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Recent Progress in Enhancing Film Condensation Heat Transfer on Horizontal Tubes

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An assessment of several promising techniques that use surface tension forces to enhance film condensation heat transfer on horizontal tubes is made. Recent progress on integral-fin tubes is stressed, including experimental findings on fin spacing, geometry, and material and the latest theoretical developments for predicting performance. Condensation heat transfer enhancement on wire-wrapped tubes is also examined, as well as the use of nonwetting strips and porous drainage devices. The effect of condensate inundation on plain and enhanced tubes is reviewed briefly, and future research directions are discussed.

INTRODUCTION

For many years, surface condensers were designed with smooth horizontal tubes and there was little motivation for improving their thermal effectiveness. However, during the past decade, when the cost of energy received worldwide attention, interest in better understanding shell-side condensation was renewed [1-4] and numerous heat transfer enhancement techniques for film condensation were proposed [5-7].

The most frequently used technique involves extended surfaces [8-11], of which low-profile integral fins are the most common. These surfaces were introduced initially to increase surface area. Fin geometries, dimensions, and spacings were limited by manufacturing techniques, and condensate flow was assumed to be governed only by viscous and gravitational forces. Surface tension forces were completely neglected until addressed by Gregorig [12] and Thomas [13] for vertical tubes. Their work generated an awareness of the importance of surface tension in the use of extended surfaces and stimulated research on the effective utilization of surface tension forces to thin the condensate film and enhance the condensation heat transfer process.

Today, low-profile integral-fin tubes are available from many manufacturers and are used commonly in the refrigeration industry with low surface tension liquids. More advanced surfaces (e.g., Hitachi Thermoexcel-C, Wolverine Turbo-C, and Wieland Gewa-TWX) have been introduced to improve on finned tube performance in this application. For years, finned tubes were thought to be inappropriate for high surface tension liquids because of condensate flooding between the fins. However, recent tests with finned tubes have shown sizable enhancements for water [14, 15] and ethylene glycol [16] provided the number of fins is chosen properly. This new information, coupled with possible advancements in manu-

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facturing techniques, has stimulated further study of integral-fin tubes to arrive at an optimum tube for a variety of fluids and applications.

This paper provides a critical assessment of the current technology of enhanced film condensation heat transfer of pure vapors on horizontal tubes. It concentrates on developments that have occurred since the 7th International Heat Transfer Conference in 1982 and provides directions for future research. Although the focus is on horizontal tubes, the techniques reviewed, which stress the use of surface tension forces, may be valuable for other geometric configurations, such as those that may occur in zero gravity of space or in the electronics industry, where small condenser surfaces are required. Because of length limitations, certain important topics had to be excluded. Condensation with binary mixtures and with noncondensing gases, the effects of vapor shear, and dropwise condensation phenomena have all been omitted.

EXTENDED SURFACES

Integral-Fin Tubes

In 1948 Beatty and Katz [17] measured condensing film coefficients for several refrigerants (i.e., low surface tension liquids) on a horizontal integral-fin tube containing 630 fins per meter. They proposed a theoretical model that neglected surface tension and considered gravity-driven condensation on both the vertical fin flanks and the cylindrical tube surface in the interfin spaces. They arrived at an empirical Nusselt-type expression for the average heat transfer coefficient that correlated their data to within 10%:

$$\bar{h} = 0.689 \left(\frac{k_f^3 \rho_f^2 g \Delta h_v}{\mu_f \Delta T_{vf}}\right)^{1/4} \left(\frac{1}{D_{eq}}\right)^{1/4} \tag{1}$$

where

$$\left(\frac{1}{D_{eq}}\right)^{1/4} = \frac{A_r}{A_{ef}} \frac{1}{D_r^{1/4}} + 1.30 \eta_f \frac{A_f}{A_{ef}} \frac{1}{\bar{L}^{1/4}}$$

and

$$\bar{L} = \pi \frac{(D_o^2 - D_r^2)}{4D_o}$$

In Eq. (1), $\bar{h}$ is the average heat transfer coefficient (W/m^2 K), k_f the thermal conductivity (W/m K), ρ_f the density (kg/m^3), g the local gravitational acceleration (m/s^2), Δh_v the specific latent heat of vaporization (J/kg), μ_f the dynamic viscosity (kg/s m), ΔT_{vf} the temperature difference across the condensate film (K), η_f the fin efficiency, A_f the fin surface area (m^2), A_r the surface area of the tube at the root or base of the fins (m^2), A_{ef} an effective total surface area of the finned tube (m^2), D_r the tube diameter at the root of the fins (m), and D_o is the outside diameter of the tube (m).

Later studies recognized the importance of surface tension forces [18–22]. Rifert [23] provided a comprehensive review of steam condensation on extended surfaces. He showed that heat transfer enhancement with finned surfaces can vary substantially, depending on fin geometry and spacing, and noted that flooding of the interfin spaces by the condensate, which increases as the fin spacing decreases, can alter the results substantially. In 1980 Rifert [24] analyzed condensation of vapor on horizontal finned tubes, including the effect of surface tension forces. In the analysis, he divided the tube into flooded and unflooded zones and solved a two-dimensional form of the energy equation for each zone. Solutions to these equations revealed that the fin temperature, in most cases, is very nonuniform. Hirasawa et al. [25] showed analytically that surface tension not only works near the fin tip to create a very thin condensate film but can also lead to thinning of the condensate film near the trough.

In recent years, a trend toward increasing fin density has occurred and more attention has therefore been placed on condensate flooding in the lower portion of the tube. In 1981 Rudy and Webb [26] made static measurements of condensate retention angles* on finned tubes with three different fin densities (748, 1024, and 1378 fins/m), using water, R-11, and n-pentane. In 1983 Honda et al. [27] reported measurements with ethanol and R-113 for both static and condensing conditions and found the results to be essentially the same. Yau et al. [28] made static measurements for a range of fin densities, using water, ethylene glycol, and R-113. The results of all these investigations were in general agreement. Rudy and Webb [29] developed a condensate retention model that was based on their experimental observations that the vertical rise height of the condenate between fins is the same for a horizontal tube and a vertical plate. For an arbitrarily shaped fin, they summed up the surface tension and body forces acting on the retained liquid between the fins and arrived at an expression for the retention angle. For rectangular fins, their expression yields

*The condensate retention angle ψ is measured from the bottom centerline of the tube to the position at which the condensate just stops filling the interfin space.

$$\psi = \cos^{-1}\left(1 - \frac{4\sigma}{D_o \rho g s}\right) \tag{2}$$

where σ is the surface tension (N/m) and s the spacing between fins (m). This result is identical to the expression derived earlier by Honda et al. [27]. Equation (2) shows that the ratio of surface tension to density (σ/ρ) for the condensate and the fin spacing s are important variables for flooding. Equation (2) is plotted in Fig. 1 for a tube with a diameter D_o of 14.7 mm and for several values of σ/ρ. Clearly, high surface tension liquids such as water can completely flood a finned tube at relatively low fin spacings, whereas for low surface tension liquids such as Freon the fin density can be increased dramatically before complete flooding occurs.

Several systematic experimental programs have been undertaken to gain more insight into film condensation heat transfer on integral-fin tubes. Rudy [30] reported data for R-11 condensing on four tubes with different fin geometries. His results indicated vapor-side coefficients exceeding Nusselt values by factors of about 7 to 9. Honda et al. [27] tested four low-fin tubes with different fin geometries, using methanol and R-113 as condensing fluids. Vapor-side enhancements between 4.8 and 9 were found. A wide

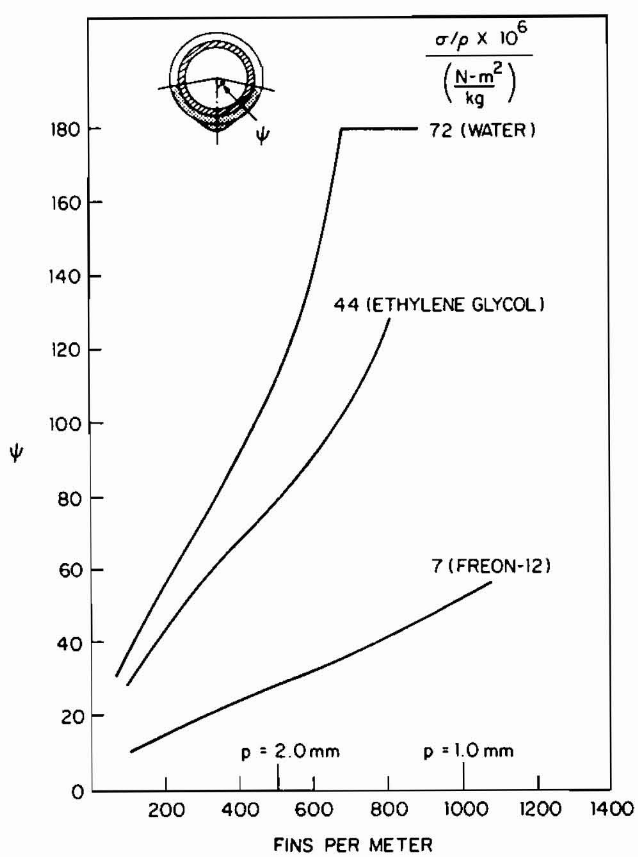

Figure 1 Predicted retention angle versus fin spacing for several values of σ/ρ.

variety of data for film condensation of steam have recently been reported [14, 31, 32]. Over 60 finned tubes with the same nominal fin root diameter of 19 mm were manufactured and tested. The effects of fin spacing, thickness, height, geometry, and material have been systematically measured both at atmospheric pressure and under vacuum conditions (~85 mmHg) to provide a data base for comparison to theoretical predictions. Figure 2 shows some representative steam-side heat transfer coefficient data [14] (based on the nominal fin root smooth tube surface area) for copper tubes with rectangular fins. Smooth tube data are also included for comparison. The data were obtained under vacuum conditions with steam flowing downward at a velocity of 2 m/s. The fin shape was held constant (with thickness and height equal to 1.0 mm) while the fin spacing was varied, and the data show that fin spacing is an important variable. An optimum spacing exists near 1.5 mm (i.e., approximately 400 fins/m or 10 fins/in.). It is also apparent that at the optimum fin spacing, enhancements of approximately 3 to 4 are possible. Results for various fin thicknesses and heights have been reported by Wanniarachchi et al. [31]. They showed that the steam-side enhancement increases with fin height; however, the rate of performance increase with increase in fin height is less than the rate of area increase, indicating a fin efficiency effect. A fin height between 1.0 and 1.5 mm is desirable. Fin thickness caused a small change in enhancement with the best thickness falling between 0.75 and 1.0 mm. Marto et al. [32] reported that fin shape can alter performance from 10 to 15%, and Mitrou [33] showed that tube wall material can alter the results considerably. Figure 3 shows Mitron's data for condensation of steam on spirally threaded tubes of copper, aluminum, copper-nickel, and stainless steel. Clearly, decreasing thermal conductivity of the fin creates an increased thermal resistance in the fin (hence, a lower fin efficiency), leading to lower film condensation heat transfer. These results are in agreement with the earlier data of Mills et al. [34] and Shklover et al. [35].

A similar systematic set of experimental measurements was made by Rose and co-workers [15, 16, 36]. Thirteen integral-fin tubes with a root diameter of 12.7 mm were manufactured with rectangular fins having the same fin thickness (0.5 mm) and fin height (1.6 mm). All tests were made at near-atmospheric pressure with downward vapor velocities of 0.5 to 1.1 m/s. The maximum vapor-side enhancement for steam was found to occur at a fin spacing of 1.5 mm, whereas for ethylene glycol and R-113 the maximum occurred at spacings of only 1.0 and 0.5 mm, respec-

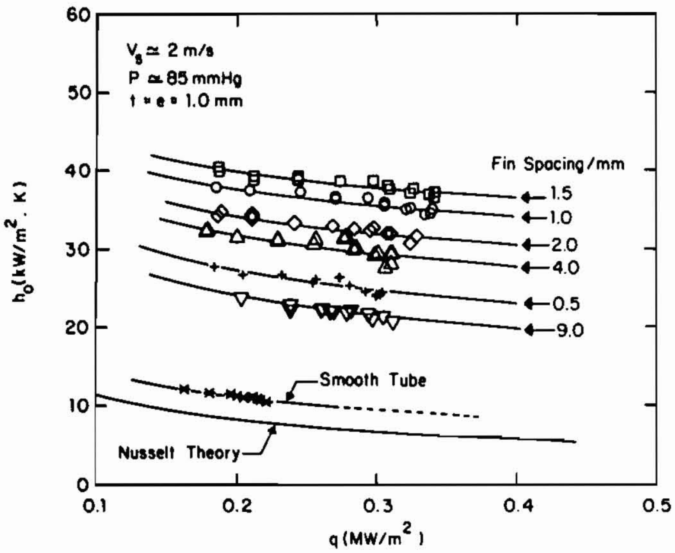

Figure 2 Effect of fin spacing on the heat transfer coefficient of steam condensing on finned tubes [14].

tively. Figure 4 shows the results of Masuda and Rose [16, 36] for the enhancement ratio of $E_{\Delta T}$ (vapor-side coefficient for finned tube to smooth tube at the same vapor-to-wall surface temperature difference) as a function of fin spacing. The percentage for each data point corresponds to the percent flooding of the tube as calculated from Eq. (2). It is interesting to note that apparently the maximum enhancement ratio increases and the optimum spacing decreases as the ratio of surface tension to density (σ/ρ) of the condensate decreases (Fig. 1). Also, for R-113 the maximum enhancement ratio occurs when only 36% of the tube is flooded, whereas for steam the maximum occurs when 53% of the tube is flooded. Coupled fluid-wall interactions are obviously occurring with these finned tubes, and additional testing is re-

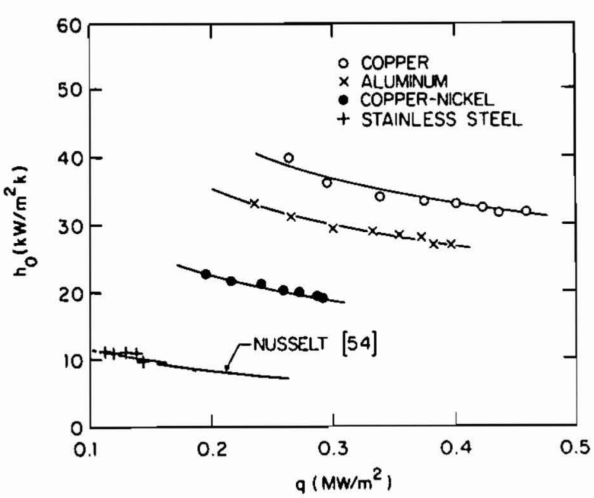

Figure 3 Effect of wall thermal conductivity on the heat transfer coefficient of steam condensing on spirally threaded tubes [33].

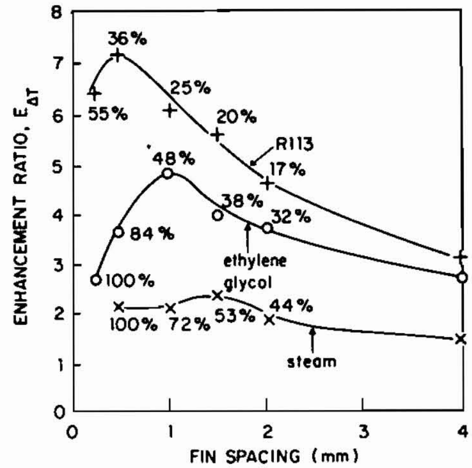

Figure 4 Dependence of enhancement on fin spacing and condensate flooding [16, 36].

quired to more fully understand the mechanisms involved.

Several important contributions have been made by Webb and co-workers. In 1983, based on their study of condensate retention, Rudy and Webb [37] proposed that the Beatty and Katz heat transfer coefficient h_{BK} [Eq. (1)] be modified to account for condensate retention, as shown by

$$\bar{h} = h_{BK}\left(\frac{\pi - \psi}{\pi}\right) \tag{3}$$

Equation (3) neglects surface tension forces on the fins and neglects heat transfer through the flooded portion of the tube. As a result, this equation underpredicted the average condensing heat transfer coefficient of R-11 by up to 35% when a significant amount of retained condensate was present. The equation was later modified by including surface tension-induced drainage on the fins in an approximate manner. This approximate model underpredicted R-11 test results for a tube having 1378 fins/m by about 10% and overpredicted the results for a tube having 748 fins/m by as much as 60%. Webb and co-workers concluded that an improved surface tension analysis was necessary.

Their latest model [38] assumes that in the unflooded region of the tube, condensate flows radially inward along the fin flanks due to surface tension forces only, and neglects gravity drainage. On the portion of the horizontal tube surface in the interfin spaces, gravity drainage occurs according to a Nusselt-type analysis. On the fin flanks, they use the heat transfer coefficient proposed by Adamek [39], who conducted a theoretical analysis for surface tension-driven condensation on a family of condensate surface profiles with local radius of curvature given by

$$\frac{1}{r} = \frac{\theta_m}{S_m} \frac{(\xi + 1)}{\xi} \left[1 - \left(\frac{s}{S_m} \right)^{\xi} \right] \tag{4}$$

In Eq. (4) ξ is a parameter that characterizes the aspect ratio of the surface, S_m the length of the convex condensate surface (m), θ_m the angle through which a normal to the fin surface rotates in going from the tip ($s = 0$) to the end of the convex surface ($s = S_m$) (rad), and s the distance along the condensate film surface measured from the fin tip (m). For the flooded region, a two-dimensional conduction analysis was used to solve for the heat flow through the fins and the condensate.

Their final expression for the average heat transfer coefficient was expressed as

$$\bar{h}_\eta = \left(1 - \frac{\psi}{\pi}\right) \left(h_h \frac{A_r}{A} + h_f \eta_f \frac{A_f}{A}\right) + \frac{\psi}{\pi} h_b \tag{5}$$

where h_f is the heat transfer coefficient on the fin flanks, h_h the heat transfer coefficient on the unflooded root surface in the interfin space, h_b the heat transfer coefficient in the flooded region, A the surface area of the tube expressed as $\pi D_o L$ (m^2), L the total length of the tube (m), and η the total efficiency of the finned surface, expressed as $1 - (1 - \eta_f)A_f/(A_r + A_f)$. Detailed procedures for calculating these quantities are provided [38]. The model predicted their heat transfer coefficient data for R-11 condensing on horizontal, integral-fin tubes with fin densities of 748, 1024, and 1378 fins/m to within 20%. When compared to the recent data of Wanniarachchi et al. [14] for steam, good agreement ($\pm 20\%$) was also observed except for the fully flooded condition where the fin spacing is 0.5 mm. For this situation, the model severely underpredicted the measured heat transfer. Webb and Kedzierski [40] continued this work by conducting experiments on vertical fluted surfaces where the flute shapes were precisely manufactured by an electrostatic discharge machining method. They manufactured six fluted surfaces: three with a Gregorig flute profile (i.e., $\xi = 2$) and three with the optimum Adamek flute profile ($\xi = -0.5$). Preliminary experimental results for several of the profiles, together with predictions, are shown in Fig. 5. In general, the slope of the data agrees with the predictions, but the data are either underpredicted

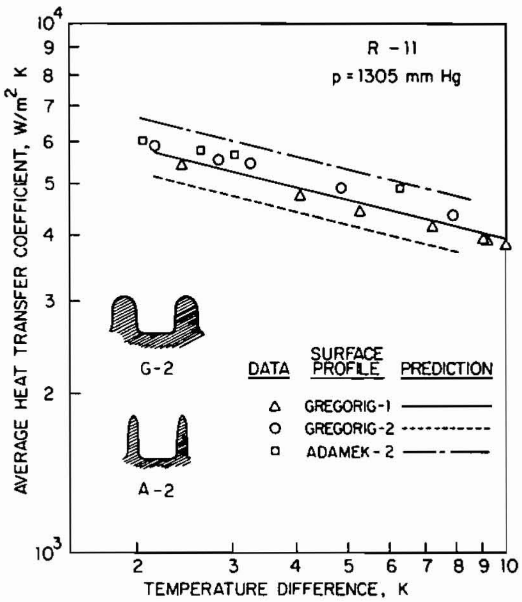

Figure 5 Film condensation of R-11 on vertical fluted plates [40].

(approximately 15% for the Gregorig-2 profile) or overpredicted (approximately 8% for the Adamek-2 profile). The results of their investigation will provide a unique opportunity to study the influence of surface tension forces more accurately.

Although the results of the Webb et al. model appear encouraging, it should be realized that it is based on the premise that the fin wall temperature is uniform, as assumed by Adamek [39]. In reality, the problem of condensation on a fin is a conjugate one, where conduction within the fins and the resultant surface temperature variations along the fins are coupled to the condensation heat transfer rates. Since the fin temperature distribution is not known a priori, wall conduction effects must be included in the analysis. In 1978 Fujii and Honda [41] carried out a numerical analysis of film condensation on a vertical fluted plate in which wall conduction effects were occurring simultaneously. In 1986 Honda and Nozu [42] reported on the most comprehensive approach to the problem. They assume that the tube is divided into unflooded and flooded regions. Their expression for the condensate retention angle, which is similar to Eq. (2), is used to calculate the flooded point. In the unflooded region, both surface tension and gravity forces are included. The effect of variable wall temperature is also included by considering overall heat transfer from the vapor to the coolant through a finned tube wall. The effect of radial wall temperature variations along the fin is taken into account by using the fin efficiency, whereas circumferential wall temperature variations are included in the numerical analysis. A detailed comparison of their analysis to

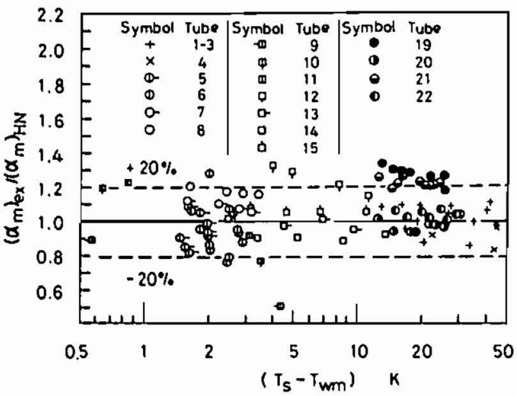

Figure 6 Comparison of measured average heat transfer coefficients on finned tubes with the Honda and Nozu prediction [42].

experimental data for refrigerants, methanol, and steam was made [42] and is shown in Fig. 6. In general, their model predicts most of the available data to within ±20%. Very recent refinements of their model [43] include axial condensate flow on the tube surface between fins due to surface tension forces, as well as axial wall temperature variations between the fin root and the tube surface. As shown in Fig. 7, these additional refinements yield very satisfactory results for steam, especially under vacuum conditions. At larger fin spacings, their refined model [43] predicts significantly higher enhancements than their original model [42] because of allowance for condensate thinning axially between fins. Under fully flooded conditions, when the fin spacing is 0.5 mm, both of their models underpredict the data by approximately 25% under vacuum conditions and 50% under atmospheric conditions. It therefore appears that the existing models

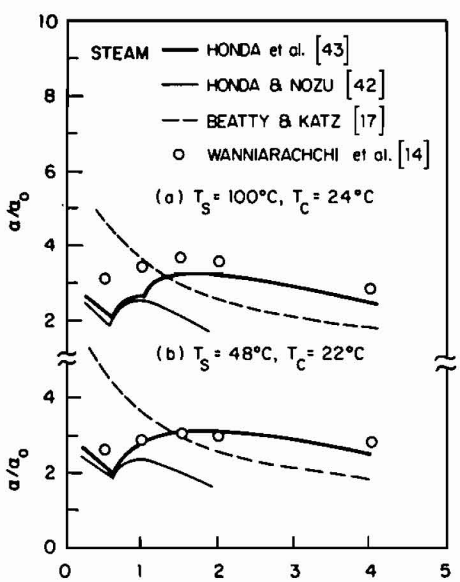

Figure 7 Comparison of measured steam condensing coefficients on finned tubes [14] with various theoretical models [43].

do not properly account for heat transfer in flooded interfin channels.

Wire-Wrapped Tubes

In 1967 Thomas [13, 44] reasoned that rectangular or circular projections fixed to a condensing surface (but not necessarily integrally attached) could be used to thin the condensate film between the projections. As a result, a substantial increase in the film condensation heat transfer coefficient is possible. (The model of Honda et al. [43] included this thinning effect.) This concept has been applied to horizontal tubes by spirally wrapping a fine wire around the tube. Such a technique has been used successfully in condensing ammonia [45, 46] and steam [47], exhibiting heat transfer enhancements of several hundred percent.

A theoretical analysis of laminar gravity-driven film condensation on a single horizontal wire-wrapped tube has been made by Fujii et al. [48]. They assumed that at any circumferential position the film thickness between wires is constant (and very thin) and that it experiences a step increase at the wire itself. In this way, the axial pressure gradient in the film, dp/dz, is zero except near the wire, where it follows a delta-function behavior. In the region beneath the wire, hat transfer was assumed to be zero. With these simplifications, they arrived at an approximate expression for the ratio of the average Nusselt number for a wire-wrapped tube, $\overline{\mathrm{Nu}}$, to that for a smooth tube as predicted by Nusselt, Nu_n:

$$\frac{\overline{\mathrm{Nu}}}{\mathrm{Nu}_n} \simeq \frac{s}{p}\left(\frac{4(1 + A)}{3}\right)^{1/4} \tag{6}$$

where s is the spacing between wires, p the wire pitch, and

$$A = \frac{4\sigma D_o}{\rho g r_s s^2}$$

In this expression for A, r_s is the local radius of curvature of the condensate surface at the base of the wire. It was approximated by Fujii et al. [48] by

$$r_s \simeq k\frac{a^3}{D_w^2} \tag{7}$$

where k is an empirical constant, a is the capillary constant $(2\sigma/\rho g)^{1/2}$, and D_w is the wire diameter (m). Measurements were made for R-11 and ethanol, using several different wire diameters and pitches.

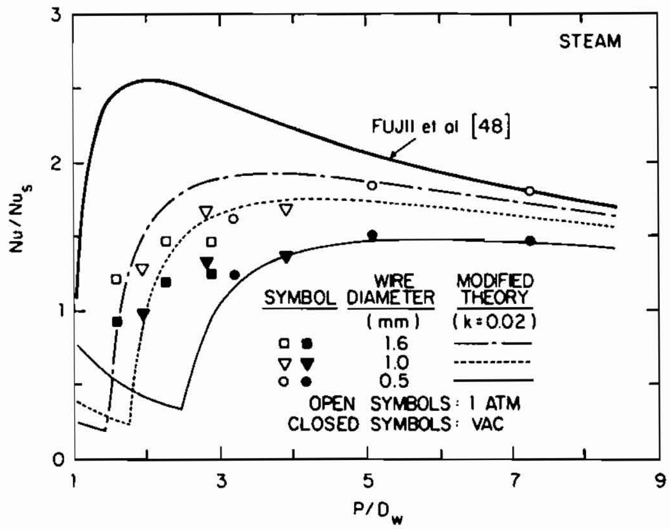

Figure 8 Comparison of measured steam condensing coefficients on wire-wrapped tubes with theoretical models [33].

Maximum enhancements of 3.5 for R-11 and 2.9 for ethanol were found, and these maximum values occurred for $p/D_w = 2.0$. Their approximate model [Eq. (6)], with an adjusted value of $k = 0.03$, showed very reasonable agreement with their data. When their model was compared to recent data of Mitrou [33] for steam condensing on wire-wrapped tubes (wire diameters of 0.5, 1.0, and 1.6 mm), the data were significantly different from the theory at low values of p/D_w (i.e., small spacings), as shown in Fig. 8. This discrepancy was thought to be caused by significant condensate retention for water between wires, which is not included in the Fujii model. When the Fujii model was corrected to include flooding (i.e., the Fujii analysis was assumed to be valid only from $\phi = 0$ to $\phi = \psi$, and heat transfer in the flooded region was computed using one-dimensional conduction through the wires and the condensate), the modified results shown in Fig. 8 were obtained. Agreement between the modified model and the data is now reasonable. An enhancement with steam of approximately 1.8 is therefore possible at atmospheric pressure, and the best enhancement appears to occur for a wire diameter of 0.5 mm and a wire pitch near 3 mm.

CONDENSATE DRAINAGE DEVICES

In an effort to reduce the average condensate thickness on a horizontal tube, Mayhew [49] examined, both theoretically and experimentally, the removal of condensate by using nonwettable longitudinal strips or "fins." His analysis showed that by setting the condensate film thickness to zero at some angle ϕ from the top of the tube and letting the condensate thickness regrow around the circumference from that point, it was possible to predict an enhancement in heat transfer. He found that the optimum location for the interruption occurred at 90°, and for this case a maximum enhancement of 19% was achievable. Experimental results, however, did not verify this claim. In 1973 Glicksman et al. [50] conducted experimental measurements of steam condensing on a single horizontal tube on which nonwetting strips of Teflon (polytetrafluoroethylene) were placed. They showed that of all the locations tested, the best performance was achieved when the strip was placed longitudinally along the bottom of the tube. Enhancements as large as 100% were observed at low temperature differences (i.e., low heat fluxes). In 1979 Desmond and Karlekar [51] used a nonwetting strip of Emralon-330, a highly durable fluoroplastic, to enhance steam condensation on a horizontal stainless steel tube. The strip was 9.5 mm wide and 0.025 mm thick. They measured a 20% enhancement in the overall heat transfer coefficient for tube-side water velocities between 1.7 and 2.4 m/s.

Yau et al. [28, 52] studied the effects of longitudinal drainage strips located at the bottom of both finned and plain tubes. Their strips were oriented vertically at the bottom of the tubes rather than horizontally as done earlier [50, 51]. They used a Teflon strip 0.5 mm wide and 8 mm high, as well as two copper strips, 0.5 mm wide and 4 mm and 8 mm high. In the case of the plain tube, the 8-mm copper strip enhanced the steam-side coefficient by approximately 10–15%. The 4-mm copper strips made no change in the data and the Teflon strip resulted in a slight decrease. When the 8-mm-high copper strip was placed on the underside of the finned tubes (both 1.5- and 2.0-mm fin spacings were used), enhancements as large as 30% were observed. This increase was attributed to a reduction of the condensate retention angle effected by the strip.

Honda et al. [27] conducted studies of heat transfer augmentation by attaching porous drainage strips to the underside of horizontal finned tubes. They measured enhancements over the finned tube results of 36% for R-113 and 108% for methanol. More recently, Honda and Nozu [53] modified their earlier analysis for condensation on a finned tube [42] to include the effect of a porous drainage strip. At the tube bottom, the condensate was assumed to be influenced by gravity as well as by capillary forces due to the presence of the porous strip. The condensate flow through the porous strip was governed by Darcy's law. A comparison between their analysis and their data for methanol is shown in Fig. 9. Tube A was a smooth tube and tube D an integral-fin tube (fin pitch

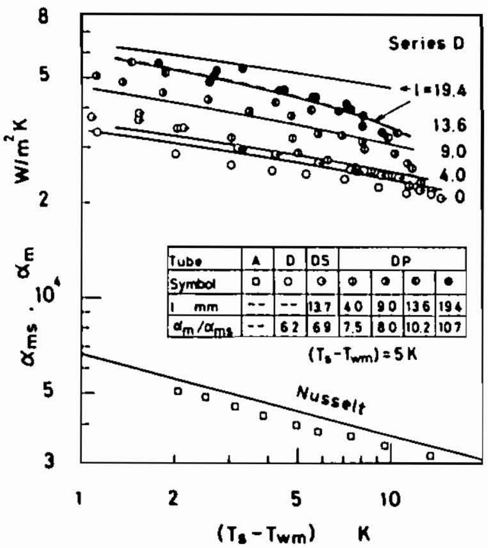

Figure 9 Influence of porous drainage strips on the measured and predicted average heat transfer coefficient of methanol condensing on finned tubes [53].

of 0.5 mm, fin height of 1.13 mm, and fin spacing of 0.39 mm); both were made of copper. Tube DS had a 1.9-mm-thick by 14-mm-high solid drainage strip of polyvinyl chloride and tubes DP had 1.9-mm-thick porous nickel strips of varying heights (4, 9, 13.6, and 19.4 mm). The results indicate that the average heat transfer coefficient for an integral-fin tube increases as the height of the porous drainage strip increases and enhancements of approximately 80% are possible. Agreement between their data and their prediction is very good.

EFFECT OF CONDENSATE INUNDATION

All the information described above pertains to a single tube. Shell-side condensation in an actual condenser may be widely different from condensation on a single tube because of the effect of condensate inundation from neighboring tubes.

Numerous studies of the effect of condensate inundation on the condensing heat transfer coefficient have been made but have been primarily concerned with plain tubes. The Nusselt analysis [54] assumes that all the condensate from a given tube drains as a continuous laminar film directly onto the top of the tube below it. It predicts that the average coefficient for a vertical row of N tubes, $\bar{h}_N$, compared to the coefficient for the top tube, h_1, is

$$\frac{\bar{h}_N}{h_1} = N^{-1/4} \tag{8}$$

The Kern [55] model, which assumes that the condensate drains as discrete droplets or columns of liquid that cause disturbances in the condensate film, predicts a less conservative relationship:

$$\frac{\bar{h}_N}{h_1} = N^{-1/6} \tag{9}$$

Fujii [56] improved the Nusselt analysis by noting that when condensate drops onto a lower tube, it does not spread much in the axial direction, as also observed by Shklover and Buevich [57] and more recently by Kutateladze et al. [58]. Evidence indicates that the degree of spreading may be influenced by the film Reynolds number and the tube spacing [59, 60]. Fujii developed two simple models to account for this effect and showed that both models predict average heat transfer coefficients that are higher than those predicted by Nusselt [Eq. (8)]. Numerous experimental studies of the effect of inundation have been made for plain tubes [3] and, in general, the data are very widely scattered, making it very difficult to postulate which of the theoretical equations mentioned above should be used to predict tube bundle performance. In general, however, the Kern expression, Eq. (9), has met with reasonable success.

The effect of inundation for finned tubes is not clearly established at present. In 1948 Katz and Geist [61] measured condensation heat transfer coefficients on six finned tubes in a vertical row, using water, R-12, n-butane, and acetone. They found that the exponent in Eq. (9) was approximately 0.04 rather than 1/6, indicating that the effect of inundation is much less on finned tubes than on plain tubes. Rifert et al. [46] studied condensation of ammonia in a bundle of wire-wrapped tubes and found that the effect of inundation was about the same as for a bundle of plain tubes. Amberger et al. [62] studied steam condensation in a bundle of 80 tubes. They showed that the use of Emralon-330 nonwetting strips on the bottom of the tubes gives an improvement in the overall heat transfer coefficient for the bundle of 50% over plain tubes at tube-side water velocities of 1–2 m/s. This increase is considerably more than the 20% increase observed for a single tube as mentioned above [51]. This additional improvement for tube bundles is presumably due to a change in condensate drainage characteristics when the tubes contain the nonwetting strips. Marto and Wanniarachchi [63] obtained inundation data for steam condensing on wire-wrapped tubes in a vertical row. Additional tubes in a bundle were simulated by flooding the top tube with condensate from a perforated tube. They showed that the inundation exponent

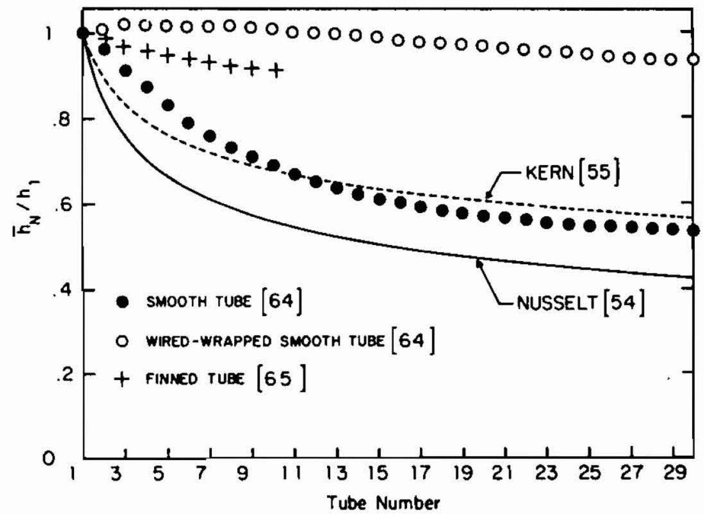

Figure 10 Influence of condensate inundation on the average heat transfer coefficient of steam condensing on smooth and enhanced tubes [64, 65].

with a wire wrap was less than 0.1 and varied with wire spacing. Brower [64] continued these measurements to include several wire diameters and pitches, and he showed that the best inundation performance was obtained when the tubes were wrapped with 1.6-mm-diameter wire at a wire pitch of 8 mm. His results are shown in Fig. 10, along with the predictions of Nusselt and Kern. Clearly, the wire-wrapped tubes experience much less deterioration than the smooth tubes. Recent unpublished data for finned tubes [65] are included in Fig. 10, and they also show that the finned tube outperforms the smooth tube. Presumably, the presence of wires or fins on the tube surface prevents the condensate that drops on the tube from spreading axially. Therefore, there are portions of the tube that are not indundated with condensate, and on the portions that are inundated, the wires or fins draw the condensate toward the fillets, thinning the film. As a result, the effect of condensate inundation is substantially reduced.

CONCLUSIONS

Significant progress has been made in understanding fluid flow and heat transfer during film condensation on integral-fin tubes. Systematic data on finned tubes with various fin shapes and spacings are being generated, and new theoretical models are being developed. However, a better understanding of heat transfer in fully flooded channels is needed to explain the high heat transfer rates that have been observed in this situation. Also, several important and fundamental questions remain unanswered: Why is the heat transfer enhancement larger for liquids with low σ/ρ? What fin and trough shapes for a particular combination of liquid and wall material will yield the maximum heat transfer coefficient? What local convective mechanisms occur in thin, surface tension-controlled condensate films? How will the combined effects of condensate inundation and vapor shear influence these results? More research is needed on wire-wrapped tubes and other extended surfaces, as well as on surfaces containing nonwetted strips, to determine condensate motion on and drainage from both single tubes and tube bundles. Compound enhancement techniques that include extended surfaces, enhanced drainage devices, and/or turbulence promoters should receive further attention. Finally, a strong effort should be made to convert the results of experiments and complex analytical models into useful design equations.

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