Enhancement of Film Condensation Using Porous Fins

A. Shekarriz* and O. A. Plumb†
Washington State University, Pullman, Washington

The effect of porous fins on film condensation on a horizontal tube was examined experimentally. It was demonstrated that the porous fins are effective in thinning the liquid film via a surface tension driven flow and hence enhancing the condensation rate. The experiments were limited to condensation of saturated vapor with surface subcooling up to 14°C. The results obtained using 95% porosity fins and R-11 revealed that, at the optimum fin spacing, the condensation rate can be increased by as much as two times that for a plain tube. It was observed that the liquid film behaves differently for different fin spacings. At small fin spacing (3.2–6.4 mm), a thin, apparently stable film was present. The film was nearly uniform and all of the condensate drained through the porous fins. At larger fin spacing (>9.5 mm) dripping from the tube itself as well as drainage through the porous fins occurred. The maximum condensation rate occurred at a fin spacing of 9.5 mm (3/8 in.).

Nomenclature

A = surface area, m^2
h = tubeside heat transfer coefficient, W/m^2-K
h_{fg} = latent heat of vaporization, J/kg
k = thermal conductivity, $W/m-K$
K = permeability, m^2
$\dot{m}$ = condensate mass flow rate, kg/s
Nu = Nusselt number
T = temperature, °C
δ = film thickness, m
μ = dynamic viscosity, kg/m-s
σ = surface tension, N/m
θ = porosity

Subscripts

cw = cooling water
D = based on diameter
f = film
i = inside/inlet
ℓ = liquid
o = outlet
w = wall

Superscript

$-$ = *average or bulk quantity*

Introduction

THE common use of the Rankine cycle for both refrigeration and power applications has resulted in considerable research on film condensation since the original theory was developed by Nusselt in 1916. Early research focused on physical understanding of the condensation process while more recently efforts have been directed toward enhancement. The desire to develop enhanced surfaces is motivated by the need for lightweight and compact equipment. Applications include condensers for space applications, where it may be desirable to design a unit which will operate effectively in both gravity and low gravity environments, condensers for compact refrigeration cycles, and condensers for use in the vapor space in immersion cooled integrated circuits. In addition, recent advances in the development of superconducting materials may lead to transportation/aerospace applications requiring onboard cryogenic refrigerators. In these types of applications both compactness and lightweight are essential.

During film condensation the dominant resistance to heat transfer is most often the condensate film. Hence, many enhancement techniques have focused on the expeditious removal or thinning of the liquid film. During the past decade, a considerable amount of work has been conducted in this area. The most successful techniques may be classified into two groups depending upon the physical phenomena involved in removing or thinning the condensate film: 1) those involving surface tension forces, and 2) those involving skin friction forces. Recent reviews of the current literature can be found in Marto[1] and Fujii[2] for enhancement caused by surface tension and in Kutateladze and Gogonin[3] for enhancement via interfacial shear. In addition, the works of Marto et al.,[4] Rifert et al.,[5] Yau et al.,[6] Honda et al.,[7] Carnavos,[8] and Fujii et al.[9] on surface tension effects, and Honda et al.,[10] Cavallini et al.,[11] and Rose[12] on interfacial shear effects on the filmwise condensation heat transfer should be noted although this list is by no means intended to be comprehensive.

In order to form a basis for judging the results to be discussed in this paper, a brief discussion of the results obtained by other investigators for various enhancement techniques is in order. The experimental investigation of Marto et al.,[4] on wire-wrapped tubes, reveals enhancement of up to 1.8 times for condensation of steam flowing at 1.0 m/s, compared to plain tube condensation. This is rather significant, since the manufacturing cost for producing such heat transfer surfaces is substantially lower than integral fin tubes. This enhancement technique is analogous to that discussed in this paper in that a surface tension driven flow toward the wire results in a thinner film and hence enhanced condensation over a portion of the tube.

Honda et al.,[10] in their experiment with high velocity R-113 vapor flowing at 8.4 m/s over a plain tube, obtained enhancement of up to 2.9 times that for a stagnant vapor condensation. Previous to that, Kutateladze and Gogonin[3] reported enhancements of up to 4 times for R-12 vapor flowing at 5.0 m/s. Although these values are considerably higher than a stagnant vapor condensation on a single tube, application to tube bundles is limited because of liquid inundation on the lower tubes of the condenser. To provide an additional point

Received Nov. 30, 1987; presented as Paper 88-0260 at the AIAA 26th Aerospace Sciences Meeting, Reno, NV, Jan. 11-14, 1988; revision received May 3, 1988.

*Graduate Student, Department of Mechanical and Material Engineering.
†Professor and Chair, Department of Mechanical and Material Engineering.

of reference, the studies by Marto[1] indicate enhancement in the neighborhood of three to four times for steam condensing on integral fin tubes.

An alternative technique for removal of condensate film from the surface, which falls in the first class of enhancement techniques described earlier is the use of capillary porous fins in much the same way as solid fins. This method was first proposed by Plumb,[13] and further investigated by Shekarriz and Plumb.[14] Using a simple one-dimensional model, it was shown that the presence of porous fins on the condensing surface has the potential to significantly increase the condensation rate. Capillary transport in the direction normal to the surface, within the fin, coupled with continuity of the liquid phase, imposes a transverse pressure gradient on the film, which under specific conditions may cause a significant reduction in the liquid film thickness.

The objective of this paper is to present the results of a series of experiments performed to demonstrate the potential for enhancement of condensation using porous fins. For these experiments, fins were constructed from porous copper and positioned on a cylindrical horizontal condenser tube. The distance between adjacent fins was varied and the optimum fin spacing was determined for a single fin thickness. The results are compared to plain tube condensation with no imposed vapor velocity.

Experimental Apparatus and Procedure

The test apparatus is schematically illustrated in Fig. 1. The primary components of the experimental setup may be classfied into four main systems: 1) Evacuation/pressurization system—to eliminate noncondensible gases and isolate the test section from the environment. 2) Cooling system—to supply and maintain a constant temperature cooling liquid to the test section. 3) Boiling system—to provide vapor for the condenser surface. 4) A system to measure the desired temperature and the condensation rate.

The evacuation/pressurization system included an octagonal aluminum base with eight inlet/outlet ports, a 6.3 mm (1¼ in.) thick glass bell jar, two longitudinal straps to securely hold the glass jar to the aluminum base, and a vacuum pump. O-rings were used at the contact surfaces between the ports and the base. All access to the chamber took place via the inlet/outlet ports. Openings were sealed either by using the proper tube or pipe fitting or by application of a nitrite-based adhesive. The glass bell jar, measuring 305 mm (12 in.) × 457 mm (18 in.) high and containing nearly 80% of the chamber volume, allowed for full observation of the condensing surface during the experiment.

The cooling system was composed of a 450 W constant temperature bath with a thermoregulator, a 1/12 hp centrifugal pump, and a rotometer to measure coolant flow rate. Copper tubing was used to connect the preceding components in series with the condenser. The condenser was made of two concentric copper tubes. The outer tube measured 31.8 mm (1.25 in.) O.D. × 4.8 mm (3/16 in.) thick and the inner tube was 13.7 mm (0.54 in.) O.D. × 1.6 mm (1/16 in.) in thickness. Details of the construction are shown in Fig. 2. The condenser length was 203 mm (8 in.) out of which 103 mm (4 in.) was used as active surface, and the remaining 100 mm was covered by 12.7 mm (1/2 in.) thick nylon insulation. The inner tube was wrapped with 1.2 mm diam copper wire to enhance the tubeside heat transfer coefficient through both the generation of swirl and turbulence. The concentric tube design coupled with the enhancement was adopted to ensure a more uniform wall temperature in the axial direction. Finally, all of the connecting tubes and cooling lines were insulated with fiberglass insulation tape.

The temperature difference between the outlet and inlet cooling liquid was measured using a differential thermocouple, and a potentiometer accurate to ± 1 μvolt, which translates to $\pm 0.025\,^{\circ}$C. This temperature difference was less than 1.5°C for all the tests that are reported. The primary objective of this measurement was to ensure that axial temperature differences along the condenser tube were minimal. Surface temperature was measured directly through four copper-constantan thermocouples (wire diam 0.005 in.), imbedded on the surface inside 1.6 mm (1/16 in.) deep holes. These holes were drilled to be slightly larger in diameter than the thermocouple beads (~1 mm). The holes were then filled with nitrite-based adhesive to keep the thermocouple wires in place and the condensate away from the thermocouple beads. Nitrite-based adhesive was used because of its compatibility with the R-11 environment. The axial location of four surface thermocouples is indicated in Fig. 2. The four thermocouples were

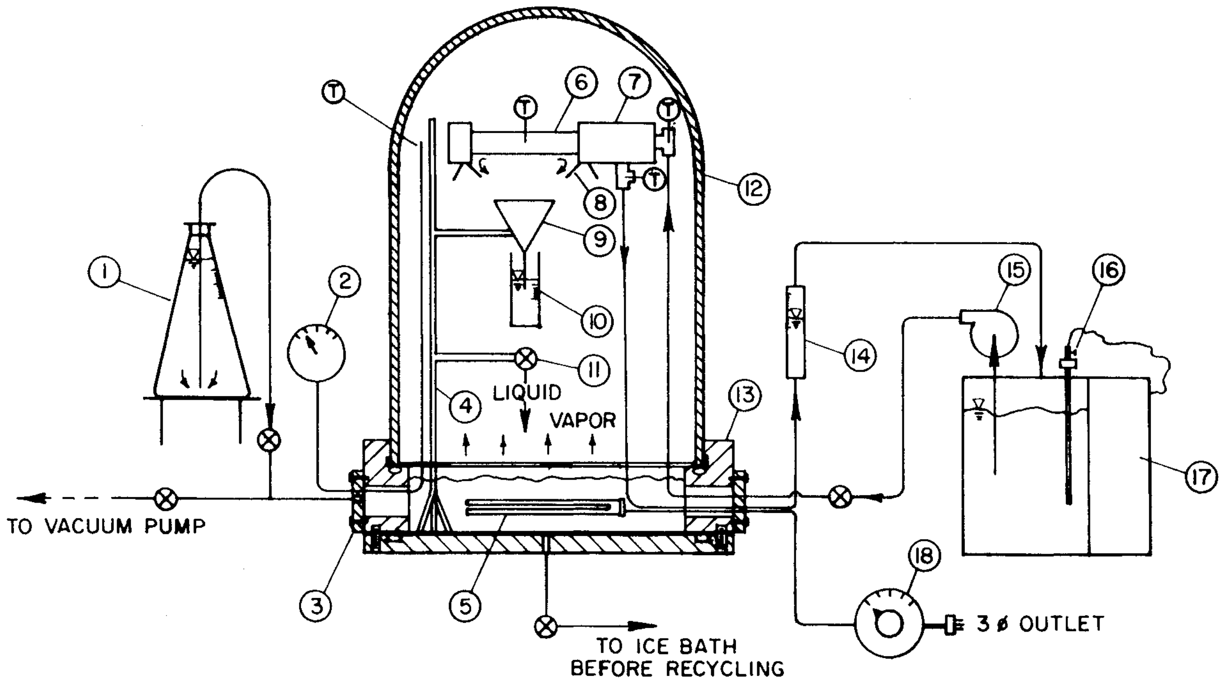


Fig. 1 Schematic diagram of the experimental apparatus.

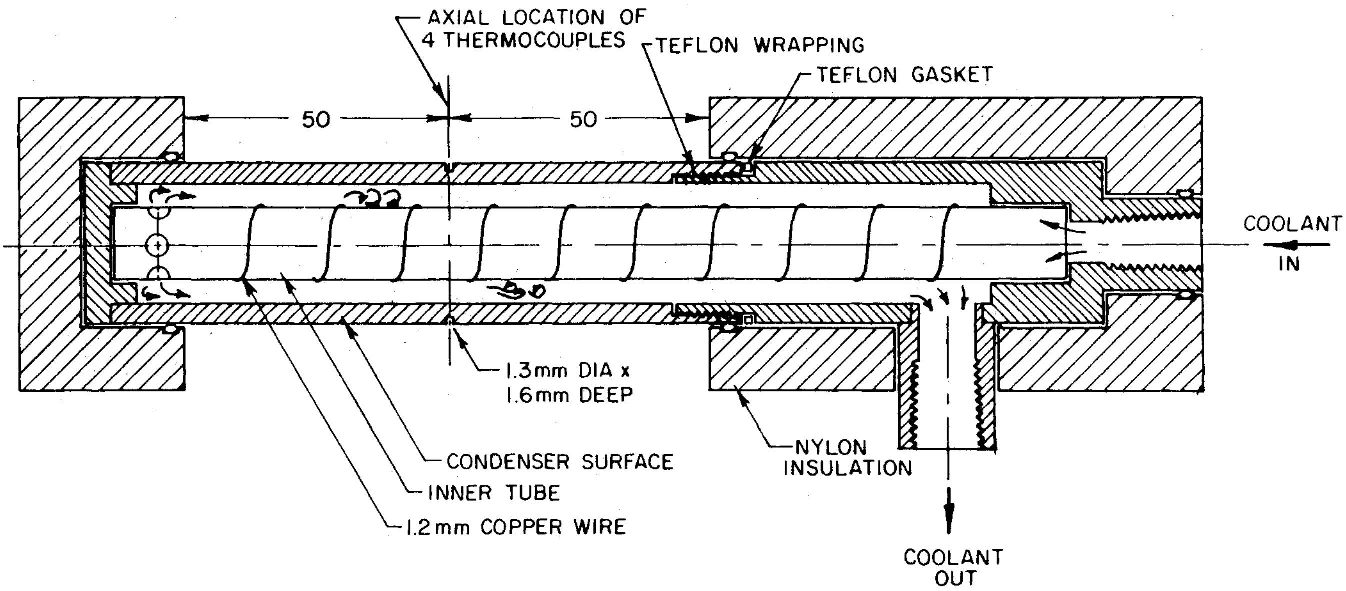


Fig. 2 Details of the test section.

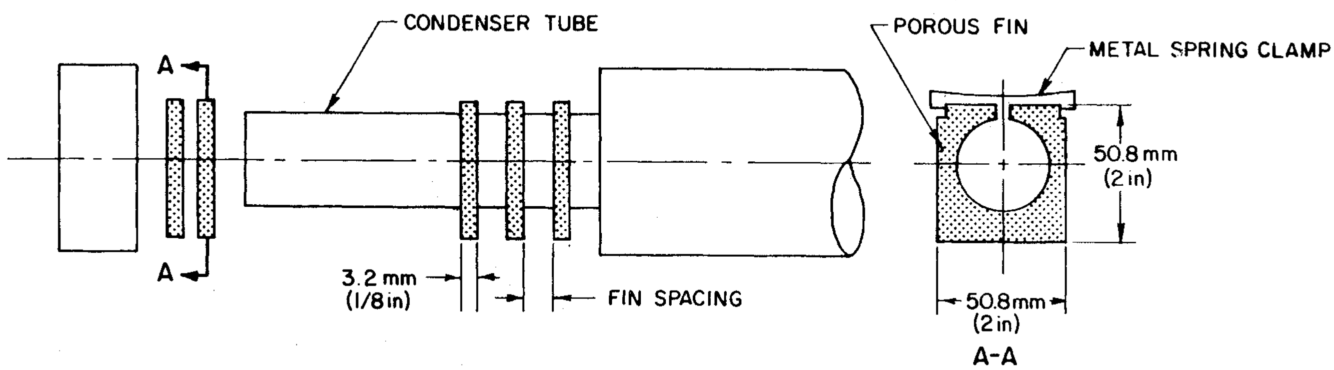


Fig. 3 Details of the porous fins.

placed at 0, 90, 180, and 270 deg from the top of the tube. The fins were mounted so that these thermocouples were always located between two adjacent fins. In addition, thermocouples were placed under the nylon insulation at each end of the active portion of the condenser tube—again to check for axial temperature variations. The average of all temperature readings was taken as the reference surface temperature for the test section. The output of these thermocouples, along with the output from the inlet and outlet cooling liquid temperature, and vapor temperature were supplied to a 10-channel digital thermometer.

The experiment was initiated with an evacuation injection procedure. The system was evacuated to ~ 3 Torr (~ 0.1 in. Hg) absolute pressure. Refrigerant-11 liquid was then injected into the system until the pressure was nearly 500 Torr (~ 20 in. Hg) absolute. The injection process was stopped, and the system was evacuated for the second time. The system was then injected with nearly 1600 ml of the refrigerant. At a pressure of 1 atm, nearly 25% by weight of the working fluid was in the form of vapor, and the rest formed a liquid pool at the bottom of the chamber completely covering the immersion heater. The advantage of evacuating and injecting the system with R-11 twice, as may be apparent, was the considerable reduction in the noncondensible gas content. This mass of noncondensibles was calculated assuming ideal gas behavior, to be less than $\sim 0.01\%$, which is sufficiently small to ensure a negligible effect on condensation rate. This was confirmed by measuring the vapor pressure and temperature before the condensation process was started and noting that both quantities correspond to the same thermodynamic state. Condensation was then initiated by allowing the cooling fluid to circulate through the condenser. Simultaneously, boiling was initiated in the liquid pool by supplying power to the 2 kW immersion heater. The condensate was collected in a nylon cup. The cup had a graduated view glass and its outlet was controlled using a solenoid valve. Condensate flow rate was measured by measuring the time required to fill the 100 ml cup. The uncertainty in this measurement is estimated to be ± 4 ml.

The porous fins were installed on the condenser tube as shown in Fig. 3. Porous fins 50×50 mm and 3.2 mm thick were cut from copper foametal (manufactured by Foametal, Inc., Willoughby, Ohio). The fin thickness was that which was available from the manufacturer and the width and length selected arbitrarily. Hence, neither dimension can be expected to be optimal. Eccentric holes 31.8 mm in diameter were cut from the fins so that they could be slid onto the condenser tube. The permeability of this material was measured to be $\sim 5 \times 10^{-9}$ m^2 in a constant head permeameter. The fins were positioned at the desired spacing and clamped to the tube as shown in Fig. 3. Data were obtained for fin spacings of 3.2, 6.4, 9.5, 12.7, and 30.2 mm. The data was highly repeatable, and the accuracy of the results was estimated to be within $\pm 4\%$ for the mass flow rate of condensate and $\pm 4\%$ for the surface subcooling.

Surface Temperature Calculation

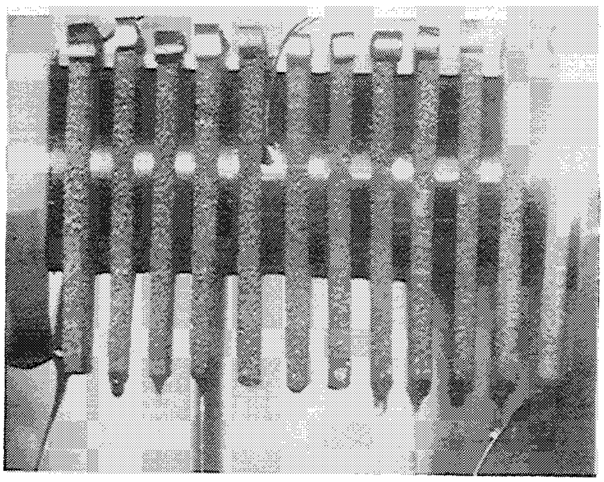
Since accurate measurement or estimation of surface temperature is critical to the accuracy of the results, an additional experiment was performed to check the accuracy of the temperature readings from thermocouples embedded in the tube surface. A trial experiment was performed, in which condensation on a plain tube was studied. The surface temperature was measured directly by using the embedded thermocouples. This reference temperature was used to evaluate the tube-side heat transfer coefficient by using the following relationship,

$$\bar{h}_i = \dot{m} h_{fg} / A_i (T_w - \bar{T}_{cw}) \tag{1}$$

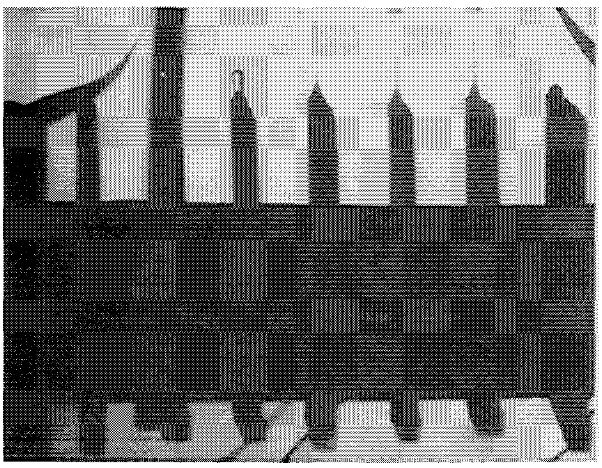
where $\dot{m}$ is the measured condensation rate in kg/s, h_{fg} is the latent heat of vaporization in J/kg, A_i is the internal surface area of the tube in m^2, and finally, T_w *and* $\bar{T}_{cw}$ are the wall temperature and coolant bulk temperature in °C. For the range of heat fluxes studied the temperature difference across the wall was calculated to be on the order of 0.10°C. The properties were evaluated using the following reference temperatures: $T_f = T_w + 0.3(T_{sat} - T_w)$ for the condensate and $\bar{T}_{cw} = (T_{cw_i} + T_{cw_o})/2$ for the internal flow. In order to ensure that the above measured quantities were acceptable, the results were compared with Nusselt's film condensation theory, and it was found that the experiment yielded 20% higher results than the theory. This is consistent with the outcome of the experiments performed by other investigators.

The measured internal heat transfer coefficient was 2–2.5 times that predicted for an internal turbulent flow in a smooth annulus. The bulk of the data was taken at Reynolds numbers around 3000, since it was desirable to operate the experiment at the upper limit of available pumping capacity to keep the coolant as nearly isothermal as possible. To evaluate the surface temperature at a particular condensation heat flux, the properties of coolant were determined at the bulk coolant temperature, and the internal convective heat transfer coefficient was determined from the experimental results discussed previously. The rearranged form of Eq. (1) was then used to calculate the surface temperature. This method was used to determine the surface temperature for the experiments on finned tubes. Direct measurement of the surface temperature corroborated the accuracy of this measurement to within ±4%.

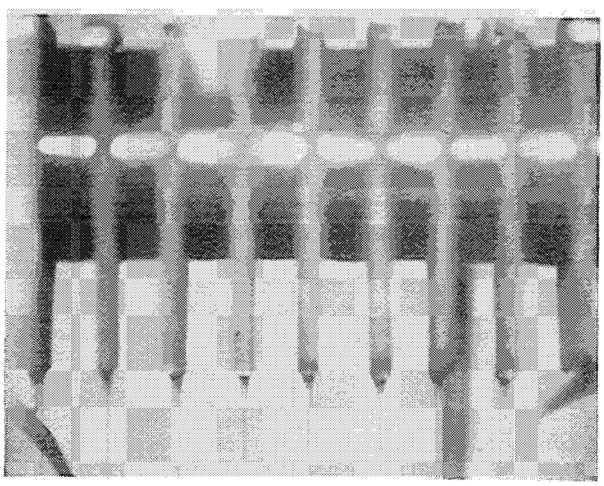
In the plain tube experiment, the circumferential surface temperature variation as reported by Kutateladze and Gog-



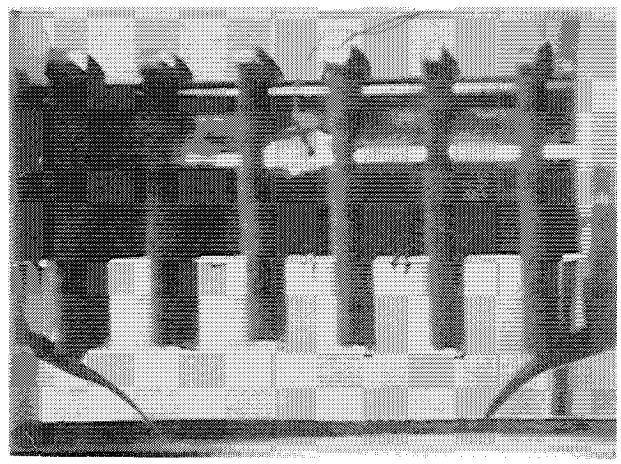
a)



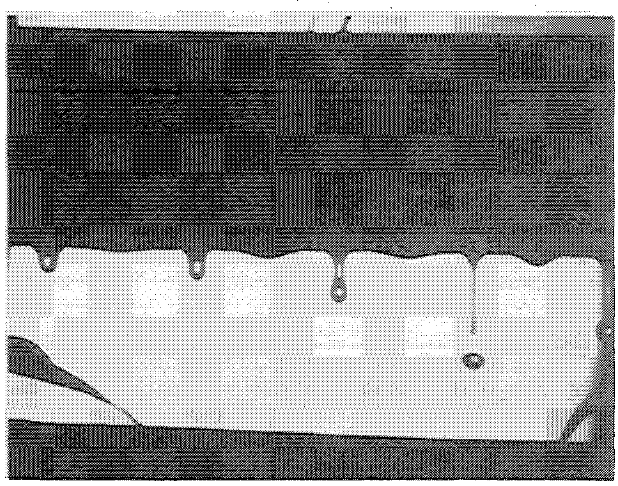
b)



c)



d)



e)

Fig. 4 Photographs of the condensation with various fin spacings: a) 3.2 mm (1/8 in.) spacing, b) 6.4 mm (1/4 in.) spacing, c) 9.5 mm (3/8 in.) spacing, d) 12.7 mm (1/2 in.) fin spacing, and e) plain tube.

onin[3] was observed. However, no substantial variation in the temperature around the circumference of the tube was measured for experiments involving small fin spacing (≤ 9.5 mm). A slight axial temperature variation was measured, although this amount was substantially lower than axial surface temperature variations reported by other investigators[9] due to the relatively thick wall of the condenser and the turbulent internal counter flow.

Discussion of Experimental Results and Observations

Experiments were conducted with fin spacings ranging from 3.2 mm (1/8 in.) to 30.2 mm (1-3/16 in.) at which point only three fins were present on the surface. The photographs in Fig. 4 for four fin spacings (3.2, 6.4, 9.5, and 12.7 mm) as well as the plain tube are useful in providing insight to the interpretation of the results. At smaller fin spacings (≤ 9.5 mm) as shown in Fig. 4a–c the liquid drainage occurred entirely through the fins for all experimental conditions except for those at low surface subcooling (<7 K). For these cases no liquid drops associated with the Taylor instability studies by Dhir and Taghavi-Tafreshi[15] were observed at the bottom of the tube, and the thickness of the film appeared to be uniform both axially and circumferentially. Evidence of the Kelvin Helmholtz instability in the form of surface waves on the liquid film was not observed. This is consistent with what one would expect. The thinning of the film caused by the presence of the porous fins would be expected to stabilize the film.

For condensation of R-11 on a 31.8 mm (1¼ in.) diam tube, the dominant wavelength as reported by Dhir and Taghavi-Tafreshi[15] becomes approximately 9.5 mm (3/8 in.). This is consistent with our observations for condensation on a plain tube (Fig. 4e). At a fin spacing of 9.5 mm (3/8 in.), drops appeared at low surface subcooling only. It is speculated that this was because the liquid film was too thin for the porous fins to become effective. Since the porous fins were slipped onto the tube and hence the fin-to-tube contact was imperfect, the effectiveness of the fin becomes questionable in the case of a very thin film. As the fin spacing was increased beyond 9.5 mm, surface waves became noticeable and the Taylor instabilities became evident as can be seen in Fig. 4d. The ratio of the amount of liquid draining from the fins to that draining from the tube surface decreased as the fin spacing was increased.

In addition to the observed effects of the fins on the film stability as discussed above, there are two additional factors related to the fin properties which deserve attention. First, the role of the fins in transferring heat to the condenser surface is worthy of mention. The effective thermal conductivity of the fins as a first order approximation may be estimated as[16]

$$k_{\text{eff}} = k_s(1-\phi) + k_\ell \phi \tag{2}$$

where subscripts s and ℓ correspond to solid and liquid phases, respectively, assuming the matrix is fully saturated by the liquid phase. Since ϕ was measured to be approximately 95% and $k_\ell << k_s$, from Eq. (4) it may be seen that k_{eff} is approximately 5% of k_s. If the effect of contact resistance at the surface of the condenser is considered, the porous fin would exhibit an efficiency considerably smaller than that of a solid fin. As a consequence of this poor thermal communication, we conclude that the fins play a very minor role if any in transferring heat to the condenser surface. Thus, it can be assumed that the portion of the test section covered by a porous fin was essentially adiabatic, and made no significant contribution to the condensation rate. The fins in essence served as insulators reducing the effective surface area.

Second, the role of the fins in imbibing and draining liquid is the key to their effectiveness. The natural tendency would be towards choosing a porous material with a high capillary pressure so that it would be effective in imbibing liquid from the film. Unfortunately, high capillary pressure corresponds to low permeability and a diminished drainage capability, leading to a saturated fin. For this reason, the optimum fin properties must lie in the region where drainage and imbibition capacities are properly balanced. Since we have yet to measure the capillary pressure of the porous fins and have conducted experiments with only a single fin material, it is at this point not possible for us to speculate on the optimum fin properties.

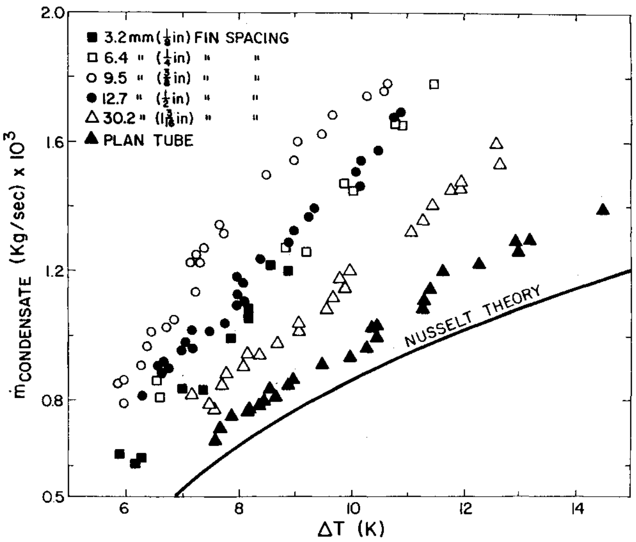


Fig. 5 Condensate mass flow rate vs ΔT subcooling for various fin spacings.

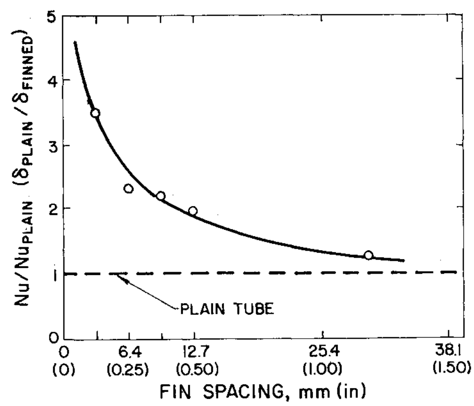


Fig. 6 Enhancement of condensation based on exposed surface area, as function of fin spacing ($\delta T = 10$ K).

Quantitative results are shown in Figs. 5 and 6. Figure 5 exhibits the direct experimental values of condensate mass flow rate as a function of temperature difference. We note that the total condensation rate, for a given surface subcooling, increases up to a value depicted by the curve given for 9.5 mm (3/8 in.) fin spacing, and then decreases to a value governed by plain tube condensation as the fin spacing approaches infinity. The maximum enhancement compared to the plain tube is approximately 1.8 times for the 9.5 mm (3/8 in.) fin spacing at a constant temperature difference of 10°C. For fin spacings less than 9.5 mm the degree of enhancement is lower. This does not mean that the fins are not effective but is a result of the insulating effect of the fins as discussed above.

In Fig. 6, the results for the degree of enhancement are presented as a function of fin spacing at a constant temperature difference of 10°C. In this case the Nusselt number is based on only the surface area on which the condensation takes place, that is, the exposed tube surface. It should be noted that the Nusselt number is inversely related to the film thickness. Thus, the vertical axis in Fig. 6 represents either a ratio of Nusselt numbers or a ratio of average film thicknesses. This figure emphasizes the effectiveness of the fins in thinning the liquid film. It clearly demonstrates that, even for small spacings where enhancement was not large, the film thinning continues to increase. The fact that the degree of enhancement continues to increase as surface subcooling is increased, as shown in Fig. 5, and as the fin spacing is decreased, as shown in Fig. 6, leads us to conclude that the fins are not saturated and have additional capacity to drain liquid. This is consistent with the calculations of the maximum liquid carrying capacity of the fins. Their performance is then limited by their capacity to imbibe liquid. Data could not be obtained at higher subcooling (higher ΔT) for two reasons. First, we were faced with a limited refrigeration capacity to provide cooling water to the condenser, and second, the safe operating pressure of the bell jar was limited to a maximum of approximately two atmospheres. Future experiments will be conducted using fins having smaller permeability (smaller pore size) and, hence, higher capillary pressure. It should be noted also that the capacity to imbibe fluid depends directly upon surface tension, and the ability to drain liquid is proportional to K/μ. If we examine the ratio of σ/μ for water as compared to R-11 we would expect the degree of enhancement for water to be in excess of twice that achieved with R-11.

Conclusions

Experimental results for film condensation with and without porous fins were reported in this paper. It was demonstrated that the fins contribute significantly toward thinning the liquid film, resulting in a 3.5 time reduction in film thickness for 3.2 mm (1/8 in.) fin spacing. Unfortunately, the fins have a very low thermal conductivity leading to an insulating effect, which reduces the effective surface area for condensation. These two opposing factors result in an optimum fin spacing of 9.5 mm (3/8 in.) at which the condensation rate is increased by ~ 1.8 times that for a plain tube. This value is, of course, dependent upon both fluid and fin properties, thus, would be expected to be different if either were changed. The measured degree of enhancement falls far below the maximum predicted by Shekkariz and Plumb[14] since these predictions represent the upper limits based on the ability of the fin to drain fluid with no resistance to imbibition from the liquid film to the base of the porous fin. These preliminary experiments demonstrate that this approach to enhancing condensation has considerable potential in situations where compact and lightweight condensers are required.

Acknowledgment

The authors would like to acknowledge the support of the National Science Foundation under Grant CBT-84-18497. In addition, the thoughtful comments by the reviewers were useful in improving the clarity of the manuscript.

References

[1]Marto, P. J., "Recent Progress in Enhancing Film Condensation Heat Transfer on Horizontal Tubes," *Proceedings of the 8th International Heat Transfer Conference*, Vol. 1, Hemisphere, New York, 1986, pp. 161–170.

[2]Fujii, T., "Importance of Vapor Flow in Condensers," *Proceedings of the 1987 ASME/JSME Thermal Engineering Joint Conference*, Vol. 4, American Society of Mechanical Engineers, New York, 1987, p. 375.

[3]Kutateladze, S. S. and Gogonin, I. I., "Heat Transfer in Condensation of Flowing Vapor on a Single Horizontal Cylinder," *International Journal of Heat and Mass Transfer*, Vol. 28, No. 5, 1985, pp. 1019–1030.

[4]Marto, P. J., Mitrou, E., Wanniarachchi, A. S., and Katsuta, M., "Film Condensation of Steam on a Horizontal Wire-Wrapped Tube," *Proceedings of the 1987 ASME/JSME Thermal Engineering Joint Conference*, Vol. 1, 1987, pp. 509–516.

[5]Rifert, V. G., Trokoz, Y. Y., and Zadiraka, V. Yu, "Enhancement of Heat Transfer in Condensation of Ammonia Vapor on a Bundle of Wire-Wrapped Tubes," *Heat Transfer-Soviet Research*, Vol. 16, Jan.–Feb. 1985, pp. 36–41.

[6]Yau, K. K., Cooper, J. R., and Rose, J. W., "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*, Vol. 38, edited by C. T. Avedisian and T. M. Rudy, American Society of Mechanical Engineers, Heat Transfer Division, New York, 1984, pp. 151–156.

[7]Honda, H., Nozu, S., and Mitsumori, K., "Augmentation of Condensation on Horizontal Finned Tubes by Attaching a Porous Drainage Plate," *Proceedings of the ASME/JSME Thermal Engineering Joint Conference*, Vol. 3, American Society of Mechanical Engineers, New York, 1983, pp. 289–296.

[8]Carnavos, T. C., "An Experimental Study: Condensing R-11 on Augmented Tubes," American Society of Mechanical Engineers Paper 80-HT-54, July 1980.

[9]Fujii, T., Wang, W. C., Koyama, S., and Shimizu, Y., "Heat Transfer Enhancement for Gravity Controlled Condensation on a Horizontal Tube by Coiling Wires," *Heat Transfer Science and Technology*, edited by B.-X. Wang, Hemisphere Publishing, Washington, DC, 1987, pp. 773–786.

[10]Honda, H., Nozu, S., Uchima, B., Fujii, T., "Effect of Vapour Velocity on Film Condensation of R-113 on Horizontal Tubes in a Crossflow," *International Journal of Heat and Mass Transfer*, Vol. 29, No. 3, 1986, pp. 429–438.

[11]Cavallini, A., Frizzerin, S., and Rossetto, L., "Condensation of R-11 Vapor Flowing Downward Outside a Horizontal Tube Bundle," *Proceedings of the 8th International Heat Transfer Conference*, Vol. 4, Hemisphere, New York, 1986, pp. 1707–1712.

[12]Rose, J. W., "Effect of Pressure Gradient in Forced Convection Film Condensation on a Horizontal Tube," *International Journal of Heat and Mass Transfer*, Vol. 27, No. 1, 1984, pp. 39–47.

[13]Plumb, O. A., "Capillary Effects on Film Condensation in Porous Media," 19th AIAA Thermophysics Conference, Snowmass, CO, June 1984.

[14]Shekarriz, A. and Plumb, O. A., "A Theoretical Study of the Enhancement of Filmwise Condensation Using Porous Fins," American Society of Mechanical Engineers Paper 86-HT-31, June 1986.

[15]Dhir, V. K. and Taghavi-Tafreshi, K., "Hydrodynamic Transitions During Dripping of a Liquid from Underside of a Horizontal Tube," American Soceity of Mechanical Engineers Paper 81-WA/HT-12, Nov. 1981.

[16]Bejan, A., *Convective Heat Transfer*, Wiley, New York, 1984, p. 353.