CONF-720307--1

RECEIVED BY IIC MAR 10 1972

MAILED

PROSPECTS FOR FURTHER IMPROVEMENT IN ENHANCED
HEAT TRANSFER SURFACES*

by

David G. Thomas

Oak Ridge National Laboratory
Oak Ridge, Tenn. 37830

Prepared for the Second Conference on
Water Desalination, Central Salt & Marine
Chemicals Research Institute, Bhavnagar 2,
India, March 1-3, 1972

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Atomic Energy Commission, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

* Research sponsored by the Office of Saline Water under Union Carbide Corporation's contract with the U. S. Atomic Energy Commission.

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

249

INTRODUCTION

The rate of evaporation and condensation from thin films of water falling down vertical tubes is markedly increased by introducing many fine axial grooves into the smooth surface.¹⁻¹² Typically the grooves have a pitch of ~ 0.3 cm and penetration of ~ 0.1 cm. Such enhanced surfaces are often referred to as fluted or corrugated and frequently overall heat-transfer coefficients as large as $1600 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$ have been reported when evaporating water at atmospheric pressure. In contrast the overall coefficient for thin film evaporation and condensation obtained with smooth tubes under similar conditions¹³⁻¹⁵ is $\sim 650 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$. Since heat transfer area contributes some 40% to overall capital costs of long tube vertical evaporator distillation plants,¹⁶ use of enhanced heat transfer surfaces affords an attractive possibility of reducing water costs. As Kays¹⁸ has pointed out, the reduction in water costs when using enhanced surfaces would be expected to occur largely through reduction of capital costs. Clearly enhanced surfaces permit a reduction in heat transfer area requirements and in addition, some plant components such as pressure vessel can be made smaller. Another option is to increase the number of stages by reducing the temperature difference across each stage, thereby increasing the plant performance ratio.¹⁷⁻¹⁸ Of course, realization of the cost reductions depends on developing economical methods for producing enhanced surfaces and operating procedures for ensuring that high heat-transfer coefficients can be maintained in practice.¹⁷

FLUTED ENHANCED SURFACES

Gregorig's original fluted surface design¹ had grooves with rounded bottoms and ridges with rounded tops, a pitch of 0.2 cm and a penetration of 0.07 cm. Since then surfaces have been tested which had grooves with round bottoms and ridges with flat tops, with vee shaped grooves, and with grooves formed by corrugating flat sheet.⁵ The pitch has been varied from 0.3 cm to 1.5 cm and the penetration from 0.06 to 0.23 cm.⁷ Despite the range of shapes and dimensions the overall heat-transfer coefficient (after subtracting the metal wall resistance) varied only from 1330 to 1900 Btu/hr·ft²·°F at atmospheric pressure, with a ΔT of 12°F and a Reynolds number, ($4 \Gamma/\mu$), of ~8000. The principal factor affecting the magnitude of the overall coefficient appeared to be⁷ the ratio of groove pitch, p , to penetration, e , with maximum values of the coefficient at a value of $p/e = 2.8$. The nature of the surface shape seemed to have only a secondary effect.⁷ Other factors which cause increased coefficients with fluted tubes are surfactants and salt at sea water concentrations,⁵ with overall coefficients in the range 2000 to 3000 Btu/hr·ft²·°F often observed in atmospheric pressure tests.

The principal development effort on enhanced surfaces for vertical tube evaporators has been on variants of the fluted tube surface in which the shape of the grooves and ridges are roughly mirror images. A variety of such shapes are now commercially available from tubing manufacturers and the performance described above can be considered the current state-of-the-art for present day plant designs.

In a recent survey,¹⁸ it was found that whereas seamless tubes cost slightly less than \$1.00 per pound, doubly fluted tube costs were roughly \$2.00 per pound. Since heat transfer coefficients for the two types of surfaces at atmospheric pressure are 650 to 700 and 1300 to 1600 Btu/hr·ft²·°F, respectively, the cost of enhancement largely cancels any benefits from the increased heat-transfer coefficient of doubly fluted tubes. Consequently, to realize the full benefits of enhanced surfaces either the cost of enhanced surfaces must be decreased or the overall heat-transfer coefficient must be increased with no additional cost penalty.

ADVANCED ENHANCED SURFACES

Subsequent to the discovery of the marked effect of fluted surfaces on the rate of evaporation and condensation from thin films on vertical surfaces, a variety of experimental surfaces were developed which had substantially larger heat-transfer coefficients than conventional fluted tubes.¹⁹⁻²⁴ The chronology of these events is illustrated in Fig. 1, in which the range of overall heat-transfer coefficients for the different surfaces is shown plotted against the approximate time that the new surface was discovered. (The references cited on the figure do not necessarily coincide with the date of discovery but are literature sources describing the surfaces.)

In 1967 Linde announced^{19,25} that a specially textured surface markedly increased the evaporating heat-transfer coefficient; little information is available on the nature of the textured surface. When the textured surface was combined with fluted, finned, or gold plated surfaces on

the condensing side, overall coefficients were 2700 to 3200 Btu/hr·ft²·°F with fresh water feed at atmospheric pressure but the coefficients decreased to ~2400 Btu/hr·ft²·°F with salt water feed.⁶

In 1967 it was also shown²⁶ that small axial wires increased the condensing coefficient up to 4.5 times when loosely strapped to the surface of vertical tubes. With this discovery it became clear that the configuration of the surface was not as important as providing regions at intervals around the tube where the bulk of the condensate could be drained off as a rivulet. Based on results of a simplified theoretical model, it was expected that further increases in condensing coefficient would be affected by reducing the amount of tube surface masked by the rivulet guide. When round wires were replaced by loosely attached fins, the condensing coefficient was increased by a factor of ~9 over the smooth tube values.²⁰ When similar fins were placed on the evaporative side of a tube, the evaporative heat-transfer coefficient increased by up to a factor of 12 over the value for smooth tubes.²¹

By combining fins on both the evaporative and condensing sides of a vertical tube, it was possible to achieve overall heat-transfer coefficients in the range 2500 to 5000 Btu/hr·ft²·°F with the higher heat-transfer coefficients occurring at a heat flux of 10⁴ Btu/hr·ft².

Finally, based on concepts developed in the finned tube studies, a general corrugated surface configuration was designed.²³ When one version was tested,²² overall coefficients for thin film evaporation and condensation were in the range 3300 to 3800 Btu/hr·ft²·°F. A sample of the corrugated surface is shown in Fig. 2, Item C. Also shown in this figure

are portions of tubes fluted with General Electric profile 9 (Item A) characterized by ridges with flat tops and the Anaconda-Grob profile (Item B) characterized by vee-shaped grooves.

Both the double finned and corrugated surfaces share the special characteristics that (1) the thin evaporating film opposes the thin condensing film so that the heat flow path through the metal wall is minimized and (2) the ratio of rivulet guide circumferential width to thin-film heat transfer circumferential width is the minimum consistent with the water carrying capacity required for the rivulet guide.

The response of different heat transfer surfaces to the temperature difference, ΔT , between the evaporating tube side and the condensing tube side is illustrated in Fig. 3. For smooth tubes,^{5,24} the value of U is substantially independent of ΔT . This is also true for both the fluted⁵ and corrugated^{22,23} tubes when the evaporative film flow rate is ~ 1400 lb/hr·ft. On the other hand the overall coefficient of fluted tubes²⁷ and double ribbed tubes²⁴ increased markedly as ΔT decreased when the evaporative film flow rates were 630 and ~ 340 lb/hr·ft, respectively. This behavior may have important consequences in the design of high performance plants in which the number of stages must be increased to improve the performance ratio.¹⁷

The object of this paper is to examine prospects for further improvement in enhanced heat transfer surfaces. From the point of view of current state-of-the-art of manufacturing capability, double fluted tubes must serve as the baseline for comparison. At atmospheric

pressure, overall heat-transfer coefficients of $\sim 1700 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$ are being specified in designs of large VTE plants²⁸ which will use doubly fluted tubes made from a Cu-Fe alloy, [$k = 150 \text{ Btu/hr}\cdot\text{ft}^2 (\cdot^\circ\text{F}/\text{ft})$].

Based on the results shown in Figs. 1 and 3, it is clear that experimental test surfaces exist which have 1.5 to 2 times greater overall heat-transfer coefficient. However, as far as is known, these surfaces are not available commercially. In the light of our poor understanding of the mechanisms responsible for these high coefficients, it would seem prudent to make additional studies of temperature dependence of the heat-transfer coefficient, susceptibility of the surface to scaling or fouling, etc., before attempting to go into commercial production.

In attempting to assess whether there are prospects for further improvement in enhanced heat-transfer surfaces, there are two somewhat different but related questions.

1. What is the optimum shape for "conventional" fluted surfaces and is there a special configuration that will lead to heat-transfer coefficients well above $2200 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$?
2. Can a surface shape be developed that will represent a marked improvement over current experimental surfaces? Of interest would be surfaces which either have heat-transfer coefficients markedly greater than the $3500\text{-}5000 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$ indicated in Figs. 1 and 3 on which are practical, inexpensive and easy to manufacture and have heat-transfer coefficients near the upper limit given by existing experimental surfaces.

In view of the large number of "conventional" fluted surfaces that have been tested,⁵ it seems likely that the most fruitful approach to answering question 1 would be from increased understanding of the mechanisms involved in producing the enhanced heat-transfer coefficients.

Although the designs of some of the experimental surfaces²⁰⁻²⁴ were based on simplified mathematical models, the models are far from complete. Consequently the answers to question 2 appear to be most readily approached through improved models based on the fundamental mechanisms involved. Further, with overall coefficients as large as 5000 Btu/hr·ft²·°F, other factors may be of more importance than improving the thin film evaporation and condensation heat-transfer coefficients. For instance, a recommended fouling factor for fluted copper alloy surfaces is 0.0002; a reasonable tube wall resistance may be as low as 0.000027. With these values and assuming that the evaporating and condensing film heat-transfer coefficients could be made infinite, the overall design coefficient would be only 4400 Btu/hr·ft²·°F. For the corrugated tube, made from 0.004-in. thick stainless steel, such a fouling factor would reduce the observed experimental coefficient from 3400 Btu/hr·ft²·°F to a design coefficient of ~2400 Btu/hr·ft²·°F. This indicates that perhaps the greatest impact on enhanced surface VTE plant performance awaits increased understanding and control of the fouling problem.

MECHANISM OF SURFACE ENHANCEMENT

Condensing Side

In his paper describing the effect of a grooved surface on thin film condensation, Gregorig¹ hypothesized that, because of the shape of the grooved tube, surface tension forces in the thin liquid film tend to draw condensate from the crests of the surface into the grooves between crests. The bulk of the condensate drains off in the grooves leaving a thin liquid film on the crests where heat-transfer coefficients can be extremely large.

The problem of obtaining an analytical solution for flow down a fluted surface is difficult, not only because of the well known problem in obtaining a solution to the equations of motion for a three dimensional flow situation but because the surface tension is introduced through the boundary condition at the vapor-liquid interface³¹

$$\Delta p - 2\mu \left(\frac{\partial u_i}{\partial x_k} + \frac{\partial u_k}{\partial x_i} \right) = \left(\frac{1}{R_1} + \frac{1}{R_2} \right) \sigma \quad (1)$$

and of course this boundary condition is just what determines the shape of the interface to which it is to be applied. Although no solutions have been reported for the equations of motion with fluted tube geometry and boundary conditions given by Equation (1), it is interesting to note that, taking into account the pressure variation with height, the equilibrium at an interface between vapor and liquid is specified by³¹

$$\rho g_L Z - \sigma \left(\frac{1}{R_1} + \frac{1}{R_2} \right) = \text{constant} . \quad (2)$$

The only parameter that can be formed from the nongeometric quantities is $(\sigma/\rho g_L)^{1/2}$ with the dimensions of length. Batchelor³¹ interprets this as being the length scale for which surface tension effects may be comparable with gravitational effects. For water this characteristic length varies from 0.27 to 0.24 cm in the temperature range 80 to 240°F. It is notable that the value of the parameter for water is substantially the same as the pitch of the grooves in many of the fluted tubes.⁷

In his analysis of the condensation phenomenon, Gregorig¹ postulated two-dimensional flow, a constant film thickness over the crest and then assumed that the pressure gradient in the liquid film was given by $\Delta p = \sigma/R$. With those assumptions Gregorig was able to calculate an "optimum" surface shape as a function of film thickness. Since film thickness was a parameter in the calculation, heat-transfer coefficients were not obtained on an a priori basis. Subsequent studies² indicate that the magnitude of the condensing heat-transfer coefficients obtainable with grooved surfaces were not so much a function of surface shape as they were of pitch and depth of the grooves.

In analyzing film condensation results obtained with small wires or fins loosely attached to the surface of vertical tubes, Thomas^{20,26} divided the flow into two regions, a thin film between the wires or fins where the heat-transfer coefficient was high and a rivulet adjacent to the wires or fins. The bulk of the condensate was assumed to flow down the tube in the rivulet and heat transfer in the rivulet region was assumed to be negligible. After introducing the surface tension on dimensional grounds, it was possible to obtain an expression for the ratio of heat-transfer coefficient with fins (or cylinders) and the

smooth tube heat-transfer coefficient, h/h_o . For finned tubes the final expression was

$$h/h_o = \left[1 - \left(\frac{Nd}{\pi D} \right) \left\{ 1 + 3.2 \left(\frac{\sqrt{f} D}{g_L \rho Nd^4} \right)^{1/4} \right\} + 1.03 \times 10^{-3} G \left(\frac{g_c \sigma L}{\sqrt{f}} \right)^{3/2} \left(\frac{\sqrt{f} N^3}{g_L \rho D^3} \right)^{5/8} \right]^{1/3} \left[1 - \left(\frac{Nd}{\pi D} \right) \left\{ 1 + 3.2 \left(\frac{\sqrt{f} D}{g_L \rho Nd^4} \right)^{1/4} \right\} \right] \quad (3)$$

The value of G , determined experimentally, was 1.72×10^{-6} . Differentiation of Equation (3) to obtain the optimum fin spacing gave

$$\left(\frac{Nd}{\pi D} \right)_{opt} \approx \frac{5}{13} \left[\frac{1}{1 + 2.7 \left(\frac{\sqrt{f} D}{g_L \rho Nd^4} \right)^{1/4}} \right] \quad (4)$$

substantially independent of the empirical constant G as well as the surface tension and viscosity of the fluid. At a heat flux of 5×10^4 Btu/hr·ft², Equation (4) indicates an optimum fin spacing of ~0.25 cm. This is about the same value as has been adopted for the pitch of grooves in a fluted surface⁷ as well as being almost the same as the characteristic length obtained from Equation (2).

Thomas²⁰ also suggested a criterion for flooding (i.e., loss of heat transfer enhancement) based on results for cylindrical wires loosely attached to the surface of vertical tubes. Flooding was predicted to occur on tubes with cylindrical wire fins when

$$4.77 \left(\frac{\sqrt{f} D}{g_L \rho Nd^4} \right)^{1/4} > 1 \quad (5)$$

This criterion, adjusted for fin-type geometry by use of the constant 3.2 [Equation (3)] instead of 4.77, was subsequently used^{22,23} to establish the height of corrugations for the surface shown as "C" in Figure 2.

Alexander and Hoffman⁷ have shown from their experimental studies that the optimum pitch-to-penetration, p/e , for fluted tubes (for maximum heat-transfer enhancement) is less than $(p/e) = 2.8$. If flooding is assumed to be responsible for the poorer performance of surfaces with $(p/e) > 2.8$, then for fluted surfaces the appropriate value of the numerical constant in Equation (5) should be ~ 2.9 . The principal reason for the differences in numerical coefficient for different surfaces is believed to be due to the different rivulet shapes.

As shown in Figure 4, the performance of the finned surface in enhancing film condensation follows the same trend with heat flux as does data for fluted tubes and furthermore the enhancement of film condensation by the finned surface is equal to or somewhat better than the highest values reported for fluted surfaces. Thus, for the two extremely different surface shapes (finned and fluted) the dimensions of the grooves or fins rather than the specific shape appear to be controlling.

Evaporating Side

The principal difference between thin film evaporation and thin film condensation on corrugated, fluted or finned surfaces is that in evaporation liquid must be supplied continuously to the thin film to balance vaporization whereas in condensation the condensate is drawn from the thin film to the rivulet where it drains away. Although it is commonly accepted that surface tension forces play a predominant role in drawing condensate to the rivulet, mechanisms on the evaporative side are not so well understood.

It does seem to be commonly accepted that on the evaporative side as on the condensing side, surface tension forces draw the liquid film into rivulets which flow down the tube either in the grooves on fluted surfaces or along the base of fins on finned surfaces. The principal problem is elucidating the evaporative side mechanism is explaining how the thin film of liquid between rivulets is continuously supplied to maintain high rates of evaporation.

Carnavos³ and Diedrich and Lotz² proposed that nucleate boiling in the rivulet in the grooves kept the crests constantly wetted by splashes from the bubbling accompanying the boiling. However, Alexander and Hoffman⁵ were unable to observe nucleate boiling on the evaporative side when the mass flow rate was 1400 lb/hr·ft. Instead they observed roll waves moving down the surface of the tube with a wave velocity of ~140 cm/sec; the surface velocity of a completely laminar film falling down a smooth tube at the same mass flow rate was calculated to be ~160 cm/sec. The maximum frequency of waves was ~16 sec⁻¹ and the amplitude of these waves was ~0.05 cm. Between the passage of waves the crests of the fluted tube surface was clearly visible indicating that after passage of a wave which wetted the crest, surface tension forces rapidly drew the bulk of the remaining liquid into the grooves permitting high rates of evaporation from the crests.

In studies of evaporation from finned surfaces mass flow rates were ~100 lb/hr·ft, much smaller than the value of 1400 lb/hr·ft used in the fluted tube studies.^{21, 24} With these small flow rates it seems unlikely that waves would form, particularly since the fins had a height of ~0.25

cm. Consequently, Thomas and Young²¹ postulated that the tube surface was continuously wetted by small waves analogous to those observed to precede dry spot formation at very low flow rates. The phenomenon has been named "roping" by Norman and McIntyre³⁰ who observed that as the flow rate down a tube was decreased to the order of $4 \text{ g}/\mu \sim 200$, the water film became thicker in several vertical bands spaced around the tubes. Although there was a tendency for dry patches to form between ropes, the patches were quickly covered over as the film became very disturbed.

Johnson et al.⁸ studied evaporation from fluted surfaces at mass flow rates up to $\sim 360 \text{ lb/hr}\cdot\text{ft}$. Under these circumstances, they postulated that Marangoni instabilities driven by surface tension gradients (which are caused by temperature and concentration gradients in the film between grooves and crests) were responsible for the rapid renewal of the thin film on the crest of the fluted surface. They estimated that surface velocities of $\sim 500 \text{ cm/sec}$ might accompany the Marangoni instabilities. Forced convection at these large lateral velocities could then account for the large evaporative heat-transfer coefficients observed with fluted tubes.

Benjamin³⁶ has shown that even in the absence of surface tension gradients a falling liquid film on a vertical wall is unstable at all Reynolds numbers. The analysis showed that as the Reynolds number becomes very small the rates of amplification also become small and the wave length becomes large. However, since the amplification is an exponential function of the square of the film Reynolds number, amplification becomes very large after a certain Reynolds number is exceeded.

This for all practical purposes would appear as a "critical" Reynolds number for wave formation. Figure 5 is a plot of amplification vs film Reynolds number with the distance for wave amplification as a parameter calculated from Benjamin's analysis for water at $\sim 100^\circ\text{C}$. Clearly amplification becomes very large at film Reynolds numbers greater than ~ 1.5 . Cerro and Whittaker³⁷ have shown experimentally that waves can be observed on water films at Reynolds numbers as low as 1.5. However, the first waves appeared ~ 1 cm below the top of the film. At a Reynolds number of 1.5 the film velocity is approximately 0.5 cm/sec; this implies an amplification time for waves at this low Reynolds number of ~ 3 seconds. This time seems much too large for the isothermal wave analysis of Benjamin to have any application to the problem at hand especially in view of the fact that Alexander and Hoffman⁵ have observed high evaporative heat-transfer rates in the presence of roll waves which inundated the surface up to 16 times per second. This observation implies that the time scale for the thin film phenomena responsible for the high heat transfer rates must be less than ~ 0.06 seconds.

Brian³⁸ has pointed out that under circumstances where a falling liquid film might develop Marangoni instabilities, there might be an interaction between the two modes of instability. Brian also suggested that such instabilities might manifest themselves in extremely short times.

With this in mind it would be instructive to attempt to estimate a characteristic time for surface tension induced flows. One such time could be obtained by dividing the length scale for surface tension effects

deduced from Equation (2) by the velocity induced in the thin film by surface tension effects:

$$t_{\sigma} = \frac{(\sigma/\rho g_L)^{1/2}}{u_{\sigma}} \quad (6)$$

The problem is to obtain a value for the surface tension induced velocity. In the model for film condensation enhancement on finned surfaces developed by Thomas,²⁰ this velocity was given by

$$u_{\sigma} = \frac{\alpha_1 g_c \sigma \delta}{2\mu R} \quad (7)$$

where α_1 = empirical constant = 4×10^{-2} and

$$\delta = \left(\frac{3}{2\alpha_1} \right)^{1/3} \left(\frac{\pi \Gamma D R \nu}{g_c \sigma N L} \right) \quad (8)$$

Assuming that $\pi D/N$, (the fin or flute pitch) is given by the characteristic surface tension length, $(\sigma/\rho g_L)^{1/2}$, then Equation (6) becomes

$$t_{\sigma} = 30 \frac{R^{1/2} L^{1/2}}{g_L^{1/2} (4\Gamma/\mu)^{1/2} (\sigma/\rho g_L)^{1/4}} \quad (9)$$

For $4\Gamma/\mu = 200$ (the value of the Reynolds number close to dry spot formation on short tubes according to Norman and McIntyre³²), $L = 1$ cm (the value at which first waves were observed by Cerro and Whittaker³⁷) and assuming that the radius of curvature of the fluid is 0.01 cm, then for water at 212°F, Equation (9) indicates a characteristic time of ~0.01 seconds. This time is sufficiently small to indicate that interactions between two (or more) mechanisms must be involved on the evaporative side of enhanced surfaces.

Further indication of this is given by examination of the temperature dependence of the overall heat-transfer coefficient in thin film evaporation and condensation, Fig. 6. This figure is a plot of $\log U$ vs T for smooth tubes with 0-salinity feeds, fluted tubes with 0-salinity feed and fluted tubes with sea water feed. It seems remarkable that, within each set of conditions, the slopes of the lines through results of different investigators are substantially the same. The straight line relation shown on Fig. 6 suggests that a temperature coefficient, B , for the different conditions might be defined by

$$\frac{\Delta U}{U} = C + B T \quad . \quad (10)$$

The results shown on Fig. 6 give for B , (when T is in degrees F):

	$B \times 10^3$
Smooth tube, 0-salinity	5.7
Fluted tube, 0-salinity	3.9
Fluted tube, sea water	5.9
Nusselt parameter, $k(g_L/v^2)^{1/3}$	5.9
(surface tension) ⁻¹	1.3

Also included in the above list is the temperature coefficient for the Nusselt parameter for film condensation and that for the reciprocal of the surface tension. The good agreement of the temperature coefficient for the Nusselt parameter and for the smooth tubes indicate that substantially all of the loss in heat-transfer coefficient with smooth tubes as temperature is reduced, may be accounted for by variation of physical properties included in the Nusselt parameter. The markedly smaller value of B observed with fluted tubes and 0-salinity feed indicates that other physical properties are contributing. The low temperature coefficient for

the reciprocal of the surface tension appears to be significant here. Finally, when feed to the evaporating side is sea water the temperature coefficient for fluted tubes is substantially greater than for the same type tubes with 0-salinity feed. This may indicate that Marangoni instabilities induced by concentration gradients are playing an additional role.

In all probability the mechanism of enhancement on the evaporative side is quite complicated with one or more of the phenomena described above making important contributions, and with the relative contribution of the mechanisms a function of mass flow rate. It has often been pointed out that in fluid mechanics discovery of a new phenomenon almost always precedes theoretical analysis. Theoretical explanations are often arrived at only after simplifying the equations of motion and boundary conditions to the greatest extent possible consistent with the particular phenomenon being analyzed. As a consequence, development of theoretical expressions which include all of the mechanisms described above (and others as yet unknown) seems to have a small probability of success. Perhaps more fruitful would be a combined experimental-theoretical approach. First, predominant mechanisms and the range of operating conditions for which they are controlling should be identified experimentally. Then simplified analytical models could be developed which are consistent with the physics of the individual processes and which, when appropriately combined, provide an adequate description of the overall process.

FUTURE PROSPECTS

At the present state of commercial development of enhanced surfaces, overall coefficients with clean fluted tubes are $\sim 1800 \text{ Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$. In the design of distillation plants it is common practice to account for all forms of fouling and introduce a factor of safety by using a fouling factor.¹⁸ The value recommended in the standards of the Tubular Exchanger Manufacturers Association is 0.0005. If this value is adopted, the design heat-transfer coefficient is reduced from 1850 to 960 $\text{Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$, almost a 50% reduction. Even if Eissenberg's²⁹ recommended value of 0.0002 is used, the design coefficient must be decreased to 1350 $\text{Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$, a 25% reduction. Because fouling factors of this magnitude would be completely controlling if higher performance surfaces were developed, some have suggested that attempts to increase currently available performance is unwarranted.³ An alternative approach would be to develop procedures for insuring that fouling factors could be maintained at the lower limit of values already observed in some tests, say 0.00006 to 0.00008, and to simultaneously develop improved enhanced surfaces to match the improved fouling control. If the latter approach is adopted, it might be instructive to estimate just how much existing performance might be improved by as yet undeveloped surfaces.

Since existing theory seems unlikely to provide guidance, resort must be made to experiment. On the condensing side film coefficients as large as 18,000 $\text{Btu/hr}\cdot\text{ft}^2\cdot^\circ\text{F}$ have been observed with both fluted and finned surfaces.^{3,20} This value will be adopted as a reasonable upper limit for performance of enhanced surfaces in film condensation.

Observed performance on the evaporating side is not in such good agreement. Carnavos³ reported a maximum value of 7000 Btu/hr·ft²·°F in the early studies of fluted surfaces. Subsequently, Thomas and Young²¹ obtained values as large as 10,000 Btu/hr·ft²·°F with finned surfaces and Johnson et al.⁸ reported a maximum value of 12,600 Btu/hr·ft²·°F with a finely corrugated surface. Because of the complex mechanisms that may be involved on the evaporating side, it is not clear that these values represent an acceptable upper limit.

Unexpectedly large evaporative film coefficients have been obtained on very short smooth tube surfaces by reducing the flow rate to values approaching that for which dry spot formation occurs.³² Norman and McIntyre's³² results, obtained with tubes 8.2 and 23 cm long, are shown on Fig. 7 as $h/(kg_L^{1/3}/v^{2/3})$ vs $4\Gamma/\mu$, coordinates suggested by Nusselt's analysis of film condensation. Also shown are new results for low flow rates with tubes of 68 and 112 cm length. Several aspects of these results are notable. All of the low flow rate results indicate a maximum in the evaporative film coefficient in the Reynolds number range 200 to 800. There is an orderly increase in the maximum heat-transfer coefficient as tube length is decreased and for the shortest tube the heat-transfer coefficient is over 6 times greater than the value expected for convection through a laminar film (i.e., the result from Nusselt's analysis).

The maximum values of the heat-transfer coefficient from Fig. 7 are plotted in Fig. 8 as h_{evap} vs tube length; also shown are mass flow rates at the top and bottom of the tubes. It is clear that for short tubes and low mass flow rates there is a marked increase in evaporative heat-transfer coefficient as tube length is decreased and that as tube length is increased the mass flow rate must be increased to attain the maximum heat-transfer coefficient.

It seems likely that the vigorous (and as yet unexplained) agitation in thin films observed by Norman and McIntyre³² is the principal reason for the unexpectedly large heat-transfer coefficients. Further, as tube length is increased feed rate to the tube must be increased to prevent dry spot formation and this increased feed rate must have a disproportionate effect on damping the agitation in the thin films.

Be that as it may, the results shown in Fig. 8 may be used to estimate the upper limit for evaporative film coefficients. The heat-transfer coefficient for zero tube length was obtained by extrapolation on a plot of $\log (1/h)$ vs tube length. The value obtained was 14,200 Btu/hr·ft²·°F at a feed rate of 30 lb/hr·ft. Combining this value with the upper limit for film condensation given above (18,000 Btu/hr·ft²·°F) and assuming a metal wall resistance of 0.00004 gives an estimated upper limit for the overall heat-transfer coefficient of ~6000 Btu/hr·ft²·°F assuming a rivulet guide can be designed which occupies a negligible amount of surface. This coefficient is about 1.5 times the value obtained with corrugated tubes^{22,23} and only about 1.2 times the value obtained with doubly finned tubes²⁴ at heat fluxes of ~10 Btu/hr·ft².

CONCLUSIONS

The mechanism responsible for high performance on the condensing side of fluted or finned tubes seems relatively well understood, at least qualitatively. The mechanism on the evaporating side is apparently much more complicated and not at all well understood.

Estimates of the upper limits on film coefficients on the condensing side and on the evaporative side indicate that an upper limit on the overall coefficient for enhanced surfaces, as they presently are conceived, might be as large as 6000 Btu/hr·ft²·°F. However, such large values will have little impact on desalination costs unless reliable procedures are developed for reducing the fouling factor to the lower limit of 0.00006 to 0.00008 sometimes observed in practice.

NOMENCLATURE

G	Constant in Equation (3) = 1.72×10^{-6} , dimensionless
c_p	Specific heat, Btu/lb _m
d	Wire diameter or fin width, ft
D	Tube diameter, ft
e	Penetration, ft
g_c	Conversion factor, (lb _m /lb _f)(ft/hr ²)
g_L	Gravitational acceleration, ft/hr ²
h	Heat-transfer coefficient with wires, Btu/hr·ft ² ·°F
h_o	Heat-transfer coefficient without wires, Btu/hr·ft ² ·°F
k	Thermal conductivity, Btu/hr·ft ² °F/ft
L	Length, ft
l	Amplification length, ft
N	Number of wires or fins, dimensionless
N_{Re}	Reynolds number, $4\Gamma/\mu$, dimensionless
N_{Pr}	Prandtl number
P	Wetted perimeter, ft
p	Pitch, ft
Δp	Pressure difference, lb _s /ft ²
q/A	Heat flux, Btu/hr·ft ²
R	Radius of curvature, ft
t_σ	Characteristic time for surface tension effects, dimensionless
u_σ	Local velocity due to surface tension effects, ft/sec
U	Overall heat-transfer coefficient, Btu/hr·ft ² ·°F
Z	Height, ft

Greek Letters

- Γ Mass flow per unit perimeter, $\text{lb}_m/\text{hr}\cdot\text{ft}$.
- δ Film thickness, ft
- μ Viscosity, $\text{lb}_m/\text{ft}\cdot\text{sec}$
- ν Kinematic viscosity, ft^2/hr
- ρ Density, lb_m/ft^3
- σ Surface tension, lb_f/ft

REFERENCES

1. Gregorig, V. R., Zeit. fur. ang. Math. Physik, 5: 36-49 (1954).
2. Diedrich, G. E. and C. W. Lotz, U. S. Patent 3,291,704, December 1966.
3. Carnavos, T. C., Proc. 1st Int. Symp. on Water Desalination, Washington, D. C., October 3-9, 1965, 2: 205 (1967).
4. Dukler, A. E. and L. C. Elliott, Office of Saline Water R&D Report 287, 1967.
5. Alexander, L. G. and H. W. Hoffman, Office of Saline Water R&D Report 644, January 1971.
6. Alexander, L. G., J. D. Joyner, and H. W. Hoffman, Symposium on Enhanced Tubes for Desalination Plants, Office of Saline Water, July 1970, 3-43.
7. Alexander, L. G. and H. W. Hoffman, Paper 71-HT-30, ASME-AIChE Heat Transfer Conference, Tulsa, Oklahoma, August 15-18, 1971.
8. Johnson, B. M., G. Jansen, and P. C. Owzarski, Paper 71-HT-33, ASME-AIChE Heat Transfer Conference, Tulsa, Oklahoma, August 15-18, 1971.
9. Khan, A. R., Paper 71-HT-34, ASME-AIChE Heat Transfer Conference, Tulsa, Oklahoma, August 15-18, 1971.
10. Kays, D. D. and W. S. Chia, Paper 71-HT-40, ASME-AIChE Heat Transfer Conference, Tulsa, Oklahoma, August 15-18, 1971.
11. Sieder, E. N. and I. Spiewak, Nuclear Desalination, Elsevier Publishing Co., 1969, p. 829.
12. ORNL-Staff, Distillation Digest, 2(1): 41 (1968).
13. Standiford, F. C. and H. F. Bjork, Advances in Chemistry, 27: 115 (1960).
14. Kroll, J. E. and J. W. McCutchan, Trans. ASME J. Heat Transfer, 90: 201 (1968).
15. Bromley, L. A. and S. M. Read, Desalination, 7: 343-391 (1970).
16. ORNL, "Conceptual Design Study of a 250-MGD Combined Vertical Tube-Flash Evaporator Desalination Plant," Office of Saline Water R&D Report 391, August 1968.

17. Silver, R. S., Desalination, 9: 235-243 (1971).
18. Kays, D. D., "Symposium on Enhanced Tubes for Desalination Plants," Office of Saline Water, July 1970, pp. 327-336.
19. Linde Company, Distillation Digest, 1(1): 63 (1967).
20. Thomas, D. G., AIChE Journal, 14: 644 (1968).
21. Thomas, D. G. and G. Young, Ind. Eng. Chem. Process Design Develop., 9: 317 (1970).
22. Thomas, D. G. and L. G. Alexander, Desalination, 8: 13 (1970).
23. Thomas, D. G., U. S. Patent 3,568,766, March 9, 1971.
24. Thomas, D. G. and P. H. Hayes, Ind. Eng. Chem., 62(2): 4 (1970).
25. Czikk, A. M. and P. S. O'Neill, "Symposium on Enhanced Tubes for Desalination Plants," Office of Saline Water, July 1970, pp. 231-245.
26. Thomas, D. G., Ind. Eng. Chem. Fundamentals, 6: 97 (1967).
27. General Electric Company, Office of Saline Water R&D Report 181, 1966.
28. Environgenics Company, 200 MGD Desalination Plant and VTE/MSF Module Conceptual Design, November 3, 1970.
29. Eissenberg, D. M., Distillation Digest, 4(2): 11-24 (1970).
30. Lustenader, E. L. and F. W. Staub, INCO Power Conf., Session II, May 5-8, 1964.
31. Batchelor, G. K., An Introduction to Fluid Dynamics, Cambridge University Press, 1967.
32. Norman, W. S. and V. McIntyre, Trans. Instn. Chem. Engrs., 38: 301 (1960).
33. Christ, A., Escher Wyss News, 35: 94 (1962).
34. Unterberg, Walter and D. K. Edwards, AIChE Journal, 11: 1073 (1965).
35. Chun, K. R. and R. A. Seban, Trans. ACME J. Heat Transfer, 93: 391 (1971).
36. Benjamin, T. B., J. Fluid Mech., 2: 554 (1957), corrigendum 3: 657 (1957).
37. Cerro, R. L. and S. Whittaker, Chem. Eng. Sci., 26: 742 (1971).
38. Brian, P. L. T., Chem. Eng. Sci., 23: 1513 (1968).

Fig. 1 Chronology of Development of Enhanced Heat Transfer Surfaces for Vertical Tube Evaporators.

Fig. 2. Commercial and Experimental Enhanced Surfaces for Vertical Tube Evaporators. A. General Electric profile 9, B. Anaconda-Grob, C. Corrugated surface, (Thomas, 1971).

Fig.3 . Effect of Overall Temperature Difference on Overall Heat Transfer Coefficient in Thin Film Evaporation and Condensation.

Fig. 4 Comparison of the Maximum Observed Effect of Fluted and Finned Tubes on the Film Condensation Heat Transfer Coefficient.

Fig.5 . The Amplification Experienced in a Distance l by the Most Unstable Wave on a Vertical Water Film at Saturation. (After Benjamin, 1957)

Fig. 6 Temperature Dependence of Overall Heat Transfer Coefficient.

Fig. 7 Heat Transfer Coefficient for Thin Evaporating Films on Vertical Surfaces.

Fig. 8 Maximum Evaporative Film Heat Transfer Coefficient Observable at Low Flow Rates on Smooth Vertical Tubes.