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A Critical Review
of Heat Transfer Enhancement Techniques
for Use in Marine Condensers

by

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A variety of heat transfer enhancement techniques are reviewed and evaluated for use in marine condensers. This review includes tube-side enhancement, as well as shell-side enhancement techniques. At present, the most promising technique to enhance heat transfer on the tube side is with one of the commercially available, mildly indented, corrugated tubes. With this type of tube, the shell-side heat transfer enhancement is not as large and is

uncertain due to the, as yet determined, complex effects of vapor shear and condensate inundation. Large bundle tests with enhanced tubing are therefore required before the details of tube-to-tube interaction will be fully understood.

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1. INTRODUCTION

Since the early days of naval steam propulsion, the surface condenser has evolved into a very reliable component of the steam propulsion machinery plant. This reliability has been achieved by providing generous design margins to insure thermal performance at full power. The penalties for this overdesign, however, are additional weight and volume (which must be carried around for the life of the ship) and crowded machinery rooms with poor accessibility. Although surface vessels and submarines have different design constraints, both types of vessels can benefit from a more compact condenser design through improved heat transfer. The actual dimensions of the condenser may have an impact on vessel performance and cost [1]. In addition, the future development of compact steam systems, both for main propulsion and secondary heat recovery purposes, will require the application of advanced technology to all system components, including the condenser.

The purpose of this report is to review promising heat transfer enhancement schemes which may be suitable for use in naval surface condensers, and to identify important areas where further research is needed.

2. SINGLE TUBE HEAT TRANSFER ENHANCEMENT

In recent years, there has been an increased awareness regarding the use of enhanced heat transfer surfaces in the design of heat exchangers [2, 3, 4, 5, 6, 7]. Webb in 1980 provided an excellent review of enhancement methods for particular use in condensers [8]. In general, these methods may be divided into tube-side enhancement (on the cooling water side) and shell-side enhancement (on the steam side) techniques. Enhancement on the tube side, where single phase turbulent forced convection occurs, has included the use of rough surfaces, internal fins, and helical flutes. Shell-side enhancement has included surface coatings, low integral fins, fluted tubes and roped tubes. A description of those techniques which appear most promising for naval condensers is provided below.

The largest thermal resistance to heat flow in conventional surface condensers is usually, though not always, on the tube side, so that tube-side enhancement may be expected to be of greater benefit. In fact, it may be feasible to simply increase cooling water velocity to increase the inside coefficient without using any other enhancement technique. With titanium tubes, for example, since there is relatively little danger of erosion at higher water velocities, this simple technique may prove to be attractive. By retaining the smooth-tube configuration there would be no internal flutes, fins or recesses with the attendant concerns about fouling and cleaning. On the shell side, it may be possible to achieve an overall enhancement with the proper use of condensate baffles to limit inundation effects low in the tube bundle. Although these standard improvement techniques must not be overlooked, the emphasis in this report is upon the less-conventional methods that show promise in the surface condenser application.

2.1 TUBE-SIDE ENHANCEMENT

2.1.1 Rough-Walled Tubes

It is well known that surface roughness, either in the form of random sand grains or regular geometric shapes, can be used to increase turbulent flow heat transfer. A recent patent by Fenner and Ragi [9] describes a method of using three-dimensional roughness to enhance heat transfer. They propose the use of a single layer of randomly distributed metal bodies bonded to the inside of a tube wall. Results of water at a Reynolds number of 35,000 and a Prandtl number of 10 are shown in Fig. 1. As the average height of the roughness bodies increases, the enhancement increases up to a value near 2.5, corresponding to e/D near 0.016. This increase in heat transfer occurs at the expense of an equivalent expenditure of energy to overcome fluid friction. Beyond e/D = 0.016, no further improvement in heat transfer was observed but the increased frictional resistance lowered the performance ratio $(h/h_s)/(f/f_s)$ below unity.

The repeated-rib surface has been studied by Webb, Eckert and Goldstein [10]. They provided generalized correlations for both heat transfer and friction which depend upon roughness element height and spacing. More recently, Han, Glicksman and

Rohsenow [11] showed that repeated ribs at a 45° angle of attack gave better performance than ribs normal to the flow. Fig. 2 shows a schematic of the rib pattern as well as their performance data for air flowing between parallel plates. It is clear that the repeated-rib roughness with a 45° angle of attack is superior to either sand grain roughness, or to ribs placed 90^{0} to the flow, and an enhancement factor, E_i, near 3.0 is possible.

Withers and Rieger [12] describe a commercially available tube (TURBO-CHIL) which has an inner surface containing multiple-helix ridging as shown in Fig. 3. Based upon experimental data obtained by Withers [13], the following tube-side correlations for this type of tubing are proposed:

$$\sqrt{\frac{f}{8}} = - \frac{1}{2.46 \ \ell n[r + (7/Re)^{m}]} \tag{1}$$

and

$$St = \frac{\sqrt{f/8}}{5.68 \ (e/p)^{-1/8} \ \sqrt{Pr}[(e/d_i) Re \ \sqrt{f/8}]^{0.136} + \gamma} \tag{2}$$

where r and m are determined empirally for each tube, and $\gamma = - [2.5 \ \ell n \ (2e/d_i) + 3.75]$. Equations (1) and (2) predict heat transfer enhancement factors as large as 2.5.

2.1.2 Helically Corrugated Tubes

A wide variety of commercially available, helically corrugated tubes exist with varying shapes and flute depths. Figure 4 shows a representative sample of these corrugated tubes. When considering these tubes for use in condensers, the mildly indented or "roped" tubes seem most appealing. These tubes, because of their mild deformations, can be manufactured out of seamless titanium tubing, which is of definite interest in Naval applications. They are easily cleanable, and can be readily furnished with smooth lands spaced at appropriate intervals to accommodate the tube support plates.

Withers [14] described one such tube (KORODENSE), Fig. 5, and found that his data [15] could be correlated by equations similar to Eqs. (1) and (2) above. Gupta and Rao [16] compared the heat transfer and friction characteristics of a similar type of indented tube to smooth tube behavior. They found that the performance ratio $(h/h_s)/(f/f_s)$ varied with a "severity factor", $\phi = e^2/pD$, and the highest performance occurred for ϕ equal to 0.002 (i.e., a very mildly indented tube), Fig. 6. For this case, the heat transfer enhancement factor h/h_s was 1.75 for water.

A similar type of "roped" tubing is available from Yorkshire Imperial Metals, Ltd. in Great Britain. Their product brochure [17] recommends the widely used correlation

$$St = C_1 \, Re^{-0.2} \, Pr^{-2/3} \tag{3}$$

where C_1 varies with the tube geometry (C_1 is 0.027 for smooth tubes). Values of C_1 for these enhanced tubes are plotted in Fig. 7, and it is easily seen that enhancements of 3-4 are possible as groove depth increases and pitch decreases (i.e., a tighter spiral).

The above mentioned tubes all have a low number of starts-between 1 to 3. A multi-fluted tube, which contains many more flutes than the other tubes described above (approximately 15-25 starts), has been proposed by Yampolsky [18], Fig. 8. This tube has been tested by Marner at HTRI [19] and by

Reilly and Ciftci at the Naval Postgraduate School [20, 21], and its internal heat transfer and friction characteristics look attractive. For a 5/8-inch OD tube, at heat fluxes corresponding to naval condenser conditions, this tube gave a heat transfer enhancement factor over 3.0 with a corresponding friction factor less than the equivalent smooth tube. As shown in Fig. 9, the friction factor data of HTRI [19] and of Reilly [20] are in close agreement. The rather surprising decrease in friction factor compared to the smooth tube is explained by Yampolsky [18] to be due to secondary flows, within the internal grooves, which are stimulated by high heat fluxes radially inward. Work on this concept should definitely continue, including an investigation of the fouling characteristics of this type of surface.

A variety of the above-mentioned tube types have been tested and reported on [21, 22], and it appears that for 5/8-inch diameter tubes, internal enhancement factors near 2 to 3 are feasible. For a given type of groove geometry and depth, there appears to be an optimum groove pitch which gives the best thermal performance. It is reasonable to expect that these spirally fluted surfaces enhance heat transfer by a mixture of effects due to turbulence and swirl. If the helix angle (with respect to the tube axis) is too large, then the flow will tend to spill over the flutes with little swirl. The optimum helix angle would, of course, depend on heat flux, number of starts, groove depth, and flow conditions.

2.1.3 Summary

At present, the most promising technique to enhance heat transfer on the tube side is with one of the commercially available helically corrugated tubes. With a mildly indented tube, internal heat transfer enhancement factors of from 1.5 to 3.0 may be expected with a larger increase in friction factor. No one "best" surface exists and final selection will depend upon fouling characteristics, costs, and structural considerations, as well as the intended application.

There remains a significant need to study the fouling characteristics of these internally enhanced tubes, which may be a controlling feature to the practical application of these advanced concepts. Care should also be taken to study cleaning techniques as well as tube noise and vibration. Additional heat transfer research should be carried out on the multiple start helically ridged tubing, such as that proposed by Yampolsky, in order to further understand the fluid flow and heat transfer mechanisms which occur in turbulent, swirling motion.

2.2 SHELL-SIDE ENHANCEMENT

Since the analysis performed by Nusselt in 1916 [23], many analytical and experimental investigations have been conducted to further understand the condensation process. From these works, it is well-known today that during condensation a large thermal resistance occurs due to conduction of heat across the condensate film, and anything that can be done to thin or disturb this film is generally beneficial to heat transfer. For horizontal condenser tubes, this thinning may occur by promoting dropwise conditions, by using finned or fluted surfaces, or by improving condensate drainage. Vertical fluted tubes, using surface tension effects to thin the film, may also be very effective. Condensation heat transfer enhancement by shell-side surface geometry modification has been recently reviewed by Cooper and Rose [24].

2.2.1 <u>Dropwise Condensation on Smooth Tubes</u>

A recent review of dropwise condensation was performed by Tanasawa [25]. In addition to work done on the basic mechanism of dropwise condensation, Tanasawa also discussed the methods of promoting dropwise conditions, and stated that finding how to promote dropwise conditions for long periods of time was one of the most important problems to be solved before practical application of this mechanism can be accomplished. He further concluded that of all the promoting techniques, the use of a thin coating of organic polymer (such as Teflon) was the most promising in regard to economic feasibility. With this technique, however, two major problems must be addressed: (1) organic coatings have poor thermal conductivities and must therefore be applied in the form of an ultra-thin film, in order to reduce their conduction resistance, and (2) techniques must be developed to apply these ultra-thin films so that they are strongly adherent to the condenser tube, and have a toughness to withstand industrial conditions during assembly and use.

Topper and Baer [26] in 1955 were the first investigators to use Teflon as a promoter of dropwise condensation.

Unfortunately, they did not obtain heat transfer data as part of their investigation. A U. S. Navy study in 1956 [27] used Teflon paint sprayed on the outside of a horizontal 5/8-inch OD copper-nickel condenser tube. Although the coating was probably guite thick, about a 10 percent increase in overall heat transfer coefficient was measured for steam condensing at 2 inches Hg absolute pressure. Depew and Reisbig [28] sprayed Teflon onto a horizontal 1/2-inch OD aluminum tube. They noted that a strong bond was formed, presumably due to a porous oxide coating on the aluminum, which allowed the Teflon to adhere to many pores on the surface. They estimated the coating thickness to be 6 μm (0.00025 inches), and measured an improvement in the overall heat transfer coefficient of about 30-50 percent when condensing steam at 1 atmosphere pressure. They attributed this overall increase to an increase in the condensing codfficient of about 100 percent.

Erb and Thelen [29] in 1965 applied ultra-thin polymer films using a vapor deposition technique. They noted that the polymers did not stick on a 90-10 copper-nickel substrate, presumably due to a loosely adhering oxide on the substrate, which had a tendency to flake off. However, if they covered the substrate first with a thin film of chromium, then the polymer adhered very well. They took heat transfer data using a 0.25 μm thick film of parylene N (a polymer of para-xylylene) on a <u>vertical</u> tube with steam condensing at 1 atmosphere pressure. Their overall heat transfer coefficient with a cooling water velocity of 9.75 ft/sec increased by 38 percent over their filmwise results. Edwards and Doolittle [30] in 1965 used a 25 μm thick film of Teflon to promote dropwise condensation on a vertical copper tube. Their results for steam condensing at 1 atmosphere showed a 16-30 percent increase in overall coefficient depending on heat flux; their calculated condensing coefficients ranged from 240 to 330 percent larger than their filmwise coefficients. They further noted that their dropwise heat transfer coefficients were about half as large as values obtained by other investigators who used chemical promoters rather than a permanent organic coating.

In 1966 Brown and Thomas [31] applied a 2.5 μm thick Teflon film to a horizontal 3/4-inch OD tube of Admiralty brass. They measured their noncondensible gas concentration and kept this less than 100 ppm throughout their tests. Their results for steam condensing at pressures between 0.35 and 8.0 inches Hg absolute, showed that at a given heat flux there was a fall in both dropwise and filmwise heat transfer coefficients with decreasing operating pressure. At 7.0 inches Hg absolute, and a flux of 19,000 BTU/hr ft^2, they measured an increase in their condensing heat transfer coefficient of approximately 180 percent. Other studies [32,33,34,35,36] have been carried out in recent years to confirm the promotion of dropwise condensation of steam using a Teflon film, but these investigations have not provided useful heat transfer data.

Two recent thesis research projects were carried out at the Naval Postgraduate School in an effort to determine the usefulness of applying a Teflon coating to promote dropwise condensation. In 1979, Manvel [37] utilized sputtered ultra-thin films of Teflon on horizontal 5/8-inch OD, 90-10 copper-nickel tubes. He obtained condensing heat transfer coefficients for steam at a pressure of 6 inches Hg absolute by using a Wilson Plot technique [38]. His results are plotted in Figure 10 and show that as the thickness of the sputtered film decreases from 0.2 to 0.08 μm the dropwise condensing heat transfer coefficient increases about 40 percent. As the film thickness is reduced further to 0.04 μm, it appears that the heat transfer coefficient may decrease, presumably due to the fact that for very thin films, the film may not give complete coverage of the surface, and the condensing surface may therefore be partially wet in regions, leading to incomplete dropwise conditions. This effect is in agreement with a recent paper by Woodruff and Westwater [39] who showed that for less than 200 layers of gold electroplated on copper, pure dropwise conditions did not occur, and deposits of 1000 layers of gold (approximately 0.2 μm) were required before "perfect" dropwise conditions occurred. In examining Figure 10, it is clear that the results must be treated as being preliminary due to the large uncertanties associated with the measurements inherent in using the Wilson Plot technique to calculate the separate coefficients based on the measurement of overall values.

A second project was therefore carried out by Perkins [40] in 1979 using vertical copper-nickel discs 1-1/4 inches in diameter. He obtained heat transfer data for steam condensing on sputtered films of Teflon at the same pressure used by Manvel. Some of Perkins' data is shown in Fig. 11 which compares his results for filmwise conditions to those with 0.13 μm and 0.08 μm thick films of Teflon. Data for dropwise condensation using n-octadecyl mercaptan in octanoic acid as the promoter is also provided for comparison. His results show that the thinner film of Teflon increases the heat transfer coefficient, but neither Teflon film is as good as the chemically promoted surface. For example, at a heat flux of 50 kW/m^2, the condensing coefficient for the 0.13 μm thick film was approximately 100% higher than the filmwise value, and the 0.08 μm thick film produced a coefficient which was 140% above the filmwise value. Perkins calculated the thermal resistance of each of the Teflon films and was not able to account for the difference between his chemically promoted surface and his Teflon coated surfaces. He attributed the discrepancy either to an error in the Teflon film thickness (a value close to 3 μm would be needed to account for the difference), or to the possibility that the Teflon coating itself outgasses, producing a zone of noncondensible gas in the immediate vicinity of the test surface, which adds an additional thermal resistance to the condensation process.

In recent years, significant technological advances have been made in the coatings industry, including the development of techniques to produce strongly adhering ultra-thin films of organic materials for wear and lubrication [41]. The use of an RF plasma to deposit Teflon films has been described by Warner, Park and Mayhan [42]. They, as well as others [43], noted that RF plasma deposited Teflon is different than conventional Teflon due to extensive cross linking which occurs in the process, making a more adherent film with fewer voids. A recent patent disclosure by NASA [44] describes an ion beam sputter deposition process for fluoropolymers which can be applied to fabrics, glass, metals, and metal oxides at high deposition rates (and therefore reduced cost). Their technique achieves more complete coverage of the substrate having surface irregularities and interstices, and has been developed to give preferred film thicknesses between 0.2 and 5 μm. The Naval Research Laboratory has developed fluoronated epoxy coatings which are hydrophobic and exhibit better resistance to abrasion than Teflon [45]. Commercially available coatings such as NEDOX [46] exist in which a porous chrome-nickel layer can be electro-deposited onto a copper substrate, and polytetrafluorethylene is then infused into the micro pores to form an abrasion resistant, hydrophobic coating. With several of these new types of coatings, it may be possible to apply an ultra-thin continuous film which has strongly adhering qualities. In so doing, long lasting heat transfer improvements of several hundred percent may become a reality.

The above results show that using an organic coating to promote dropwise conditions may give enhancements of the steam condensing heat transfer coefficient for <u>single horizontal tubes</u> in the range of from 40 to perhaps 200 percent. This range of increase is substantially less thatn the 1000 to 2000 percent increase obtained for monolayer promoters on vertical surfaces [25, 47], and suggests perhaps that the potential steam-side enhancement for organic-coated tubes has yet to be achieved. Therefore, one would expect that it may be possible to improve significantly on the modest enhancements so far reported for organic-coated tubes.

2.2.2 Film Condensation on Finned Tubes

Since the early work of Beatty and Katz [48] in 1948, the use of externally finned tubes in surface condensers has received much attention, although most of the efforts have been devoted to condensing refrigerants. As originally described by Gregorig [49] in 1954, the fins generate surface tension forces which tend to thin the condensate film on the convex tips of the fins and to thicken the film in the concave channels, or troughs, between fins. In so doing, the condensing heat transfer coefficient is increased over the smooth tube case.

Staub [50] reported on some experimental work in 1961 which included film condensation of steam on horizontal finned tubes at both atmospheric and sub-atmospheric pressures. All data were obtained with 5/8-inch OD, copper tubes. His external heat transfer coefficients for a fine pitch tube (26 fins/inch) at 1 1/2-inches Hg absolute were about 2.5 to 3 times his smooth tube values at a heat flux comparable to surface condenser designs (approximately 25,000 BTU/hr ft^2). Results with a coarse pitch tube (17 fins/inch) for the same conditions were only 2 times his smooth tube results, indicating that fin pitch is an important variable. Nabavian and Bromley [51] used a finned condenser tube (8 fins/inch) to take measurements of the condensation coefficient of water in 1963. They machined the fins into a 1/4-inch nominal OD, schedule 80, copper pipe. The profile of the fins was chosen "to yield a constant and very high heat transfer coefficient along the top part of the fin" [51]. Karkhu and Borovkov [52]obtained condensation data for steam on four horizontal tubes containing different configuration of trapezoidally shaped fins. All their data was taken at pressures slightly higher than atmospheric, and they discovered that for transverse fins with large Weber numbers (i.e., large surface tension forces in relation to gravity forces), the average condensation heat transfer coefficient increased by 50 to 100 percent, whereas for low Weber numbers, there was little, or no improvement.

Carnavos [53] tested a wide variety of finned tubes using R-11 and showed gains in the heat transfer coefficient as much as 5 times the smooth tube result. Some recent work in Japan [54], for condensing R-113, showed that a finned surface which was covered with a porous metal coating gave more than a tenfold increase in the heat transfer conductance per unit length of tube. The development of the Hitachi THERMEXCEL-C condenser tube for refrigerants [55] has provided further proof that finned condenser tubes have promise in surface condensers. Figure 12 shows that an optimum groove spacing (i.e., pitch) exists for this tube based upon data for R-12. The best performance occurs for a tube with 35 fins/inch.

Of course, using steam, due to differences in surface tension, thermal conductivity, latent heat of vaporization and viscosity when compared to the refrigerants, a different pitch should exist for optimum performance.

In fact, in designing finned tubes for use with steam, the relatively large surface tension of water requires that special care must be exercised to avoid bridging of the gap between fins by the condensate. As shown schematically in Fig. 13, the fin geometry must be chosen with the particular working fluid in mind. In the case of steam, the fins must be spaced far enough apart to avoid this bridging phenomenon, otherwise, the thermal performance will be severely deteriorated. Some recent data for steam indicate that the optimum fin spacing may be near 10 fins/inch [56].

Several theoretical investigations [57, 58, 59, 60, 61, 62] show that in addition to groove spacing, variables such as wall material, fin shape and groove or trough dimensions are all significant during film condensation with these finned surfaces. A detailed theoretical analysis of film condensation on finned horizontal tubing has yet to be performed, however, without making significant simplications.

All of the above results are based upon the Gregorig premise that the surface contains numerous small fins and troughs to allow surface tension forces to be important. These forces act to thin the condensate film on the convex portions of the fins (where heat transfer is high) and to thicken the film in the troughs (where the heat transfer is low). In marked contrast to this model is the scheme proposed by Thomas [63] for a vertical tube which uses projections from the tube to draw the condensate away from the tube surface into the fillets, thereby thinning the film <u>between the fins</u>. This technique does not rely on conduction through the fins, and is even effective when loosely fitting wires are used. Figure 14, from the patent of Thomas [63], shows that an optimum number of fins/wires exists depending upon heat flux or condensate mass flux. Figure 15 shows the relative performance which occurs as the wires are moved away from the condenser surface. At high heat flux, displaced wires can do better than those placed on the surface, apparently allowing more space for condensate to collect and drain from the fillets at the base of the wires. Even though this work was done for vertical surfaces, this same principle may be applied to horizontal tubes. For example, some recent data for <u>ammonia</u> condensing on the outside of horizontal, wire-wrapped tubes (about 4 wraps per inch) gave external heat transfer coefficients about 3 times the Nusselt value for smooth tubes [64].

2.2.3 <u>Film Condensation on Corrugated Tubes</u>

In 1971, Withers and Young [65] compared film condensation performance of horizontal corrugated ('roped') tubes to smooth tubes. Tests were carried out for both 5/8-inch OD copper tubes and 1.0 inch OD 90-10 copper-nickel tubes for atmospheric as well as sub-atmospheric steam conditions. The corrugations were produced on the tube by spiralling a single mild indentation, 0.03 inches deep, along the tube with a pitch of 1/4-inch.

This configuration provided four corrugations per inch of tube length. Their external heat transfer coefficients were determined by a Wilson plot technique from overall measurements and showed considerable scatter. Their 1.0 inch OD corrugated tube showed a 35 percent increase in external heat transfer coefficient when compared to a smooth tube. Their 5/8-inch OD corrugated tube showed an 8 percent <u>decrease</u>. Palen, Cham and Taborek [66] in 1971 tested 1.0 inch OD corrugated tubes made of 97.5 percent copper. The tubes were corrugated by placing four deep indentations, 0.19 inches deep, around the tube circumference, and twisting the tube to a desired pitch of 0.56 inches. This created a tube with almost two corrugations per inch. The tests were made for a bundle of 196 tubes. Steam pressure was kept greater than atmospheric pressure to eliminate air leaks into the condenser. External heat transfer coefficients were calculated using the Wilson plot method, and results showed an average enhancement factor on the outside of 2.1 when compared to the smooth tube.

In 1975, Young, Withers and Lampert [67] reported on some additional measurements using mildly corrugated tubes with a groove depth near 0.03 inches. Three 1.0 inch OD 90-10 copper-nickel tubes with 2, 3, and 4 flutes per inch were tested at atmospheric and sub-atmospheric steam conditions.

The Table below summarizes their findings.

TUBE	FLUTES/INCH	PRESSURE	ENHANCEMENT FACTOR (h_a/h_s)
II	4	atm	1.67
		vac	1.33
III	3	atm	1.62
		vac	1.41
IV	2	atm	1.31
		vac	1.25

This data shows that the heat transfer improvement of these mildly corrugated tubes is better at atmospheric conditions than sub-atmospheric, or vacuum, conditions. The number of flutes per inch also influences the tube behavior, presumably due to different drainage characteristics.

In 1976, Catchpole and Drew [68] tested a family of corrugated tubes having different groove depths and spacings. All tests were on 5/8-inch OD 70-30 copper-nickel tubes at a steam pressure of 2 psia. They were able to correlate their external enhancement factor by the following relationship:

$$h_a/h_s = 1.17\ (We \cdot \cos\alpha)^{0.076} \qquad (4)$$

where

We = Weber Number, $\frac{\partial p/\partial x}{\rho g}$ = a function of condensate density and surface tension, as well as enhancement geometry and dimensions.

α = Helix angle of the grooves measured from the perpendicular to the tube axis.

They therefore concluded that it is desirable to design the corrugation so that a large Weber number exists, and a small helix angle exists. In fact, when $\alpha = 0^0$, i.e., when the grooves are vertical, the best performance would occur. The highest enhancement factor they measured was 1.7, corresponding to a tube with grooves 0.02 inches deep, spaced 0.08 inches apart (i.e., 12 grooves per inch). Their results are reproduced in Figure 16.

In 1978, Cunningham and Milne [69] studied the effect of helix angle on mildly indented ('roped') tubing. The tubes had a nominal OD of about 3/4 inches, an indentation depth near 0.008 inches, and were prepared with two different numbers of starts to give the same groove spacing but different helix angles of 18 and 44 degrees. Their data showed no condensation heat transfer enhancement when compared to their smooth tube. However, they pointed out that dropwise conditions were never completely eliminated from their smooth tube, implying that their measured heat transfer coefficients for the smooth tube were larger than expected. Consequently they were probably getting enhancement on their 'roped' tubes. In 1979, Mehta and Rao [70] obtained data for a family of mildly indented tubes having an OD near 3/4 inch, groove depths from 0.0005 to 0.056 inches, and spacings from 2 to 8 grooves/inch. Their results for steam at atmospheric pressure showed that the outside coefficient was dependent on a 'severity factor'. $\phi = e^2/pD_i$ where e is the groove depth, p is the groove pitch and D_i is the tube inside diameter. Marto, Reilly and Fenner [22] obtained data for 5/8 inch OD tubes having a variety of configuration shapes and sizes. Their results for steam at 3 psia for three types of corrugations are shown in Figure 17. Their data indicate that both pitch/diameter and depth/diameter are important variables. For each type of tube tested, the performance increased as pitch/diameter decreased, implying that grooves should be near vertical to promote good condensate drainage. Other recent studies [21,71] show that microgrooves on the surface of a corrugated tube can improve upon the enhancement that occurs for steam at low pressure, giving an enhancement factor near 2.0.

2.2.4 Condensate Film Drainage and Removal

During film condensation on a smooth horizontal tube, the Nusselt theory predicts that the condensate film is thinnest at the top of the tube, and thickens around the tube until at the bottom it becomes infinitely thick as the film drains off the tube in a continuous sheet. With this model, the thinner film on the top of the tube causes better heat transfer to occur. For example, Jakob [72] states that the Nusselt theory predicts that the top half of the tube will transmit 60 percent of the heat compared to only 40 percent through the bottom half. Therefore, it is reasonable to expect that if the thick film is interrupted on the lower part of the tube, creating the opportunity for a new thin film to be generated, heat transfer will increase. Mayhew [73] analyzed this possibility and found that if the condensation film is interrupted at $\phi = 90^0$ from the top of the tube, and if a new film is assumed to grow from this point as seen in Figure 18, the average heat transfer coefficient increases by 19 percent. This augmentation technique was experimentally studied in 1973 by Glicksman, Mikic and Snow [74]. They interrupted the condensate film on copper tubing by using Teflon tape 0.125 inches wide and 0.006 inches thick. They found however, and perhaps somewhat surprisingly, that the best position for the tape was along the bottom of the tube, and this location gave an average heat transfer coefficient which was 1.6 times the smooth tube value with no tape. Desmond and Karlekar [75] tested a 1.25 inch OD stainless steel tube which had a 0.36 inch wide and 0.001 inch thick film of Emralon, a non-wetting fluoroplastic, attached at the bottom. They pointed out that this location of the tape gave the greatest incremental increase in heat transfer, which amounted to a 20 percent increase in the overall heat transfer coefficient.

As pointed out above, the Nusselt analysis assumes that the condensate film drains from a horizontal tube in a continuous sheet. In reality, this does not occur. The condensate collects at the bottom of the tube and forms drops which depart from the tube at discrete points when they grow large enough for the force of gravity to overcome surface tension. In a recent paper by Yung, Lorenz, and Ganic [76], they postulate that the thin film on the underside of a horizontal tube is similar to the classical Taylor instability which occurs when a heavier fluid is on top of a lighter one. They therefore predict that the spacing between departing drops is given by the Taylor wavelength λ:

$$\lambda = 2\pi \sqrt{\frac{2\sigma}{\rho_\ell g}} \tag{5}$$

where σ is the surface tension of the condensate. The above equation agrees with experimental data for water to within 15 percent, and predicts a droplet spacing of about 1.0 inches for water at 100^0F. When this surface tension effect is included in the analysis of film condensation from a horizontal tube, agreement with experimental data is improved [77]. Shklover and Buevich [78], using high speed filming, measured the time it takes for a new droplet to begin forming to be about 0.04 sec. They also measured the time it takes for this newly formed drop to depart from the tube. This time was 0.37 sec. Their data points out that to keep the condensate film thin, it is extremely important to remove the large droplets as often as possible. Also, it may be of considerable merit to generate drop departure sites using spines or flutes. These sites should be spaced closer together than the Taylor wavelength in order to provide more locations to which the film on the tube can drain. The minimum spacing would be approximately the same as the departure diameter of a droplet, which according to Yung, Lorenz and Ganic [76] would be approximately 0.3 inches for water at 100°F. This would say that 3 departure sites per inch of tube would be desirable (i.e., 2 more than occur naturally). Condensate retention of horizontal integral-fin tubing was recently reported on by Rudy and Webb [79]. They performed a series of experiments to measure static liquid retention in integral-fin tubing having variable fin densities, and showed that surface tension forces play a significant effect in film drainage.

2.2.5 Enhancement on Vertical Tubes

It is well-known in the heat transfer literature that the film condensation heat transfer coefficient for a vertical, smooth tube is less than the equivalent horizontal tube. The Nusselt theory gives the following ratio:

$$\frac{h_V}{h_H} = 1.295 \left(\frac{D}{L}\right)^{1/4} \tag{6}$$

where D is the outside diameter of the tube, and L is the tube length. For a tube with a diameter of 1.0 inch and a length of 15 ft, Eq. (6) predicts that a vertical tube would be only about 1/3 as effective as a horizontal tube. As mentioned earlier, the use of round or rectangular wires spaced around the circumference of a vertical tube has been shown [63] to effectively thin the condensate film, and enhance condensation heat transfer significantly. For steam, as shown in Fig. 14, these wires can increase the condensing heat transfer coefficient by as much as 9 times that of a smooth, vertical tube.

A wide variety of vertical fluted surfaces have been tested at Oak Ridge National Laboratory using refrigerants [80,81,82,83] Fig. 19 shows the enhancement ratio of one such tube (tube F) compared to a smooth tube (tube A) for R-113 refrigerant. This figure shows that rubber drain-off skirts placed at various axial intervals can significantly improve upon fluted tube performance at high heat fluxes. These skirts strip the condensate away from the flutes, thin the film and prevent flooding. The uppermost curve in Fig. 19 is for 7 skirts placed 0.5 ft. apart, whereas the lowest curve is for tube F with no skirts.

Newson [84] has proposed a helically fluted tube which, in the vertical orientation, gives enhancements of 3-4 times the vertical smooth tube value. A recent patent by Notaro [85] has proposed a single layer of randomly distributed metal particles bonded to a vertical tube wall. These particles serve to create tortuous paths for the condensate rather than the straight, open, unimpeded drainage channels commonly found in vertical fluted tubes. Surprisingly, these paths do not impose a severe restriction to the flow of condensate, while thinning the condensate film around each particle. Fig. 20 is a sketch of the condensate around these particles, and shows that for steam condensing on a 20 ft. long tube, the presence of the particles can increase the heat transfer coefficient by as much as 18 times the smooth tube value.

Although a savings in surface condenser area is possible with vertical fluted tubes [86], much more work remains to be performed before vertical condensers will be used aboard naval vessels.

2.2.6 Summary

The use of permanent plastic coatings in the form of ultra-thin films to promote dropwise condensation may give shell-side enhancements on a single tube of 2 to 3. This is far less then the values of 10-20 quoted for a monolayer of chemical promoter on a vertical surface, and further research is needed to investigate which methods of applying strongly-adhering, ultra-thin films lead to the best thermal performance.

Low-finned tubing for steam condensation use has not been pursued due to difficulties with condensate bridging across the fins. However, there is some experimental data showing external enhancement factors of 2-3, and work is therefore needed to select an optimum low-fin tube for steam use.

Mildly indented ("roped") tubes which are available commercially give modest enhancements of 1.2-1.5. The use of micro-grooves on these indented surfaces (similar to low fins) can increase the enhancements near 2.0. A wire wrap on these tubes may also provide additional enhancement.

Vertical fluted tubes, with properly designed drain-off skirts, and/or vertical tubes coated with metal particles may give significant enhancements over 3 times smooth horizontal performance. Further theoretical and experimental work with steam should be performed to study optimum flute shape, skirt spacings, metal coated surfaces, etc.

3. ENHANCEMENT IN TUBE BUNDLES

It is well-known that the thermal performance of a condensing tube may be dramatically affected by the surrounding two-phase multi-component flow. Condensate loading, local vapor velocity and local noncondensible gas content all have important effects upon the thermal resistance on the steam side of the tube. It is therefore, very important to study tube performance in a bundle environment.

Nunn and Marto [87] provide a detailed discussion of the main factors that are likely to be significant in bundle performance. Most of the literature cited in their discussion however, pertains to smooth tubes, and little information exists in the literature regarding the performance of enhanced tubes in large tube bundles.

With dropwise conditions, there are data available which show that condensate falling on a horizontal tube may not deteriorate the heat transfer coefficient (as is well-known during filmwise conditions [88,89]), and in certain circumstances, may actually increase the heat transfer coefficient [90,91,92]. This reversal in trend with the two modes of condensation is presumably due to the fact that condensate falling on a tube condensing in the dropwise mode may sweep the tube of the large, ineffective drops, thereby creating the opportunity for new microscopic drops to form and improve upon the performance. Data obtained during filmwise condensation on a horizontal, wire-wrapped smooth tube show a similar trend. In this latter case, the condensate falling on the lower tubes does not deteriorate the thermal performance of these tubes because the helically-wrapped wires draw the condensate very rapidly toward the fillets formed between the wire and the tube wall [93]. Because of this mechanism (which doesn't exist for the smooth tube), the condensate is always thinned between the wires; and a large portion of the tube wall is therefore available for high heat transfer rates.

There is evidence that during film condensation on corrugated tubing the effect of condensate inundation is not as pronounced as for the smooth tube case [65,68]. For a 5/8 inch diameter tube, Withers and Young [65] show that the mean heat transfer coefficient for a vertical column of n tubes can be represented by:

$$k_m = C_n \, h_{Nu} \tag{7}$$

where h_{Nu} = the mean heat transfer coefficient according to the Nusselt theory,

= const. $n^{-1/4}$,

and

$$C_n = \begin{cases} 1.11 \, n^{0.20}, & \text{KORODENSE} \\ 1.20 \, n^{0.006}, & \text{smooth.} \end{cases}$$

From this result, it is clear that the effect of condensate inundation is less with the KORODENSE tube than with the smooth tube.

It should be pointed out however, that the inundation trends mentioned above were obtained with low velocity steam. The exact behavior with high velocity steam moving in various directions with respect to the tube bundle remains to be determined, and will require a comprehensive set of bundle tests for a variety of test conditions.

4. CONCLUSIONS

(a) With the present state-of-the-art, the most promising technique to enhance heat transfer is with one of the commercially available, helically corrugated tubes.

(b) With a mildly indented tube, internal enhancement factors of between 1.5 to 3.0 may be expected at the expense of a larger increase in friction factor.

(c) Mildly indented tubes do not show as much enhancement on the shell side. External enhancement factors of between 1.1 to 1.5 may be reasonably expected with steam.

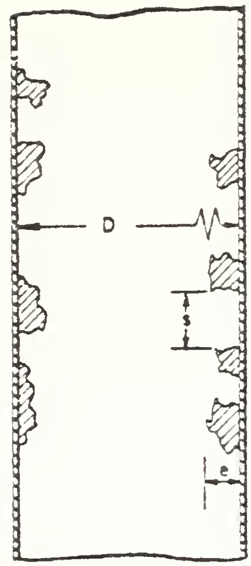
(d) In the future, with the use of multiple-start ridging on the tube side, internal enhancement factors of 3.0 or greater are possible. By using dropwise condensation or a wire wrap, external enhancement factors of 2.0 to 3.0 are possible.

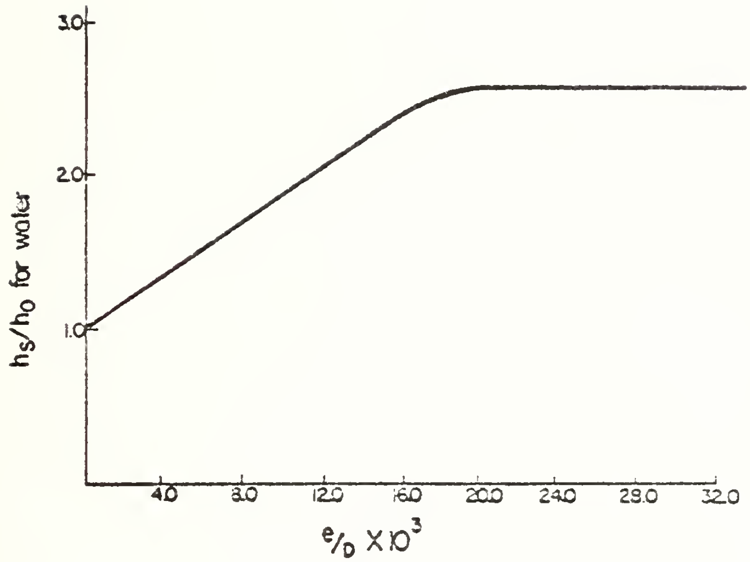
(e) The influence of vapor shear, condensate inundation and non-condensible gas concentration upon external enhancement factors must be determined by conducting a series of carefully planned experiments.

(f) There remains a significant need to study long term fouling characteristics of internally enhanced tubes. Care should also be taken to study cleaning and inspection techniques for use with enhanced tubes.

(g) A comparison of vertical, fluted tubes to horizontal fluted, or finned, tubes should be made by conducting large bundle tests with each orientation.

5. FIGURES





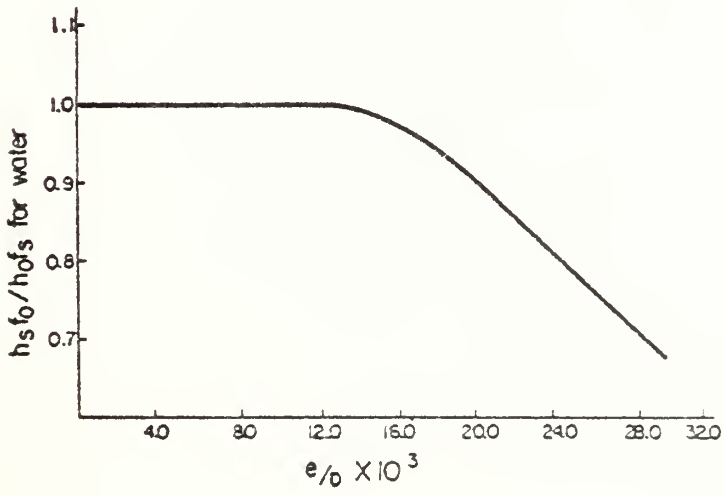
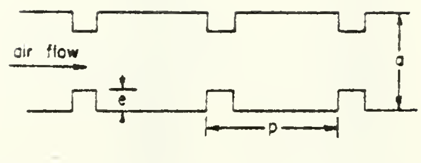
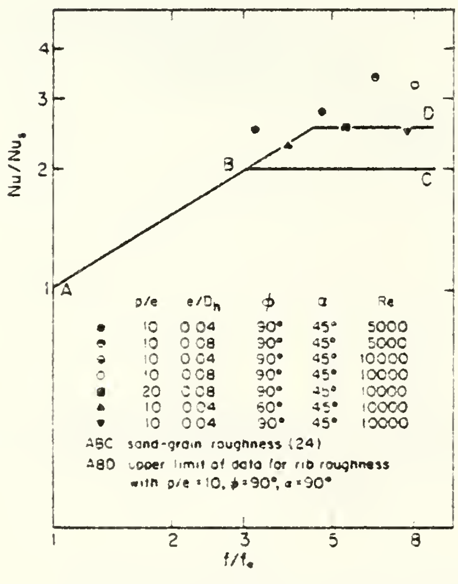


Figure 1. Performance characteristics of three-dimensional roughness [from Ref. 9]



Schematic of Rib Roughness



Performance of different types of rib roughness.

Figure 2. Enhancement with two dimensional rib roughness [from Ref. 11]

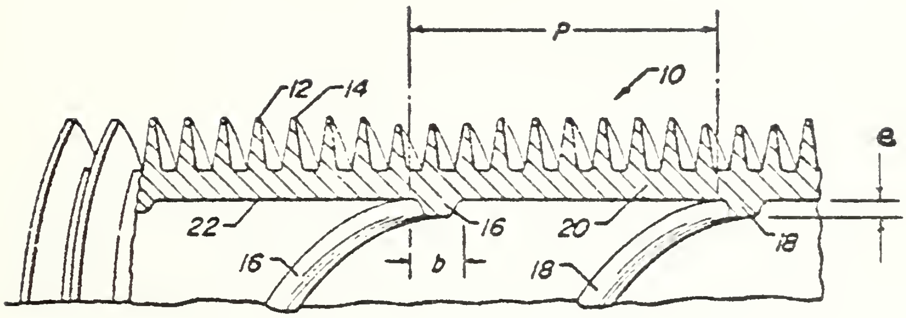
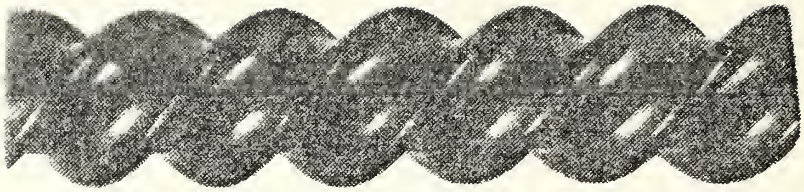
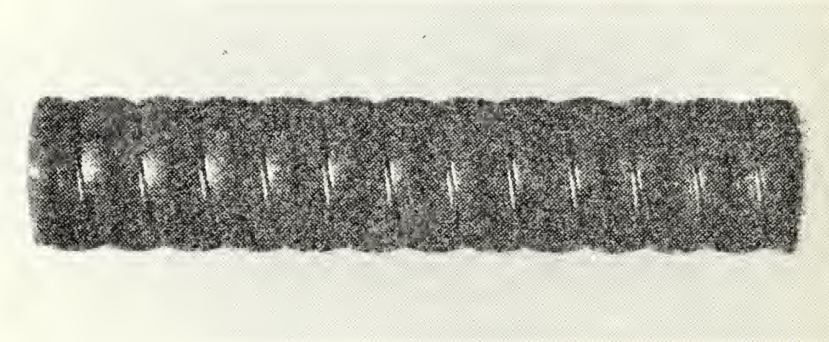


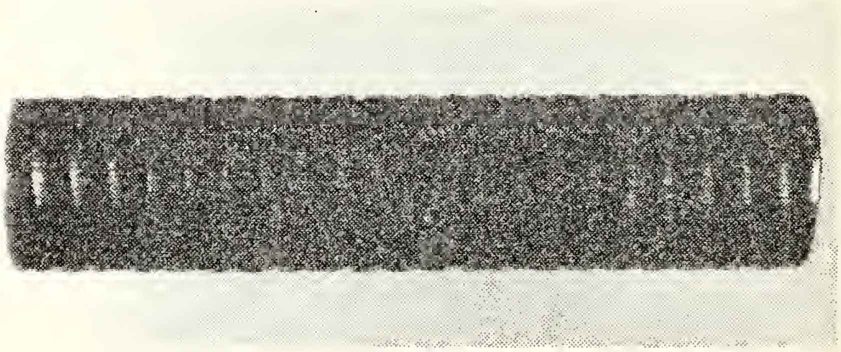
Figure 3. Cross sectional view of TURBO-CHIL tubing [from Ref. 12]



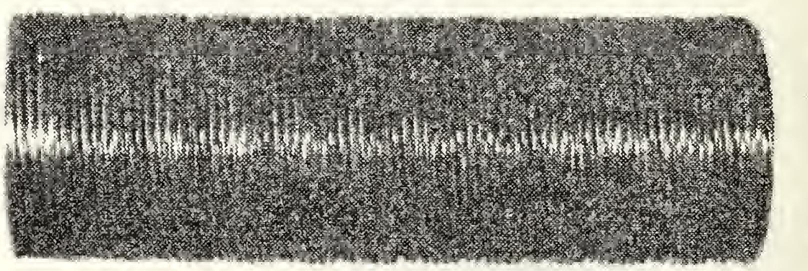
(a) Turbotec tube



(b) Korodense tube



(c) Yorkshire roped tube



(d) Yorkshire roped tube with enhanced profile on the outside

Figure 4. Photographs of corrugated tubes

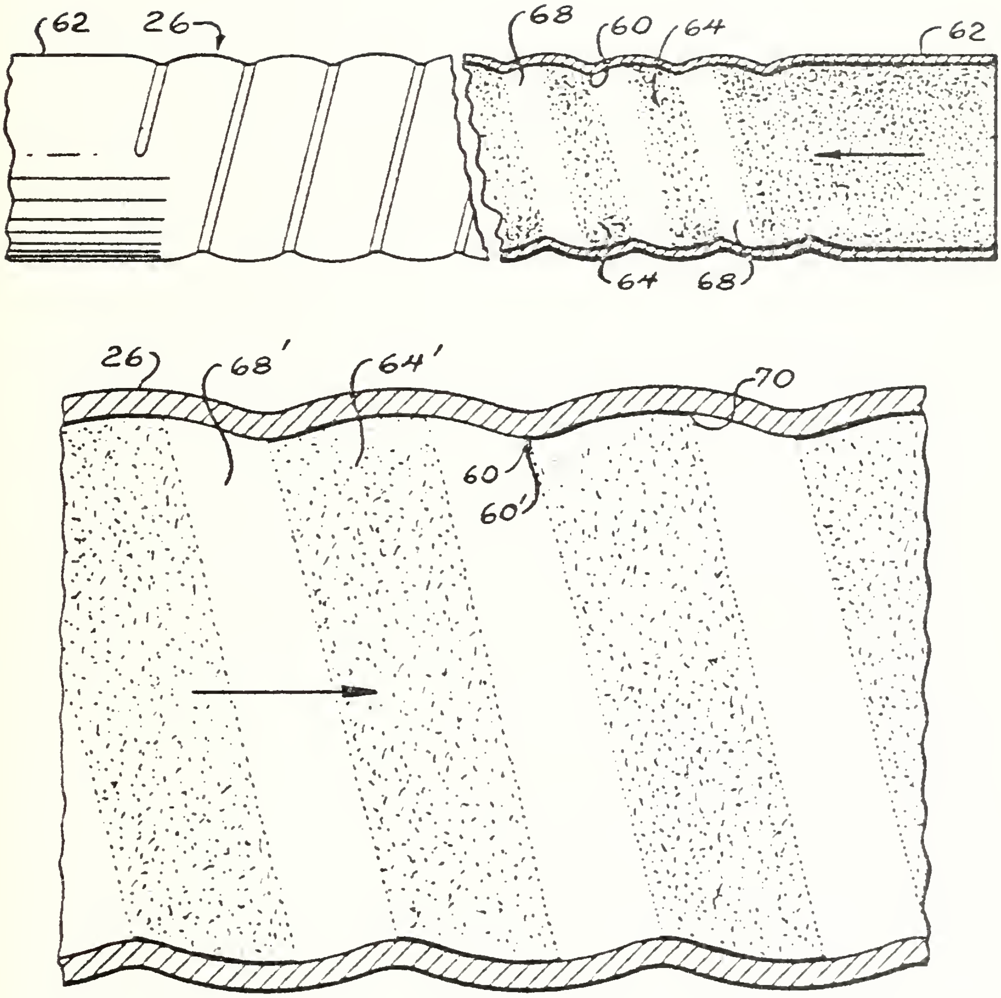


Figure 5. Cross sectional views of KORODENSE tubing [from Ref. 14]

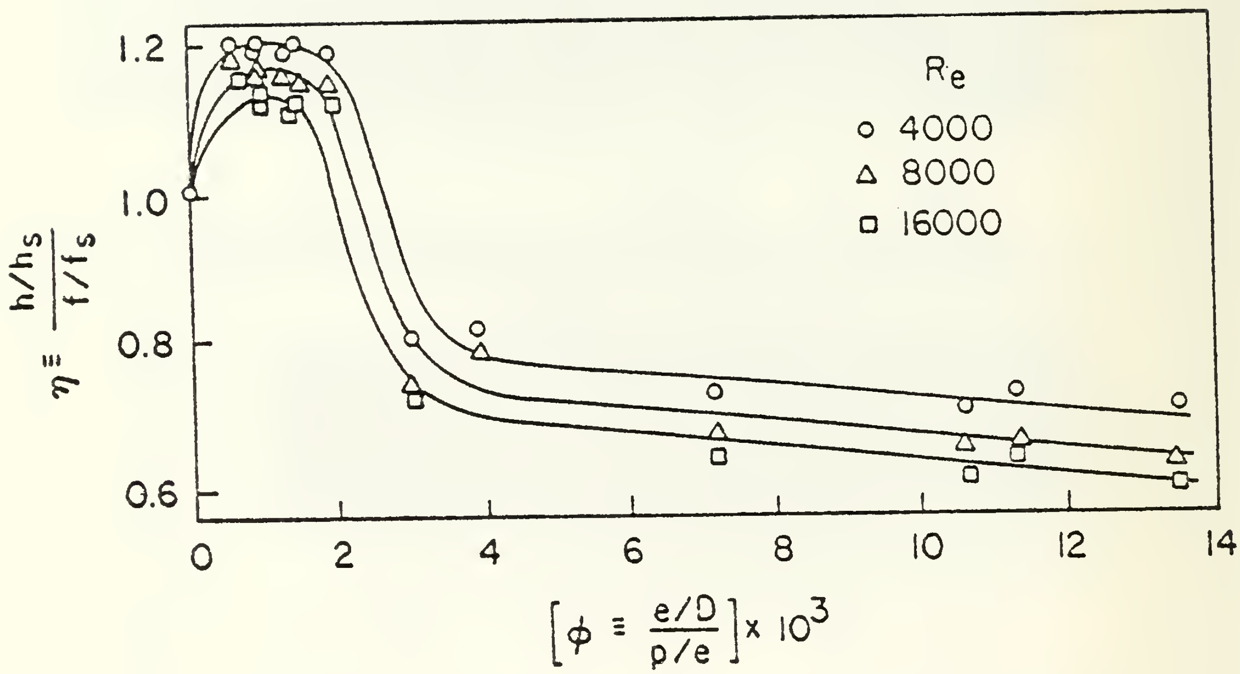


Figure 6. Variation of performance ratio with severity factor [from Ref. 16]

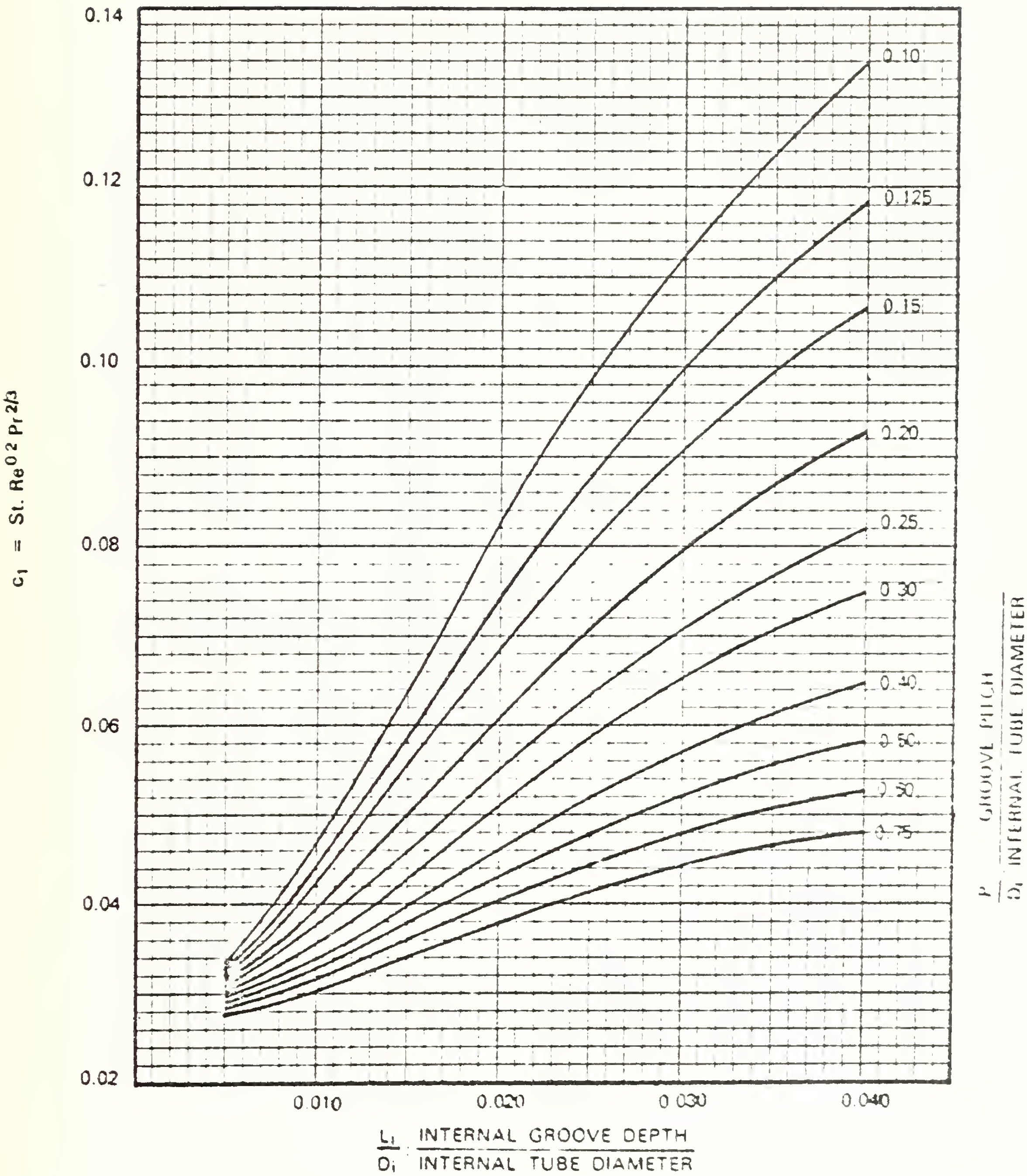


Figure 7. Estimated values of convective coefficient C_1 for a range of tube profiles [from Ref. 17]

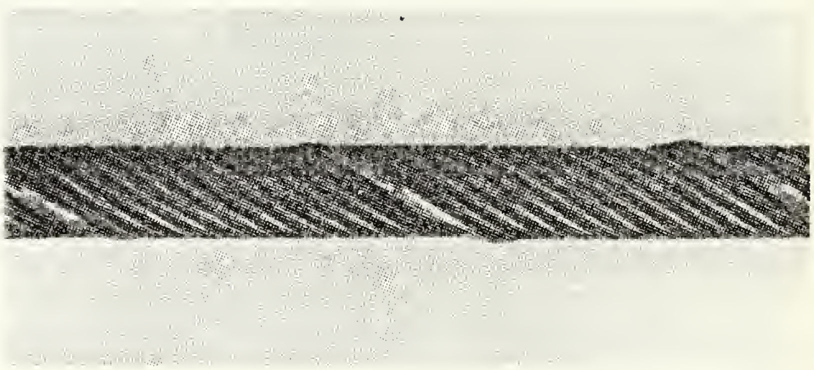


Figure 8. Photograph of General Atomic multiply-fluted tube.

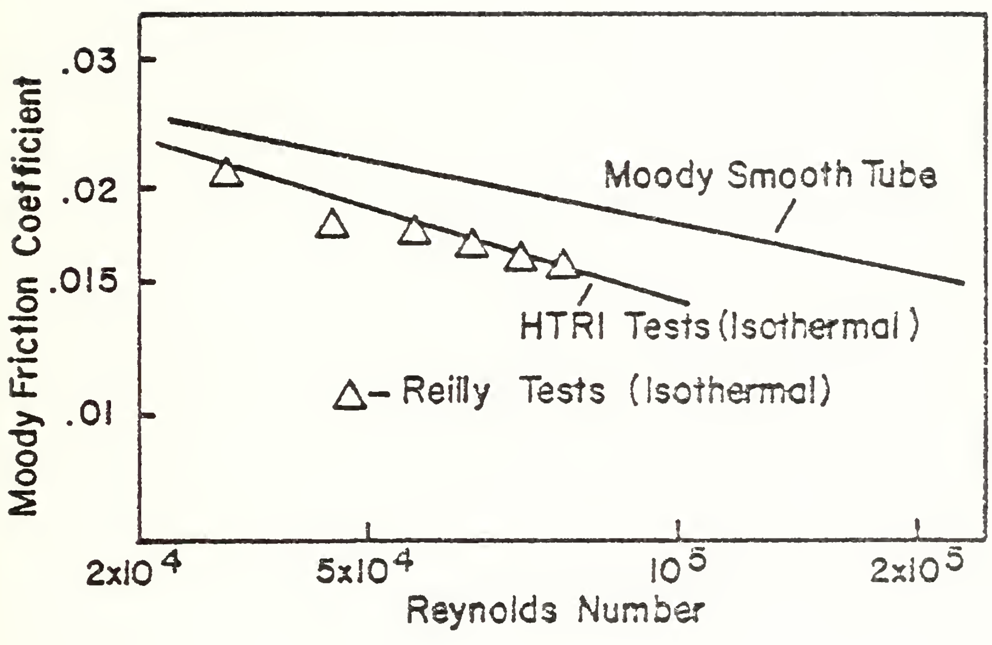


Figure 9. Moody friction coefficient versus Reynolds number for helically fluted tube [from Ref. 18].

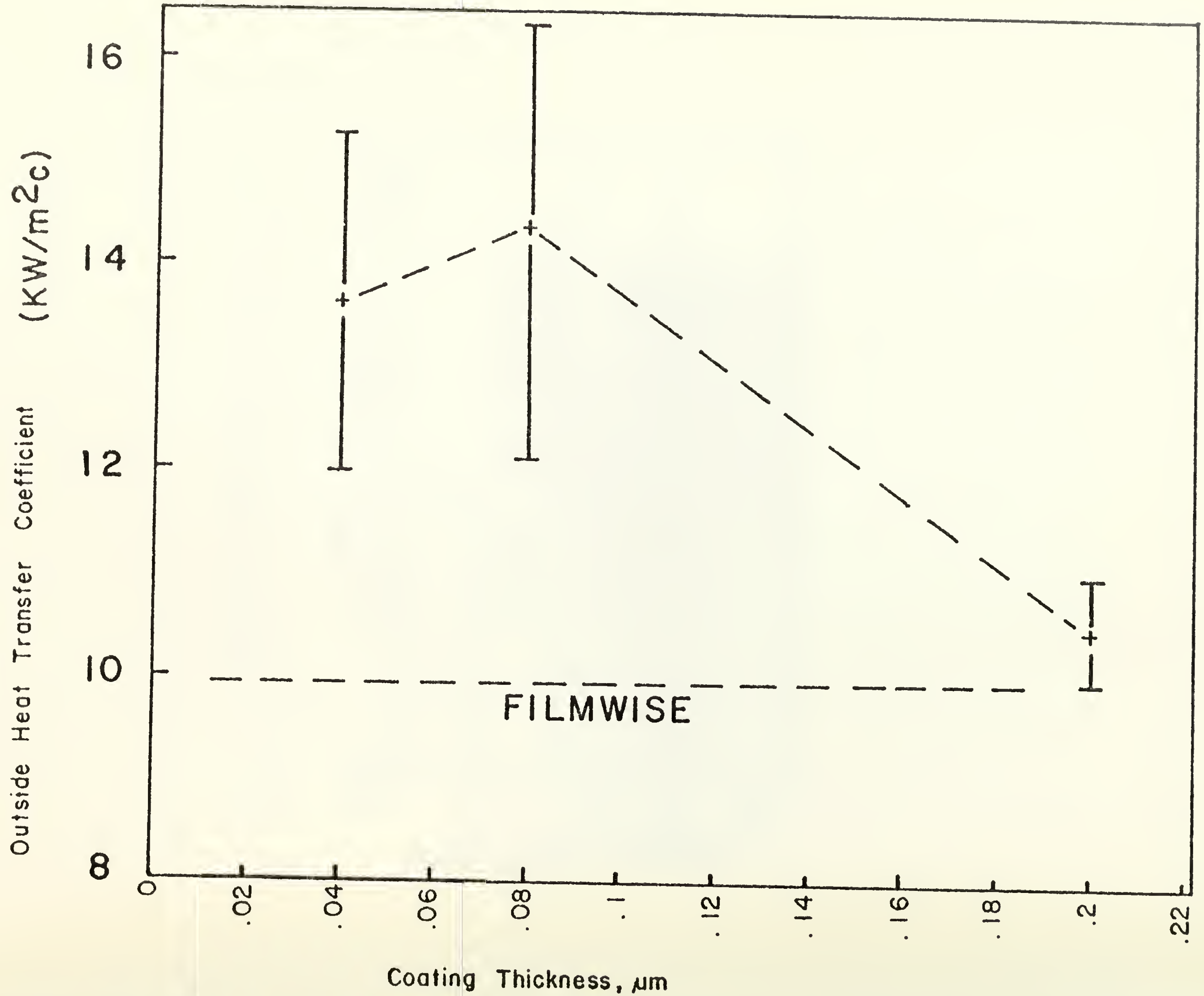


Figure 10. EFFECT OF TEFLON COATING THICKNESS ON DROPWISE HEAT TRANSFER COEFFICIENT ON A HORIZONTAL TUBE [from Ref. 37].

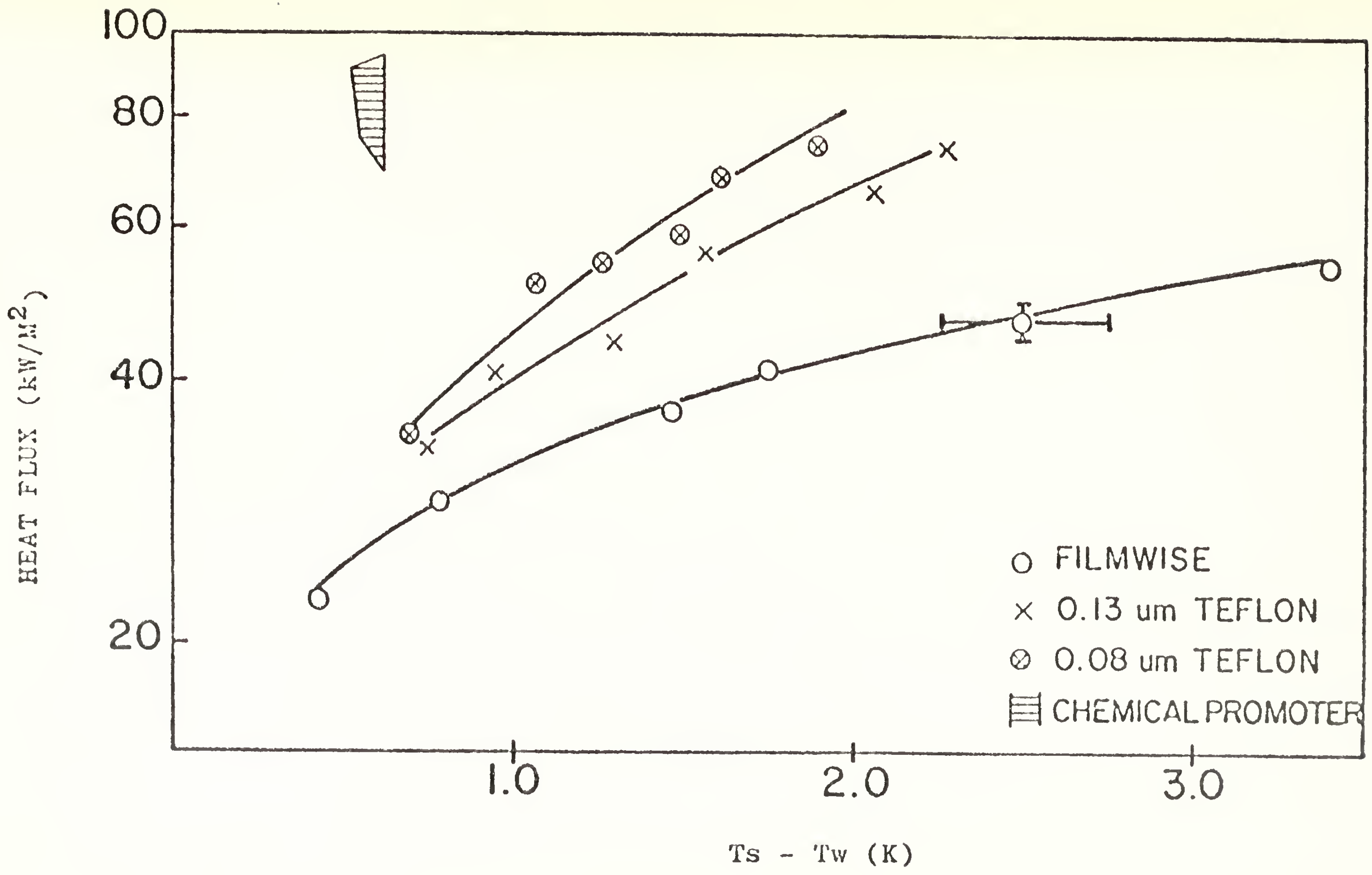


Figure 11. INFLUENCE OF PROMOTER CHARACTERISTICS OF CONDENSATE HEAT TRANSFER [from Ref. 40].

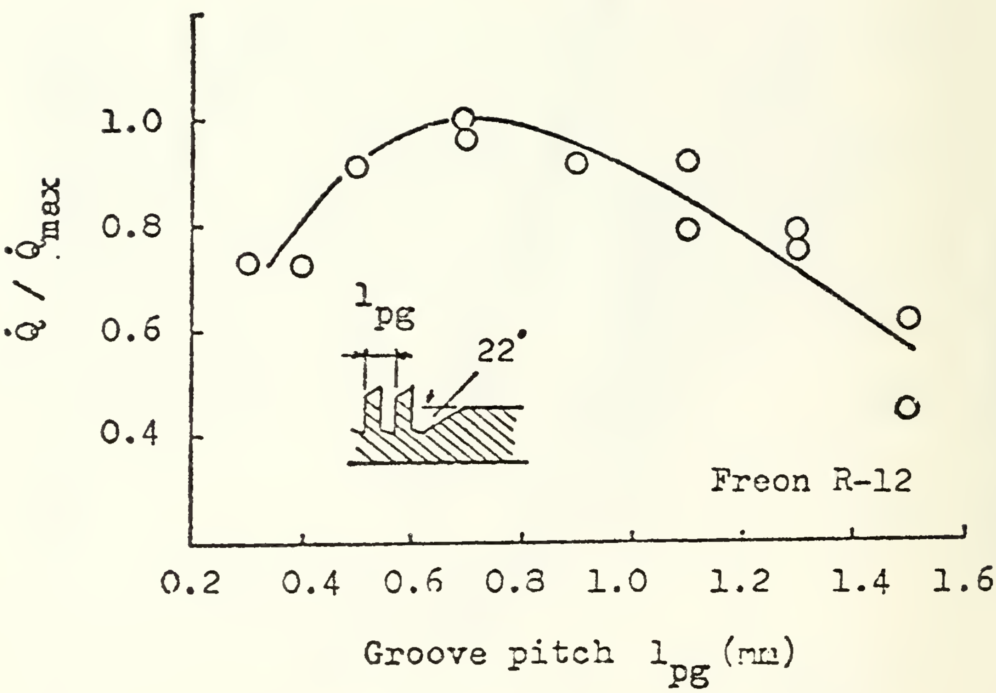


Figure 12. VARIATION OF HEAT FLOW RATE ON A UNIT LENGTH OF THE CONDENSER TUBE WITH THE GROOVE PITCH [FROM REF. 55].

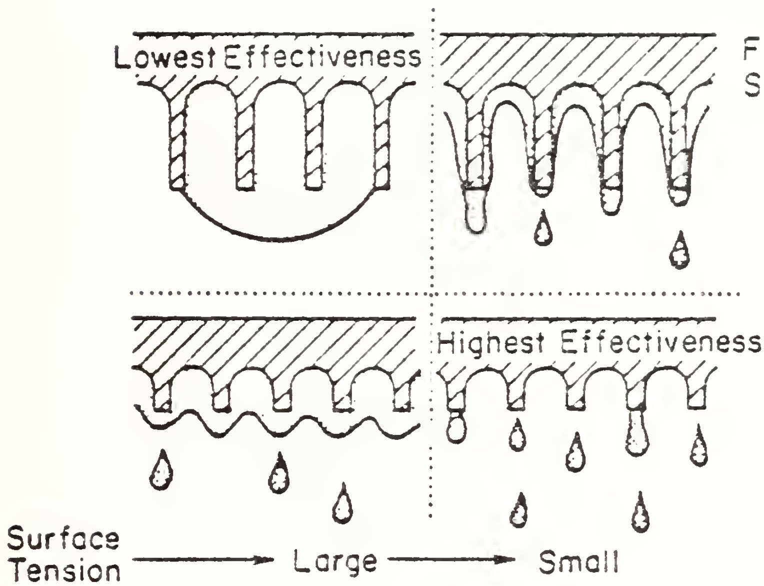


Figure 13. Condensate retention as a function of fin geometry and condensate surface tension .

Dec. 19, 1967 D. G. THOMAS 3,358,750

CONDENSER TUBE

Filed Aug. 10, 1966 4 Sheets-Sheet 1

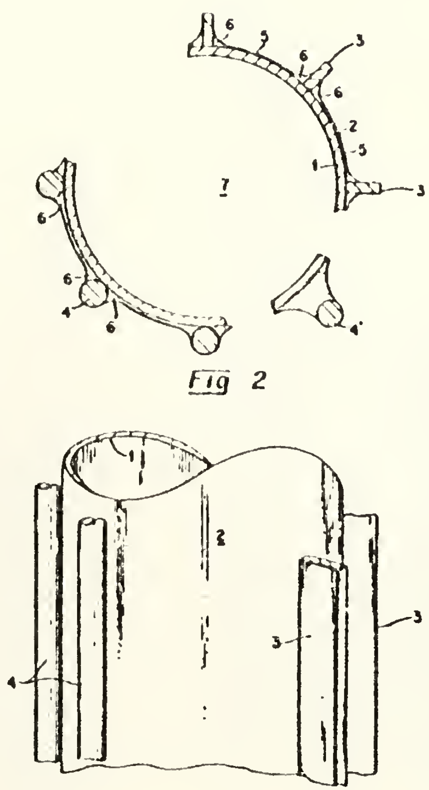


Fig 2

INVENTOR
David G. Thomas
BY
ATTORNEY.

Dec. 19, 1967 D. G. THOMAS 3,358,750

CONDENSER TUBE

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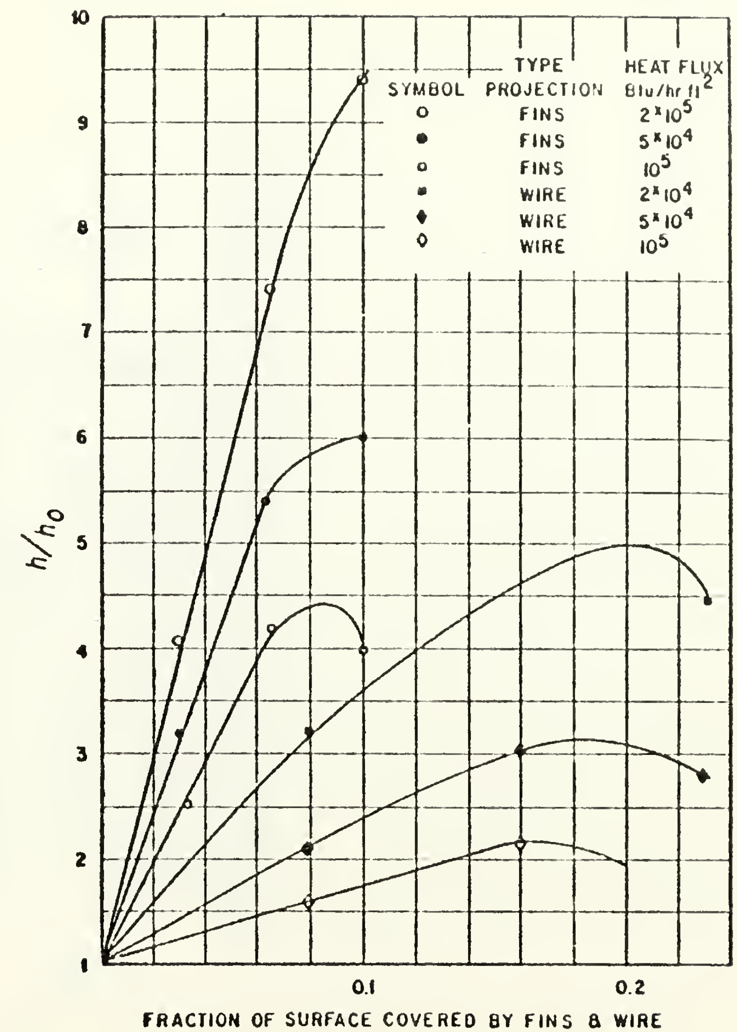

Figure 14. THE INFLUENCE OF FINS OR WIRES UPON HEAT TRANSFER ON A VERTICAL TUBE [FROM REF. 63].

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CONDENSER TUBE

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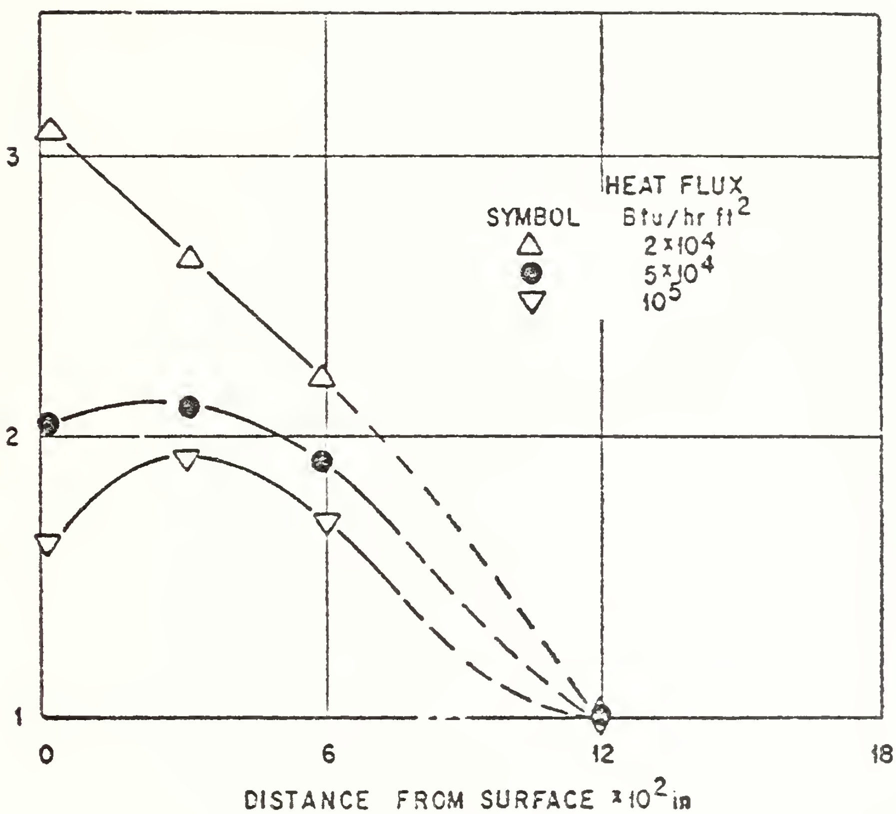

Figure 15. THE EFFECT OF WIRE DISTANCE AWAY FROM CONDENSING SURFACE UPON HEAT TRANSFER PERFORMANCE [from Ref. 63].

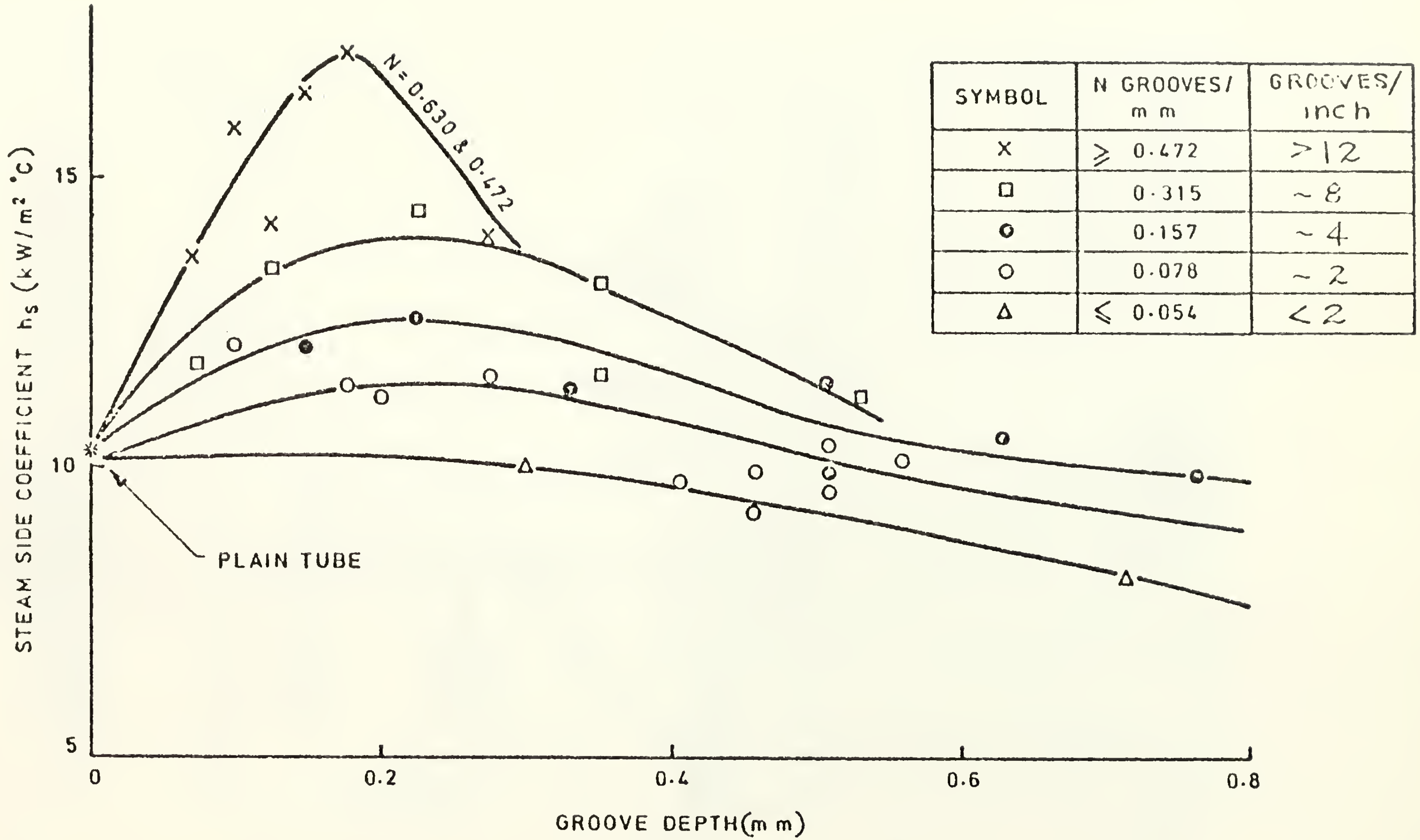

Figure 16. STEAM SIDE COEFFICIENT VERSUS GROOVE DEPTH [from Ref. 68].

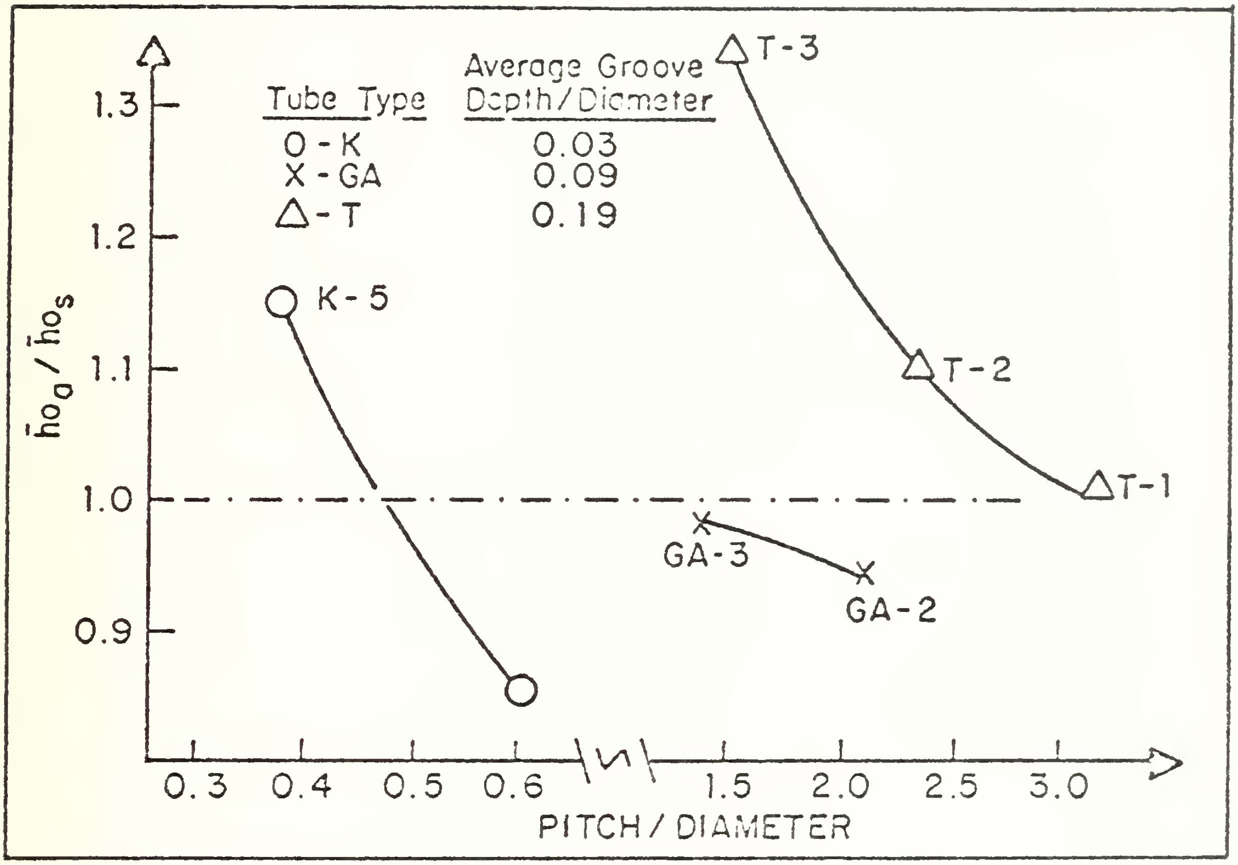

Figure 17. CONDENSING ENHANCEMENT FACTOR VERSUS TUBE GEOMETRY [from Ref. 22].

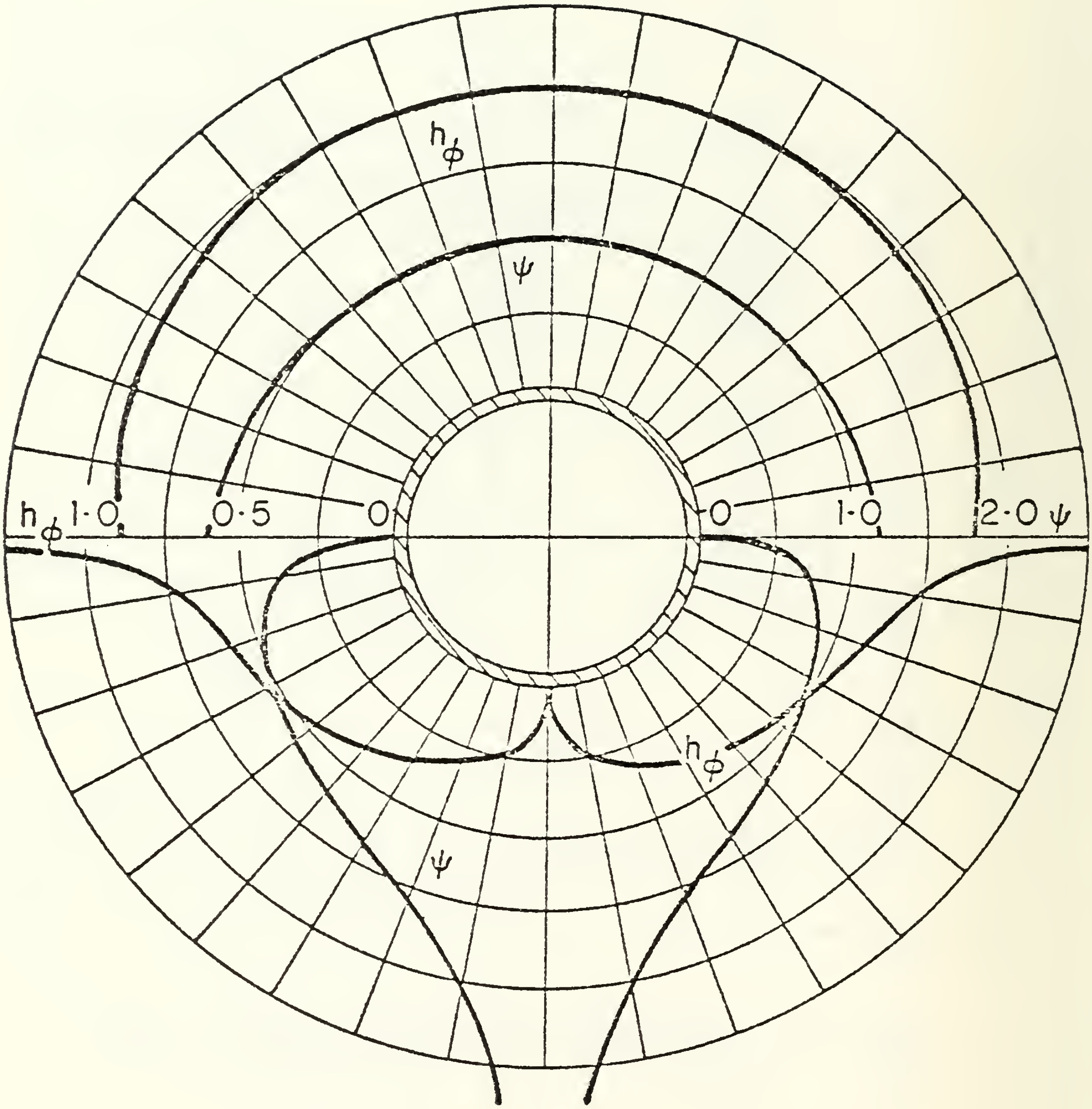

Figure 18. LOCAL FILM PROFILE AND HEAT TRANSFER COEFFICIENT FOR A SINGLE HORIZONTAL TUBE WITH CONDENSATE SPLITTERS AT 90° [from Ref. 73].

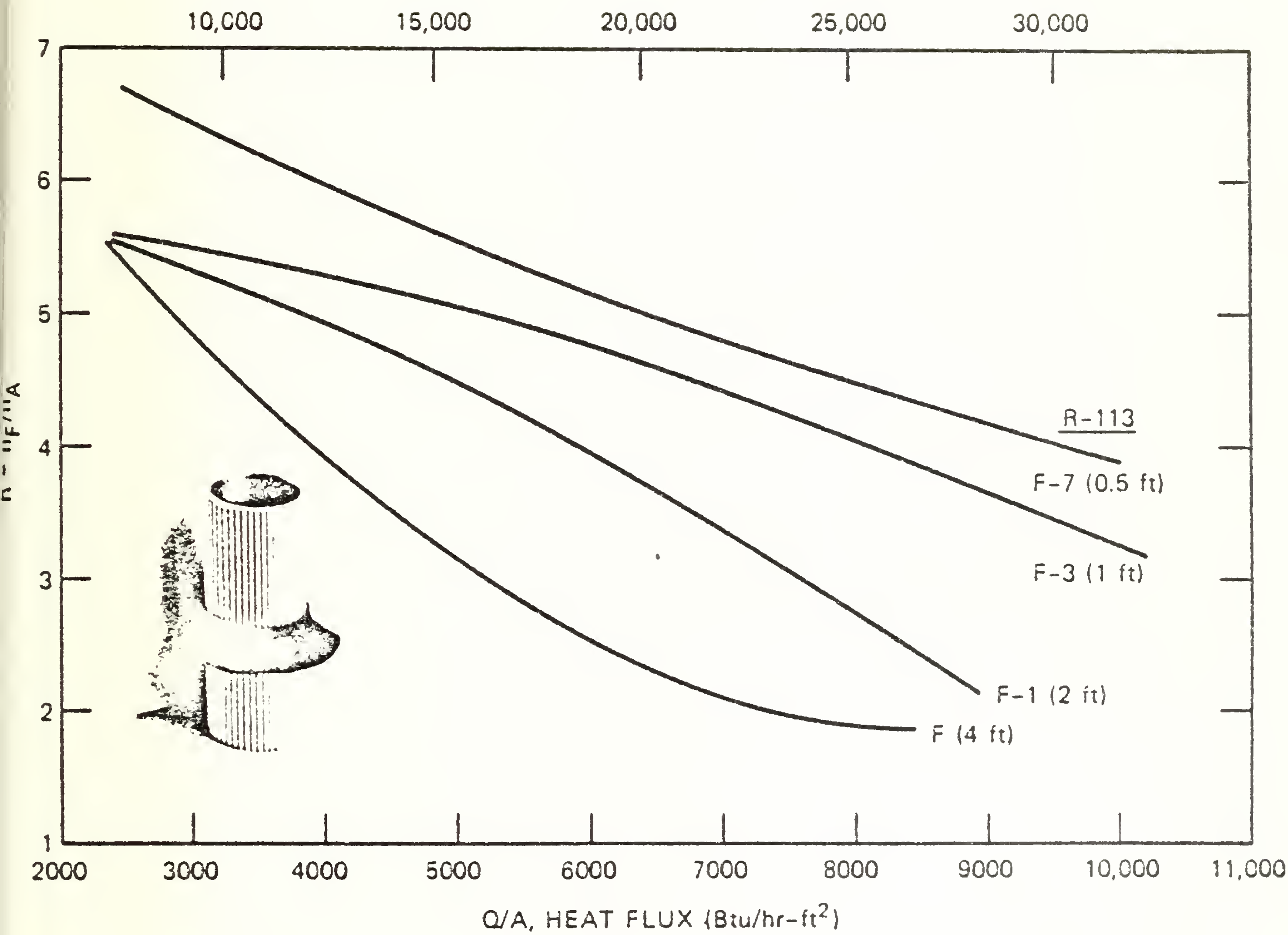

Figure 19. Enhancement ratio for a vertical fluted tube, and the effects of drain-off skirts [from Ref. 82].

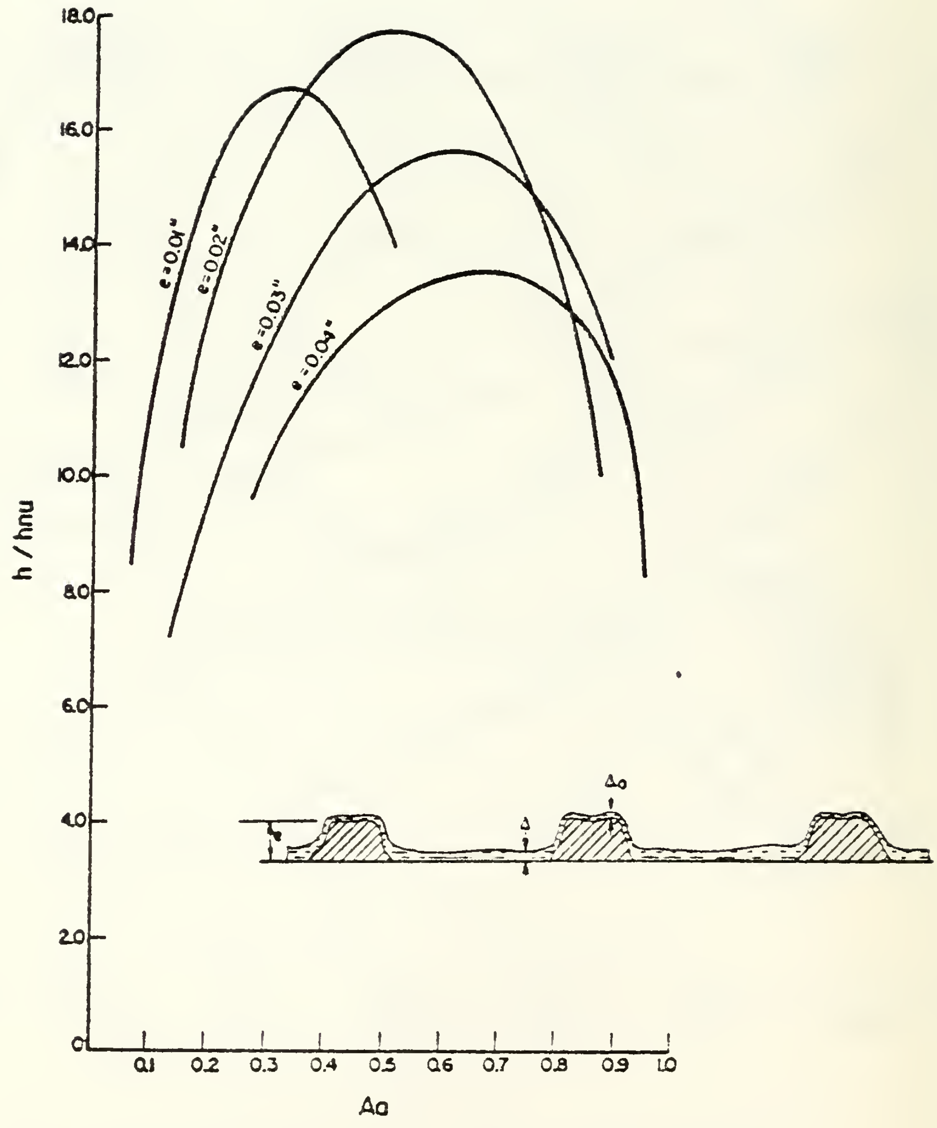

Figure 20. The influence of randomly distributed metal particles on film condensation on a vertical tube [from Ref. 85].

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