PROGRESS REPORT

CONCURRENT STUDIES OF ENHANCED
HEAT TRANSFER AND MATERIALS FOR
OCEAN THERMAL EXCHANGERS

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ABSTRACT

To demonstrate the enhancement of heat transfer in OTEC exchangers by means of fine, axial flutes on the heat transfer surfaces, experiments have been performed on single internally or externally fluted tubes of 1-inch diameter.

Internal flutes provide water-side enhancement of both heat transfer and friction to about the same extent. At moderate water velocities heat transfer coefficients and friction factors are greater than those of the comparable smooth tube by a multiplying factor about equal to the fluted-to-smooth surface area ratio. Area ratios from 1.0 to 1.5 have been tested using two styles of flutes. Flute heights have ranged from 3 to 35 mils, tube lengths from 2 to 8 ft.

Heat transfer experiments have been performed on Freon-11 moving in a thin layer under gravity down the outer surface of a smooth tube and of a tube having 26-mil external, axial flutes. Test lengths have averaged 4-1/2 ft. Enhancement of the local coefficient by 3 to 5 times has been observed during both evaporation and condensation. The effectiveness of the flutes is greatest in the range of laminar-turbulent transition.

Work is continuing in order to establish the separate effects of flute style and size on the water side and to extend the findings to ammonia on the power-fluid side of the heat transfer surface.

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1. Background

The heat exchangers of an OTEC plant constitute the major part of its capital expense. To produce power at a competitive cost, it is necessary to seek means of enhancing heat transfer while maintaining low parasitic power losses. The hallmark of OTEC exchangers is the low temperature potential ($\leq 10^\circ\text{F}$) available for transferring heat between the seawater and the power fluid (assumed to be ammonia). Under such a temperature difference, the two exchangers in the OTEC power cycle function as an evaporator and a condenser rather than as a boiler and a condenser. Only when the micro-structure of the heating surface is altered (as in the case of Linde surfaces) can true boiling of the power fluid be anticipated.

The works of Alexander and Hoffman [1] and many others have demonstrated that both evaporation and condensation can be enhanced by means of axial flutes on the heat transfer surface. Most of these studies have dealt with layer-flow of water down the surfaces of fluted tubes, under the action of gravity. In the case of vertical tubes, there is general agreement that heat transfer coefficients on the enhanced side, or sides, of the wall can be improved several times over by the flutes.

Alexander and Hoffman suggested that the efficacy of axial flutes is related to the presence of waves and ripples in the falling liquid layer. The crests of the waves provide means of intermittently lavng the crests of the flutes, while the troughs permit surface tension to draw the liquid from the crests of the flutes into the intervening rills as proposed by Gregorig [2]. Support for this view is found in the relationship between the heat transfer coefficient and the Reynolds number of the liquid layer. In both evaporation and condensation the water coefficients obtained by Alexander

and Hoffman reached a local maximum in the range of laminar-turbulent transition [3] where frequent, large-amplitude waves containing disturbance-eddies are separated by troughs which are in pseudo-laminar motion [4,5]. No hydraulic model exists to permit other than empirical conclusions.

When there is no vapor-liquid interface to be intercepted by the flutes, the physical situation is quite different. When a conduit with axially-fluted surfaces is running full of fluid, enhancement of heat transfer on that side must, as in the case of ordinary fins, be due mainly to the increase of surface area. If the flutes are fully effective the heat transfer coefficient defined on the smooth-tube ("rubber band") area in the usual way should be greater than the smooth-tube coefficient by the ratio of fluted surface area to smooth-tube surface area. That is the maximum enhancement to be expected. But there is no model through which to predict whether the maximum degree of enhancement can be attained and, if so, under what operating conditions. Empirical data are necessary.

2. Objectives

With the preceding background, a common design can be proposed for the evaporator and condenser. Both units are of the shell-and-tube type with vertical tubes having fine, axial flutes on their inner and outer surfaces. Seawater flows inside the tubes and fills the whole interior cross-section. Ammonia flows under gravity in layer form down the outside surface of the tubes. In the condenser, the ammonia flow is solely condensate; there is no load at the top of the tubes. In the evaporator, liquid ammonia from a supply plenum is applied through apertures to the top of the tubes. The unevaporated portion is recycled. The liquid flow rate of ammonia in the evaporator, unlike that in the condenser, is more a matter of choice than a matter of latent heat load.

On that basis, the objectives of the present investigation can be formulated:

1. To demonstrate whether water-side, axial flutes can attain full heat transfer effectiveness at practical water velocities.
2. To assess the penalty of increased pressure loss associated with water-side flutes.
3. To demonstrate the extent to which thin-layer heat transfer can be enhanced by axial flutes in non-aqueous power fluids and to relate this enhancement to the mode of fluid motion through the Reynolds number.

3. Experimental Approach

To attain objectives 1 and 2, experiments were made on single tubes through which water at essentially room temperature flowed steadily at predicted OTEC velocities. The water was heated electrically by means of tapes surrounding the outer wall.

To attain objective 3, experiments were made on a single-tube evaporator and a single-tube condenser connected by vapor lines. One unit had a smooth tube, the other a fluted tube; either could be used as the evaporator and the other as the condenser. Initial experiments were run on Freon-11; current experiments are being run on ammonia.

Details of the water and Freon experiments are treated separately in the following sections.

4. Water-Side Experiments

4.1. Experimental Tubes

Heat transfer and fluid friction in water flowing steadily inside single, horizontal tubes were studied experimentally. In all cases, the tubes had smooth outer surfaces, but the inner surfaces were smooth in certain cases and axially fluted in others. The smooth tubes, which were used as the basis of comparison for the fluted ones, were of nominal 1-inch, 16 BWG, seamless tubing. That is, the outer diameter was 1.000 inch and the inner diameter 0.870 inch, corresponding to a wall thickness of 0.065 inch (65 mils).

The specifications of all tubes used in the experiments are listed in Table 1. Two basic flute shapes were produced, depending on the method of fabrication. Flutes produced by a forming tool had the generic shape shown, simplified, in Figure 1a. Flutes produced by milling a flat strip, then rolling and welding to form a tube took almost the constant-radius shape shown in Figure 1b. It was observed that tools forced through the interior of seamless tubing did very little cutting; instead, the flutes were produced through forming of the metal. As a result, the original amount of metal remained in the wall. The cross-sectional area of the water stream in the fluted tube was therefore the same as that in the original smooth tube. This afforded a basis for defining the inside diameter of the fluted tube as the diameter of the smooth tube having the same internal cross-sectional area as the fluted tube.

The diameter thus defined is useful in comparing the behavior of smooth and fluted tubes. On that basis, flutes having crests and rills of the same constant radius (Figure 1b) lead to an internal tube diameter passing through the mid-height of the flutes. The same is also approximately true when the flutes are shaped as shown in Figure 1a. It is therefore

Table 1

Specifications of Tubes Used in Water-Side Experiments

Tube No.	Material	Tube Diam. (in.)		Average Height (Y) of Flutes on Inner Wall (in.)	Method of Flute Production	Remarks	Test Length (Inside Dia.)	Fluted Surface Area $\pm$ Smooth-Tube Surface Area	Crest Width (X) of Flutes (in.)	No. of Flutes on Periphery of Tube
		Outer	Inner*							
a 1 b c	Cu	1.000	0.870	Smooth	-	Commercial Tubing	110 55 28	1.00	-	-
2	Cu	1.000	0.870	Axial Scratches	Coarse Sanding	Hydraulically Smooth	28	1.00	-	-
3	Cu	1.000	0.870	0.003	Brooching	-	55	1.23	0.012	180
4	Cu	1.000	0.875	0.014 (0.008 to 0.020)	Forming	Flutes Irregular in Height	39	1.30	0.037***	52
5	Cu	1.000	0.898	0.035	Forming	-	28	1.47	0.045	32
6	Al	1.000	0.870	Smooth	-	Commercial Tubing	116	1.00	-	-
7	Al	1.000	0.943	0.026	Milling	Rolled Strip with Welded Seam	97	1.50	0.026	55

Notes:

* This diameter is very close to the diameter measured to the mid-height of the flutes.

** This tube was used at ORNL by Alexander and Hoffman (Their No. 19).

*** Corresponds to measured area ratio.

(A)

(B)

Fig. 1 . Profile of flutes: (A). Copper tubes; (B), Aluminum tube.

reasonable to compare any fluted tube with the smooth tube having the same internal cross-sectional area. That is the basis of comparison adopted in the present work.

4.2 Experimental Apparatus

The general layout of the experimental equipment is shown in Figure 2. Pittsburgh tap water was brought into the horizontal experimental tube through a mixing chamber and a 2.7 ft long calming section. After flowing through the test section, which was 2 to 8 ft. long, the water was discharged through a mixing tee to the drain or to a tank where the flow rate could be measured volumetrically. Manometer taps at the inlet and outlet of the test section permitted measurement of the pressure drop due to friction.

The water was heated by means of heating tapes wrapped around the outer wall of the test section. Veriac control was used to adjust the electrical power input which, for most runs, amounted to 1.5 kw. Wall temperatures along the test section were measured by five calibrated iron-Constantan thermocouples, spaced about evenly along the length. Thermocouples in the supply and discharge mixing chambers measured the bulk-average inlet and outlet water temperatures, thus permitting the energy balance to be closed. The test section was heavily insulated and the energy balance typically closed within 2 per cent.

4.3 Scope of Experiments

Inlet water temperatures varied with the season from roughly 40°F to 70°F, a range quite compatible with OTEC needs. Thermal power input was varied from 0.5 kw to 1.5 kw with most runs at the upper figure. Water flow rates were adjusted to produce average linear velocities from 1 ft/sec to 10 ft/sec in keeping with projected OTEC rates.

Fig. 2. Schematic drawing of apparatus.

Measurements of isothermal and nonisothermal pressure drop, heat flux and temperature difference were translated into Fanning friction factors and heat transfer coefficients. These were, in turn, correlated with the bulk Reynolds number of the flowing water.

5. Correlation of Water-Side Data

5.1 Fluid Friction

The Fanning friction factor (f) is a dimensionless conductance for momentum transfer. It is related to the frictional pressure drop (Δp), average fluid velocity (V), fluid density (ρ), tube length (L), inside tube diameter (D) and Newton's law conversion factor (g_c) through the Fanning equation

$$f = \Delta p g_c D / 2 \rho V^2 L \quad (1)$$

In smooth tubes, f is uniquely related to the dimensionless Reynolds number (N_{Re}) which is commonly defined as

$$N_{Re} = DV\rho/\mu \quad (2)$$

μ is the fluid viscosity and the other symbols are the same as indicated above. At Reynolds numbers below 2100, laminar flow prevails and from first principles $f = 16/N_{Re}$. The onset of disturbance eddies occurs at a Reynolds number of about 2100, introducing laminar-turbulent transition which is essentially completed, from the standpoint of friction, at a Reynolds number of about 4000. At higher Reynolds numbers, the fluid is in fully turbulent motion. In the Reynolds number range from 4000 to 100,000 the friction factor is represented empirically to about ± 2 per cent by the Blasius formula

$$f = 0.079/N_{Re}^{0.25} \quad (3)$$

In rough tubes or commercial pipes dimensional considerations indicate that at least one additional dimensionless group is needed to characterize the roughness. It is fortuitous that commercial pipes can be assigned a roughness height (e), different for each pipe-wall material such that the "relative roughness" (e/D) is an adequate dimensionless representation of the effect of roughness on the friction factor. Symbolically,

$$f = \phi(N_{Re}, e/D) \quad (4)$$

where ϕ is an empirical function.

Axially fluted tubes cannot be described by means of a single, dimensionless parameter, since both size and style must be taken into account. In view of that fact, and in the absence of an hydrodynamic model, separate correlations of friction factor as a function of Reynolds number were produced from the experimental data for each tube investigated. The smooth tube correlations were compared with the Blasius formula to eliminate the possibility of gross error. They were then used as the basis of comparison for the fluted tubes.

5.2 Heat Transfer

The water-side heat transfer coefficient (h) is conventionally related to the rate of heat transfer (q), the area of the water-side heat transfer surface (A) and the temperature difference ΔT between the tube wall and the mixing-cup temperature of the water through the equation

$$h = q/A\Delta T \quad (5)$$

Since the heat transfer coefficient is dimensional, it is customary to reduce it to a dimensionless conductance. One means of doing this is to form the dimensionless Stanton number (N_{St}) by dividing the heat transfer coefficient by the product of the fluid heat capacity (C_p), the fluid density and the average linear fluid velocity. Thus by definition,

$$N_{St} = h/C_p \rho V \quad (6)$$

Dimensional analysis indicates that for fully turbulent flow in smooth tubes

$$N_{St} = \psi_1(N_{Re}, N_{Pr}) \quad (7)$$

provided that radial differences in temperature can be neglected when the fluid properties are evaluated. The symbol ψ_1 represents an empirical function; the symbol N_{Pr} represents the Prandtl number, a dimensionless group defined as

$$N_{Pr} = C_p \mu / k \quad (8)$$

where k is the thermal conductivity of the fluid. Allowance for the effect of radial temperature changes in Equation 7 can be made in any one of several generally agreed-upon ways. Sieder and Tate [6] have suggested the form

$$N_{St} = \psi_2(N_{Re}, N_{Pr}, \mu/\mu_w)$$

where the Stanton, Reynolds and Prandtl numbers are evaluated at the bulk fluid temperature. The dimensionless group (μ/μ_w) is simply the ratio of the fluid viscosities at the bulk temperature and at the tube wall temperature respectively.

On the basis of empirical data in the fully turbulent range, Sieder and Tate [6] proposed the correlation

$$N_{St} = 0.027 N_{Pr}^{-2/3} N_{Re}^{-0.2} (\mu/\mu_w)^{0.14} \quad (9)$$

This can be rewritten as

$$N_{St} N_{Pr}^{2/3} (\mu_w/\mu)^{0.14} = 0.027 N_{Re}^{-0.2} \quad (10)$$

The left-hand-side of the equation is commonly known as the Sieder and Tate form of the Colburn j -factor for heat transfer (j_H). With this terminology,

$$j_H = 0.027 N_{Re}^{-0.2} \quad (11)$$

The j_H factor for fully turbulent flow in smooth tubes is therefore a unique function of the Reynolds number.

The region of laminar-turbulent transition indicated by heat transfer data extends over a wider range of Reynolds numbers than that observed through frictional data. It appears, empirically, that transitional heat transfer occurs in the Reynolds number range from 2100 to 8,000 or even 10,000. Over that range the heat transfer coefficient is influenced by the length-to-diameter ratio of the tube, since the development of the temperature profile has its effect even when the velocity profile is fully developed at all points along the tube. In the present work, the length-to-diameter ratio was constant, so a single correlation of j_H as a function of Reynolds number could be established from the smooth-tube data.

As in the case of friction, the heat transfer data for the smooth tubes were compared with the Sieder and Tate correlation in the fully turbulent range and with graphical information in the transitional range to eliminate the possibility of gross error. The actual smooth-tube data were then used as the basis of comparison for the fluted tubes. Each fluted tube was treated separately in terms of the relationship between j_H and the Reynolds number; as in the case of friction, there was no á priori basis for attempting to include flute size and style in a generalized correlation.

6. Results and Discussion of Water-Side Experiments

6.1 Pressure Drop

Correlations of isothermal friction factors are shown in Figures 3, 4 and 5; corresponding non-isothermal friction factors are shown in Figures 6 and 7.

In all cases, the friction factors associated with the smooth tubes are in close agreement with the Blasius formula (Equation 3). Axial scratches such as produced by wire-brushing do not increase the friction beyond the precision of measurement.

Regardless of their shapes, the fluted tubes exhibit the same general type of relationship between friction factor and Reynolds number. In all cases the fluted tubes behave almost as smooth tubes at low Reynolds numbers. At sufficiently high Reynolds numbers, the friction factor exceeds that of the corresponding smooth tube by a factor equal to the ratio of fluted-to-smooth surface area. As discussed previously, that is the behavior to be expected from "fully effective" flutes. Lines corresponding to full effectiveness can be seen on the appropriate graphs. Between the limits of smooth-tube behavior at low Reynolds numbers and fully effective behavior at high Reynolds numbers, there is in all cases a region of transition across which the friction factor is nearly independent of Reynolds number. Although characteristic of the flutes and fluted-to-smooth area ratios (1.00 to 1.50) investigated, there is no a priori reason to predict that the constancy of friction factor over the transition range will hold for other flutes or higher area ratios.

As a practical approximation, a Reynolds number of 50,000 in the present experiments corresponds to an average linear water velocity of about 10 ft/sec and the Reynolds number and velocity are directly proportional

Fig. 3 Isothermal friction factor.

Fig. 4. Isothermal friction factor.

Fig. 5. Isothermal friction factor: Tube No. 7.

Fig. 6 . Non-isothermal friction faction.

Fig. 7 . Non-isothermal friction factor.

to one another. That means that the investigated flutes attain full effectiveness at, say, 4 to 6 ft/sec, a water velocity quite in keeping with OTEC predictions based on probable constraints imposed by fouling and corrosion.

In view of the experimental data, there is no reason to anticipate that there is a fluted-to-smooth area ratio at which full effectiveness of the flutes cannot be attained. Conversely, the data suggest that full effectiveness can be expected at typical OTEC Reynolds numbers for area ratios of 2.0 or 2.5.

A fully effective flute of course represents the greatest friction factor and therefore the largest parasitic power loss to be encountered at a given Reynolds number in a given fluted tube. Whether the penalty is offset by economic benefits depends on whether the flutes are also fully effective from the standpoint of heat transfers.

6.2 Heat Transfer

Correlations of j -factors for heat transfer are presented in Figures 8 and 9. Although the smooth copper and aluminum tubes exhibit small differences, they are in substantial agreement with the Sieder and Tate correlation in the fully turbulent range. The deviation is well within the spread of data used to support the correlation initially.

As in the case of pressure drop, the fluted tubes move from smooth-tube behavior at low Reynolds numbers to full effectiveness at sufficiently high Reynolds numbers. Unlike the friction factor, the j -factor depends to some degree on Reynolds number in the intervening range. Lines corresponding to full effectiveness have been placed on the graphs one fluted-to-tube area ratio factor above the smooth tube line.

Fig. 8. Colburn j-factor for heat transfer.

Fig. 9. Colburn j -factor for heat transfer.

In general, both the transitional and fully effective regions are attained from the standpoint of heat transfer at lower Reynolds numbers than from the standpoint of friction. Full effectiveness is attained at water velocities of 3 to 5 ft/sec, well within projected OTEC limits.

If an analogy exists between heat transfer and momentum transfer in a fluid flowing in fully turbulent motion through a smooth tube, Colburn's reasoning leads to the simple relationship

$$j_H = f/2 \quad (12)$$

Preliminary studies [3] have suggested that such a relationship yields a beneficial economic trade-off between enhanced heat transfer and power loss. Thus if $2j_H/f \geq 1$, the situation is favorable. Table 2 shows the value of this ratio as a function of Reynolds number for fluted aluminum tube No. 7. At all Reynolds numbers above 10,000, the ratio is greater than unity. Above a Reynolds number of 20,000, the ratio reaches a plateau roughly 30 per cent greater than unity. Since both j_H and f are influenced in direct proportion by the fluted-to-smooth area ratio at high Reynolds numbers, the 30 per cent additional advantage must stem from the $(2j_H/f)$ ratio of the smooth tube.

It can be concluded that fine, axial flutes on the water side produce notable enhancement of heat transfer and friction at moderate water velocities. The relative enhancements of heat transfer and friction are such that a favorable economic trade-off can be anticipated.

Table 2

Values of $2j_H/f$
 for Fluted Aluminum
 Tube No. 7 (26-mil Flutes)

N_{Re}	$j_H \times 10^3$	$f \times 10^3$	$2j_H/f$
6,000	4.3	9.6	0.90
8,000	4.4	9.0	0.98
10,000	4.8	8.5	1.13
12,000	5.2	8.8	1.18
14,000	5.6	9.0	1.24
16,000	5.8	9.1	1.27
18,000	5.9	9.2	1.28
20,000	5.8	9.0	1.29
25,000	5.7	8.8	1.30
30,000	5.6	8.7	1.29
40,000	5.3	8.2	1.29
50,000	4.8	7.4	1.30

7. Generalization of Water-Side Data

It has been mentioned previously that at low Reynolds numbers the friction factors of fluted tubes are almost equal to those of corresponding smooth tubes. They do, however, appear to exceed the smooth-tube factors by a few per cent. Dr. L. Dykstra of our staff has proposed that at low Reynolds numbers the flutes are totally immersed in the laminar sublayer which exists adjacent to the wall of the tube, even when the main stream is in full turbulence. On that basis, he has shown that the friction factor is increased by the presence of constant-radius flutes. His equation is

$$f_{\text{fluted}} / f_{\text{smooth}} = 1 + (2n-3) \left(\frac{\delta}{D} \right)^2 \quad (13)$$

where δ is the flute height and n the total number of flutes. The 26-mil, constant-radius flutes of aluminum tube No. 7 should have a friction factor 8 per cent greater than smooth aluminum tube No. 6 according to Dr. Dykstra's equation. The corresponding line added to Figure 5 shows his prediction to be roughly correct.

It is reasonable to hypothesize that the fluted tube might first break away from the nearly-smooth-tube behavior visualized by Dr. Dykstra when the crest of the flute penetrates the laminar sublayer and projects into the region of turbulent motion. Since the level of turbulence changes continuously with distance from the wall, the "turbulent region" must be defined arbitrarily.

A zero-order approximation involves the old "film theory" of transport. In that model, all of the resistance to momentum transfer is assumed to lie in a thin laminar film of thickness δ_M adjacent to the wall. Since the velocity gradient in such a thin film is essentially constant and equal to V/δ_M , and since the shearing stress within the film is essentially

constant and equal to $f\rho V^2/2$,

$$\delta_M/D = 2/N_{Re} f \quad (14)$$

In the lower turbulent range where f is given by the Blasius formula,

$$\delta_M/D = 25.3/N_{Re}^{0.75} \quad (15)$$

Table 3 shows the hypothetical laminar film thickness for a smooth tube of 0.870-inch inside diameter. Flutes of 26-mil height penetrate this film at a calculated Reynolds number of 8000. Figure 5 indicates that the friction factor of the aluminum tube fitting these specifications departed from Dr. Dykstra's pseudo-smooth line at a Reynolds number of about 9000. This constitutes very good agreement with the film model.

Table 3
Thickness of Hypothetical
Laminar Film Thickness
for Momentum Transport
in a 0.870-inch I.D. Tube

N_{Re}	δ_M/D	δ_M (mils)
4,000	0.050	44
6,000	0.037	32
8,000	0.030	26
10,000	0.025	22

Similar application of the film theory to the turbulent transport of heat leads to a simple definition of the hypothetical thermal film thickness δ_H , namely

$$\delta_H = k/h \quad (16)$$

In terms of the j -factor for heat transfer,

$$\delta_H/D = 1/j_H N_{Pr}^{1/3} N_{Re} \quad (17)$$

Comparison with Equation 14 immediately yields

$$\delta_M/\delta_H = N_{Pr}^{1/3} (2j_H/f) \quad (18)$$

The quantity $(2j_H/f)$ refers, of course, to the smooth tube, in keeping with the preceding calculation of δ_M . When Equation 18 is applied to the fluted aluminum tube No. 7, the Reynolds number at which j_H departs from smooth-tube behavior is predicted to be about 5000. The experimentally observed Reynolds number is between 6000 and 7000. Thus, the simple film model represents both thermal and hydraulic behavior to an extent adequate for most purposes of design.

When the Reynolds number at which flutes attain full effectiveness from the standpoint of friction is known through experiment, Equation 18 can be used to predict the Reynolds number at which the flutes become fully effective from the standpoint of heat transfer. When the equation is applied to aluminum tube No. 7, the flutes are predicted to become fully effective with respect to heat transfer at a Reynolds number of 18,600. The observed value from Figure 6 is about 18,000, constituting excellent agreement with the film model. It is of interest to note that the hypothetical film thickness at the point of full effectiveness is about 8.5 mils, almost exactly one-third of the actual 26-mil flute height.

8. Power-Fluid Experiments

8.1 Experimental Tubes

Two tubes were used in this part of the study. One was a nominal 1-inch, 16 BWG, aluminum tube, the same as Tube No. 6 of the water-side experiments. The other was an externally fluted tube obtained from ORNL. This tube had been used by Alexander and Hoffman who designated it as their Tube No. 3. The fluted tube was seamless with 26-mil deep, essentially constant-radius flutes milled into the outer surface. The shape of the flutes was basically that shown in Figure 1b. Since the flutes were milled into a 1-inch, 16 BWG, aluminum tube, the externally fluted tube was in every respect the companion of internally-fluted tube No. 6 used in the water-side experiments. The effective lengths of the tubes varied with the liquid levels at the bottom of the twin units, but the average length was about 4-1/2 ft .

8.2 Experimental Apparatus

The layout of the experimental equipment is shown in Figure 10. One vertical member contained the smooth tube, the other the fluted tube. The heat source and sink could be reversed, permitting either tube to function as the evaporator while the other functioned as the condenser. A single run therefore produced data on the smooth evaporator and fluted condenser or on the fluted evaporator and smooth condenser, as the case might be.

Heating of the evaporator and cooling of the condenser were accomplished by water which flowed upward through the tubes. The warm water was heated electrically in an external supply tank. The cold water was taken at tap conditions for most of the Freon experiments but could be cooled by ice in an external supply tank on demand.

FIG. 10 SCHEMATIC DIAGRAM OF THE FREON-11 LOOP

Freon-11, flowing in closed-loop fashion was the test fluid in the experiments reported here. Currently ammonia experiments are being run in the same apparatus, only slightly modified. Freon-11 was chosen for preliminary runs because it has a vapor pressure close to 1 atmosphere at about room temperature. Therefore, it offers a minimum of hazard due to elevated pressure.

Liquid Freon was pumped from the bottom of each test section to a reservoir at the top. It was then discharged downward through a 1 ft long annulus surrounding the tube under test. The outer radius of the annulus was 0.010-inch greater than the nominal radius of the tube. Liquid flowed under gravity through the annular aperture onto the external surface of the tube below. The same arrangement existed on both sides of the twin unit. Thus, both the evaporator and condenser could be loaded independently at the top with a controlled flow rate of incoming liquid. Vapor generated in the evaporator flowed through any of three cross-over lines to the condenser. Unevaporated liquid was recycled to the top of the evaporator after being supplemented by make-up condensate from the condenser. The remaining liquid from the bottom of the condenser was recycled to the top of that unit. Liquid levels at the bottom of the test sections were used to indicate when steady-state had been achieved.

Calibrated iron-Constantan thermocouples (now replaced by thermistors) were used to measure inlet and outlet water temperatures, and water flow rates were measured by means of calibrated rotameters. Rotameters also measured the rates of supply of Freon to the tops of the test sections. Thermocouples were used to measure the Freon supply temperature and vapor temperatures at several positions in the vapor space, near the test tubes and also in the vapor cross-over lines. The apparatus was heavily insulated against heat loss. The heat balance was established by comparing the

heating load on the evaporator with the cooling load on the condenser, both based on water flow rates and temperatures. In most cases, the heat balance closed within 5 per cent .

8.3 Scope of Experiments

Freon-11 temperatures averaged roughly 95°F in the evaporator and 85°F in the condenser. Typical heat fluxes in each unit were from 600 to 900 Btu/(hr)(sq. ft.) . Liquid flow rates onto the tops of the test sections were such as to produce average Reynolds numbers in the evaporator from 200 to 4000 and in the condenser from 90 to 3500 . Only the lowest Reynolds numbers in the condenser were associated with zero loading at the top of the unit. In all other cases liquid was fed onto the tubes from the overhead reservoirs.

Except in a few runs with near-zero loading at the top of the tube, operating conditions were such that only a small fraction of the applied liquid was evaporated and condensed. The change of liquid Reynolds number between the top and bottom of the test section (on the average a distance of about 4 1/2 ft.) was therefore relatively small. Consequently, the heat transfer coefficient thus obtained was closely associated with the average Reynolds number and in that respect was a "local" value.

It is important to note the "local" characteristic of the present results. Their possible relationship to "average" quantities is taken up in a later section of this report.

Unlike ammonia coefficients, the heat transfer coefficients for Freon-11 were low enough to permit the backing-out of the water-side coefficient. The latter was well-established by the water-side experiments previously described. Therefore, measurement of the tube-wall temperature could be avoided in the preliminary runs with Freon. (Current experiments on ammonia employ wall-temperature measurements.)

The experimentally determined quantities permitted computation of the overall heat transfer coefficient. Knowledge of the water-side coefficient therefore allowed calculation of the tube-wall temperature. That, in turn, led to direct calculation of the power-fluid heat transfer coefficient, since both the heat flux and temperature potential on the power-fluid side were then known.

9. Correlation of Power-Fluid Data

9.1 Condensation

When a liquid layer of uniform thickness δ falls down a smooth, vertical surface under the action of gravity, a simple force balance on the layer shows that

$$\delta = C_1 \left(\frac{\mu^2}{\rho^2 g} \right)^{1/3} (N_{Re} \sqrt{f})^{2/3} \quad (19)$$

where C_1 is a constant and g is the local acceleration due to gravity. The Reynolds number of the layer on a surface having a large radius of curvature relative to δ is

$$N_{Re} = 4\delta V \rho / \mu \quad (20)$$

If the liquid layer is in completely laminar motion, the heat transfer coefficient is $h = k/\delta$ where k is the thermal conductivity of the liquid. At the same time $f = 24/N_{Re}$ for practical purposes. Therefore,

$$h/\phi = C_2 N_{Re}^{-1/3} \quad (21)$$

where C_2 is a constant and

$$\phi = (k^3 \rho^2 g / \mu^2)^{1/3} \quad (22)$$

If the liquid layer is in fully turbulent motion, analogy with flow between parallel plates yields

$$h/\phi = C_3 N_{Pr}^{1/3} N_{Re}^{0.22} \quad (23)$$

in the lower turbulent range. No generalized correlation is available for the range of laminar-turbulent transition.

Equations 21 and 23 indicate that heat transfer coefficients for a given fluid can be expressed as dimensional functions of the Reynolds number. The local coefficients obtained in the present experiments have been graphed on that basis for both smooth and fluted tubes.

9.2 Evaporation

If evaporation is the mirror-image of condensation, the preceding considerations should apply equally well to both. However, the literature neither confirms nor rejects the mirror-image concept. In view of this uncertainty, it can only be assumed that a dimensional graph of the local heat transfer coefficient against Reynolds number can adequately portray the difference between a smooth tube and a fluted tube of given size and style.

10. Results and Discussion of Power-Fluid Data

Heat transfer coefficients for the smooth and fluted condenser are presented in Figure 11. The coefficients for the smooth and fluted evaporator are presented in Figure 12.

The coefficients in the smooth-tube condenser are in almost exact agreement with Nusselt's theory in the laminar range. The coefficients in the smooth-tube evaporator are substantially higher, indicating that the mirror-image view of condensation and evaporation may not be correct.

FIG. 11 FILM HEAT TRANSFER COEFFICIENT
CONDENSER

FIG. 12 FILM HEAT TRANSFER COEFFICIENT
EVAPORATOR

(Current experiments on ammonia show the same degree of difference between the two processes.) An extensive search of the literature has shed no light on the situation. No explanation of the observed difference is available at the present time.

Consistent with the ORNL data of Alexander and Hoffman, the heat transfer coefficients for the fluted-tube condenser exhibit a local maximum in the range of laminar-turbulent transition. Although not spectacular, the peak is significant. It occurs precisely in the Reynolds number region where the smooth-tube coefficients pass through a minimum.

The fluted-tube evaporator has an interesting behavior. At low Reynolds numbers, the fluted tube is less effective than a smooth tube operated at the same Reynolds number. The explanation of this phenomenon lies in the fact that at low Reynolds numbers the liquid is confined to the rills of the flutes, leaving the crests dry. Unlike the crests of the flutes in the condenser, those of the evaporator have no opportunity to be wetted until laved with descending waves.

The heat transfer coefficient in the fluted-tube evaporator rapidly catches up with and surpasses the smooth-tube coefficient as the Reynolds number is increased. The local coefficient in the evaporator, in fact, exhibits a much more pronounced peak in the laminar-turbulent transition range than does the condenser coefficient. It is notable that the percentage increase of the coefficient over that of a smooth tube is essentially the same in the evaporator and condenser. That implies, but does not prove, that the difference between evaporation and condensation on the smooth tube is real and not merely an artifact of the present experiments.

11. Generalization of Power-Fluid Data

As noted previously, generally-accepted correlations of heat transfer coefficients for layer flow with a phase change indicate that h/ϕ is a rough function of Reynolds number, at least on smooth surfaces. Numerous studies of condensation have confirmed that the ϕ factor provides a satisfactory, albeit not perfect, means of proceeding from one fluid system to another, in the present case from Freon-11 to ammonia. That is to say, h/ϕ for condensation on a smooth tube is essentially the same function of Reynolds number for ammonia and for Freon-11. Although not as well confirmed by experiment, the same is probably true for evaporation on a smooth tube. In fact, experimental data being taken on ammonia at the present time in this laboratory indicate that both condensation and evaporation on smooth tubes can be correlated adequately in terms of the dimensionless ratio h/ϕ expressed as a function of the liquid-layer Reynolds number. That is, for both fluids, neglecting the effect of Prandtl number in full turbulence,

$$h/\phi = \psi_3(N_{Re}) = \psi_3(4\Gamma/\mu) \quad (24)$$

where ψ_3 is a unique empirical function, and Γ is the mass rate of liquid flow in the layer per unit outer perimeter of the tube.

Equation 24 implies that the effect of waves and ripples in the liquid layer falling down a smooth, vertical surface is roughly the same for all fluids. Although one study [7] has indicated a significant effect of surface tension (correlated through the dimensionless Weber number), the vast majority of experimental points scatter so much that adjustment of Equation 24 to include wave effects cannot be justified.

Fluted tubes introduce additional complications. First, it is not clear that flutes of different heights produce the same wave action at the

same mass flow rate (i.e. at the same Reynolds number) in all fluids. Second, it is equally unclear that the Reynolds number is the sufficient parameter for all fluids when the size and style of the flutes are varied, even when wave phenomena are ignored. Additional experimental data are therefore needed before the fluted-tube results can be generalized. Current experiments on ammonia are being performed in order to answer this need. In the meantime, the ϕ -factor can be used as the tentative link between Freon-11 and ammonia.

For purposes of design, it is convenient, if not necessary, to have available the mean heat transfer coefficient (h_m) over a finite tube length (L_0) rather than the local coefficient (h) over a differential tube length (dL). The mean coefficient is the length-mean if temperatures are constant:

$$h_m = \frac{1}{L_0} \int_0^{L_0} h dL \quad (25)$$

Since the local coefficient (h) is a function of the local Reynolds number (Equation 24) which is, in turn, a function of position (L) along the tube, the mean coefficient (h_m) can be established once the function ψ_3 of Equation 24 has been determined by means of experiment.

The results of the present work yield values of the local heat transfer coefficients. These coefficients are "local", however, only in the sense that they are procured at essentially constant Reynolds number. They are associated with a particular flow rate and a particular wave pattern that are essentially constant over the whole 4-1/2 ft length of the experimental tube.

In the prototype evaporator or condenser, the Reynolds number changes from point to point along the length. If the local conditions at any point are not influenced by the nature of the flow higher on the tube, the experimental coefficients of the present work can be inserted into Equation 25

and the mean coefficient can be determined directly. If, however, the local coefficients are sensitive to the wave forms existing at points higher on the tube, the coefficients measured in the present experiments cannot adequately account for the history of the liquid layer. It remains for current experimental work to clarify the situation.

12. Conclusions

1. Water-side, axial flutes significantly enhance the heat transfer coefficient. At moderate water velocities the coefficient is increased by a multiplying factor equal to the ratio of fluted-to-smooth surface area.
2. Water-side, axial flutes also increase the pressure loss due to friction. At the maximum, the parasitic loss is increased by a multiplying factor equal to the fluted-to-smooth surface area ratio. The trade-off between increased heat transfer coefficient and increased power loss is economically favorable.
3. Axial flutes on the power-fluid side greatly enhance heat transfer coefficients of Freon-11. Similar enhancement is anticipated for ammonia.

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