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Measurements of Enhanced Film Condensation Utilizing a Porous Metallic Coating

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Condensation experiments are performed on vertical isothermal porous metallic-coated plates which are immersed in saturated steam containing noncondensable gas at atmospheric pressure. The experimental results of a porous-coated plate system demonstrate as much as a 200% increase in heat flux (represented by an average Nusselt number) as compared to the experimental data of a plain copper surface. A comparison of the experiments with a theoretical model based on porous/fluid composite condensation that has been previously published shows relative agreement. The results of this investigation highlight the potential that exists for enhanced condensation when a porous metallic coating is utilized.

Nomenclature

b = width of plate surface, m
C_p = specific heat of coolant, J/kg-K
dT = finite temperature difference across test section, K
dy = finite thickness across test section, m
h_{fg} = modified latent heat of vaporization, J/kg
h_L = average heat transfer coefficient, W/m^2-K
k = thermal conductivity, W/m-K
k_l = thermal conductivity of condensate, W/m-K
L = length of plate surface, m
m = coolant mass flow rate, kg/s
Nu_L = average Nusselt number
q = heat transfer rate, W
q'' = heat flux, W/m^2
Re = condensate Reynolds number
T_{ci} = coolant inlet temperature, K
T_{co} = coolant outlet temperature, K
T_s = saturation temperature, K
T_w = surface temperature, K
w = condensate mass flow rate, kg/s
$\delta(x)$ = film condensation thickness,
ε = porosity
μ_c = dynamic viscosity of condensate, kg/m-s

Introduction

CONDENSATION on a surface typically commences when vapor is cooled below its saturation temperature and the vapor molecules undergo a phase change. Condensation is defined as the removal of heat transfer from a system in such a manner that vapor is converted into a liquid. It is a complex problem which involves the interactions of fluid motion, heat and mass transfer, as well as surface and interfacial phenomena.

There are two forms of condensation: 1) dropwise condensation; and 2) filmwise condensation (corresponding to the analogous cases in evaporation of nucleate boiling and film boiling). Filmwise condensation is more common since it is difficult to promote dropwise condensation over long periods of time. During filmwise condensation, condensate forms a continuous film on a cooled, easily wetted surface. During dropwise condensation the condensed liquid coalesces into droplets. The latent heat released by the condensation is conducted through the liquid from the interface where the condensation occurs and is removed through the wall. Because of its importance to technology (especially thermal engineering), a considerable body of literature has been developed concerning these phenomena. A general review of published works on condensation is summarized by Refs. 1–4.

Recent interest in the enhancement of film condensation has been generated by industrial demands for more efficient and more compact-size thermal equipment. Examples of thermal engineering applications in which condensation occurs and which stand to benefit from the enhancement of heat transfer are exemplified by condensers, dehumidifiers, heat pipes, heat exchangers, cooling electronic devices, nuclear reactors, and nuclear waste disposal systems. Substantial advances have been made in film condensation enhancement by way of finned surfaces and are summarized in Refs. 5–7. This investigation focuses on the recent findings of a technique that promotes the mechanism of heat transfer enhancement. This alternative method employs a very permeable, highly conductive porous coating which is interfaced to a condensing plate. The majority of studies using a porous-coated surface to augment thermal communication have been devoted to film, nucleate, or pool boiling. Relevant theoretical and experimental studies include those discussed in Refs. 8–21. Two unanimous conclusions drawn by these authors are that the employment of a porous surface structure 1) produces high-heat transfer coefficients with small temperature differences when compared to the classical plain surface model; and 2) makes it a very attractive and viable heat transfer augmentation alternative.

A review of the literature reveals that there has been very few studies directly related to the present investigation. A remarkable lack of published results—especially experimental—in all phases exists for the problem of condensation within a porous metallic-coated surface. A few of the more interesting studies are now described. The work of Woodruff and Westwater[22] discovered that gold surfaces promoted dropwise condensation (DWC) of steam at atmospheric pressure. Rifert et al.[23] reported that wire-finned tubes enhanced the heat transfer coefficients in condensation by approximately 60–100% as compared with the case of smooth tubes. Other works have used organic coatings[24] or scratched rough surfaces[25] to promote dropwise and filmwise condensation.

More relevant to the present study is the work of Shekarriz and Plumb.[26] Their experimental investigation focused on the effect of porous fins on film condensation on a horizontal tube. It was demonstrated that the porous fins contribute

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significantly toward thinning the liquid film, and therefore, enhancing the condensation rate. Recently, Renken et al.[27] presented results for a simple model of the present experimental research. The work investigated the possibility of heat transfer enhancement in laminar film condensation by coating a vertical plate with a conductive porous material. It was shown that a conductive coating may yield a considerable heat transfer enhancement during condensation when compared to a plain surface case.

This article presents the preliminary results of a series of experiments that demonstrate the potential of condensation enhancement utilizing a porous metallic coating. In these experiments, the condensation heat transfer data is obtained by condensing saturated steam at atmospheric pressure on vertical isothermal copper plates that are plain and porous-coated. The experimental findings are then compared to the original theory developed by Nusselt[28] and the theoretical model of Renken et al.[27]

Mathematical Model

Figure 1 shows a schematic of the problem of interest, namely film condensation within a porous metallic coating. More specifically, we have a vertical, isothermal, and condensing surface of length L which is coated with a porous metallic 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, whereas, the film condensation thickness is designated as $\delta(x)$.

Assuming a steady, laminar, incompressible, and two-dimensional flow, Renken et al.[27] utilized the Darcy-Brinkman flow model to describe the flow process in the porous metallic region while classical constant-property boundary-layer equations for film condensation were used in the pure liquid region. An energy balance was made on the porous/fluid composite and a fourth-order equation for $\delta(x)$ was generated and solved by numerical methods. The following expression for the local heat transfer coefficient in terms of problem parameters was reported:

$$h_x = k_1/[H(k_1 - 1) + \delta(x)] \tag{1}$$

From Eq. (1) the average heat transfer coefficient can be determined by numerically integrating over the plate length. For brevity, the results are not given here.

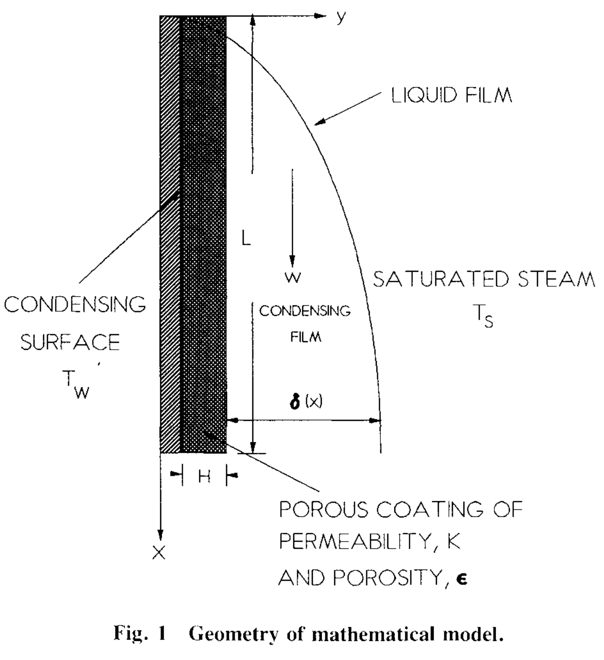

Fig. 1 Geometry of mathematical model.

Experimental Apparatus and Procedure

Figure 2 shows a schematic drawing of the experimental apparatus. The experiment consisted of several interconnected subsystems. Specifically, these were the boiler, the recirculating chiller, the data acquisition system, the condensing chamber, and the test surfaces. The boiler was a cylinder of stainless steel 304 and measured 15.2-cm o.d. and 45.7 cm. It contained a temperature-controlled 5-kW immersion heater. With the system operating at atmospheric conditions, the controller was set at maximum temperature to provide continuous production of steam. The water used in the boiler was distilled to eliminate any impurities. Make up water was added from a reservoir by an electronic control valve and a photoelectric sensor which measured the level of water (by means of a float) seen through a sight-glass indicator. Excess condensate in the condensing chamber was also drained back into the boiler by several drain ports. In this way the same conditions throughout the experiment were maintained.

The recirculating chiller had a heat removal capacity of 9 kW at 20°C, which provided ample cooling for the 5-kW heater. A throttling and pressure relief valve were used to accurately control the coolant flow rate. Maximum flow rates were used to maintain the constant surface temperature requirement. Surface temperatures between 20–80°C were achieved by adjusting the coolant set point between 10–70°C. The recirculating coolant was a mixture of 50% distilled water and 50% ethylene glycol by volume.

As shown in Fig. 2, the temperature data from the 36 thermocouples within the condensing system were processed by a Hewlett-Packard 75000 data acquisition system. Twenty-two copper-constantan, 30 American wire gauge (AWG) and Teflon® insulated thermocouples measured the temperature in the condensing test plate which had a maximum deviation of 1.0°C. In addition, four 1.59-mm-diam type-T thermocouple probes were used to measure the temperature difference within the coolant inlet and outlet ports across the plate. Steam temperatures were obtained by averaging 10 probes and thermocouples that were placed both in the boiler and around the condensing surface. Data acquisition software (LabTech Notebook) was used to program, display, and save the thermocouple outputs and to format the raw data. Calculations of the heat flux and the average Nusselt number using the temperature data was accomplished by a spreadsheet software (Lotus 1-2-3).

The condensing chamber consisted of a 45.7-cm and 76.2-cm-high Pyrex® glass bell jar. The outer surface of the bell jar was covered by 10 cm of fiberglass insulation which greatly reduced the amount of condensation lost to the jar's inner walls. Operating conditions within the chamber were limited to atmospheric pressure by the use of exhaust ports in the

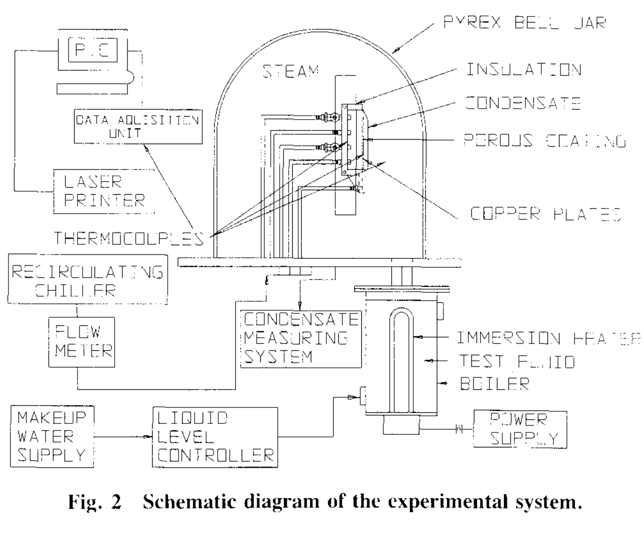

Fig. 2 Schematic diagram of the experimental system.

base plate of the bell jar. Figure 3 shows the details of the condensing surface test section. It consisted of two parts: 1) the coolant block plate; and 2) the test surface plate. The coolant block plate contained four insulated flow passages for the recirculating chiller coolant and was made of oxygen-free 101 copper. It measured 7.62 × 12.7 × 1.91 cm (width × length × thickness). The removable condensing surface was a 7.62 × 12.7 × 1.27-cm plate of oxygen-free 101 copper. The edges of the copper were insulated by 1.27-cm lexan. The 22 thermocouples that were used to measure temperatures very close (0.79 mm) to the outer surface and temperature gradients within the condensing plate were embedded by thermocouple plugs which were placed into thermocouple wells as shown in Fig. 3. The thermocouple wires were then led through channels milled into the back side of the condensing plate through the insulation and out of the steam chamber to the data acquisition unit. The condensing plate was attached to the coolant block by machine screws on the reverse side of the condensation surface. A thermally conductive silicone grease was also applied between the two surfaces to eliminate contact resistance. To assure constant surface temperatures, flow rates in the four horizontal channels of the coolant block were controlled by valves.

Three test surfaces were examined in the present investigation. The first was a plain surface which was a removable 7.62 × 12.7 × 1.27-cm oxygen-free 101 copper plate. The surface of the plate was sanded with successively higher grain emery paper to achieve a mirror finish to promote maximum condensation. The second surface tested was a copper foametal surface. This porous metallic matrix which listed a manufacturer's porosity of 0.95 and permeability of 10^{-8} m^2 was 1.52-mm in thickness. It was attached to an identical oxygen-free 101 copper plate by means of seven 2-56 × 0.125 machine screws and butt welding. The third and last surface tested was a thermal spray self-bonding aluminum bronze copper aluminum composite. The manufacturer specifications listed a porosity of 0.50 and a permeability of 10^{-10} m^2. The thickness of this coating was approximately 2.54×10^{-2} mm. Figures 4 and 5 display scanning electron microscope photographs of the copper foametal surface and the thermal spray coating, respectively. These figures are useful in verifying the manufacturer's claims of porosity and permeability as well as providing insight into the complexities that are involved in modeling a porous/fluid composite system.

The procedure to attain steady-state condensation conditions was as follows. Distilled water was supplied to the boiler by the reservoir tank. The boiler was turned on and allowed to produce steam at atmospheric pressure until a constant temperature was reached. The recirculating chiller was programmed for a coolant set point and pumped the coolant through the coolant block's four channels to create a surface subcooling temperature difference which allowed the formation of film condensation on the porous metallic coating or plain plate surface. The entire system then ran for approximately 1 h. After this time the volume within the bell jar was made up of primarily steam and the temperatures and the condensate flow rate were constant. Once this condition was reached it was considered steady state and experimental data was logged. The temperature data (for both the condensation plate and the coolant) and coolant flow rates were sent to a data acquisition file for analysis. System pressures were measured by a pressure gauge to assure atmospheric conditions. For each run an average value of condensation volume flow rate was calculated by means of a condensation

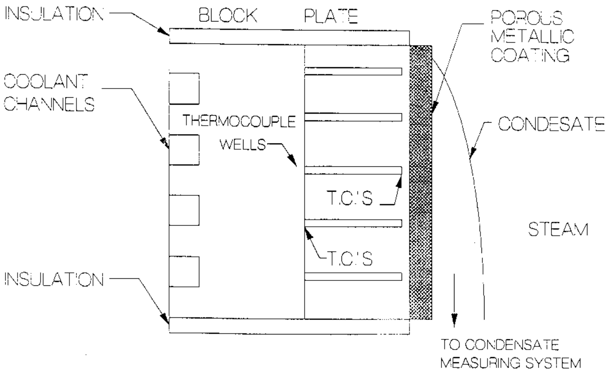

Fig. 3 Details of the condensing surface test section.

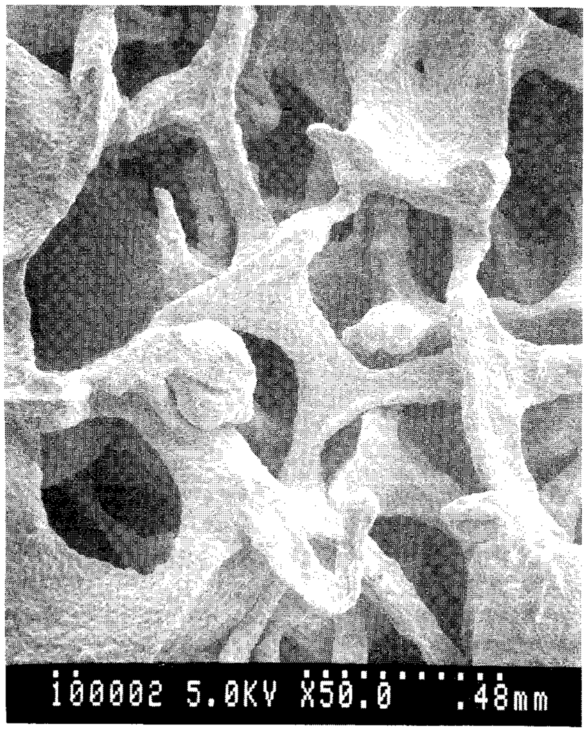

Fig. 4 Photograph of copper foametal covering. Top view of surface: 50× —0.48-mm legend length.

Fig. 5 Photograph of thermal spray self-bonding aluminum bronze copper aluminum coating. Top view of surface: 500× —0.48-μm legend length.

collection system which measured the volume of condensation from the test surface for a 600-s interval.

The experimental data provided accurate information on average heat flux (Nusselt number), surface subcooling temperature differences, and condensate mass flow rates. The Hewlett-Packard 75000 data acquisition unit and its 16-channel thermocouple relay multiplexers with thermocouple compensation and terminal block provide a $\pm 0.01°C$ sensitivity and an accuracy of $\pm 0.1°C$. The overall experimental uncertainty of the heat flux varied with the following dominate parameters in order of magnitude: 1) material and fluid properties; 2) temperature measurements; 3) coolant flow rate; 4) volume; 5) length; and 6) time measurements. A conservative estimate of experimental uncertainty in the average Nusselt number was 8%.

Discussion of Experimental Results

Experiments were conducted at atmospheric pressure and with a surface subcooling temperature difference range from 17–75°C. The experimental conditions could produce observable dropwise, transition, and filmwise condensation on the surfaces. This investigation was interested in the net effects that a porous metallic coating produced, primarily heat transfer enhancement as compared to a plain surface. The average Nusselt number and the surface subcooling temperature difference were measured and compared for a plain surface, a foam metal covered surface, and a thermal spray coating.

The heat transfer rate was measured by the following three methods. The first measured the temperature difference between the coolant inlet and outlet

$$q = mC_p(T_{co} - T_{ci}) \tag{2}$$

The second method made use of Fourier's law with the measurement of temperature gradient by way of thermocouple wells

$$q'' = -k\left(\frac{dT}{dy}\right) \tag{3}$$

The final method utilized the measurement of condensation rate and a modified heat of vaporization which accounts for liquid crossflow within the condensate[29]

$$q = mh_{fg} \tag{4}$$

These methods allowed for the heat transfer rate across the lexan insulation of the top, bottom, and sides of the test block. These values were estimated separately.

The experimental heat transfer coefficient and average Nusselt number were calculated by the following two equations:

$$q'' = h_L(T_s - T_w) \tag{5}$$

$$Nu_L = h_L L/k_l \tag{6}$$

The values of heat transfer rate from the three methods were in good agreement (within 10%). The experimental data was tested for turbulent flow conditions by use of a condensate Reynolds number defined as

$$Re = 4w/\mu_c b \tag{7}$$

For $Re \leq 30$, the film condensation flow is laminar and wave-free.[30] Our experimental data contained values of Re less than 10.5.

Quantitative experimental measurements of enhanced filmwise condensation are given in Fig. 6. The experimental values of average Nusselt number as a function of surface subcooling temperature difference ($T_s - T_w$) are exhibited. We note that the average Nusselt number decreases with increasing surface subcooling for all three test cases. For a given surface subcooling the thermal spray-coated surface performed the best as it produced the largest values of Nusselt number. At low surface subcooling we have as much as a 200% increase in heat transfer as compared to the uncoated case. At high subcooling we have approximately a 50% increase. Differences in this heat transfer enhancement can be attributed to minimal film thickness at low values of subcooling. For the foametal porous coating which was approximately 60 times as thick as the thermal spray coating, we find that the average Nusselt number compared closely with the plain surface case at all values of surface subcooling. The fact that the thinner porous coating produced higher heat transfer rates leads us to conclude that the porous region in the thicker coating was not fully saturated and contained vapor. The experimental results also quantitatively support the prediction of an optimal coating thickness and structure which was predicted by Renken et al.[27] They reported that the heat transfer enhancement produced by a porous coating would be nullified if the thickness exceeded a maximum value and henceforth acted as an insulator.

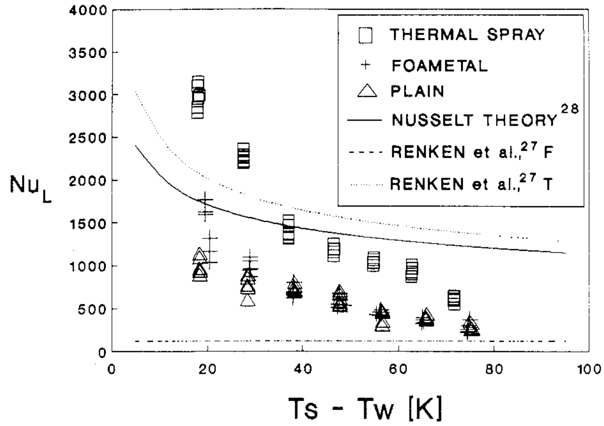

Fig. 6 Comparison of experimental heat transfer results with analytical and numerical predictions.

Also in Fig. 6 is a comparison of the experimental results with the Nusselt Theory[28] which provides an order-of-magnitude comparison with the experimental data. Nusselt's predictions were obtained under the assumption of laminar film condensation free of noncondensable gas and surface waves for a vertical plain plate. Here, the thermal spray results show a heat transfer enhancement as compared to the perfect situation (pure saturated vapor) of filmwise condensation on a plain surface for surface subcooling less than 35 K. The foametal and plain test cases demonstrate a reduction in heat transfer (as compared to the Nusselt prediction) for all surface subcooling, but follow the general trend of a decrease in Nusselt number with an increase in surface subcooling. This result is expected since other experimental results have found as much as a 50% reduction in heat transfer by the presence of only 5% noncondensable gas.[30] It is speculated that if the experiments are repeated with an evacuation of the condensing chamber, thereby reducing the noncondensable gas content, there would be an appreciable increase in condensation rate in all three test cases.

The experimental results of the porous-coated test cases are compared to a simple theoretical model. Porosity and permeability values supplied by the manufacturer as well as a volumetric-averaging technique to calculate the effective thermal conductivities of the copper foametal and the copper-aluminum thermal spray coatings were used. The predictions of an average Nusselt number for the theoretical case of a thermal spray coating (Renken et al.[27] T) show relative agreement for all values of surface subcooling. Again, the general trend is for a decrease in Nusselt number with an increase in surface subcooling, and a significant heat transfer increase relative to an uncoated surface. The solution for the foametal case (Renken et al.[27] F) exhibits poor agreement for all values of surface subcooling. The analytical prediction demonstrates an independence on surface subcooling and a very strong relationship with coating thickness. The theoretical model appears to be nonrepresentative of the real case due to the extreme thickness of the porous coating. It is hypothesized that the theoretical model has surpassed the upper limit of porous coating thickness for film condensation. The foametal coating thickness was 60 times the thickness of the thermal spray case.

It should be noted that the experimental results presented here are only preliminary. The results clearly demonstrate that the employment of a porous metallic coating to a condensing surface needs to be considered as a viable alternative for heat transfer enhancement in condensation problems. Future experimentation and modeling work is needed on the effects of porous-coating type, thickness, permeability, and thermal conductivity, as well as the presence of noncondensables in the steam.

Acknowledgments

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