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INVESTIGATION OF HEAT TRANSFER PERFORMANCE OF A SERIES OF U. S. NAVY FREON-12 CONDENSERS

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ABSTRACT

This heat transfer investigation was made to determine the minimum number of Freon-12 finned tube condensers required to cover a range of heat duties from 21,000 to 1,180,000 Btu per hour for the Navy Department. The design procedure is outlined and the unit specifications for sixteen condensers are presented using a finned tube 5 feet long with land sections at 2, 3, and 4 feet from one end.

INTRODUCTION

The condensation of Freon refrigerants provides a favorable application for finned tubes. Compact units using integral finned tubes have found wide usage throughout the refrigeration industry. The Navy Department has indicated interest in the use of Trufin type S/T tubes for replacing the plain tube condensers currently being used.

Test data on the condensation of Freon-12 have been obtained by the York Corporation on six condensers tubed with 5/8-inch O.D. 18 BWG Cupro-Nickel plain tubes and three condensers tubed with 3/4-inch O.D. 16 BWG 90-10 Cupro-Nickel Trufin tubes having 19 fins per inch. The plain-tube test data had been used to prepare the standardized program currently being used by the Navy Department. Figure 1 presents the predicted heat-transfer performance of 24 plain-tube condensers with shell diameters varying from 6 to 20 inches and tube lengths varying from 1.5 to 8 feet. The number of units required to cover the desired range of heat duties from 21,000 to 1,180,000 Btu per hour can be reduced effectively by using finned tube units with plain ends of the same O.D. as the plain tubes. These tubes can be rolled into tube sheets and supported in the same manner as the plain tubes.

The York Corporation had analyzed their original test data obtained from the three finned tube condensers and had prepared rating curves from which the heat transfer performance of a given unit could be determined. On the basis of these rating curves, the Navy Department has completed a preliminary program which includes seven finned tube condensers with shell diameters varying from 6 to 16 inches and tube length of 2.5 and 5 feet. Table I gives the shell and tube specifications for the Navy Department condensers. Figure 2 gives the predicted performance of each unit within the range of water flow rates of 2 and 6 feet per second.

The following are the important considerations in numerical order of importance that the Navy Department desires to be included in a standardized finned tube program:

1. minimum number of tube lengths to carry in inventory,
2. minimum number of head sizes required,
3. minimum number of shell diameters, and
4. maximum length of tube 6 feet, and preferably shorter.

This investigation was made to determine the minimum number of finned tube condensers required to cover the range of heat duties from 21,000 to 1,180,000 Btu per hour.

DESIGN PROCEDURE

The procedure used to determine the performance of the recommended condensers is that suitable for the design of condensers using Trufin type S/T tubes. This method has been indicated previously.[1]

Equation (1) relates the heat duty, Q, of a heat exchanger with the overall heat-transfer coefficient, U_o, the total outside area, A, and the mean temperature difference, ΔT_{av}:

$$Q = U_o A \Delta T_{av} \quad . \tag{1}$$

For a given condenser the heat duty is also related to the water throughput and the temperature rise:

$$Q = W_t C_p \Delta t \quad . \tag{2}$$

In equation (1), U_o including an inside fouling factor, r_i, is defined by equation (3):

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{X_f A_o}{k_m A_m} + \frac{A_o}{A_i} r_i + \frac{A_o}{A_i h_i} \quad . \tag{3}$$

The essential steps of the design involve the determination of the correct ΔT driving force and the overall coefficient U_o from the individual resistances in equation (3).

For this application

$$\Delta T_{av} = \Delta T_{LM} = \frac{(T_{sv}-t_1) - (T_{sv}-t_2)}{\ln \dfrac{T_{sv} - t_1}{T_{sv} - t_2}} \quad . \tag{4}$$

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[1] Balekjian, G. and Young, E. H., "Use of Finned Tubes in Condensing Butyl Heads and Isopropyl Alcohol," University of Michigan, Engineering Research Institute Report No. 33 for Wolverine Tube Division of Calumet and Hecla, Inc., June, 1954.

The water-film coefficient is obtained from

$$h_w = 150\ (1+0.011\ t_w)\ \frac{V_t^{0.8}}{d_i^{0.2}} \qquad (5)$$

Evaluation of the tube wall resistance is shown in Appendix B. The outside condensing film coefficient is calculated from the following equation[2]:

$$h_o = 0.725\ C_N \left(\frac{k_f^3\ \rho_f^2\ g_c\ \lambda}{\mu_f\ N\ \Delta t_{cf} D_{eq}} \right)^{1/4} . \qquad (6)$$

In equation (6), $(1/D_{eq})^{1/4}$ is defined as follows:

$$\left(\frac{1}{D_{eq}}\right)^{1/4} = 1.3\ e_f\ \frac{A_f}{A_o}\left(\frac{1}{L}\right)^{1/4} + \frac{A_r}{A_o}\left(\frac{1}{D_r}\right)^{1/4} . \qquad (7)$$

C_N and N are functions of the number of tubes in the condenser and the tube layout. For a triangular tube layout

$$N = 0.40\ x^{0.54} . \qquad (8)$$

C_N is a correction factor applied to the condensing coefficient for the average tube in the bundle as predicted by the Nusselt equation, and it varies with N and the material condensed. In this investigation the variation of C_N with N for Freon-12 was established from the test data obtained by the York Corporation for 8-, 12-, and 16-inch shell condensers and given in Figure A of their test data. Evaluation of C_N is shown in Appendix A for run No. 28 of Figure A. Figure 3 indicates the C_N values calculated from the test data, along with the suitable lines used for subsequent designs. The two points appreciably below the upper line represent run Nos. 6 and 29 (Figure A of test data) at 1.97 feet per second water velocity. The lower line was used to determine the condensing coefficient values with water velocities of 2 feet per second, while the upper line was used for water velocities above 2 feet per second.

In this investigation shell diameters and header baffle clearances conform to the Bureau of Ships Specification No. 5000-S5902-F-841623-B. The tube selected is 90-10 Cupro-Nickel with 18 BWG (0.049 inch) wall thickness,

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2 Young, E. H., et al., "Condensation of Freon-114 on a Horizontal 19 Fin/Inch Tube," University of Michigan, Engineering Research Institute Report No. 32 for Wolverine Tube Division of Calumet and Hecla, Inc., January, 1954.

Trufin type S/T 195049-53.

Evaluation of h_o from equation (6) and the determination of U_o from equation (3) involve a process of successive approximation. This procedure is facilitated by using Figures 4 and 5. The preparation of Figure 4 includes fin efficiency corrections as is illustrated in the calculation given in Appendix C. Figure 5 presents the variation of the property group in the modified Nusselt equation (6) with the mean condensate film temperature, T_f. Appendix D presents the calculation of the total outside area for the 4-foot-long finned tube having 2-inch-long land sections at 2 and 3 feet from one end, and 1-inch-long plain ends.

The details of the design calculations are given in Appendix E for unit No. 15 with 16 inches I.D. shell and 4-foot-long tubes. The heat-transfer performance of each unit was determined at water velocities of 2, 3, 6, and 10 feet per second for a condensing temperature of 105°F, and an inlet water temperature of 85°F. Table II gives the characteristics of Trufin tube type S/T No. 195049-53. Table III gives the shell and tube specifications for the 16 Trufin units necessary to cover the specified range of heat duties. This table includes also additional units considered for possible use. Figure 6 presents the tube sheet layout for the 6-, 8-, 10-, 12-, and 16-inch condensers. Figure 7 presents the performance of the 16 units selected for the recommended standardized program. The units were selected on the basis of the overlap obtained with the performance at 3 to 6 feet per second water velocity indicted by the solid lines. The performance of each unit from 2 to 3 feet per second and 6 to 10 feet per second water velocity is shown by dashed lines.

Figure 8 is the rating chart determined for the condensers designed in this investigation. It is based on the water temperature rise and flow rates, and the heat duties summarized in Table III. The nominal heat duty for a ton of refrigeration is taken as 15,000 Btu per hour.

A comparison of the design procedure used in this investigation with that used by York Corporation for their recommendations indicates that the two methods give comparable results. The present method is preferred because in the evaluation of the condensing coefficient it accounts for the variation of the condensing film coefficient with the selected inside fouling resistance, water velocity and temperature, and number of tubes in the bundle. In their designs, York Corporation used a constant condensing coefficient for a given tube bundle obtained by averaging the four values calculated from the test data.

DISCUSSION AND RECOMMENDATIONS

In order to cover the wide range of heat duties, sixteen condensers were designed with five shell diameters and four different tube lengths. Table III summarizes the shell and tube specifications for the sixteen condensers covering a range of heat duties from 21,000 to 1,300,000 Btu/hr. Figure 7 presents the performance of each unit as a function of the water throughput in gal/min.

The desirable features of the designs summarized in Table III are:

1. Only five different head sizes and shell diameters are required.
2. The range of water velocities and temperature rise is suitable to prevent excessive tube-side fouling.
3. The sixteen condensers enable the selection of a unit which for a specific requirement provides a reasonable capacity range.
4. In order to accommodate the various tube and shell lengths it is recommended that a single tube length be carried in inventory. A tube 5 feet long with lands at 2, 3, and 4 feet from one end will provide all the necessary tube lengths with plain ends of 1 inch and land sections of 2 inches length.

The recommended standardized program is based on the predicted performance of units within the water velocity range of 3 to 6 ft/sec. The required number of units is more than that specified in Table I based on the performance of units within 2 to 6 ft/sec water velocity. In general, fouling on the tube side tends to become appreciable below 3 ft/sec, while erosion at the tube entrance and pressure drop became limiting factors at water velocities above 10 ft/sec. The Cupro-Nickel alloy tubes specified in this application are suitable for use with tube-side water velocities up to 10 ft per second. This water velocity range increases appreciably the range of performance of each unit so that the required heat duties may be obtained satisfactorily with fewer units than those specified in Figure 7. The use of water velocities inside the tubes of up to 10 ft./sec. is recommended for this particular application.

APPENDIX A

EVALUATION OF C_N FOR RUN NO. 28
FIGURE A OF YORK TEST DATA

1. TUBE CHARACTERISTICS

Trufin No. 195065-53   19 fins per inch

$d_o = 0.739$ inch

$d_r = 0.624$ inch

$d_i = 0.494$ inch

wall thickness = 0.065 inch (16 BWG)

mean fin thickness = 0.016 inch

$A_o = 0.496\ ft^2/ft$

$A_i = 0.1292\ ft^2/ft$

$A_o/A_i = 3.84$

$A_{cs} = 0.00133\ ft^2/tube$

$k_m = 27.5\ \frac{Btu}{hr\text{-}°F\text{-}ft}$

2. EXPERIMENTAL CONDENSING FILM COEFFICIENT

a. Metal Resistance - r_m

d_m, mean diameter, $= \dfrac{d_r - d_i}{l_n \dfrac{d_r}{d_i}}$

$$d_m = \frac{0.624-0.494}{l_n \frac{0.624}{0.494}} = 0.560 \text{ inch}$$

$$A_m = \frac{(3.14)(0.560)}{(12)} = 0.1468\ ft^2/ft$$

$$r_m = \frac{X_f A_o}{k_m A_m} \qquad (9)$$

$$r_m = \frac{(0.065)(0.496)}{(12)(27.5)(0.1468)} = 0.000665\ \frac{hr\text{-}°F\text{-}ft^2}{Btu}$$

b. Water Film Resistance - $\dfrac{A_o}{A_i h_w}$

Number of tubes in bundle = 46.
Water velocity, V_t = 5.96 ft/sec.
Average water temperature, $t_w = \frac{84.98+92.48}{2} = 88.73°F$
From equation (5),

$$h_w = 150\ (1+0.011x88.73)\ \frac{(5.96)^{0.8}}{(0.494)^{0.2}}\ ,$$

$$h_w = 1422\ \frac{Btu}{hr\text{-}°F\text{-}ft^2}\ ,\ \text{and}$$

$$\frac{A_o}{A_i h_w} = \frac{3.84}{1422} = 0.0027\ \frac{hr\text{-}°F\text{-}ft^2}{Btu}\ .$$

c. Experimental Condensing Film Coefficient - h_o

$$\frac{1}{h_o} = \frac{1}{U_o} - \frac{A_o}{A_i h_w} - r_m\ . \qquad (10)$$

$$U_o = 186.0\ ,\quad \frac{1}{U_o} = 0.00538\ .$$

$$\frac{1}{h_o} = 0.00538 - 0.0027 - 0.000665 = 0.002015\ .$$

$$h_o = \frac{1}{0.002015} = 496\ \frac{Btu}{hr\text{-}°F\text{-}ft^2}\ .$$

3. EVALUATION OF C_N

From equation (6)

$$h_o = 0.725\ C_N \left(\frac{k_f^3 \rho_f^2 g_c\ \lambda}{\mu_f N\ \Delta t_{cf} D_{eq}} \right)^{1/4}\ .$$

For, x = 46 tubes in the bundle,
$N = 0.40\ (46)^{0.54} = 3.16$,
$N^{1/4} = (3.16)^{1/4} = 1.3335$,
$\Delta T_{LM} = 16.04°F$ (From Figure A of test data),
$\Delta t_{cf} = (16.04)\left(\frac{0.002015}{0.00538}\right) = 6.0°F$,
$\Delta t_{cf}^{1/4} = (6.0)^{1/4} = 1.566$,

$$T_f = T_{sv} - 1/2\ \Delta t_{cf}\ ,\ \text{and} \qquad (11)$$

$$T_f = 105.06 - \frac{6.0}{2} = 102.06\,°F.$$

From Figure 5,

$$\left(\frac{k_f^3 \rho_f^2 g_c \lambda}{\mu_f}\right)^{1/4} = 406.7 \quad .$$

From Figure 4,

$$\text{at } h_o = 496, \left(\frac{1}{D_{eq}}\right)^{1/4} = 2.775 \quad .$$

Substituting in equation (6),

$$h_o = \frac{(0.725)(406.7)(2.775)}{(1.3335)(1.566)}\, C_N = 392\; C_N \quad .$$

From section (2c),

$$392\; C_N = 496 \quad \text{and}$$

$$C_N = \frac{496}{392} = 1.27 \quad .$$

This result is shown by the circular point at N = 3.16 in Figure 3.

APPENDIX B

EVALUATION OF THE TUBE WALL RESISTANCE FOR TRUFIN NO. 195049-53

1. TUBE CHARACTERISTICS - Trufin No. 195049-53

   Summarized in Table II

2. METAL RESISTANCE - r_m $\dfrac{X_f A_o}{K_m A_m}$

$$d_m = \frac{0.624-0.526}{\ln \dfrac{0.624}{0.526}} = 0.583 \text{ inch} .$$

$$A_m = \frac{(3.14)(0.583)}{(12)} = 0.1528 \text{ ft}^2/\text{ft} .$$

$$r_m = \frac{(0.049)(0.496)}{(12)(27.5)(0.1528)} = 0.000482 \frac{\text{hr-°F-ft}^2}{\text{Btu}} .$$

APPENDIX C

CALCULATION FOR VARIATION OF $(1/D_{eq})^{1/4}$ WITH h_o FOR TRUFIN NO. 195049-53 TUBE

The following calculation is made for one point on Figure 4. It involves evaluation of the outside film coefficient (h_o) in Btu/hr-°F-ft^2 outside area and the equivalent diameter $(1/D_{eq})^{1/4}$ for an assumed value of the outside film coefficient (h_o') in Btu/hr-°F-ft^2 equivalent outside area.

In equation (7),

$$\left(\frac{1}{D_{eq}}\right)^{1/4} = 1.3\ e_f\ \frac{A_f}{A_o}\left(\frac{1}{L}\right)^{1/4} + \frac{A_r}{A_o}\left(\frac{1}{D_r}\right)^{1/4} \quad ,$$

$$A_f = (0.80)\ (0.496) = 0.397\ \text{ft}^2/\text{ft} \quad ,$$

$$A_r = (0.20)(0.496) = 0.099\ \text{ft}^2/\text{ft} \quad , \text{ and}$$

$$L, \text{ mean effective fin height} = \frac{(0.397)(12)}{(12)(19)(2)(0.739)} = 0.01414\ \text{ft.}$$

Substituting in equation (7),

$$\left(\frac{1}{D_{eq}}\right)^{1/4} = \frac{(1.3)(0.397)}{(0.496)}\left(\frac{1}{0.01414}\right)^{1/4} e_f + \frac{(0.099)}{(0.496)}\left(\frac{12}{0.624}\right)^{1/4} \quad , \text{ and}$$

$$\left(\frac{1}{D_{eq}}\right)^{1/4} = 3.02\ e_f + 0.417 \ .$$

To evaluate the fin efficiency from Figure 5 of Reference 1,

$$\frac{d_o}{d_r} = \frac{0.739}{0.624} = 1.182 \quad \text{and}$$

$$\text{fin height, } H = \frac{(0.739 - 0.624)}{(2)(12)} = 0.0048\ \text{ft.}$$

For an outside fouling factor of $r_o = 0$, the abscissa reduces to

$$H\sqrt{\frac{2h_o'}{k_m Y}} = 0.0048\sqrt{\frac{(2)\ h_o'\ (12)}{(27.5)(0.016)}} = 0.0355\sqrt{h_o'} \ .$$

For an assumed value of

$$h_o' = 800 \frac{\text{Btu}}{\text{hr-°F-ft}^2 \text{ equivalent outside area}} \quad ,$$

$$\text{abscissa} = 0.0355\sqrt{800} = 1.003 \quad .$$

$$\therefore \quad e_f = 0.735 \quad \text{and}$$

$$\left(\frac{1}{D_{eq}}\right)^{1/4} = (3.02)(0.735) + 0.417 = 2.637 \quad .$$

$$A_e = e_f\, A_f + A_r \quad , \tag{12}$$

$$A_e = (0.735)(0.397) + 0.099 = 0.391 \text{ ft}^2/\text{ft} \quad ,$$

$$h_o = \frac{A_e}{A_o} h_o' \quad , \text{ and} \tag{13}$$

$$h_o = \frac{(0.391)}{(0.496)}(800) = 630 \frac{\text{Btu}}{\text{hr-°F-ft}^2 \text{ outside area}} \quad .$$

This calculation gives one point on Figure 4 and is repeated for other values of the assumed coefficient h_o'.

APPENDIX D

CALCULATION OF TOTAL OUTSIDE AREA OF 4-FOOT-LONG TUBE

Total tube length = 4 feet
2-inch -long land sections at 2 and 3 feet
1-inch -long end sections, and
Length rolled into tube sheet = 3/4 inch.

At every land and end the reduction in the outside area because of imperfect finning is:

0.86 sq. inches at start of finning operation, and
1.88 sq. inches at end of finning operation.
Total reduction in area = 0.86 + 1.88 = 2.74 sq. inches.

The total outside area of the tube is obtained from the following equation:

$$A = L_1 A_o + L_2 A_o' - \frac{2.74}{144}(n+1) \;, \qquad (14)$$

where:

A = total outside area, ft^2 ,
n = number of land sections ,
L_1 = total length - $\frac{2(n+1)}{12}$, ft ,

$$L_2 = \frac{2n}{12} + \frac{1}{24} \;, \text{ ft} \;,$$

$$A_o = 0.496 \text{ ft}^2/\text{ft} \;, \text{ and}$$

$$A_o' = 0.1963 \text{ ft}^2/\text{ft} \;.$$

For the 4-ft tube,

$$L_1 = 4 - \frac{6}{12} = 3.5 \text{ ft} \quad \text{and}$$

$$L_2 = \frac{(2)(2)}{12} + \frac{1}{24} = 0.375 \text{ ft} \;.$$

Substituting in equation (14),

$$A = (3.5)(0.496) + (0.375)(0.1963) - \left(\frac{2.74}{144}\right)(2+1) \;,$$

$$A = 1.737 + 0.0736 - 0.057 = 1.754 \text{ ft}^2 .$$

The outside area of the other tubes was calculated similarly:

Tube length, ft	No. of Lands	Outside Area, ft^2
5	3	2.318
4	2	1.754
3	1	1.325
2	0	0.8992

APPENDIX E

EVALUATION OF THE PERFORMANCE OF UNIT NO. 15

1. CONDENSER SPECIFICATIONS

Shell I.D. = 16 inches,
No. of tubes = 222,
No. of tube passes = 2,
Total outside area = (222)(1.754) = 389.5 ft^2, and
Tube layout: $\frac{15}{16}$-inch triangular pitch.

2. HEAT-TRANSFER PERFORMANCE - at 3 ft/sec water velocity

No. of tubes per pass = $\frac{222}{2}$ = 111,

A_{cst} = (111)(0.00151) = 0.1676 ft^2/pass,

$$\text{Water flow rate} = (0.1676)(3)(3600)(62) = 112{,}200 \text{ lb/hr} = \frac{(112{,}200)}{(8.33)(60)} = 224.4 \ \frac{\text{gal}}{\text{min}} .$$

From section (2) of Appendix B,

$$r_m = 0.000482 \ \frac{\text{hr-°F-ft}^2}{\text{Btu}} .$$

For the selected inside fouling factor,

$$r_i = 0.0005 \quad \text{and}$$

$$\frac{A_o}{A_i} r_i = (0.0005)(3.59) = 0.001795 \ \frac{\text{hr-°F-ft}^2}{\text{Btu}} .$$

The overall coefficient is determined by successive approximation in the following manner:

saturated vapor temperature = 105°F,
inlet water temperature = 85°F,
assume Δt water = 6.8°F,
t_2 = 85 + 6.8 = 91.8°F ,

$$\Delta T_{av} = \Delta T_{LM} = \frac{(105-85) - (105-91.8)}{\ln \frac{105.85}{105-91.8}} = 16.4°\text{F, and}$$

$$t_w = \frac{85 + 91.8}{2} = 88.4°\text{F}.$$

From equation (5),

$$h_w = (150)(1+0.011x88.4)\ \frac{(3)^{0.8}}{(0.526)^{0.2}} \quad ,$$

$$h_w = 811 \ \frac{\text{Btu}}{\text{hr-°F-ft}^2} \quad .$$

Water-film resistance $= \dfrac{A_o}{A_i h_w} = \dfrac{3.59}{811} = 0.00442\ \dfrac{\text{hr-°F-ft}^2}{\text{Btu}}$ and

$$R_t - \frac{1}{h_o} = 0.00442 + 0.001795 + 0.000482$$
$$= 0.006697\ \frac{\text{hr-°F-ft}^2}{\text{Btu}} \quad .$$

Assume $h_o = 595\ \dfrac{\text{Btu}}{\text{hr-°F-ft}^2}$,

$$\frac{1}{h_o} = \frac{1}{595} = 0.00168 \ ,$$

$$R_t = 0.00168+0.006697 = 0.00838 \quad ,$$

$$\Delta t_{cf} = \left(\frac{0.00168}{0.00838}\right)(16.4) = 3.29\,°\text{F} \quad ,$$

$$\Delta t_{cf}^{1/4} = (3.29)^{1/4} = 1.347 \quad \text{, and}$$

$$T_f = 105 - \frac{3.29}{2} = 103.4\,°\text{F} \ .$$

From Figure 5,

$$\left(\frac{k_f^3 \rho_f^2 g_c \lambda}{\mu_f}\right)^{1/4} = 405.6 \quad .$$

From Figure 4,

$$\left(\frac{1}{D_{eq}}\right)^{1/4} = 2.67 \quad .$$

From equation (8),

$$N = 0.40(222)^{0.54} = 7.4 \quad \text{and}$$
$$N^{1/4} = (7.4)^{1/4} = 1.65 \quad .$$

From the upper line of Figure 3,

$$C_N = 1.685 \quad .$$

Substituting in equation (6),

$$h_o = \frac{(0.725)(1.685)(405.6)(2.67)}{(1.65)(1.347)} = 596 \ \frac{\text{Btu}}{\text{hr-°F-ft}^2} \quad .$$

Since the assumed and calculated values of h_o agree satisfactorily,

$$U_o = \frac{1}{R_t} = \frac{1}{0.00838} = 119.4 \ \frac{\text{Btu}}{\text{hr-°F-ft}^2 \text{ outside area}} \quad .$$

For the assumed Δt water, condenser heat duty from equation (1),

$$Q = (119.4)(16.4)(389.5) = 763{,}000 \text{ Btu/hr} \quad \text{, and}$$

$$\text{corresponding } \Delta t \text{ water} = \frac{763{,}000}{112{,}200} = 6.8\text{°F} \quad .$$

Calculated Δt water checks satisfactorily with the assumed value of 6.8°F.

3. HEAT-TRANSFER PERFORMANCE - at 6 ft/sec water velocity

$$\begin{aligned} \text{Water flow rate} &= (0.1676)(6)(3600)(62) \\ &= 224{,}400 \text{ lb/hr} \\ &= \frac{224{,}400}{(833)(60)} = 448.8 \ \frac{\text{gal}}{\text{min}} \quad . \end{aligned}$$

Assume Δt water = 4.6°F ,

$$t_2 = 85+4.6 = 89.6\text{°F} \quad ,$$

$$\Delta T_{LM} = \frac{(105-85)-(105-89.6)}{\ln \dfrac{(105-85)}{(105-89.6)}} = 17.52\text{°F} \quad ,$$

$$t_w = \frac{85+89.6}{2} = 87.3\text{°F} \quad ,$$

$$h_w = 150(1+0.011 \times 87.3) \ \frac{(6)^{0.8}}{(0.526)^{0.2}} = 1405 \quad ,$$

$$\frac{A_o}{A_i h_w} = \frac{3.59}{1405} = 0.00256 \text{ , and}$$

$$R_t - \frac{1}{h_o} = 0.00256 + 0.001795 + 0.000482 = 0.004837 \text{ .}$$

Assume $h_o = 550$,

$$\frac{1}{h_o} = \frac{1}{550} = 0.00182 \text{ ,}$$

$$R_t = 0.004837 + 0.00182 = 0.006657 \text{ ,}$$

$$\Delta t_{cf} = \left(\frac{0.00182}{0.006657}\right)(17.52) = 4.79°F \text{ ,}$$

$$\Delta t_{cf}^{1/4} = (4.79)^{1/4} = 1.48 \text{ , and}$$

$$T_f = 105 - \frac{4.79}{2} = 102.6°F \text{ .}$$

From Figure 5,

$$\left(\frac{k_f^3 \rho_f^2 g_c \lambda}{\mu_f}\right)^{1/4} = 406.2 \text{ .}$$

From Figure 4,

$$\left(\frac{1}{D_{eq}}\right)^{1/4} = 2.715 \text{ .}$$

Substituting in equation (6),

$$h_o = \frac{(0.725)(1.685)(406.2)(2.715)}{(1.65)(1.48)} = 551 \text{ .}$$

The assumed value of h_o checks with the calculated value

$$U_o = \frac{1}{0.006657} = 150 \frac{\text{Btu}}{\text{hr-°F-ft}^2 \text{ outside area}} \text{ .}$$

For the assumed Δt water,

$$Q = (150)(17.52)(389.5) = 1{,}023{,}000 \frac{\text{Btu}}{\text{hr}} \text{ .}$$

Corresponding,

$$\Delta t \text{ water} = \frac{1{,}023{,}000}{224{,}400} = 4.56°F \text{ .}$$

Calculated Δt water checks satisfactorily with the assumed value of 4.6°F.

TABLE I

SHELL AND TUBE SPECIFICATIONS
FOR NAVY DEPARTMENT FINNED TUBE CONDENSERS

Unit No.	Shell O.D., In.	Tube Length, Ft	No. of Tubes	No. of Tube Passes	Total Outside Area, Ft^2	Water Flow Rate Gal/Min		Heat Duty Btu/Hr	
						2 Ft/Sec	6 Ft/Sec	2 Ft/Sec	6 Ft/Sec
1	6-5/8	2.5	12	4	13.9	3.6	10.7	21,000	41,500
2	6-5/8	2.5	20	4	23.2	6.0	18.0	34,500	69,000
3	6-5/8	5	20	2	46.4	12.0	31.0	65,000	115,000
4	8-5/8	5	38	2	88.0	21.5	68.0	115,000	222,000
5	10-3/4	5	68	2	157.2	36.0	107.0	195,000	385,000
6	12-3/4	5	102	2	236.0	68.0	190.0	360,000	645,000
7	16-13/16	5	196	2	454.0	96.0	350.0	515,000	1,180,000

Tubes: Trufin Type S/T No. 195042-53.

TABLE II

TUBE CHARACTERISTICS OF
TRUFIN TYPE S/T NO. 195049-53

MATERIAL: 90-10 CUPRO-NICKEL

19 fins per inch
d_o = 0.739 inch
d_r = 0.624 inch
d_i = 0.526 inch
wall thickness - 0.049 inch (18 BWG)
mean fin thickness = 0.016 inch
A_o = 0.496 ft^2/ft
A_i = 0.138 ft^2/ft
A_o/A_i = 3.59
A_{cs} = 0.00151 ft^2/tube
$k_m = 27.5 \dfrac{Btu}{hr\text{-}°F\text{-}ft}$

TABLE III

SHELL AND TUBE SPECIFICATIONS FOR FREON-12 CONDENSERS

Unit No.	Shell I.D., in.	Tube Length, ft	No. of Tubes	No. of Tube Passes	Total Outside Area, ft^2	Water Velocity: 2 ft/sec			3 ft/sec			6 ft/sec			10/ft/sec		
						Δt, °F	GPM	Q_1 Btu/hr	Δt, °F	GPM	Q_1 Btu/hr	Δt, °F	GPM	Q_1 Btu/hr	Δt, °F	GPM	Q_1 Btu/hr
1	6	2	13	2	11.7	2.4	17.5	20,900	1.9	26.3	25,400	1.2	52.5	32,800	0.85	87.5	37,100
2	6	3	13	2	17.2	3.5	17.5	30,440	2.8	26.3	36,500	1.8	52.5	46,600	1.2	87.5	53,700
3	6	4	13	2	22.8	4.5	17.5	39,000	3.6	26.3	47,200	2.3	52.5	61,500	1.6	87.5	70,100
4	6	5	13	2	30.1	5.6	17.5	49,500	4.6	26.3	60,900	3.1	52.5	80,400	2.1	87.5	93,400
--	6	2	26	2	23.4				3.7	26.3	48,200	2.4	52.5	62,900			
5	6	3	26	2	34.4	6.2	17.5	54,400	5.2	26.3	68,800	3.4	52.5	90,700	2.4	87.5	104,000
6	6	4	26	2	45.6	7.8	17.5	68,700	6.6	26.3	87,100	4.4	52.5	117,000	3.1	87.5	137,500
7	6	5	26	2	60.2	9.6	17.5	84,400	8.3	26.3	108,800	5.6	52.5	148,500	4.0	87.5	175,800
	6	6	26	2	69.6				9.2	26.3	121,500	6.4	52.5	168,000			
8	8	3	48	2	63.6	6.2	32.3	100,700	5.3	48.5	128,300	3.5	97.0	172,000	2.5	161.5	198,200
9	8	4	48	2	84.3	7.6	32.3	122,000	6.7	48.5	161,800	4.5	97.0	219,000	3.2	161.5	256,000
10	8	5	48	2	111.0	9.6	32.3	155,000	8.3	48.5	202,000	5.7	97.0	276,000	4.1	161.5	330,000
--	8	6	48	2	128.4				9.3	48.5	226,000	6.5	97.0	314,500			
--	10	3	76	2	100.6				5.3	77.0	202,000	3.6	154.0	274,000			
11	10	4	76	2	133.2	7.8	51.1	199,200	6.8	77.0	259,500	4.5	154.0	346,000	3.2	256.0	410,000
12	10	5	76	2	176.0	9.6	51.1	246,000	8.4	77.0	322,000	5.8	154.0	443,000	4.1	256.0	526,000
--	10	6	76	2	203.5				9.3	77.0	358,000	6.5	154.0	501,000			
--	12	3	120	2	159.0				5.4	121.5	324,500	3.5	243.0	431,000			
13	12	4	120	2	210.5	7.8	81.0	316,000	6.8	121.5	411,000	4.6	243.0	555,000	3.2	404.0	658,000
14	12	5	120	2	278.0	9.6	81:0	389,000	8.5	121.5	515,000	5.8	243.0	710,000	4.2	404.0	850,000
--	12	6	120	2	321.0				9.4	121.5	567,000	6.6	243.0	805,000			
15	16	4	222	2	389.5	7.7	150.0	580,000	6.8	224.4	763,000	4.6	448.8	1,023,000	3.3	750.0	1,222,000
16	16	5	222	2	514.5	9.6	150.0	716,000	8.5	224.4	952,000	5.9	448.8	1,320,000	4.2	750.0	1,570,000
--	16	6	222	2	594.0				9.4	224.4	1,060,000	6.6	448.8	1,490,000			

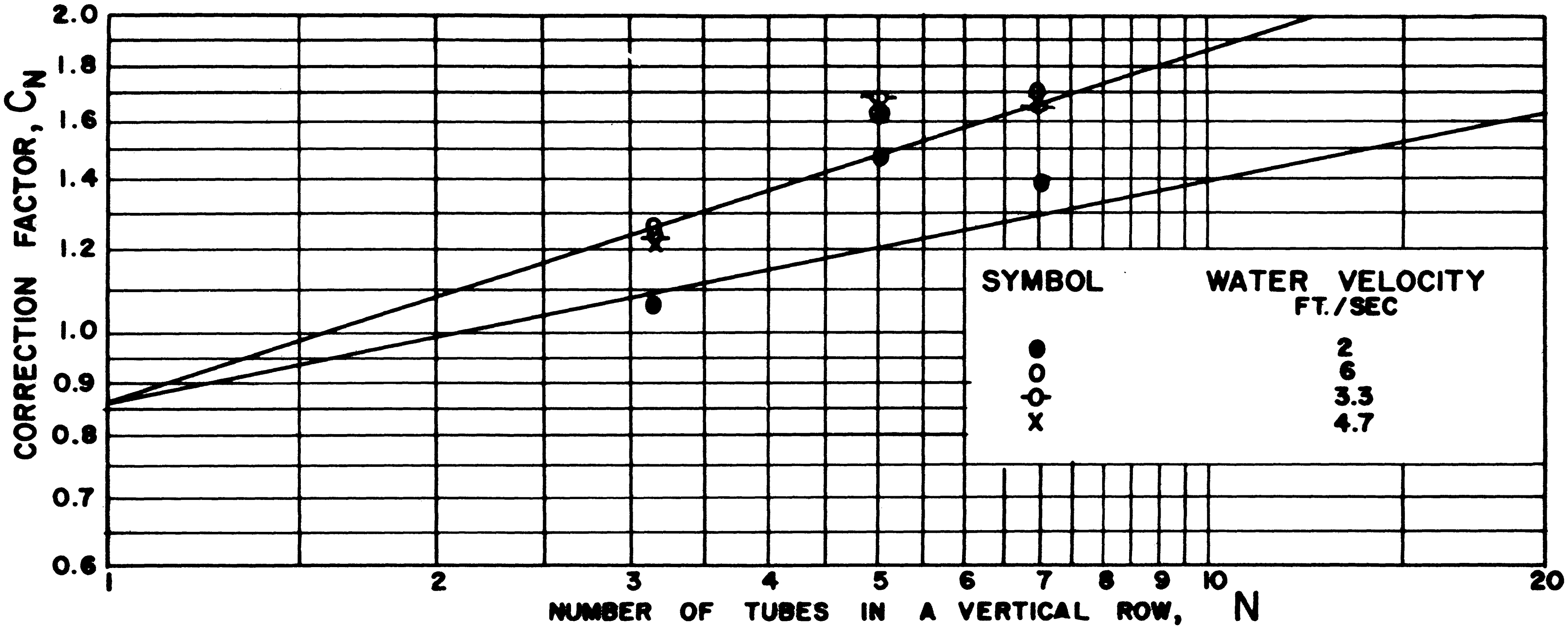

FIGURE 3 RATIO OF EXPERIMENTAL TO THEORETICAL CONDENSATION COEFFICIENTS FOR FREON-12

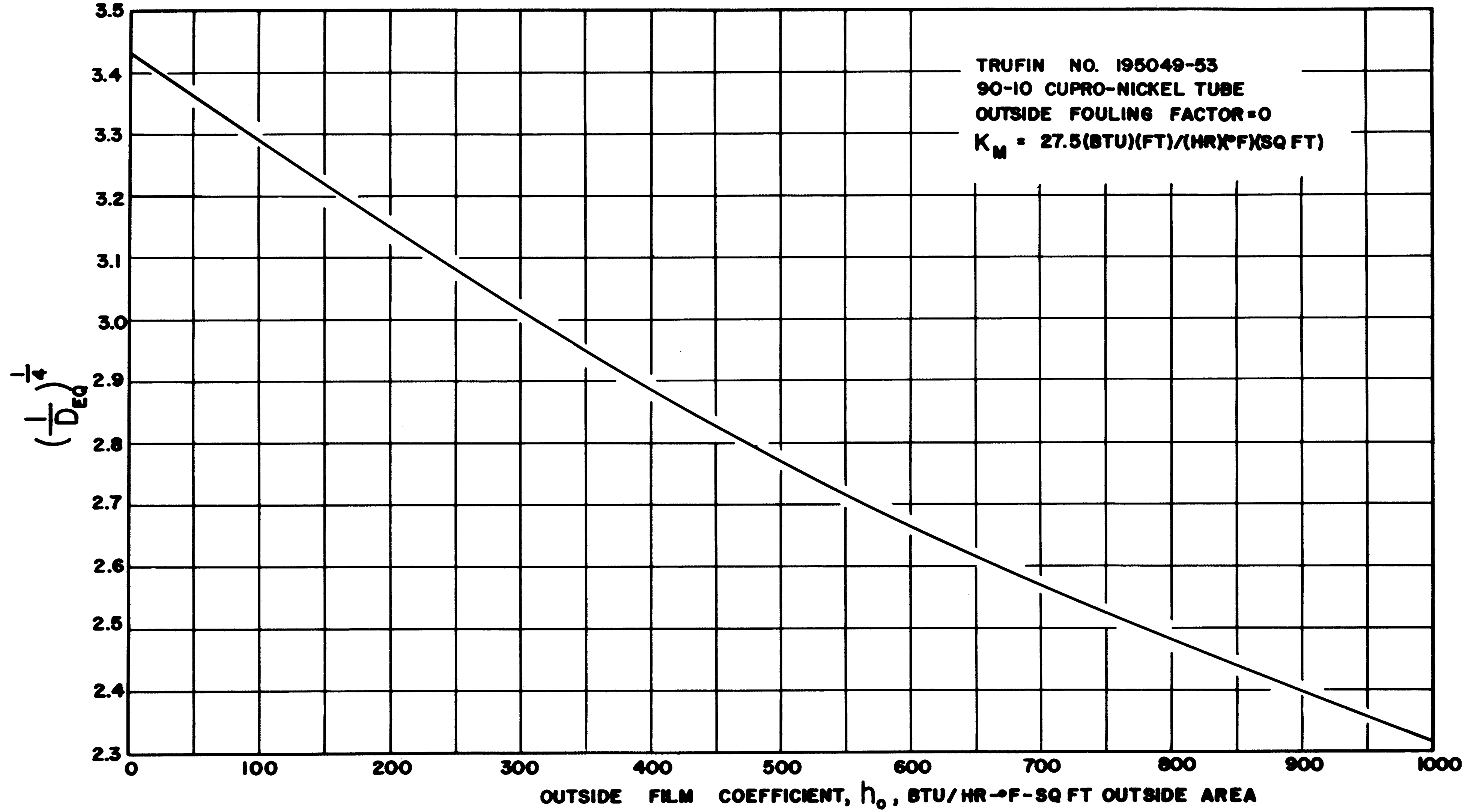

FIGURE 4 VARIATION OF $\left(\frac{1}{D_{EQ}}\right)^{\frac{1}{4}}$ WITH OUTSIDE FILM COEFFICIENT

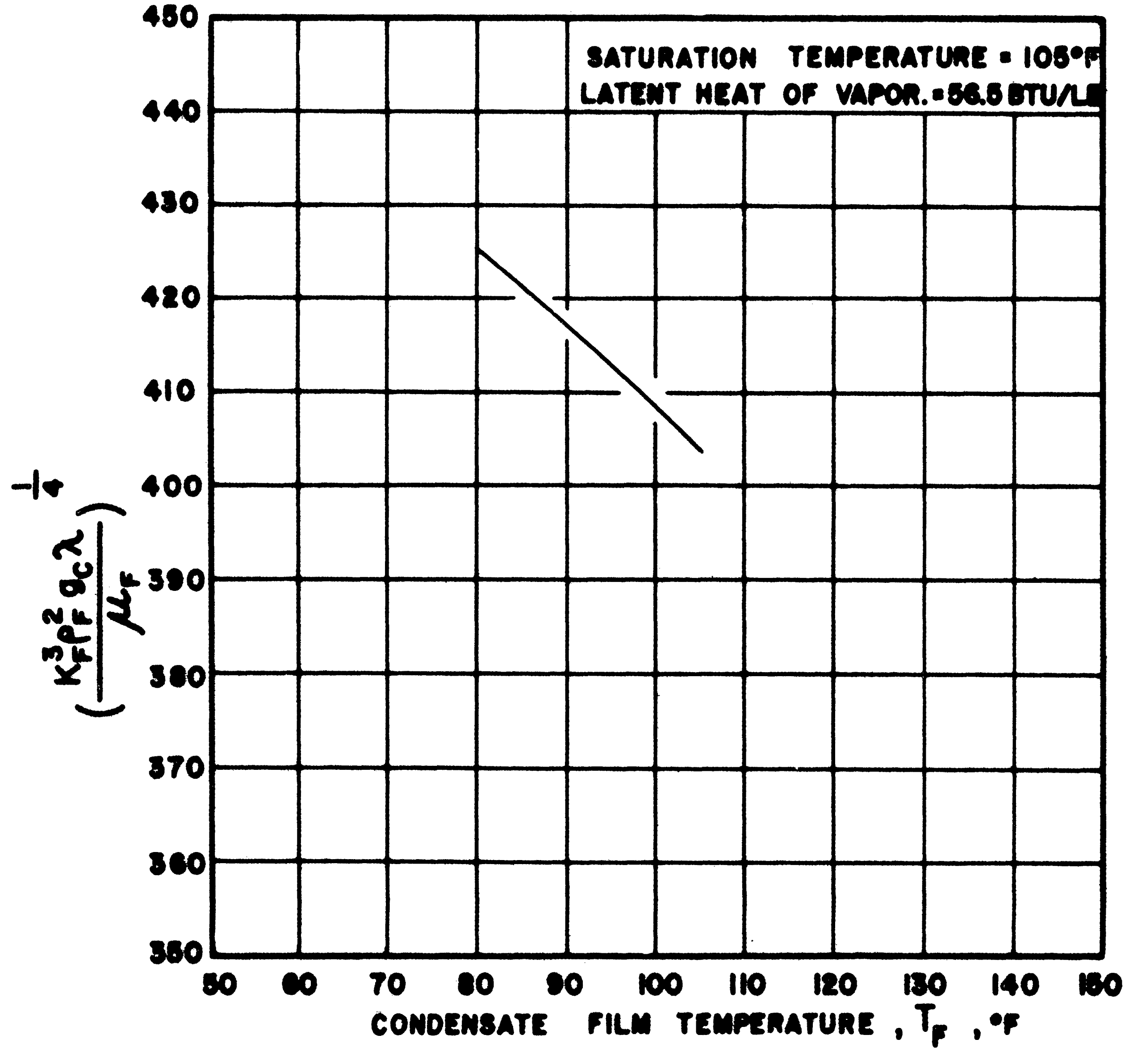

FIGURE 5 VARIATION OF PHYSICAL PROPERTY GROUP IN NUSSELT EQUATION WITH CONDENSING AND CONDENSATE FILM TEMPERATURES FOR FREON-12

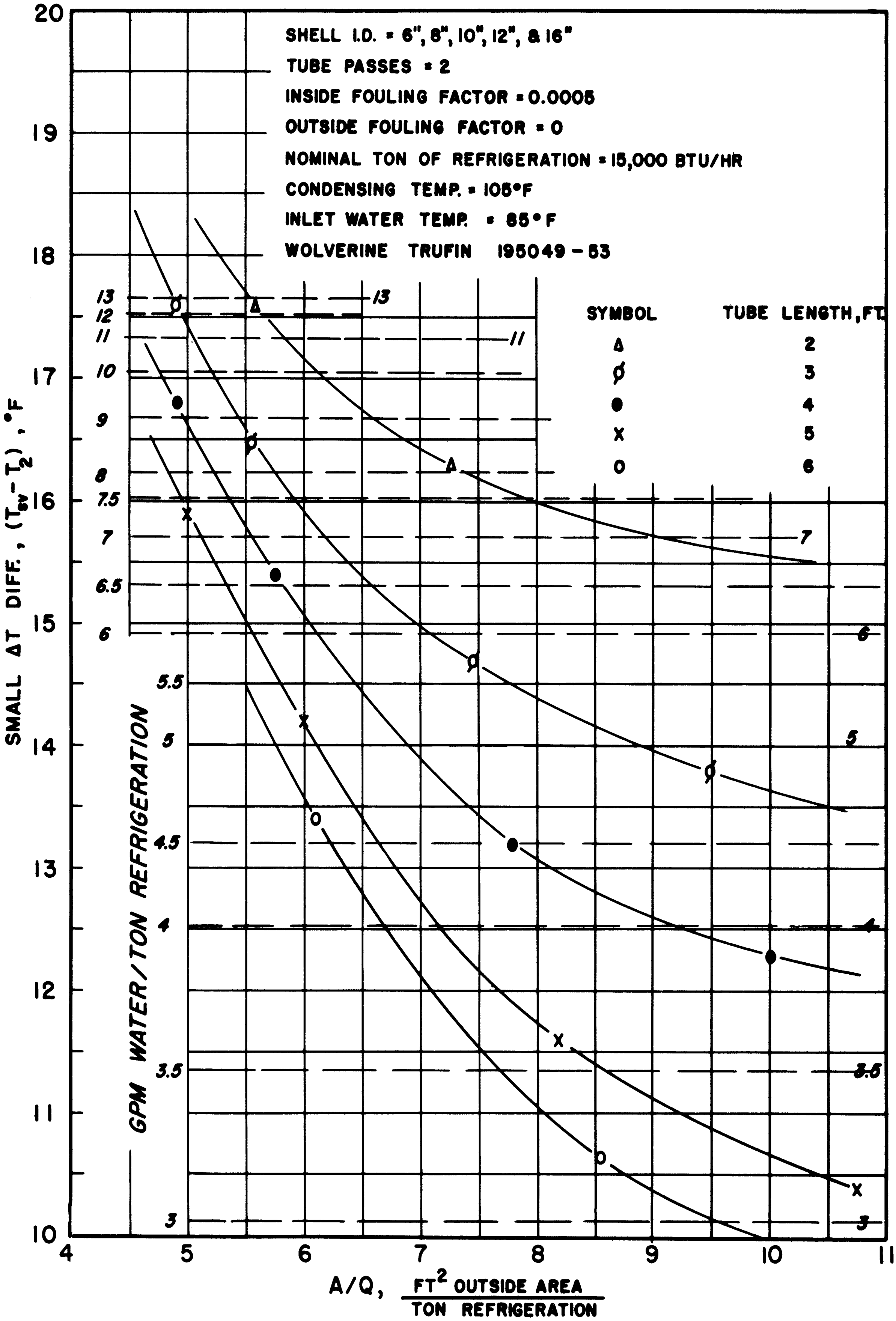

FIGURE 8 DESIGN CHART FOR FREON-12 CONDENSERS

NOMENCLATURE

A Total outside area, ft^2

A_{cs} Cross section area of tube for water flow, ft^2

A_f Outside area of fins, ft^2/ft

A_i Inside heat transfer area, ft^2/ft

A_m Mean metal heat transfer area

$$A_m = \frac{\pi(d_m)}{12} \qquad ft^2/ft$$

A_o Outside area, ft^2/ft

A_r Outside root metal area, ft^2/ft

C_N Correction factor for condensing coefficient with N tubes in a vertical row

C_p Mean heat capacity, Btu/lb-°F

d_i Inside diameter of tube, inches

d_m $\dfrac{d_o - d_r}{l_n \dfrac{d_o}{d_r}}$, inches

d_o Diameter over fins, inches

d_r Root diameter of tube, inches

D_o Diameter over fins, feet

D_{eq} Equivalent diameter of finned tube for condensing, feet

e_f Fin efficiency, defined as $\dfrac{T_{sv}-t_f}{T_{sv}-t_r}$ where t_f and t_r are the temperatures of the fin and root respectively

h_i	Inside heat transfer coefficient $\dfrac{\text{Btu}}{\text{hr-°F-ft}^2\text{ (inside area)}}$
h_o	Outside condensing coefficient $\dfrac{\text{Btu}}{\text{hr-°F-ft}^2\text{ (outside area)}}$
h_o'	Outside condensing coefficient, $\dfrac{\text{Btu}}{\text{hr-°F-ft}^2\text{ (equivalent area)}}$
K_M	Thermal conductivity of metal, Btu/hr-°F-ft
$\left(\dfrac{k_f^3 \rho_f^2 g_c}{\mu_f}\right)$	Nusselt physical property group
L	Length of finned condensing surface $L = \dfrac{(12)A_f}{d_o(N)(2)(12)} = \text{ft}$
L_1	Length of finned portion of tube, ft
L_2	Length of plain or unfinned portion of tube, ft
N	Average number of tubes in a vertical row
Q	Total heat duty, Btu/hr
r_i	Inside fouling resistance, hr-°F-ft^2(inside area)/Btu
R_t	Total resistance to heat transfer, $1/U_o$
t_1	Inlet water temperature, °F
t_2	Outlet water temperature, °F
T_{sv}	Temperature of saturated vapor, °F
t_w	Average water temperature, °F
T_f	Average condensate film temperature, °F
Δt	Temperature rise of water
Δt_c	Temperature drop across condensate film, °F
ΔT_{av}	Mean temperature difference, °F
ΔT_{LM}	Logarithmic temperature difference, °F

U_o	Overall heat transfer coefficient, Btu/hr-°F-ft^2 (outside area)
V_t	Water velocity inside tubes, ft/sec
W_t	Water side flow rate, lb/hr
X	Wall thickness $\frac{d_r-d_i}{24}$, feet
x	Number of tubes in bundle
Y	Mean fin thickness, feet
λ	Latent heat of condensing, Btu/lb