is inclined with the vertical axis by an angle of ϕ. The impermeable surface is coated with a very thermally conductive (metallic) and permeable porous substrate of thickness H, permeability K, and porosity ε. The mass flow rate of the condensing liquid, which flows through the porous/fluid composite by gravitational forces, is expressed as w, and the film condensation thickness is represented by $\delta(x)$. The surrounding environment is at atmospheric pressure and contains pure saturated steam at temperature T_{sat}.

It is assumed that the flow is steady, laminar, incompressible, and two-dimensional (2-D). In addition, we assume that (1) a distinct boundary exists separating the liquid zone from the vapor zone, thus eliminating the difficulties associated with the use of relative permeability; (2) the boundary layer is so thin that all the usual approximations common in thin film analysis can be adopted; (3) the thermophysical properties of the condensate and the porous coating are evaluated at the film temperature $(T_{sat} + T_w)/2$; (4) the condensate-saturated porous substrate is considered homogeneous and isotropic and in local thermodynamic equilibrium with the fluid; and (5) the porous coating is thinner than the condensation film and therefore covered by it.

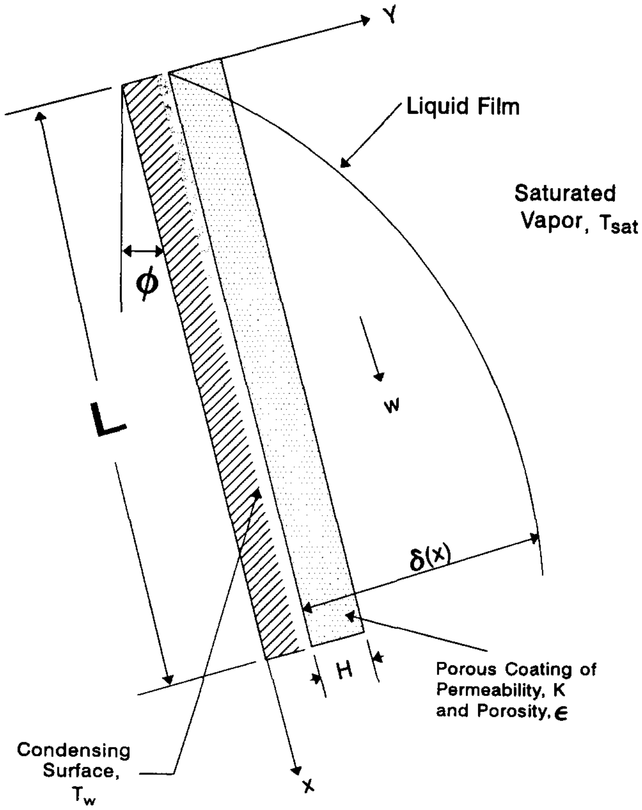

Figure 1 Schematic of problem: film condensation within an inclined surface coated with a porous material of thickness H, permeability K, and porosity ε

In the liquid region, traditional laminar film condensation conservation equations of mass, momentum, and energy are used. The conservation equations for the porous region are based on the Darcy–Brinkman model, which accounts for friction caused by macroscopic shear and satisfies the no-slip

Notation

b	Width of surface
C_p	Specific heat of fluid
g	Gravitational acceleration
H	Thickness of porous coating
h_L	Average heat transfer coefficient
h_x	Local heat transfer coefficient
k	Thermal conductivity
K	Permeability of porous coating
L	Length of surface
Nu_L	Average Nusselt number
Nu_x	Local Nusselt number
p	Pressure
q''	Local heat flux
Q''	Average heat flux
Re	Condensate Reynolds number
T	Temperature
u	x-component of velocity
v	y-component of velocity
V	Velocity
w	Mass flow rate per unit depth
x	Vertical Cartesian coordinate
y	Horizontal Cartesian coordinate

Greek symbols

α	Thermal diffusivity
δ	Film-condensation thickness
ε	Porosity
θ	Dimensionless temperature
λ_{fg}	Modified latent heat of vaporization
μ	Dynamic viscosity
ν	Kinematic viscosity
ρ	Density
ϕ	Plate inclination angle

Subscripts

eff	Effective porous/fluid
f	Fluid
H	Porous/fluid layer interface
l	Pure liquid layer
s	Solid
sat	Saturation
v	Pure vapor
w	Wall

Superscripts

*	Dimensionless quantity

condition on a solid surface. The governing equations for the porous layer are

$$\nabla \cdot \mathbf{V} = 0 \quad (1)$$

$$\mathbf{V} \cdot \nabla \mathbf{V} = \frac{-1}{\rho_f} \nabla p + \nu_{eff} \nabla^2 \mathbf{V} - \mathbf{V} \frac{\nu_f}{K} + \mathbf{g} \quad (2)$$

$$\mathbf{V} \cdot \nabla T = \alpha_{eff} \nabla^2 T \quad (3)$$

where the effective thermal diffusivity is defined as $\alpha_{eff} = k_{eff}/\rho_f C_p$ and ρ_f and C_p refer to the density and the heat capacity of the fluid. The boundary conditions used for the present study are

$$x = 0: \quad u = 0, \qquad v = 0, \quad (4)$$

$$y = 0: \quad u = 0, \qquad v = 0, \qquad T = T_w \quad (5)$$

$$y = \delta(x): \quad \frac{du}{dy} = 0, \qquad v = 0, \qquad T = T_{sat} \quad (6)$$

The second and third boundary conditions refer, respectively, to the no-slip condition at the isothermal wall and the assumption of no interfacial shear at the condensate/vapor interface, which is kept at a constant temperature of T_{sat}. The matching conditions at the interface of the porous/fluid layer are

$$y = H: \quad u(y = H-) = u(y = H+) \quad (7a)$$

$$\mu_{eff}\left(\frac{du}{dy}\right)\bigg|_{y=H-} = \mu_f\left(\frac{du}{dy}\right)\bigg|_{y=H+} \quad (7b)$$

$$T(y = H-) = T(y = H+) \quad (7c)$$

$$k_{eff}\left(\frac{dT}{dy}\right)\bigg|_{y=H-} = k_f\left(\frac{dT}{dy}\right)\bigg|_{y=H+} \quad (7d)$$

These conditions include the continuity of velocity, shear, temperature, and heat flux at the condensate-saturated porous-coating/liquid interface. In these equations, we have set the effective viscosity of the condensate-saturated porous substrate equal to the viscosity of the fluid and have used two models to calculate an effective thermal conductivity for the porous coating. These include the volumetric averaging technique and the Maxwell upper-bound model (Hadley 1986), respectively:

$$\text{Model 1:} \quad k_{eff} = \varepsilon k_l + (1 - \varepsilon) k_s \quad (8a)$$

$$\text{Model 2:} \quad k_{eff} = \frac{2(k_s^2/k_f)(1 - \varepsilon) + (1 + 2\varepsilon)k_s}{(2 + \varepsilon)k_s/k_f + (1 - \varepsilon)} \quad (8b)$$

Numerical solutions

The numerical scheme uses an iterative procedure whereby the solutions of the separate governing equations and the interface conditions are matched. To render compact results, the following dimensionless variables are introduced:

$$x^* = \frac{x}{K^{1/2}} \qquad y^* = \frac{y}{K^{1/2}} \quad (9a)$$

$$H^* = \frac{H}{K^{1/2}} \qquad \delta^*(x^*) = \frac{\delta(x)}{K^{1/2}} \quad (9b)$$

$$\mu^* = \frac{\mu_{eff}}{\mu_l} \qquad k^* = \frac{k_l}{k_{eff}} \quad (9c)$$

$$u^* = u \frac{\mu_{eff}}{[K(\rho_l - \rho_v) g \cos\phi]} \quad (9d)$$

$$\theta = \frac{(T - T_w)}{(T_{sat} - T_w)} \quad (9e)$$

The dimensionless boundary and matching conditions become

$$x^* = 0: \quad u^* = 0, \qquad v^* = 0 \quad (10a)$$

$$y^* = 0: \quad u^* = 0, \qquad v^* = 0, \qquad \theta = 0 \quad (10b)$$

$$y^* = \delta^*(x^*): \quad \frac{du^*}{dy^*} = 0, \qquad v^* = 0, \qquad \theta = 1 \quad (10c)$$

$$y^* = H^*: \quad u(y^* = H^*-) = u^*(y^* = H^*+)$$

$$\mu^*\left(\frac{du^*}{dy^*}\right)\bigg|_{y^*=H^*-} = \left(\frac{du^*}{dy^*}\right)\bigg|_{y^*=H^*+}$$

$$\theta(y^* = H^*-) = \theta(y^* = H^*+)$$

$$k^*\left(\frac{d\theta}{dy^*}\right)\bigg|_{y^*=H^*-} = \left(\frac{d\theta}{dy^*}\right)\bigg|_{y^*=H^*+} \quad (11)$$

Upon inserting the above dimensionless variables into the governing equations of the porous and fluid regions and solving the system of differential equations with the above boundary and matching conditions in the same fashion as Renken *et al.* (1989), the following velocity and temperature profiles result:

$0 < y^* < H^*$:

$$u^*(y^*) = \sinh y^*(\delta^*(x^*) - H^* + \sinh H^*)/\cosh H^* + 1 - \cosh y^* \quad (12)$$

$$\theta(y^*) = k^* y^*/(\delta^*(x^*) + H^*(k^* - 1)) \quad (13)$$

$H^* < y^* < \delta^*(x^*)$:

$$u^*(y^*) = 0.5(H^{*2} - y^{*2}) + \mu^* \delta^*(x^*)(y^* - H^*) + \tanh H^*(\sinh H^* - H^* + \delta^*(x^*)) + 1 - \cosh H^* \quad (14)$$

$$\theta(y^*) = (y^* + H^*(k^* - 1))/(\delta^*(x^*) + H^*(k^* - 1)) \quad (15)$$

The mass flow of condensate through any vertical position of the film is obtained by integrating the product of the density and velocity over each flow region. The resulting total mass flow rate through a cross section of the film is a third-order polynomial in terms of the dimensionless thickness, $\delta^*(x^*)$.

The heat removed by the porous/fluid composite system over an incremental distance dx can be approximated by

$$q'' = \lambda_{fg} b \frac{dw}{dx} \quad (16)$$

where λ_{fg} represents a modified heat of vaporization (Rohsenow 1956) that accounts for liquid crossflow within the film. From Fourier's Law and the assumption that we have linear temperature distributions within the porous layer and pure liquid regions, the heat flux can be expressed as

$$q'' = \frac{-k_{eff}}{H}(T_w - T_H) = \frac{-k_l}{(\delta - H)}(T_H - T_{sat}) \quad (17)$$

where T_H is the temperature at the porous/fluid interface. Combining Equations 16 and 17 and integrating produces a fourth-order equation for $\delta^*(x^*)$, which is solved numerically using a Newton-Raphson method. For the special case of $H^* = 0$ and $\theta = 0$, the solution reduces to Nusselt's solution of an uncoated inclined surface.

The heat transfer results are reported by the local heat transfer coefficient and the local Nusselt number, which are respectively defined as

$$q'' = h_x(T_{sat} - T_w) \quad (18)$$

$$\mathrm{Nu}_x = \frac{h_x x}{k_1} \tag{19}$$

Using an energy balance, the local heat transfer coefficient is expressed in terms of the problem parameters as

$$h_x = \frac{k_1}{[K^{1/2}(\delta^*(x^*) + H^*(k^* - 1))]} \tag{20}$$

Integrating over the length of the surface produces the average heat transfer coefficient, which is used to define the average Nusselt number and average surface heat flux:

$$\mathrm{Nu}_L = \frac{h_L L}{k_1} \tag{21}$$

$$Q'' = h_L(T_{sat} - T_w) \tag{22}$$

It should be noted that the Nusselt number is based on the thermal conductivity of the fluid to make for a more substantial comparison with the noncomposite system. It was also assumed that the effective dynamic viscosity of the condensate-saturated porous substrate (μ_{eff}) was equivalent to the dynamic viscosity of the pure liquid (μ_l), or (μ^*) = 1.

Discussion of results

The results of this investigation are illustrated in Figures 2 to 6. Figure 2 exhibits the numerical predictions of the average Nusselt number as a function of surface subcooling temperature difference ($T_{sat} - T_w$) and inclination angle (ϕ). For these cases, the dimensionless porous substrate thickness (H^*) is 0.25 and the porosity of the porous coating (ε) is 0.50; the dimensionless thermal conductivity (K^*), which varies with the film temperature, has a representative value of 3.29×10^{-3} based on a 75°C film temperature. We note that the average Nusselt number decreases with increasing surface subcooling for all three test cases. It is observed that the value of average Nusselt number also decreases with increased inclination angle for a particular value of surface subcooling. A comparison between a vertical surface ($\phi = 0°$) and surfaces at 30- and 60-degree inclines reveal average differences in Nusselt numbers of 5 and 20 percent, respectively. This result is expected, since the porous/fluid composite system is experiencing the effect of inclination. Another comparison in Figure 2 is between the present model and Nusselt's correlation. The present model shows an average of 25 percent heat transfer enhancement over the range of surface subcooling temperatures as compared to Nusselt's theory under the conditions specified. Nusselt's predictions were obtained under the assumption of laminar film condensation with no surface waves for an inclined uncoated surface (Nusselt 1916). For the range of parameters used in our calculations, the laminar flow assumption is valid (Re < 30).

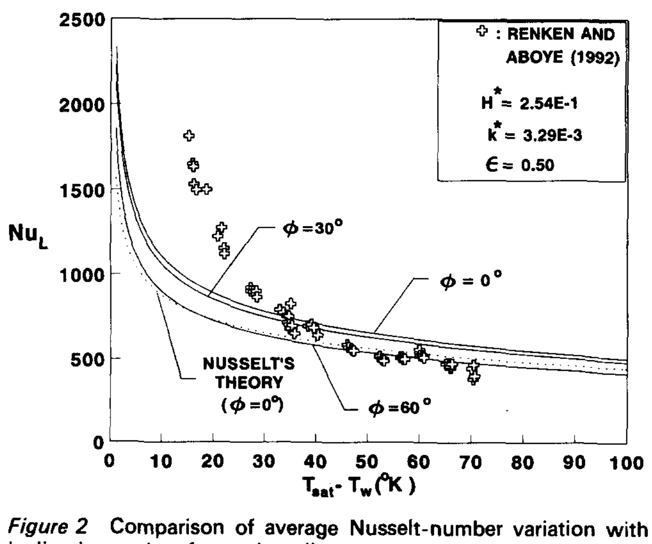

Figure 2 Comparison of average Nusselt-number variation with inclination and surface subcooling

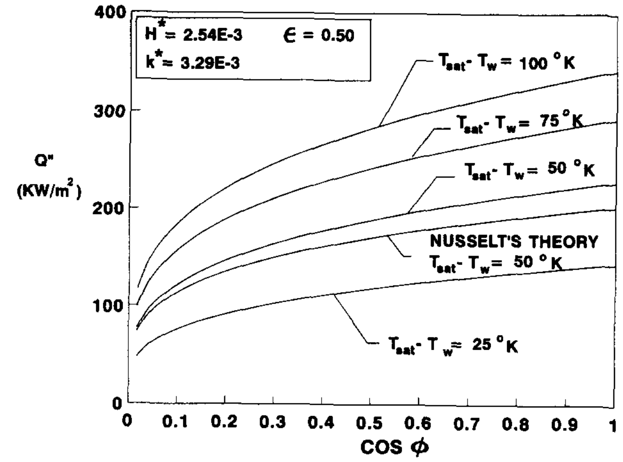

Figure 3 Comparison of heat transfer with surface subcooling and effective body force

Figure 2 also contains a comparison of the present model predictions with recent experimental data for the limiting case of film condensation of saturated steam at atmospheric pressure on a vertical isothermal porous coated plate (Renken and Aboye 1992). The theoretical and experimental values for H^*, k^*, and ε are essentially equivalent. The porous substrate used in this experimental run was a thermospray self-bonding aluminum-bronze-copper coating of approximately 50 percent porosity that was adhered to a $7.62 \times 12.7 \times 1.27$ cm (width × length × thickness) oxygen-free 101 copper plate. The thickness of this porous coating was approximately 2.54×10^{-2} mm (0.001 in). At the low end of the surface subcooling, we have as much as a 70 percent increase in the average Nusselt number as compared to the model prediction. It was observed that under these low surface subcooling temperatures when we had minimal liquid film thickness, particular sections of the porous coated plate (especially near the leading edge) were not completely saturated. We have concluded that a two-phase porous region existed that produced transitional (dropwise/filmwise) condensation over a small length of the porous coating. This resulted in higher heat transfer rates and deviation from the model. At the higher end of surface subcooling where the film saturates the porous substrate, we have good agreement between the model and the experimental data. It was estimated that the experimental uncertainty of the average Nusselt number was within 8 percent.

In Figure 3, the numerical predictions have been displayed in a compact form to compare the effects of angle of inclination and surface subcooling temperature differences on film-condensation heat transfer rates for characteristic values of porous coating thickness, effective thermal conductivity, and porosity. We find that the average heat flux increases with reduced angle of inclination ($\phi \to 0°$) for all four cases. More specifically, for ($T_{sat} - T_w$) = 100°K, we observe an 83-percent increase in heat transfer between effective body forces of 0.1 g and 1 g. We also notice heat transfer augmentation as the driving force (the surface subcooling temperature difference) increases. There exists as much as a 137-percent increase in heat flux when comparing surface subcooling temperature differences of 25°K and 100°K. Figure 3 also compares the results of the present model with Nusselt's theory. We have an average increase of 10 percent in average heat flux over the range of body forces tested as compared to the Nusselt prediction.

Figures 4 to 6 examine the dependence of film-condensation heat transfer enhancement on the thin porous-layer coating characteristics. Figure 4 shows the effect of porous substrate thickness for a metallic (copper) coating with a porosity of 0.50 and surface subcooling of 50°K. This type of porous coating with a dimensionless thickness of $H^* = 1.27 \times 10^{-2}$ produces an extremely noticeable increase (approximately 60 percent) in heat flux when compared to the uncoated case (Nusselt's theory). It is observed that a curtailment in heat transfer enhancement exists when the dimensionless porous coating thickness exceeds $H^* = 1.27 \times 10^{-2}$ for the conditions specified. This lower value of enhancement can be attributed to the concept of an optimal porous coating thickness, suggested by Renken *et al.* (1989) and Renken and Aboye (1992). The physical significance of this result is that the porous substrate begins to act as an insulation.

Figure 5 depicts the importance of the thin porous-layer coating composition and porosity. In our simulations, we have utilized two different models that are used exclusively in the porous-coating phase-change literature to calculate the effective thermal conductivity for the porous region. The dimensionless effective thermal conductivity subscript 1 refers to the volume-averaging model (Equation 8a), while subscript 2 refers to the Maxwell upper-bound model (Equation 8b). These test cases used a surface subcooling temperature difference of 50°K, a dimensionless porous coating thickness of 2.54×10^{-3}, and a porous substrate porosity of 0.50. In Figure 5, we compare the results of both models to the uncoated case (Nusselt's theory) when pure copper ($k_1^* = 3.29 \times 10^{-3}$ and $k_2^* = 4.11 \times 10^{-3}$) and pure aluminum ($k_1^* = 6.55 \times 10^{-3}$ and $k_2^* = 8.17 \times 10^{-3}$) porous substrates are used. We find excellent agreement between the models and significant heat transfer augmentation when compared to the noncoated case. The figure also contains an order-of-magnitude analysis of the porous-coating effective thermal conductivity with the condensate thermal conductivity. Here, we have calculated results for $k_{eff} = k_l$ ($k_1^* = 1.0$), $k_{eff} = 5 \times k_l$ ($k_1^* = 0.2$), and $k_{eff} = 50 \times k_l$ ($k_1^* = 0.02$) with $\varepsilon = 0.50$. The results show that when

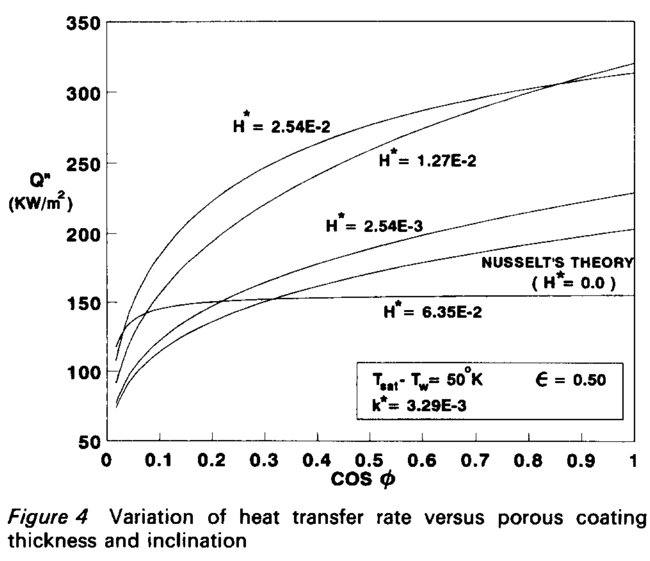

Figure 4 Variation of heat transfer rate versus porous coating thickness and inclination

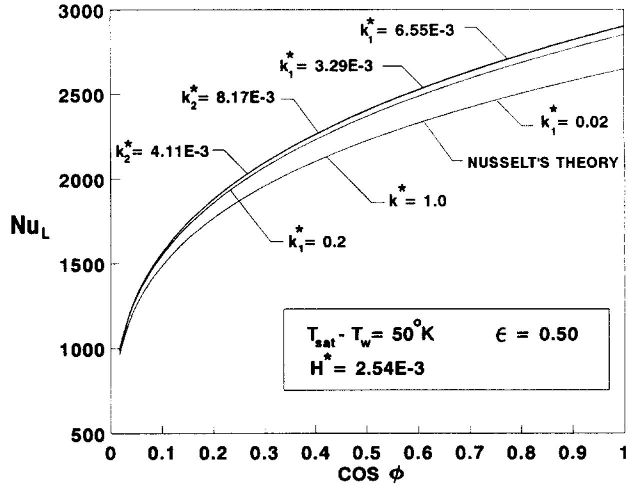

Figure 5 Variation of average Nusselt number versus porous coating effective thermal conductivity and inclination

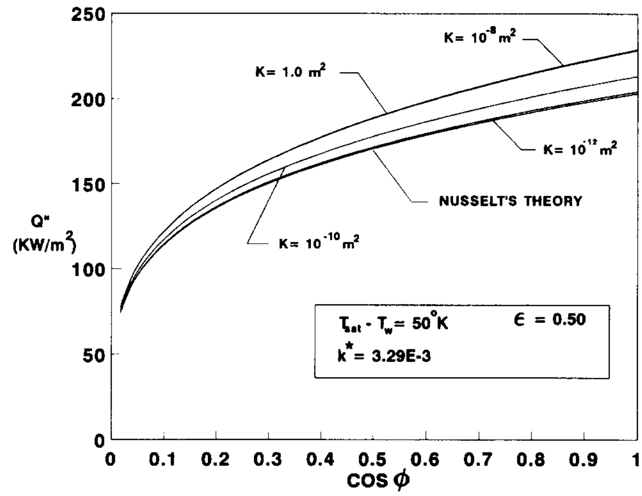

Figure 6 Variation of heat transfer versus permeability and inclination

$k^* = 1.0$, the heat transfer rate is equivalent to Nusselt's correlation, while when $k^* < 1.0$, major increases in heat transfer are observed.

Finally, Figure 6 displays the variation in average heat flux with inclination and permeability for the case of $(T_{sat} - T_w) = 50°K$, $k_1^* = 3.29 \times 10^{-3}$, and $\varepsilon = 0.50$. It is observed that the average heat flux increases with permeability and effective body force. A closer examination of the results show that as $K \to 0$, the heat transfer rate approaches the prediction of Nusselt's correlation. This is expected, since the less penetrable the porous-layer coating becomes, the more it resembles an impermeable or uncoated surface.

Conclusions

The problem of film condensation within an inclined thin porous-layer coated surface has been modeled numerically by a porous/fluid composite system. The Darcy–Brinkman flow model was utilized to describe the flow and heat transfer processes in the porous region. Two sets of governing equations for the porous and fluid regions were employed, and matching interface conditions were used to solve the problem. The characteristics of the velocity and temperature fields have been presented, and exemplary heat transfer results were obtained. Comparisons of the theoretical predictions of the thin porous metallic-layer coated surface system and the noncoated case revealed a significant increase in heat transfer rate.

Recent experimental findings exhibit an order-of-magnitude verification of the present work and call for advanced modeling of condensation within porous substrates. The effects of inclination, surface subcooling, porous substrate thickness, effective thermal conductivity, and permeability were documented. In conclusion, the results of this fundamental study demonstrate that the employment of a porous metallic coating on a condensing surface needs to be considered as a viable alternative for heat transfer enhancement in condensation problems.

Acknowledgment

Financial support for this research, provided by the National Science Foundation through Grant No. CTS-8909410, is greatly appreciated.

References

Afgan, N. M., Jovic, L. A., Kovalev, S. A. and Lenykov, V. A. 1985. Boiling heat transfer from surfaces with porous layers. *Int. J. Heat Mass Transfer*, **28**, 415–422

Avedisian, C. T. and Koplik, J. 1982. Liedenfrost boiling of methanol droplets on hot porous/ceramic surfaces. *Int. J. Heat Mass Transfer*, **30**, 379–393

Avedisian, C. T., Ioffredo, C. and O'Connors, M. J. 1984. Film boiling of discrete droplets of mixtures of coal and water on a horizontal brass surface. *Chem. Eng. Sci.*, **39**, 319–327

Bergles, A. E. and Chyu, M. C. 1982. Characteristics of nucleate pool boiling from porous metallic coatings. *J. Heat Transfer*, **104**, 279–285

Carnavos, T. C. 1980. An experimental study: condensing R-11 on augmented tubes. ASME paper 80-HT-54, 1–7

Cheng, P. and Verma, A. 1981. The effect of subcooled liquid on film boiling about a vertical heated surface in a porous medium. *Int. J. Heat Mass Transfer*, **24**, 1151–1160

Hadley, G. R. 1986. Thermal conductivity of packed metal powders. *Int. J. Heat Mass Transfer*, **29**, 909–920

Holden, K. M., Wanniarachi, A. S., Marto, P. J., Boone, D. H. and Rose, J. W. 1987. The use of organic coatings to promote dropwise condensation of steam. *J. Heat Transfer*, **109**, 768–774

Ito, T., Nishikawa, K. and Tanaka, K. 1982. Enhanced heat transfer by nucleate boiling at sintered metal layer—discussion on sintered layer and experiments by piled layer and form layer. *Refrigeration*, **57**, 77–81

Izumi, M., Yamakawa, N., Shinmura, T., Isobe, Y., Ohtani, S. and Westwater, J. W. 1989. Drop and filmwise condensation on horizontally scratched rough surfaces. *Heat Transfer Jpn. Res.*, **18**, 1–14

Kajikawa, T., Takazawa, H. and Mizuki, M. 1983. Heat transfer performance of metal fiber sintered surfaces. *Heat Transfer Eng.*, **4**, 57–66

Kaviany, M. 1991. *Principles of Heat Transfer in Porous Media*. Springer-Verlag, New York, 561–572

Konev, S. V. and Mitrovic, J. 1986. An explanation for the augmentation of heat transfer during boiling in capillary structures. *Int. J. Heat Mass Transfer*, **29**, 91–94

Konev, S. K., Polasek, F. and Horvat, L. 1987. Investigation of boiling in capillary structures. *Heat Transfer Soviet Res.*, **19**, 14–17

Kovalev, S. A., Solov'yev, S. L. and Ovodkov, O. A. 1987. Liquid boiling on porous surfaces. *Heat Transfer Soviet Res.*, **19**, 109–120

Marto, P. J. and Lepere, V. J. 1982. Pool boiling heat transfer from enhanced surfaces to dielectric fluids. *J. Heat Transfer*, **104**, 292–299

Marto, P. J. 1986. Recent progress in enhancing film condensation heat transfer on horizontal tubes. *Proc. 8th Int. Heat Transfer Conf.*, 161–170

Marto, P. J., Mitrou, E., Wanniarachi, A. S. and Katsuta, M. 1987. Film condensation of steam on a horizontal wire-wrapped tube. *Proc. 2nd ASME-JSME Thermal Eng. Joint Conf.*, 509–516

Nakayama, W., Daikoku, T., Kuwahara, H. and Nakajima, T. 1980a. Dynamic model of enhanced boiling heat transfer on porous surfaces. Part 1: Experimental investigation. *J. Heat Transfer*, **102**, 445–450

Nakayama, W., Daikoku, T., Kuwahara, H. and Nakajima, T. 1980b. Dynamic model of enhanced boiling heat transfer on porous surfaces. Part 2: Analytical modeling. *J. Heat Transfer*, **102**, 451–456

Nakayama, W., Daikoku, T. and Nakajima, T. 1982. Effects of pore diameters and system pressure on saturated pool nucleate boiling heat transfer from porous surfaces. *J. Heat Transfer*, **104**, 286–291

Nield, D. A. and Bejan, A. 1992. *Convection in Porous Media*. Springer-Verlag, New York, 343–344

Nusselt, W. 1916. Die oberflachenkondensation des wasser dampfes. *Zeitschrift des Vereins Deutsches Ingenieure*, **60**, 541–575

Renken, K. J., Soltykiewicz, D. J. and Poulikakos, D. 1989. A study of laminar film condensation on a vertical surface with a porous coating. *Int. Comm. Heat Mass Transfer*, **16**, 181–192

Renken, K. J. and Aboye, M. 1992. Condensation measurements within inclined thin porous-layer coated surfaces. *Fundamentals of Heat Transfer in Porous Media 1992*, HTD-Vol. 193, pp. 1–6, presented at The 28th National Heat Transfer Conference and Exposition, San Diego, California, August 9–12, 1992.

Rifert, V. G., Trokoz, Y. Y. and Zadiraka, V. Y. 1984. Enhancement of heat transfer in condensation of ammonia vapor on a bundle of wire-finned tubes. *Heat Transfer Soviet Res.*, **16**, 36–41

Rohsenow, W. M. 1956. Heat transfer and temperature distribution in laminar film condensation. *Trans. ASME*, **78**, 1645–1648

Shekarriz, A. and Plumb, O. A. 1986. A theoretical study of the enhancement of filmwise condensation using porous fins. ASME Paper No. 86-HT-31, presented at the Joint AIAA/ASME Thermophysics and Heat Transfer Conference, Boston, Massachusetts, June 2–4, 1986, 1–5

Styrikovich, M. A., Malyshenko, S. P., Andrianov, A. B. and Talaev, I. V. 1987. Investigation of boiling on porous surfaces. *Heat Transfer Soviet Res.*, **19**, 23–29

Webb, R. L. 1981. The evolution of enhanced surface geometries for nucleate boiling. *Heat Transfer Eng.*, **2**, 46–69

Webb, R. L. 1983. Nucleate boiling on porous coated surfaces. *Heat Transfer Eng.*, **4**, 71–82

Woodruff, D. W. and Westwater, J. W. 1979. Steam condensation on electroplated gold: effect of plating thickness. *Int. J. Heat Mass Transfer*, **22**, 629–632

Woodruff, D. W. and Westwater, J. W. 1981. Steam condensation on various gold surfaces. *J. Heat Transfer*, **103**, 685–692

Yau, K. K., Cooper, J. R. and Rose, J. W. 1984. Effects of drainage strips and fin spacing on heat transfer and condensate retention for horizontal finned and plain condenser tubes. *Fundamentals of Phase Change: Boiling and Condensation*, presented at the Winter Annual Meeting of the American Society of Mechanical Engineers, New Orleans, Louisiana, December 9–14, 1984, 151–156