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Forced convection steam condensation experiments within thin porous coatings

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Abstract—This paper reports the experimental findings of forced convective condensation heat transfer on plates with a thin porous coating. The composite system consists of a relatively thin, highly conductive and permeable porous coating bonded to a cold isothermal condensing surface which is placed parallel to saturated steam flow. The porous coatings ranged in thickness from 0 to 254 μm and are used as a passive technique for heat transfer augmentation. The inclined and isothermal plates were exposed to bulk vapor velocities of 0.9–6.5 m s^{-1} resulting in free-steam Reynolds numbers of 5.0×10^4–2.7×10^6. The forced convective heat transfer coefficients show a substantial heat transfer enhancement (250% increase) when compared to noncoated surface under the prescribed conditions. The experimental data exhibits as much as a 600% increase over previously measured free convection condensation results. The effect of subcooling temperature and plate inclination (vertical to horizontal) is also documented. The results of this study provide valuable fundamental predictions of condensation heat transfer that can be used in a number of thermal engineering applications which require heat transfer enhancement or retardation of an impermeable surface. Copyright © 1996 Elsevier Science Ltd.

1. INTRODUCTION

Condensation heat transfer has been researched extensively since the original work of Nusselt [1], who had analysed laminar film condensation on vertical plates under free convective conditions. Subsequently, because of its predominance, the effects of forced convective condensation have been given necessary attention. Investigators such as Cess [2] and Koh [3] have analysed flowing vapor over flat plates where the motion of the condensate film is influenced only by the *sweeping* effect of the passing vapor. Recent interest in the enhancement of phase-change heat transfer (e.g. condensation) has been stimulated by the analysis and application of heat transfer promotion in heat pipes, electronic cooling equipment, mechanical refrigeration and nuclear reactor cooling.

Previous research has produced a great variety of heat transfer enhancement techniques, involving both *active* and *passive* methods. Included in these methods are various techniques to promote dropwise condensation. One such study by Woodruff and Westwater [4] discovered that gold surfaces promoted condensation of steam. A more recent paper by Zhao *et al.* [5] documented a surface treatment technique utilizing ion-plating technology, whereby the condensing surface energy is reduced by implanting certain elements (i.e. Cr, N and C) to cause the onset of dropwise formation. A study by Yamauchi *et al.* [6] reported increased heat transfer by the use of gold-deposited copper surfaces. Furthermore, Izumi *et al.* [7] have investigated the effects of artificially scratched roughened surfaces during mixed condensation.

An alternate method for the enhancement of condensation heat transfer was first introduced by Renken *et al.* [8]. Further experimental and modeling work was performed [9–11]. The method involves the thermal spraying of copper surfaces with an Al–Cu–Br composition which results in a highly conductive yet permeable porous coating which can be described as a thin porous-layer metallic coated surface. The majority of heat transfer work involving these types of surfaces has been devoted to boiling and evaporation and is summarized in refs. [12, 13]. Thome [14] and Webb [15] have also documented work that involved enhancement techniques used in boiling heat transfer that are similar to the current technique.

It is the objective of this paper to report the results of forced convective steam condensation heat transfer experiments that utilized a thin porous coating on isothermal plates. The heat transfer enhancement due to the forced steam flow is compared to free convection data and to the results of other analogous heat transfer enhanced surfaces. The variation of free-stream steam velocity (Reynolds number), subcooling temperatures (Jakob number), and the effective body force on heat transfer enhancement is also documented. This documentation provides fundamental information in the prediction of heat transfer characteristics for complex configurations. Such potential applications include: thermal insulations, electronic cooling systems, direct contact heat exchangers, sol-

NOMENCLATURE

C_p	specific heat [J kg^{-1}K^{-1}]
f	average surface roughness [μm]
g	gravitational acceleration [m s^{-2}]
h_{fg}	heat of vaporization [J kg^{-1}]
h_L	average heat transfer coefficient [W m^{-2}K^{-1}]
H	thickness of porous coating [μm]
Ja	Jakob number, $C_p(T_\infty - T_w)/h_{fg}$
k	thermal conductivity [W m^{-1} K^{-1}]
K	permeability [m^{-2}]
L	test plate length [m]
Nu_L	average Nusselt number, $h_L L/k_f$
q''_L	average heat flux [W m^{-2}]
Ra	Rayleigh number, $g \cos\phi\ \beta\ (T_\infty - T_w)\ L^3/\nu \cdot \alpha$
Re_L	Reynolds number, $U_\infty L/\nu$
t	test plate thickness [m]
T_w	test plate wall temperature [K]
T_∞	ambient vapor temperature [K]
ΔT_{sub}	subcooling temperature, $T_\infty - T_w$[K]
U_∞	free stream (bulk) vapor velocity [m s^{-1}]
x	vertical coordinate [m]
y	horizontal coordinate [m].

Greek symbols

α	thermal diffusivity [m^2 s^{-1}]
β	thermal expansion coefficient [K^{-1}]
δ	condensate boundary layer thickness [m]
ε	porosity of porous matrix
ν	kinematic viscosity [m^2 s^{-1}]
ϕ	test plate inclination angle.

idification of castings, drying processes in the textiles, pulp, paper and food industries, etc.

2. EXPERIMENTAL APPARATUS AND PROCEDURES

The following is a description of the experimental systems and procedures used in the investigation. Complete details of the experimental apparatus and preparations are contained in [11].

2.1. *Experimental apparatus*

Figure 1 illustrates the problem of interest. More specifically, saturated vapor at a free-stream velocity, U_∞ and a temperature, T_∞, flows over an isothermal condensing surface of temperature, T_w. Affixed to the surface is a thin porous coating of thickness, H, permeability, K and porosity, ε.

Figure 2 shows a general schematic of the entire experimental apparatus used to make the heat transfer measurements of forced convective steam condensation on inclined, plain and thin porous flat plates. The steam was supplied via a 15.2 mm (6 in) diameter main at 690 kPa (100 psig) where it was regulated down to 138 kPa (20 psig). The steam passed through a vortex mass flow meter, a needle valve for mass flow adjustments, and then to the flow channel where the test plates were positioned.

As shown in Fig. 3, the flow channel, which was made of 304-stainless steel, was comprised of four sections: the upstream calming/straightening section, the test section, the downstream flow channel, and the exiting converging nozzle. The main flow area dimensions were 15.2 cm × 7.60 cm. The upstream section (0.9 m long) contained a number of stainless steel and aluminum honeycomb inserts to maintain uniform flow prior to the 23 cm long test section. The hollow downstream section was 46 cm long. The upstream and downstream sections were designed to be interchangeable to accommodate gravity-opposing flow as well as the current, gravity-aiding flow case. The entire flow channel was insulated with a double layer of standard 5.1 cm (2 in) thick blanket insulation. At the flow channel's outlet was a converging nozzle which was connected to a shell-and-tube heat exchanger/condenser. Excess steam that exited the flow channel was passed through a 2.4 m flexible steam hose, and then into the tube side of a single-pass (91 cm) condenser. Cooling water ran from the laboratory tap, counter-current to the vapor flow on the condenser's shell side.

As shown in Fig. 3, the test section was equipped with a 76.2 cm (3 in) diameter non-fogging, stainless steel flanged sight glass so that visual observations, high speed video, and conventional photography of the forced convective film condensation process could be observed and recorded. Also shown in Fig. 3 is a right-angle speed reducer, which allowed for the flow channel to be rotated from 0° to 90° (vertical to horizontal positions). An electronic LCD angle-lever indicator was used to measure inclination angles. The entire apparatus: the flow channel assembly, the inclination system, and other sub-systems were supported by a 3.25 m high, 66 cm × 66 cm angle-iron frame, that stood atop a 1.2 m × 1.2 m wooden platform.

During the experiments, three test plates were used. Each measured 76.2 mm × 127 mm × 12.7 mm thick (3 in × 5 in × 0.5 in). The *reference plate* (noncoated), made of oxygen free 101 copper, contained a smooth, mirror-like finish while the remaining two plates (identical to the reference plate) were thermal sprayed with a uniform coating thickness of an aluminum, bronze, and copper mixture. Figure 4 displays a SEM photograph of a thin porous coated plate (H = 254

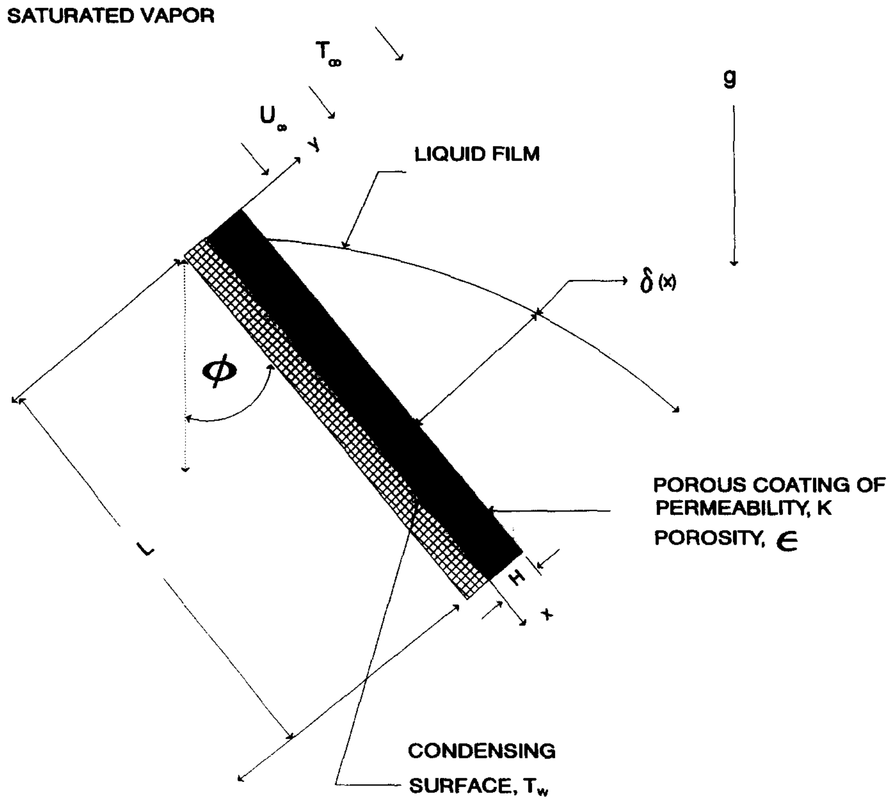

Fig. 1. Physical description of condensation phenomenon.

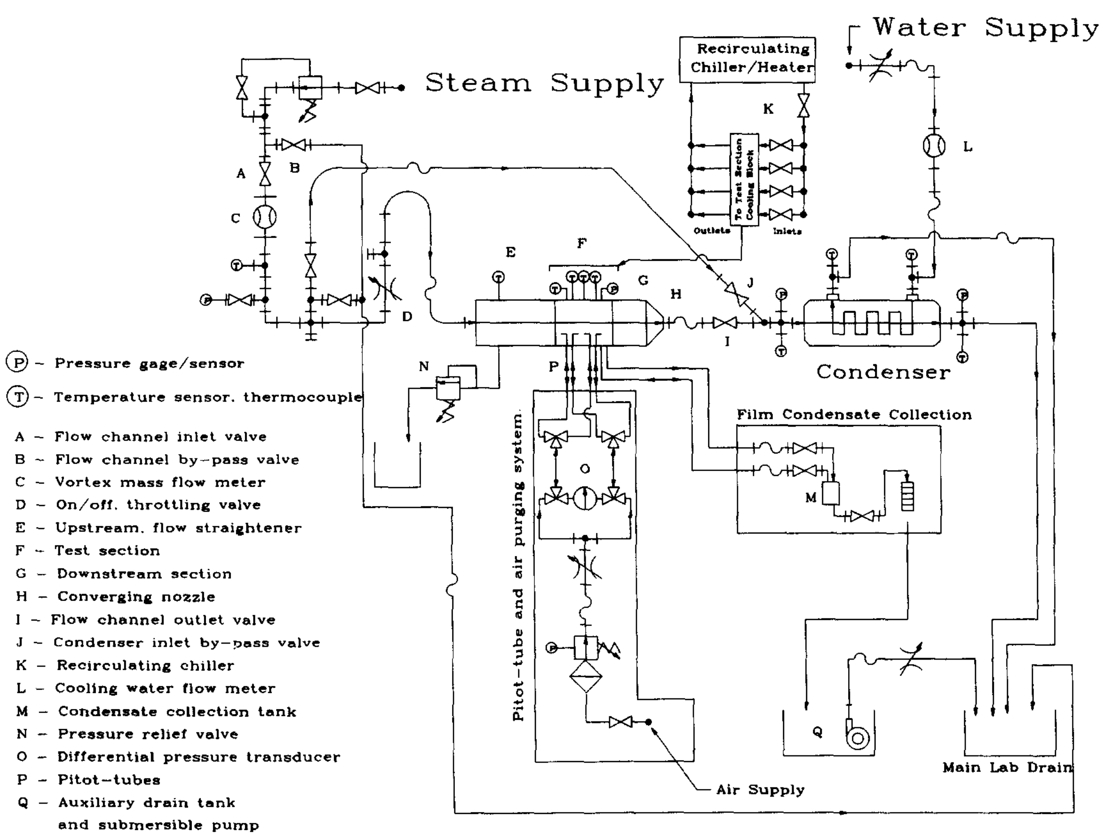

Fig. 2. Schematic of experimental apparatus.

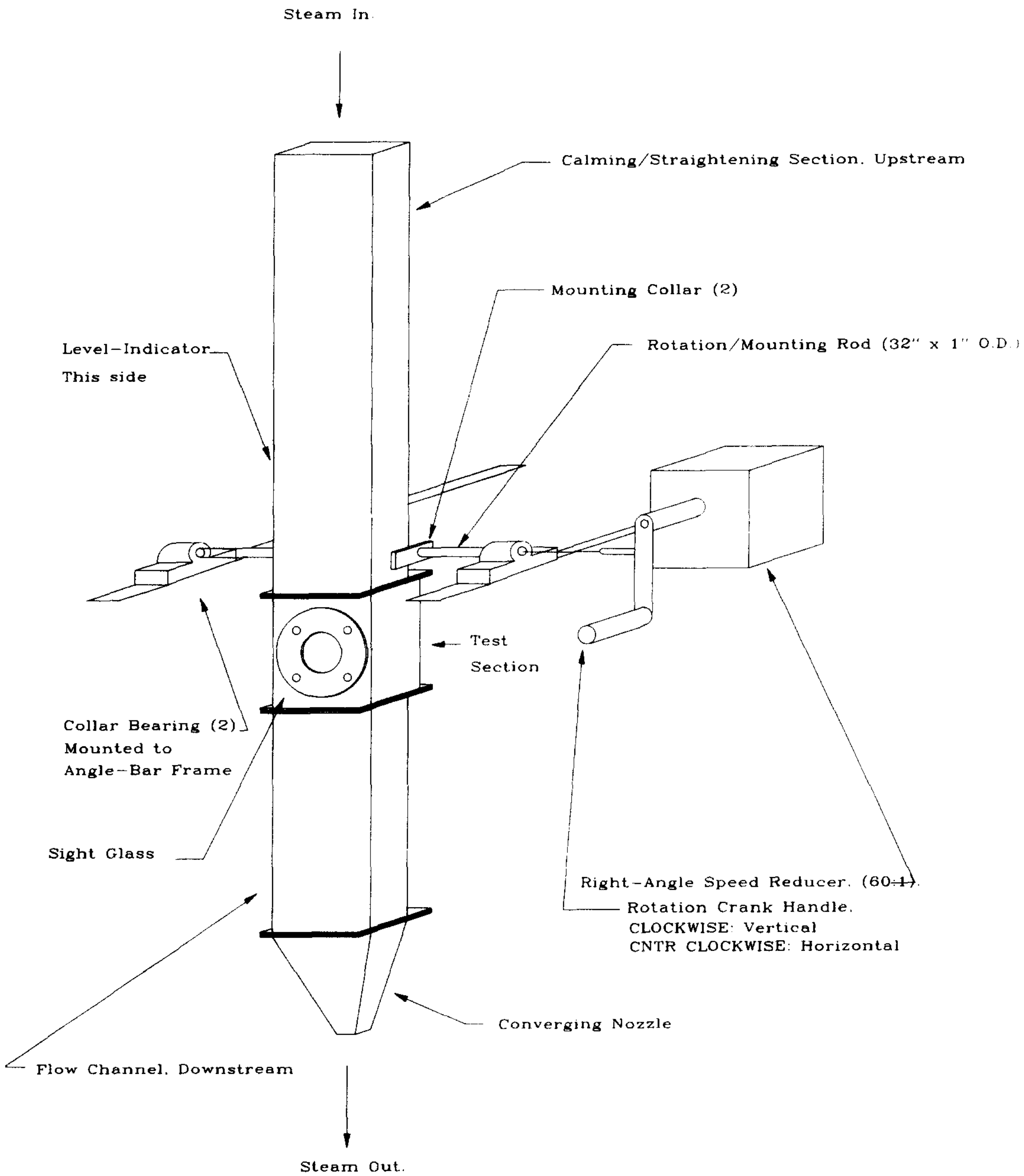

Fig. 3. Details of flow channel and plate inclination system.

μm) at $50\times$ magnification. The porous structure resembles a roughened surface that simulates a porous–fluid composite system. The thin porous-layer coating thicknesses ranged in value from 25.4 to 254 μm. The porosity and permeability of the test surfaces were estimated utilizing scanning electron microscopy. The range in values of porosity and permeability were: 45–49% and 2.4–3.6×10^{-9}, respectively.

The test plates were indirectly cooled by the use of a recirculating chiller which supplied a cooled ethylene glycol–water mixture (60/40% by volume) to the underside of a coolant block. The coolant block, which had dimensions of 76.2 mm $\times$ 127 mm $\times$ 19.1 mm thick (3 in $\times$ 5 in $\times$ 1.5 in), was made of oxygen free 101 copper and contained four insulated coolant channels to provide adequate heat transfer and subsequent condensation on the test surfaces. Coolant passed through four independent ball valves which controlled flow into the four isolated sections of the coolant block. This assured that the test plate's surface was maintained at the uniform temperature, T_w. After exiting the coolant block, coolant flowed back into the recirculating chiller's 19 l coolant reservoir after being monitored by a paddle-wheel flow sensor. The recirculating chiller provided a maximum volumetric

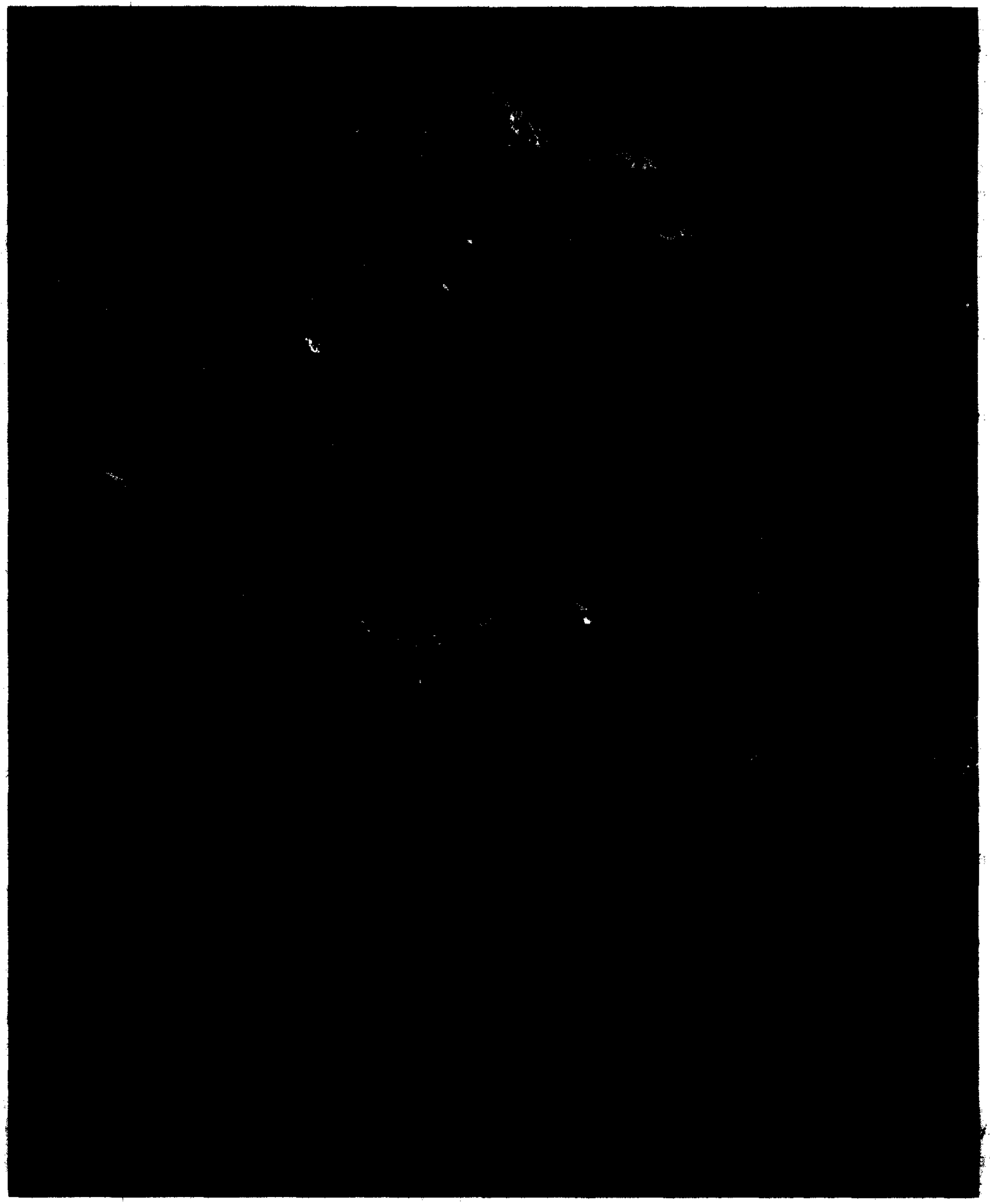

Fig. 4. SEM photograph of a 25.4 μm thick thermal spray coating.

flow rate of 53 LPM and had a cooling capacity of 9.6 kW at 20°C.

Fifteen type-T thermocouples were embedded into 3.2 mm o.d. thermowells that protruded 12 mm into the back of the 12.7 mm thick test plates. These thermocouples measured the surface temperatures, while seven thermocouples placed between the test plate and the coolant block measured the interface temperatures. These temperatures yielded the temperature differences across the plate which were utilized in Fourier's Law and Newton's law of cooling to determine the condensation heat transfer coefficient, h_L.

Condensation that formed on the test plate was collected by the condensate collection system which incorporated a main drain line (12.7 mm dia.), a bleed line (6.4 mm dia.), and a 1500 ml collection tank. Each of these two lines contained a ball valve so that condensate was not collected between data runs, when condensate was measured. Also, the collection tank had a main exit valve that was opened after the two line-valves were closed for condensate measurement. This sub-system provided an accurate verification method (within 10%) in estimating the heat transfer coefficient.

Velocity measurements of the steam over the test plates were determined by two independent methods. The first utilized the volumetric flow rate of steam (measured by the vortex meter) and the flow channel's flow area to obtain a bulk vapor velocity. The second method incorporated two stainless steel pitot tubes and a bakeable differential pressure transducer to measure the velocity directly over the test plates. The pitot tubes were 12.7 cm (5 in) long with 12.7 mm (0.5 in) o.d. bases that were secured in the test section by thermocouple fittings. Each S-type probe had two 3.2 mm (0.125 in) o.d. sensing tubes, one for the upstream stagnation pressure and the other for the downstream static pressure. The pitot tubes were connected to a 3-way ball valve to allow compressed-air purging of the steam condensation that filled the lines during the experimental runs. Cessation of in-tube condensation was accomplished by wrapping the stainless steel tubes with electrical heating tape.

Other instrumentation and sensors within the system included a pressure transducer which read the ambient test section vapor pressure and four thermocouple probes which measured the flowing vapor temperatures. The rigid probes were stainless steel-sheathed, type-T thermocouples that measured 15.2 mm in length, and 1.6 mm in outer diameter (6 in $\times$ 1/16 in). A pressure relief valve was installed for safety and was set for a 103.4 kPa cracking pressure. All condenser inlet and outlet temperatures were also measured by thermocouple probes, while cooling water volumetric flow rates were determined by a paddle-wheel transducer.

Data collection was made via a state-of-the-art PC-data acquisition system (PC-DAS) which contained high speed relay multiplexers for the various sensors in the experimental setup. A commercial data acquisition software package was used to program, record, and display the sensor signals and to format the raw data. Instantaneous calculations and graphical displays of the average heat flux and average Nusselt number were accomplished by popular spreadsheet and plotting software packages, respectively.

2.2. *Experimental procedures*

After each test plate had been instrumented with thermocouples, they were mounted to the cooling block. This was done by four type 10-24 brass screws and by applying a thin layer of high-temperature, highly conductive silicone paste to the back of the test plate. This allowed uniform thermal communication between the two surfaces and improved thermocouple response time. The paste was allowed to set overnight and the test plates were thoroughly cleaned. The uncoated plate was soaked with a soapy-water mixture and polished with ultra-fine emery paper to a mirror-like finish, while the thermal sprayed plates were cleaned with hydrochloric acid–water solution and a semi-rigid brush. The bakeable differential pressure transducer and the trace-line heating tapes were turned on. The condensate collection receptacle and the collection reservoir were flushed while the steam-side sight glass was washed and sealed.

Next, the plate inclination was adjusted by rotating the crank-handle to the desired angle. The main air supply for the pitot tube purging was switched on and the temperatures of the heated trace-lines and differential pressure transducer were monitored on the PC-DAS. The condenser cooling water was turned on and the recirculating chiller was set to the desired coolant temperature, henceforth plate surface temperature. After system-wide constant conditions were achieved, the main steam line valve was slowly opened. After a 5–10 min condensate/steam flush through an auxiliary hose to the lab drain, the flow channelling throttling valve was opened to allow steam to enter the flow channel. The pressure-regulating valve was adjusted and the temperature probe readings were monitored to determine when saturation conditions prevailed.

The pressurized steam entered the flow channel where it diffused in the calming section. It then proceeded to the test section where condensation occurred on the test plates. The excess steam was then discarded through the downstream section, the converging nozzle, and the external condenser. After approximately 20–25 min, steady-state conditions were reached and data logging was initiated. After each test run, the condensate that was collected during the run was drained from its reservoir and measured in a graduated cylinder. The next independent variable setting (test plate, surface temperature, inclination angle, etc.) was then designated. Between consecutive run series, the test section cover was removed and the test plate was cleaned.

3. DISCUSSION OF EXPERIMENTAL RESULTS AND VISUAL OBSERVATIONS

Figure 5 is a photograph through the test section's sight glass of the actual forced convective film condensation phenomenon within a 25.4-μm thick porous coating. As shown in the photo, there exists an extremely thin condensate layer which covers the length of the test plate due to the flowing vapor. Visual observations of the condensation process were made throughout all test runs so that irregularities such as dropwise condensation could be documented.

In Fig. 6, the variance of the average Nusselt number with porous coating thickness for a range of vapor velocities (Reynolds number) and a relatively constant Jakob number (subcooling temperature) is documented. As expected, we see that the average heat transfer coefficient increases with faster vapor flows or higher Reynolds numbers in each of the cases for the conditions tested. It is quite obvious that the very thin metallic coating exhibited the best condensation heat transfer for the experienced conditions. There exists an overall average increase of 259% when one compares the noncoated ($H = 0.0$ μm) plate with the thinnest coated plate ($H = 25.4$ μm). Also there is a significant decrease in Nu_L as the metallic layer increases from 25.4 to 254 μm. This result suggests that the thin porous-layer may act as a thermal resistance at certain thickness. It is hypothesized that this

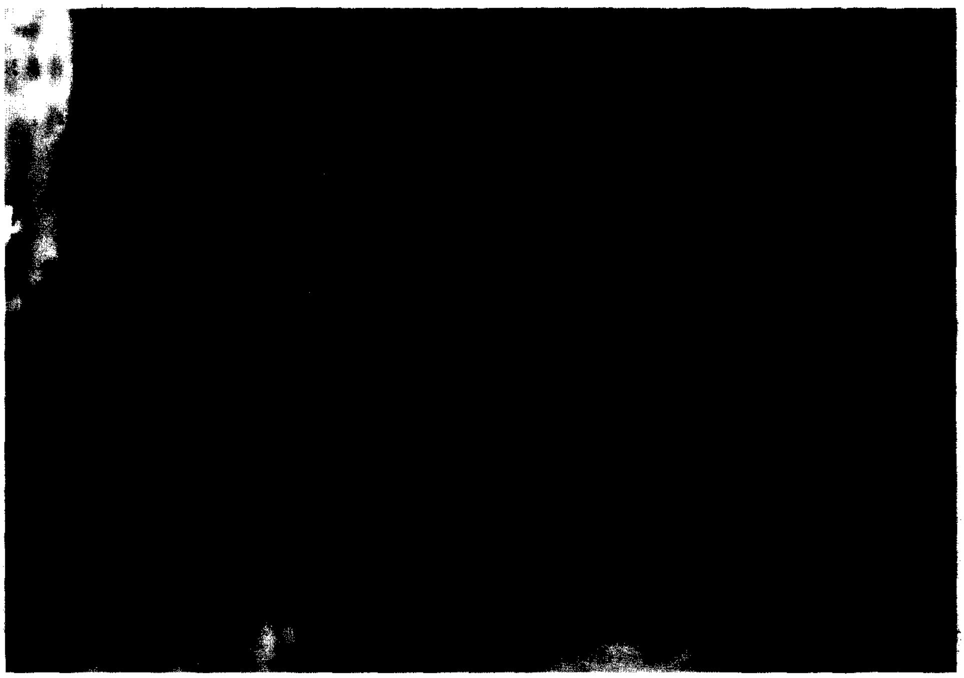

Fig. 5. Photo of forced convective condensation within a 25.4 μm thick porous coating.

critical thickness may vary depending on the flow and the make-up of the porous material.

A comparison of the present experiments with the limiting case of free convection within thin porous-layer metallic coatings is made in Fig. 7. Here, the results of Renken and Aboye [9] for the same test plate thickness and inclination angle are contrasted to current data with a Reynolds number of the order of magnitude of 10^6. Again as expected, the Nu_L increases by almost tenfold over the tested range of subcooling temperatures.

The variation of the average surface heat flux with the Jakob number for a Re of 2.7×10^6 is displayed in

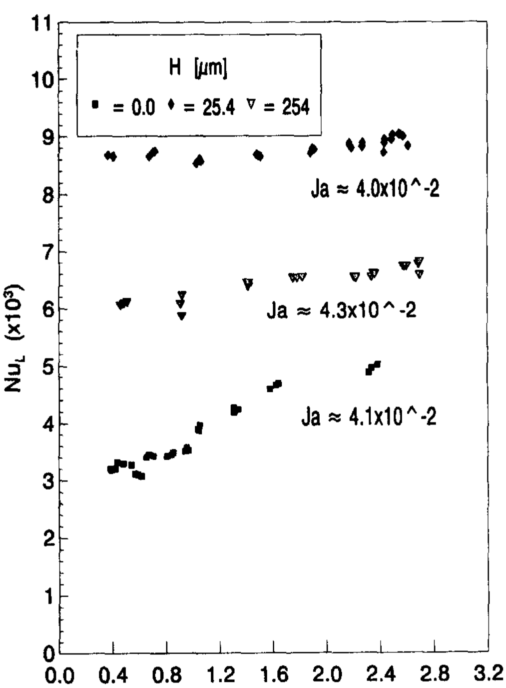

Fig. 6. Experimental results of average Nu_L vs Reynolds number for three porous coating thicknesses.

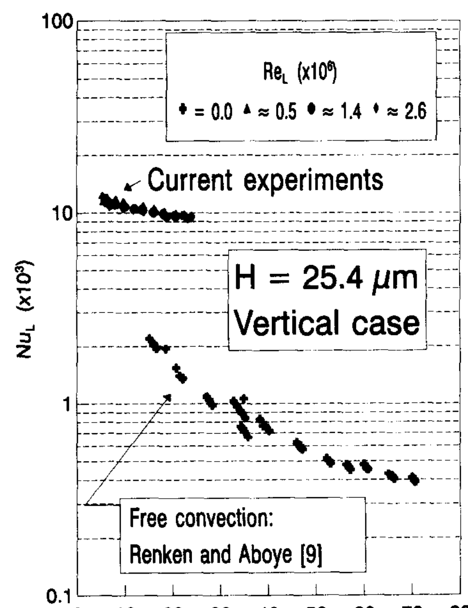

Fig. 7. Comparison of experimental results with free convection condensation data of Renken and Aboye [9].

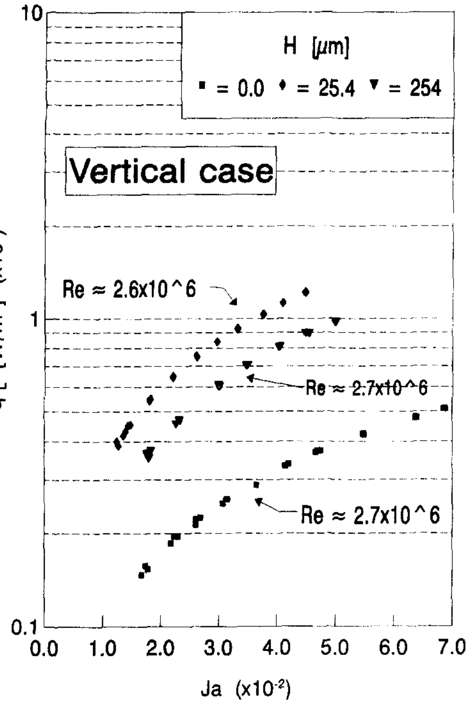

Fig. 8. Experimental results of average heat flux vs Jakob number for three porous coating thicknesses.

Fig. 8. Here, there exists smooth increases in the three test cases as the Ja increases for the vertical test runs. There exists an overall average increase in q''_L of 144% and 111% when comparing the plain plate to the thin ($H = 25.4$ μm) and thicker ($H = 254$ μm) coatings, respectively.

In Fig. 9, the current experimental results were compared to the results of Izumi *et al.* [7], who condensed

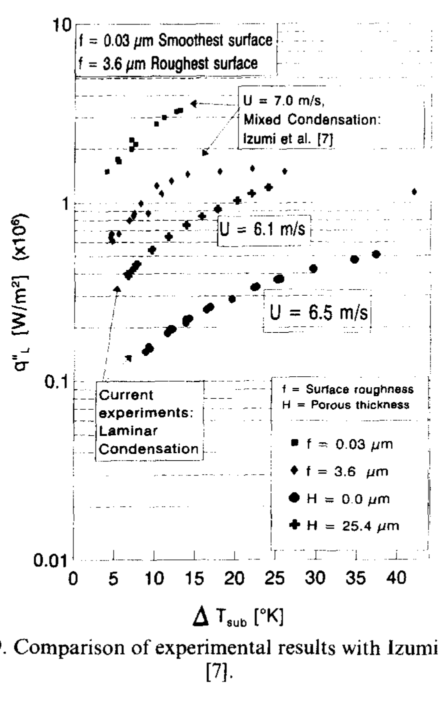

Fig. 9. Comparison of experimental results with Izumi *et al.* [7].

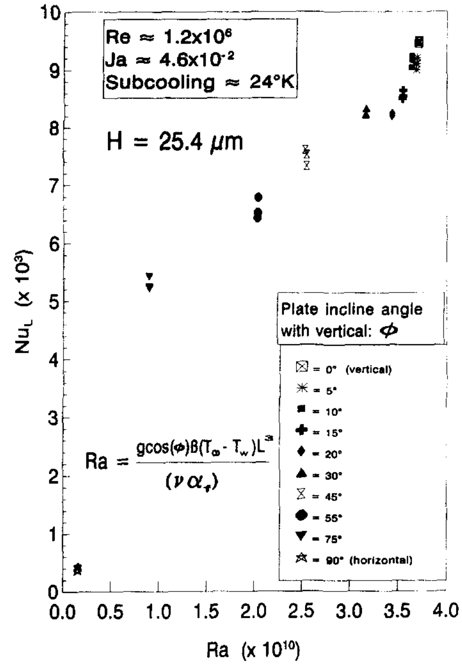

Fig. 10. Effect of plate inclination on forced convective condensation heat transfer.

steam on horizontally scratched surfaces under mixed condensation (dropwise and filmwise) conditions. Four copper test plates were used in their experiments, each were scratched with various grades of sandpaper to yield different surface roughnesses. The smoothest plate had an average surface roughness, $f = 0.03$ μm, and the roughest plate had a value of $f = 3.6$ μm. The average surface roughness was measured by traversing a contact needle-type roughness meter in the direction of condensation flow. The smooth plate of [7] gave an average increase in q''_L of 204% when compared to the present study's thin porous metallic coated plate ($H = 25.4$ μm) and a 627% increase over the current plain plate ($H = 0.0$ μm). This great increase in q''_L can be attributed to the mixed condensation conditions observed by the experiments of [7], since the differences in vapor velocities in the two experiments is almost negligible. Additionally, the current experimental data experienced laminar (non-wavy) film condensation.

The work of Slegers and Seban [16] was also reviewed to verify the accuracy of the current experiments with regards to film condensation on plain plate test surfaces. This comparison showed that the trends and experimental values were very similar. For brevity, this comparison is not displayed here, but can be viewed in [11].

As a representative case, a number of experimental test runs were made on the thin porous metallic coating ($H = 25.4$ μm) to highlight the effect of plate inclination. Figure 10 demonstrates the change in Nu_L over the full 90° rotation. Average Nusselt number is plotted against the Rayleigh number, Ra, where Ra is proportional to the effective body force, $g\cos\phi$. For these test runs, Ja and Re_L were held constant as the flow channel was rotated. An overall decrease in Nu_L of 96% occurred for the horizontal case ($\phi = 90°$) over the vertical case ($\phi = 0°$). Thus, as anticipated, the condensation heat transfer is diminished as the angle of inclinations increased. This reduction in effective body force slows the flow of liquid film which results in a thicker film layer and reduced thermal communication between the thermal engineering surface and the fluid.

It was conservatively estimated that the experimental uncertainty in the heat transfer was 7.1%, while the uncertainty in the steam velocity measurements was approximately 9.5%. Details of the error analysis are found in [11].

4. CONCLUSIONS

Experiments were conducted to document the heat transfer enhancement produced by utilizing thin porous coatings of various thicknesses on isothermal surfaces under the condition of forced convective condensation. Three test surfaces and a reference (plain) plate were exposed to a wide range of vapor velocities, surface temperatures, and plate inclinations. It was shown that the thinnest porous layer exhibited greater than a 250% increase in heat transfer as compared to the reference plate, while the forced convective steam condensation heat transfer rates overwhelmingly exceeded those of the stagnant case (in some cases by over 600%). Thus, the experiments demonstrated the advantages of using a thin-layer porous coating in a convective heat transfer environment when enhanced condensation is desired.

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REFERENCES

1. W. Nusselt, Die oberflaechenkondensation des wasserdampfes (The surface condensation of water vapor), *Z. Vereines Deutscher Ingenieure* **60,** 541–575 (1916).
2. R. D. Cess, Laminar-film condensation on a flat plate in the absence of a body force, *Z. Agnew. Math. Phys.* **11,** 426–433 (1960).
3. J. C. Y. Koh, Film condensation in a forced-convection boundary-layer flow, *Int. J. Heat Mass Transfer* **5,** 941–954 (1962).
4. D. W. Woodruff and J. W. Westwater, Steam condensation on various gold surfaces, *ASME J. Heat Transfer* **103,** 685–692 (1981).
5. Q. Zhao, D. Zhang, X. Zhu, D. Xu, Z. Lin and J. Lin, Industrial application of dropwise condensation, *Proceedings of the Ninth International Heat Transfer Conference*, Vol. 4, pp. 391–394. Hemisphere, Washington, DC (1990).
6. A. Yamauchi, S. Kumagai and T. Takeyama, Condensation heat transfer on various dropwise-filmwise coexisting surfaces, *Heat Transfer—Jap. Res* **16,** 50–67 (1987).
7. 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—Jap. Res.* **18,** 1–14 (1989).
8. 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).
9. K. J. Renken and M. Aboye, Experiments on film condensation promotion within thin inclined porous coatings, *Int. J. Heat Mass Transfer* **36,** 1347–1355 (1993).
10. K. J. Renken and C. D. Mueller, Measurements of enhanced film condensation utilizing a porous metallic coating, *J. Thermophys. Heat Transfer* **7,** 148–152 (1993).
11. M. R. Raich, Experiments on forced convection steam condensation within thin-layer porous metallic coated surfaces, M. S. Thesis, University of Wisconsin-Milwaukee, Milwaukee, WI (1994).
12. M. Kaviany, *Principles of Heat Transfer in Porous Media* (1st Edn), pp. 560–604. Springer, New York (1991).
13. D. A. Nield and A. Bejan, *Convection in Porous Media* (1st Edn), pp. 343–344. Springer, New York (1992).
14. J. R. Thome, *Enhanced Boiling Heat Transfer* (1st Edn), pp. 39–53, 70. Hemisphere, New York (1990).
15. R. L. Webb, The evolution of enhanced geometries for nucleate boiling, *Heat Transfer Engng* **2,** 46–99 (1981).
16. L. Slegers and R. A. Seban, Laminar film condensation of steam containing small concentrations of air, *Int. J. Heat Mass Transfer* **13,** 1941–1947 (1970).