Int. J. Heat Mass Transfer. Vol. 36, No. 5, pp. 1347–1355, 1993
Printed in Great Britain

0017-9310/93 \$6.00+0.00

Experiments on film condensation promotion within thin inclined porous coatings

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(*Received* 26 *March* 1992 *and in final form* 22 *May* 1992)

Abstract—This paper presents the results of an experimental investigation of condensation within thin inclined porous coatings. The porous substrate system consisted of a condensate region overlaying a thin, metallic, and permeable coating which was adhered to an isothermal copper block. Reduced gravity measurements were obtained by condensing saturated steam containing small concentrations of non-condensables at atmospheric pressure on surfaces with effective body forces between 0.3 and 1 g. The effects of surface subcooling and porous coating thickness were also investigated. The heat transfer data shows substantial enhancement when compared to a noncoated surface and Nusselt's correlation. A theoretical model based on porous/fluid composite condensation is also evaluated.

INTRODUCTION

THE PROBLEM of increasing condensation heat transfer rates has been given considerable attention in the heat transfer literature since the initial analysis by Nusselt [1]. Recent interest in the enhancement of condensation by innovative techniques has been energized by the design and operation of heat pipes, space satellite radiators, electronic equipment cooling, thermal insulations, and the cooling of nuclear reactors.

Past investigations have used a wide variety of surface geometries and modifications to promote a more rapid removal of the condensate. A few of the more interesting studies include the works of Woodruff and Westwater [2, 3], who discovered that gold surfaces promoted dropwise condensation (DWC) of steam at atmospheric pressure; Carnavos [4] who tested augmented tubes for overall R-11 condensing performance; Rifert *et al.* [5], Marto [6] and Marto *et al.* [7] who reported that wire-finned tubes enhanced the heat transfer coefficients in condensation by approximately 60–100% as compared with the case of smooth tubes; Yau *et al.* [8] who studied the effects of drainage strips for horizontal finned condenser tubes; Shekarriz and Plumb [9, 10] who theoretically and experimentally demonstrated that capillary porous fins can have a significant effect on filmwise condensation rates; and others who have used organic coatings [11] and scratched rough surfaces [12] to promote dropwise and filmwise condensation.

An alternative technique for the removal of condensate utilizes a porous/fluid composite medium which produces high heat transfer coefficients with small temperature differences. This method was first introduced by Renken *et al.* [13] and further investigated by Renken and Aboye [14] and Renken and Mueller [15]. It employs a relatively thin, very permeable, and highly conductive porous coating which is bonded to a condensing surface. The majority of research using this type of system has been devoted to evaporation and is summarized by Kaviany [16] and Nield and Bejan [17].

The objective of this paper is to present the results of a series of experiments that were performed to demonstrate the possibilities that exist for condensation enhancement with the employment of a thin, porous, and metallic substrate system. For these experiments, the effects of body force and porous coating composition thickness were investigated. The results are compared to the condensation measurements of an inclined plain surface, a theoretical model based on porous/fluid composite condensation, and Nusselt's correlation.

THEORETICAL RESULTS

Previous theoretical results, which will be compared with the present experimental results, will now be summarized. Figure 1 illustrates schematically the problem of film condensation within an inclined thin porous layer coated surface. In more detail, we have an isothermal cold surface of length L, which is immersed in saturated steam and inclined with the vertical axis by an angle of ϕ. The solid surface is coated with a thermally conductive and porous substrate of thickness H, permeability K and porosity ε. The mass flow rate of the condensing liquid flows through the porous/fluid composite by gravitational

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NOMENCLATURE

g	gravitational acceleration [m s^{-2}]
h_L	average heat transfer coefficient [W m^{-2} K^{-1}]
H	thickness of porous substrate [m]
k_{eff}	effective thermal conductivity of porous substrate [W m^{-1} K^{-1}]
k_l	thermal conductivity of condensate [W m^{-1} K^{-1}]
K	permeability of porous coating [m^2]
L	length of plate surface [m]
Nu_x	local Nusselt number
Nu_L	average Nusselt number
q''	average heat flux [W m^{-2}]
T_{sat}	saturation temperature [K]
T_w	condensing surface temperature [K]
w	condensate mass flow rate [kg s^{-1}]
x	vertical coordinate [m]
y	horizontal coordinate [m].

Greek symbols

δ	film condensation thickness [m]
ε	porosity of porous coating
ϕ	plate inclination angle [deg].

forces and is expressed as w. The local film condensation thickness is represented by $\delta(x)$.

For steady, laminar, incompressible, and one-dimensional flow, Renken and Aboye [14] utilized the Darcy–Brinkman flow model to describe the flow process in the porous region, while traditional laminar film condensation conservation equations were used in the fluid region. The assumption of linear temperature distributions within the porous and pure liquid layers as well as the employment of matching conditions of velocity, shear, temperature, and heat flux at the porous/fluid layer interface resulted in the following expression for the local Nusselt number:

$$Nu_x = x/(\delta(x)+H(k_l/k_{eff}-1)). \quad (1)$$

Since the present experiment measures the overall heat transfer rate, equation (1) can be numerically integrated to give the average Nusselt number. The result is not given here, for brevity (the solution for $\delta(x)$ is a fourth-order equation which is solved numerically).

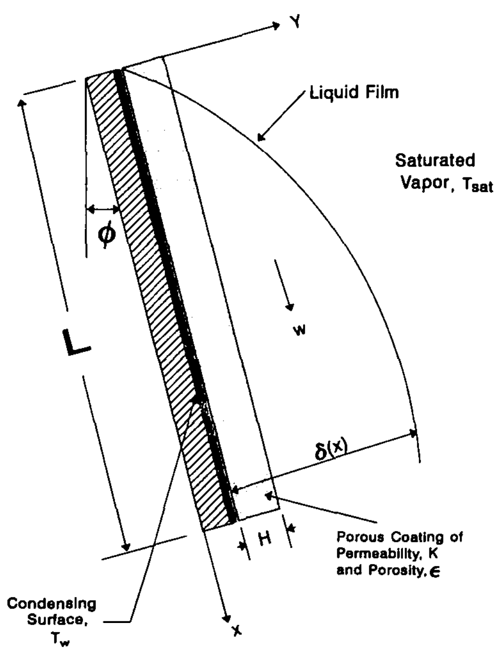

FIG. 1. Film condensation within an inclined thin porous-layer coated surface.

EXPERIMENTAL APPARATUS AND PROCEDURE

Figure 2 shows a schematic of the experimental apparatus by which the condensation of steam on the plain plate and within the inclined thin porous layer coated surfaces was measured. The test apparatus consisted of several major components which included the boiler, the condensing chamber, the recirculating chiller, the PC-data acquisition system and the test surfaces.

The boiler was a 304 stainless steel sealed cylinder with dimensions of 15.2 cm o.d. and 45.7 cm length. It contained a 5 kW immersion heater and a microprocessor based temperature controller which provided for the continuous production of steam at a temperature of 100°C. The water used in the boiler was distilled to eliminate any impurities and to reduce noncondensables. Make-up water was added from a gravity feed reservoir tank and controlled by an electronic control valve which used a photoelectric sensor, a sight glass indicator and a metallic float to maintain a consistent level of water in the boiler. Condensate within the condensing chamber was also drained back into the boiler by several drain ports. In this fashion, test conditions throughout experiments were maintained. Steam temperatures in the boiler and around the condensing test plates were measured by averaging ten Type T thermocouple probes.

The recirculating chiller had a heat removal capacity of 9 kW at 20°C and provided ample cooling for the cold test surfaces. A throttling and pressure relief valve were used to accurately control the coolant flow rate which was measured by a paddlewheel sensor with an accuracy of $\pm 1\%$. During the majority of the experiments, maximum flow rates (up to 6 g.p.m.) were used to maintain an isothermal surface. The

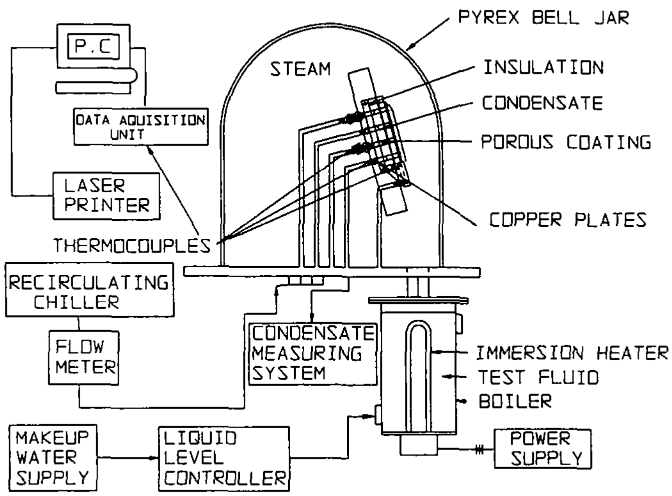

FIG. 2. Schematic diagram of the experimental apparatus.

recirculating coolant was a mixture of 60% ethylene glycol and 40% distilled water by volume.

The condensing chamber was a pyrex bell jar which has dimensions of 45.7 cm o.d. and 76.2 cm height. The outer surface of the bell jar was covered by 2.54 cm of Armoflex insulation and surrounded by a lexan tank. This configuration greatly reduced the heat transfer to the environment and the amount of condensation lost to the jar's inner walls. The double chamber system was clamped onto a 0.64 cm surgical silicone rubber gasket and 1.91 cm anodized aluminum plate which contained functional tapped holes. Exhaust ports in this base plate permitted atmospheric operating conditions which were monitored by a compound pressure gauge. The corrugated FEP Teflon tubing allowed for the coolant fluid from the recirculating chiller to circulate into and out of the coolant block plate. This type of tubing was chosen because of its flexibility and endurance under extreme temperature conditions. It was insulated by KAO-TEX tape to reduce condensation formation.

Figures 3(a) and (b) show the details of the condensing surface test section, which consisted of a coolant block, a test surface plate, as well as a temperature and condensate measurement system. The coolant block was made of oxygen free 101 copper and measured $7.62 \times 12.7 \times 1.91$ cm. It contained four insulated coolant channels which were designed to provide adequate heat transfer and subsequent condensation on the test plates. Four 1.59 mm diameter Type T thermocouple probes were used to measure the temperature differences between the coolant inlets and outlets which were located across the width of the coolant block plate.

Four identical interchangeable condensing test plates were used during the experiments. The surface of each plate contained a different thickness of porous coating. Each plate had a dimension of $7.62 \times 12.7 \times 1.27$ cm and was made of oxygen-free 101 copper. This material was chosen because of its excellent resistance to corrosion. The surface temperature and temperatures within the plate (heat flux estimate) were measured with 30 AWG Type T thermocouples, which were calibrated with an accuracy

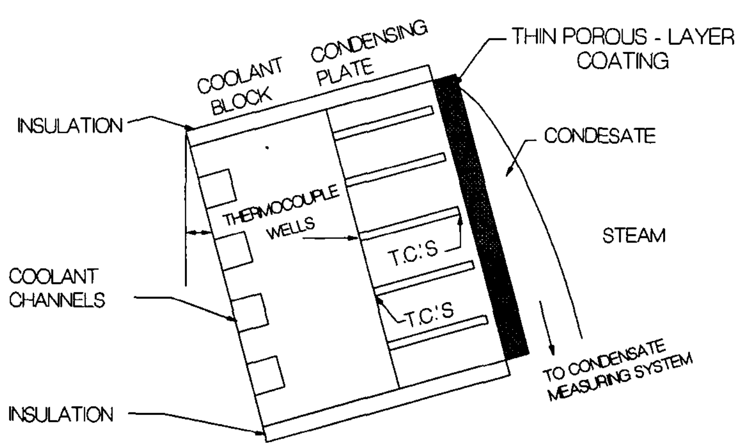

FIG. 3(a). Details of the temperature and condensate measurement system.

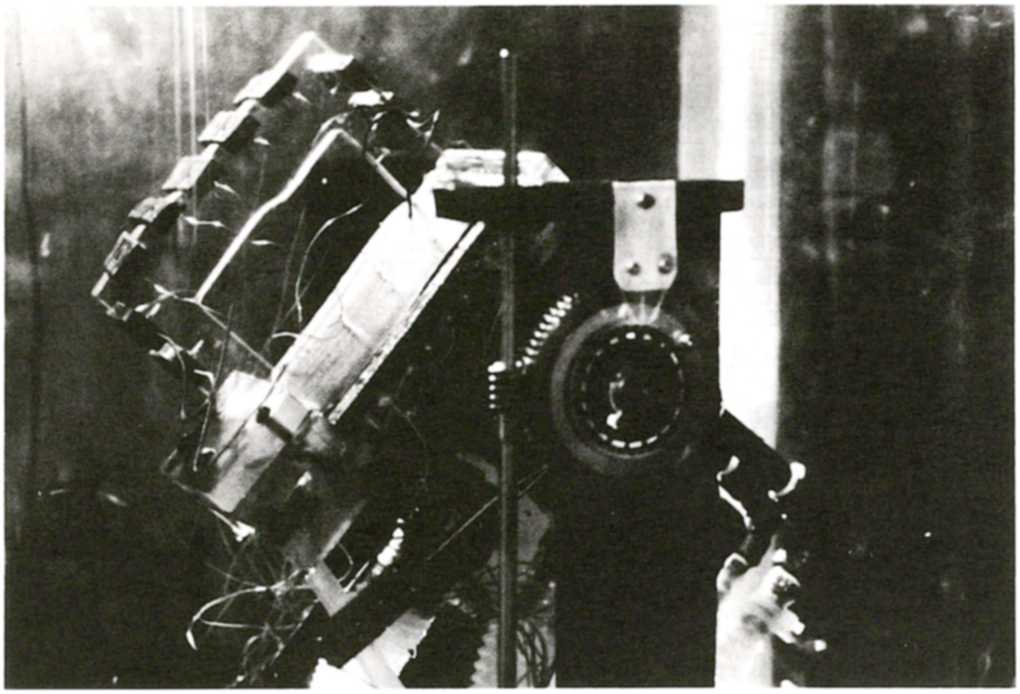

FIG. 3(b). Photograph of condensing surface test section.

of 0.1°C. There were 22 thermocouples embedded in each plate by employing thermocouple plugs which were placed into thermocouple wells, as seen in Fig. 3(a). A thermally conductive silicone grease was applied between the test plates and the coolant block to provide for a more uniform heat passage, and the two were tightly attached by machine screws on the reverse side of the coolant block. The edges of the coolant block and the test plate were then insulated by 1.27 cm thick lexan.

Four test surfaces were examined in this fundamental investigation. Three test surfaces had a thermal spray self-bonding aluminum bronze copper aluminum coating (Fig. 4(a)) with thicknesses of 25.4, 127 and 254 μm. Figure 4(b) displays a scanning electron microscope photograph of an exemplary thermal spray coating which was used to approximate the porosity and permeability. This figure provides insight into the complexities that are involved in modelling a porous/fluid composite system during condensation heat transfer. The estimated values of the porous coating porosity and permeability used in the experiments were calculated from porosity and permeability models [18] that are based on the shape and average diameter of the porous coating pores. The porosity and permeability values were found to be 0.36 and 4.1×10^{-10} m^2, respectively.

The fourth test surface was an uncoated oxygen-free 101 copper plate. The surface of this plate was sanded with successively higher grain emery paper to achieve a mirror finish to promote maximum film condensation.

The condensing surface test section was mounted and connected to a rotary motion feedthrough device that could provide rotation and reduced gravity conditions parallel to the test surface. The position of the entire unit gave nominal gravitational accelerations between 0.3 and 1 g which were measured by a front surface mirror and a high resolution compass card.

Data collection under steady state conditions was accomplished via a Hewlett-Packard System 10 PC-data acquisition system which contained high-speed relay multiplexers for the various sensors. For each test run, the system ran for approximately one hour before the temperatures and the condensate flow rates were relatively constant and considered steady state. The temperatures, coolant flow rate and system pressure were then measured. An average value of condensation volume flow rate from the test surface was also measured by a condensation collection system which was located at the surface base.

A commercial data acquisition software (LabTech Notebook) was used to program, display and record sensor signals and to format the raw data. Instantaneous calculations of the average heat flux and the average Nusselt number were accomplished by a popular spreadsheet software (Lotus 1-2-3).

DISCUSSION OF EXPERIMENTAL RESULTS AND OBSERVATIONS

Several preliminary experiments were performed to check the accuracy of the surface temperature from the thermocouples that were embedded in the test plates and to ensure that the heat transfer results obtained were consistent with Nusselt's film condensation theory and previous experimental studies of film condensation from an isothermal surface. One trial experiment tested the thermocouples' response to a known temperature source while other tests compared the methods of measuring the heat transfer rate. Here, we measured the temperature difference between the coolant fluid inlet and outlet, the coolant fluid mass flow rate, and based the specific heat of the water/ethylene glycol solution on an average bulk temperature. A second method used Fourier's law and the measurement of an average temperature gradient via thermocouple wells that were embedded in the copper test plates. The final method used the measured condensation rate and a modified heat of vaporization [19] which accounts for liquid crossflow within the film. These three methods took into consideration the heat transfer rate from the top, bottom, and sides of the condensing surface test section.

It was determined that the surface temperature measurements of these experiments has a 1% experimental uncertainty. It was also found that the three methods of heat transfer rate were in good agreement with one another (within 10%). The overall experimental uncertainty of the average heat flux varied with measurements of coolant flow rate, temperatures, condensate volume, length and time as well as composite and fluid properties. It was conservatively estimated that the experimental uncertainty in the average Nusselt number was 8%. Details of the error analysis are found in ref. [20]. The maximum value of film Reynolds number for all the test runs was 10.5 which is within the range for laminar film condensation.

The experimental average heat transfer coefficient and average Nusselt number were defined as:

$$h_L = q''/(T_{\text{sat}} - T_{\text{w}}) \tag{2}$$

$$Nu_L = h_L L / k_1. \tag{3}$$

The average Nusselt number is based on the liquid thermal conductivity to make for a more substantial comparison with the noncoated surface.

Figure 5 is a photograph of film condensation within a 25.4 μm thick porous/fluid composite system. An extremely thin condensate layer is visible along the length of this test section. Visual observations of the condensation process were made during prescribed intervals of time to check for irregularities (dropwise condensation). In all cases, filmwise condensation heat transfer measurements were made after steady state temperatures and condensation rates were achieved.

Quantitative experimental results for three porous substrate thicknesses immersed in saturated steam are shown in Figs. 6 and 7. These figures emphasize the effectiveness of the thin porous-layer in producing a higher heat transfer rate when compared to an uncoated ($H = 0$) plate. From Fig. 6 it is seen that experimental values of Nu_L for $H = 25.4$ μm are 45–56% higher than for $H = 0$ at $T_{\text{sat}} - T_{\text{w}} = 35$ K. For a porous substrate thickness which is five times as thick (127 μm), we find an average heat transfer enhancement of approximately 42%. In Fig. 7 ($T_{\text{sat}} - T_{\text{w}} = 70$ K), we find average heat transfer augmentations of 56 and 30% for $H = 25.4$ μm and 127 μm, respectively. The fact that the thinner porous

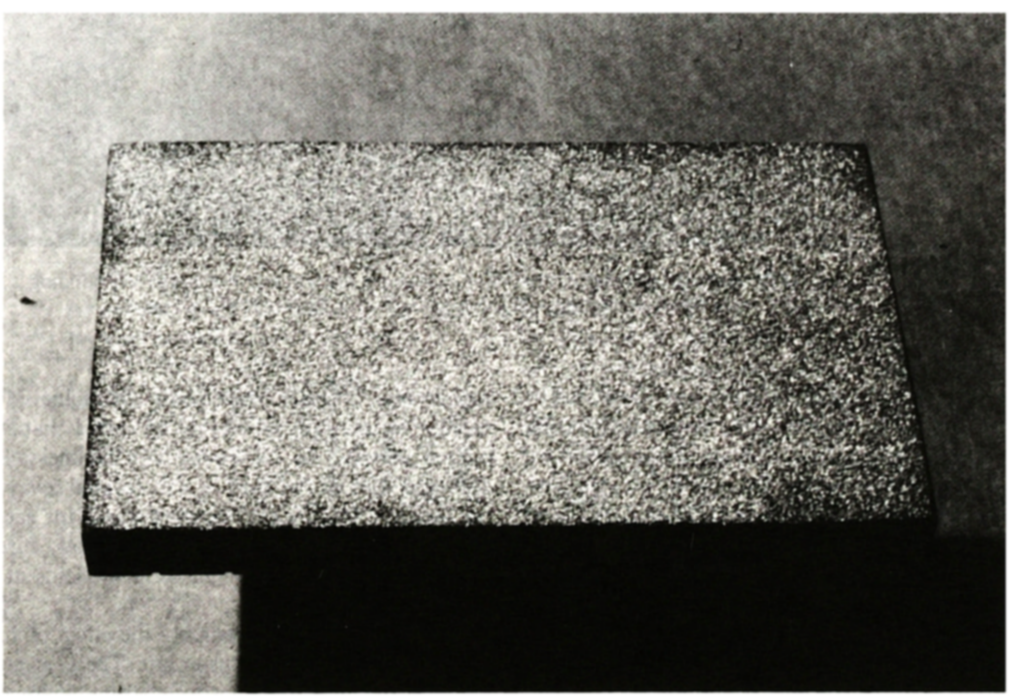

FIG. 4(a). Photograph of a typical thin porous-layer coated surface.

FIG. 4(b). Scanning electron microscope photograph of an exemplary thermal spray coating: self-bonding aluminum bronze copper aluminum with a porous substrate thickness of 25.4 μm

coating produces a greater enhancement leads us to conclude that the porous region in the thicker coating is not fully saturated and contains vapor. Since we have not yet considered the concept of a two-phase flow region in our investigations, it is not possible to speculate on the optimum porous substrate characteristics (composition, porosity, permeability, structure, etc.) that generate maximum heat transfer.

Figures 6 and 7 demonstrate clearly the effect of gravitational force. From this plot, it is concluded that the degree of enhancement continues to increase as the effective body force is increased. As expected, higher heat transfer coefficients are achieved with smaller inclinations. It should be noted that Nu_L is related inversely to the liquid film thickness. It was observed that as the surface was rotated away from the vertical, the film thickness increased for each test surface. Comparing Nu_L for effective body forces of 0.34 and 1.0 g with $T_{sat}-T_w = 35$ K yields at 75% difference.

Figures 6 and 7 also compare the effect of surface subcooling on the heat transfer rate. The average Nusselt number, for a given porous coating thickness and effective body force, increases by as much as 182%.

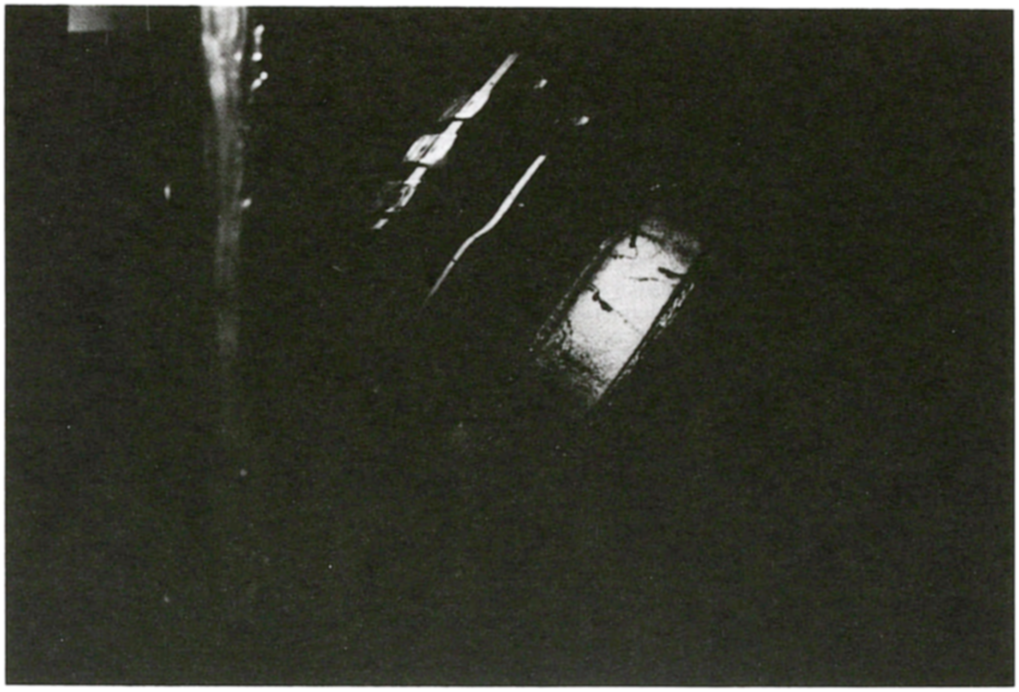

FIG. 5. Film condensation on a 25.4 μm thick thermal spray self-bonding aluminum bronze copper aluminum coating.

It was observed that this effect was most pronounced when the liquid film thickness was very thin as compared to the porous substrate.

Figures 8 and 9 compare our experimental results with the Nusselt correlation. For the purpose of illustration, we have plotted data points for one porous coating thickness ($H = 254\ \mu$m) and two gravitational forces. We note that our experimental data exhibits significantly higher heat transfer coefficients as compared to the theoretical plain surface case at lower values of $T_{sat} - T_w$ for both effective body forces.

It is worth noting that our experimental results are based on saturated steam with noncondensables while the Nusselt correlation predicts laminar film condensation in pure saturated steam. Previous investigations [21] have shown that small volume amounts of noncondensables can decrease heat transfer coefficients by as much as 50%. An estimate of the mass fraction of noncondensables was made even though the noncondensable concentration during the experiments did not vary. A comparison of our measured results for the plain copper surface with the numerical results of Minkowycz and Sparrow [22] indicates a bulk mass fraction estimate of the noncondensable gas of 0.001–0.010. Therefore, we would expect the degree of enhancement for our experimental measurements to be even more consequential if these tests used pure saturated steam.

From Figs. 8 and 9, it is seen that the numerical predictions of Renken and Aboye [14] are in relative

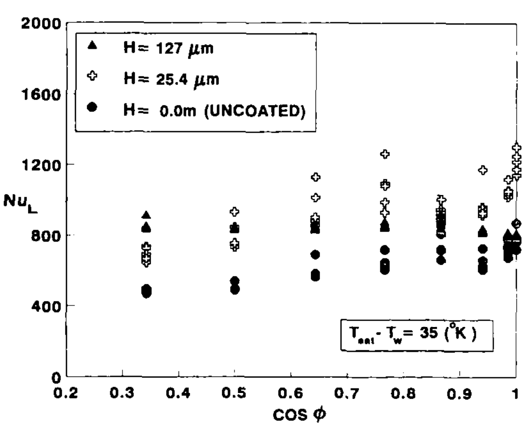

FIG. 6. Experimental results of average Nusselt number versus inclination for three porous substrate thicknesses at a surface subcooling temperature difference of 35 K.

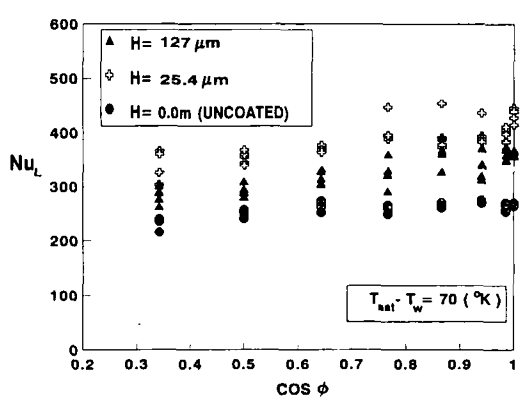

FIG. 7. Experimental results of average Nusselt number versus inclination for three porous substrate thicknesses at a surface subcooling temperature difference of 70 K.

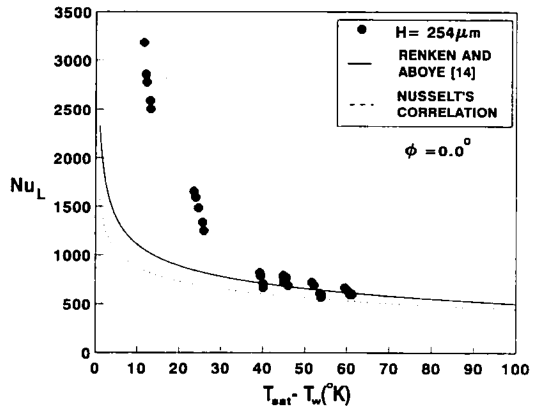

FIG. 8. Comparison of experimental results with the Nusselt theory and Renken and Aboye [14] for $\phi = 0°$.

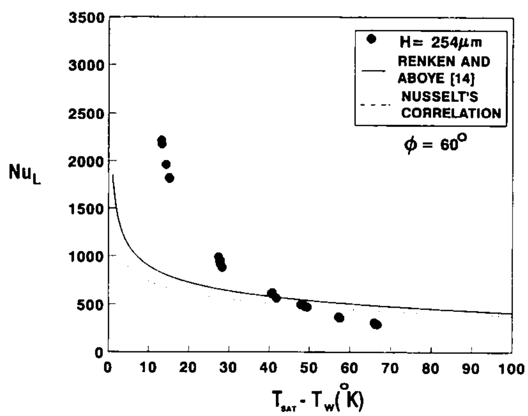

FIG. 9. Comparison of experimental results with the Nusselt theory and Renken and Aboye [14] for $\phi = 60°$.

agreement with the experimental data. It is illustrated that an increase in surface subcooling and a decrease in angle of inclination produces an increase in heat transfer performance. The deviations of the experimental values from the model tend to be higher at lower subcooling temperatures where the liquid film thickness is at its minimum. The model assumes the absence of noncondensable gas and two regions of condensate that are not contaminated with vapor. The test surface used here contained a thicker porous coating ($H = 254$ μm) than in Figs. 6 and 7. During the experimental runs, it was observed visually for small values of $T_{sat} - T_w$ that particular sections of the surface were not completely saturated. This resulted in a two-phase porous layer where capillary forces are important and should be included in the model.

CONCLUSIONS

Experiments were conducted to measure the film condensation promotion within thin inclined porous coatings. The results were compared with experimental data for an uncoated surface, the Nusselt correlation and a recent film condensation model of a porous/fluid composite system. It was demonstrated that the thin porous coating (i) produces significant increases (as much as 56%) in heat transfer rate during condensation as compared to a plain surface, (ii) exhibits a marked degree of enhancement even with noncondensables as compared to the ideal theoretical case of film condensation on a plain surface, and (iii) experimental data is in relative agreement with the Renken and Aboye [14] predictions over the range of test conditions.

Acknowledgements—Financial support for this research provided by the National Science Foundaton through Grant No. CTS-8909410 is greatly appreciated. The authors would like to thank Mr Vincent Lanza, Metco Corporation for supplying the thermal spray coating, Marcelle Gillott, UWM Department of Biological Sciences, Electron Microscopy Laboratory, and Genbe Menzel, UWM CEAS Photographer.

REFERENCES

1. W. Nusselt, Die oberflachenkondensation des wasser dampfes, *Zeitschrift des Vereins Deutshes Ingenieure* **60,** 541–575 (1916).
2. W. D. Woodruff and J. W. Westwater, Stream condensation on electroplated gold: effect of plating thickness, *Int. J. Heat Mass Transfer* **22,** 629–632 (1979).
3. W. D. Woodruff and J. W. Westwater, Steam condensation on various gold surfaces, *J. Heat Transfer* **103,** 685–692 (1981).
4. T. C. Carnavos, An experimental study: condensing R-11 on augmented tubes, ASME Paper 80-HT-54, 1–7 (1980).
5. V. G. Rifert, Y. Y. Trokoz and V. Y. Zadiraka, Enhancement of heat transfer in condensation of ammonia vapor on a bundle of wire-finned tubes, *Heat Transfer—Soviet Res.* **16,** 36–41 (1984).
6. P. J. Marto, Recent progress in enhancing film condensation heat transfer on horizontal tubes, *Proc. 8th Int. Heat Transfer Conf.*, pp. 161–170 (1986).
7. P. J. Marto, E. Mitrou, A. S. Wanniarachi and M. Katsuta, Film condensation of steam on a horizontal wire-wrapped tube, *Proc. 2nd ASME–JSME Thermal Engineering Joint Conf.*, pp. 509–516 (1987).
8. K. K. Yau, J. R. Cooper and J. W. Rose, 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, 9–14 December, pp. 151–156 (1984).
9. A. Shekarriz and O. A. Plumb, A theoretical study of the enhancement of filmwise condensation using porous fins, ASME Paper No. 86-HT-31, *Proc. Joint AIAA/ASME Thermophysics and Heat Transfer Conf.*, Boston, Massachusetts, 2–4 June, 1–5 (1986).
10. A. Shekarriz and O. A. Plumb, Enhancement of film condensation using porous fins, *J. Thermophys. Heat Transfer* **3,** 309–314 (1989).
11. K. M. Holden, A. S. Wanniarachchi, P. J. Marto, D. H. Boone and J. W. Rose, The use of organic coatings to promote dropwise condensation of steam, *J. Heat Transfer* **109,** 768–774 (1987).
12. M. Izumi, N. Yamakawa, T. Shinmura, Y. Isobe, S. Ohtani and J. W. Westwater, Drop and filmwise condensation on horizontally scratched rough surfaces, *Heat Transfer—Japanese Res.* **18,** 1–14 (1989).
13. K. J. Renken, D. J. Soltykiewicz and D. Poulikakos, A study of laminar film condensation on a vertical surface

with a porous coating, *Int. Commun. Heat Mass Transfer* **16,** 181–192 (1989).
14. K. J. Renken and M. Aboye, Analysis of enhanced film condensation within inclined thin porous-layer coated surfaces, *Int. J. Heat Fluid Flow* (in press).
15. K. J. Renken and C. D. Mueller, Measurements of enhanced film condensation utilizing a porous metallic coating, *J. Thermophys. Heat Transfer* 7(1), (January–March 1993).
16. M. Kaviany, *Principles of Heat Transfer in Porous Media*, pp. 561–604. Springer-Verlag, New York (1991).
17. D. A. Nield and A. Bejan, *Convection in Porous Media*, pp. 305–344. Springer-Verlag, New York (1992).
18. J. Bear, *Dynamics of Fluids in Porous Media*, pp. 119–189. American Elsevier Publishing Company, Inc., New York (1972).
19. W. M. Rohsenow, Heat transfer and temperature distribution in laminar film condensation, *Trans. ASME* **78,** 1645–1648 (1956).
20. M. Aboye, Theoretical and experimental investigation of film condensation heat transfer enhancement using a highly conductive porous coating, M.S. Thesis, University of Wisconsin-Milwaukee, Milwaukee, Wisconsin (1992).
21. L. C. Burmeister, *Convective Heat Transfer*, pp. 670–674. Wiley, New York (1983).
22. W. J. Minkowycz and E. M. Sparrow, Condensation heat transfer in the presence of noncondensables, interfacial resistance, superheating, variable properties, and diffusion, *Int. J. Heat Mass Transfer* **9,** 1125–1144 (1966).