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USE OF FINNED TUBES IN CONDENSING BUTYL HEADS AND ISOPROPYL ALCOHOL

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SUMMARY

This study was carried out to investigate the applicability of low-fin 19-fins-per-inch tubes in exchangers designed for the condensation of butyl heads and isopropyl alcohol.

Six plain-tube and six finned-tube condensers were designed to establish a basis for comparison of the economics of plain-tube and finned-tube units. The results of the study are summarized in Table I. The results indicate that the condensation of butyl heads and isopropyl alcohol are good applications for finned tubes. Trufin tube No. 196049-01 is specifically recommended for this purpose. For the specific heat duties and flow rates under consideration the finned-tube units cost about 30 percent less than units utilizing plain tubes of the same nominal size. The more compact finned-tube condensers are identified as designs 1 and 3 (Tables IV and V) for butyl heads and designs 7 and 9 (Tables IV and VI) for isopropyl alcohol.

This report presents in detail a recommended procedure for the design of condensers utilizing finned tubes.

TABLE I

SUMMARY OF DESIGN AND COST CALCULATIONS

Design No.	Shell-Side Fluid	Tube	Overall Coefficient U_o, Btu/hr-°F-ft^2 outside area	Length of Tubing, ft	Cost, Dollars
1	butyl heads	7/8-inch OD finned copper	134	1150	2590
2	butyl heads	7/8-inch OD plain copper	180	2320	3650
3	butyl heads	7/8-inch OD finned copper	70	2208	3432
4	butyl heads	7/8-inch OD plain copper	100	3980	4635
5	butyl heads	3/4-inch OD finned admiralty	70	3162	4005
6	butyl heads	3/4-inch OD plain admiralty	100	4640	5467
7	isopropyl alcohol	7/8-inch OD finned copper	125	1582	3236
8	isopropyl alcohol	7/8-inch OD plain copper	149	3400	4730
9	isopropyl alcohol	7/8-inch OD finned copper	80	2472	3760
10	isopropyl alcohol	7/8-inch OD plain copper	100	5100	5540
11	isopropyl alcohol	3/4-inch OD finned admiralty	80	3520	4277
12	isopropyl alcohol	3/4-inch OD plain admiralty	100	5920	6495

USE OF FINNED TUBES IN CONDENSING BUTYL HEADS AND ISOPROPYL ALCOHOL

INTRODUCTION

Finned tubes have been successfully used in a wide variety of industrial vapor-condensation applications. It has been suggested that finned tubes could be used to advantage in the condensation of isopropyl alcohol and butyl heads in petrochemical processes. The investigation of the possible usage of finned tubes in such applications requires a comparison of the economics involved in plain- and finned-tube units designed to handle specific heat duty requirements. This report presents design procedures, sample calculations, and specific recommendations for plain-tube and finned-tube exchangers for specified conditions of isopropyl-alcohol and butyl-heads condensation.

PROCESS REQUIREMENTS

The processes require the design of a shell-and-tube condenser to handle 27,000 lb per hour of isopropyl alcohol coming from an evaporator and an overhead shell-and-tube condenser to handle 18,670 lb per hour of butyl heads coming from a fractionating column. Table II gives the condenser requirements in detail for the two applications.

DESIGN PROCEDURE

The condensation heat duty, Q(Btu/hr), is related to the shell side heat transfer area A(sq ft) and to the mean overall temperature difference driving force ΔT(°F) by the following relationship:

$$Q = U_o A \Delta T \quad , \qquad (1)$$

where U_o is the overall heat transfer coefficient in Btu/hr-°F-sq ft outside area and is defined as follows:

$$\frac{1}{U_o} = \frac{1}{h_o} + r_o + \frac{X_f}{k_m}\frac{A_o}{A_m} + r_i\frac{A_o}{A_i} + \frac{A_o}{h_i A_i} \qquad (2)$$

where all other symbols used in this and subsequent equations are defined in the nomenclature table at the end of this report. The evaluation of each of the terms in equation (2) is carried out in detail for the condensation of butyl heads and is summarized for isopropyl alcohol in Appendices A and D.

TABLE II

CONDENSER REQUIREMENTS

	(1)	(2)
Service of Unit	Isopropyl alcohol	Butyl heads
Heat duty, Btu/hr	8,680,000	6,560,000
Log mean ΔT, °F	75	72
Shell side:		
vapor	98% IPOH	Butyl heads
quantity, lb/hr	27,000	18,670
latent heat of vaporization, Btu/lb	320	352
inlet temperature, °F	180	184
outlet temperature, °F	165	160
allowable ΔP, psi	0.5	0.5
condenser pressure, psi	atmospheric	atmospheric
Tube side:		
fluid	water	water
quantity, lb/hr	348,000	263,000
inlet temperature, °F	85	85
outlet temperature, °F	110	110
allowable ΔP, psi	10	10
Design pressure, psi:		
shell side	25	25
tube side	200	200
Design temperature, °F:		
shell side	250	250
tube side	160	160

The condensing coefficients were computed from a modified form of Nusselt's equation for finned and plain tubes. A modified form of Nusselt's equation which gives the condensing coefficient for the average tube in a multitube condenser is:

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

where for plain tubes D represents the outside diameter (D_o) of the tube; for finned tubes this length dimension is the equivalent diameter (D_{eq}) which varies with the fin efficiency and condensing coefficient, and C_N is a condensate correction factor which brings the thepretical relationship of Nusselt into aggreement with experimental results.

Figure 4 presents C_N as a function of the average number of tubes in a vertical row (N) for condensing Freon-12. Nusselt's theory, which depends upon laminar flow of the condensate, depreciates the performance of the single tube by a factor $(1/N)^{1/4}$ to obtain the average coefficient for N tubes in a vertical row. Experimental work has shown that this depreciation factor is too severe, evidently because the flow of the accumulated condensate on the tubes is not laminar. It should be realized that the use of C_N in equation (3) will therefore result in the design of a more compact condenser as compared to a unit designed on the basis of the Nusselt equation.

The formulas suggested for finding the value of N for exchangers having more than twenty tubes are:[1]

$$N = 0.815\ x^{0.52}, \text{ for square pitch,} \qquad (4)$$

$$N = 0.40\ x^{0.54}, \text{ for triangular pitch.} \qquad (5)$$

The latent heat of vaporization of the vapor is the value at the condensing temperature. The other physical properties identified by subscript f are obtained at the condensate film temperature, T_f, which is defined as follows:

$$T_f = T_{sv} - 1/2\ \Delta t_{cf} \ . \qquad (6)$$

Figures 2 and 3 were prepared for butyl heads and isopropyl alcohol respectively to facilitate the computation of the theoretical overall coefficients by multiple trial-and-error solutions.

Inside water-film coefficients were calculated by use of the following relationship given by McAdams[2]:

$$h_w = 150\ (1 + 0.011\ t_w)\ \frac{V_t^{0.8}}{d_i^{0.2}} \ . \qquad (7)$$

Figure 5 presents fin efficiency as a function of the outside film coefficient h_o', the outside fouling factor r_o, the finned-tube dimension, and the thermal conductivity of the metal.[3] To facilitate the successive-approximation computation required for determining condensing film coefficients, Fig. 1 was prepared for Trufin tube No. 196049-01. This figure was obtained by plotting a series of values of $(1/D_{eq})^{1/4}$ as a function of the outside film coefficient h_o. The difference between h_o' and h_o is that h_o' is based on the equivalent outside area and h_o is based on the outside area. The equivalent

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[1] Katz, D. L., and Williams, R. B., Oil and Gas Journal, July 28, 1949.

[2] McAdams, W. H., Heat Transmission 2nd edition, McGraw-Hill Book Company, 1942, p.183.

[3] Gardner, K. A., Trans. A.S.M.E., Vol 67 No. 8, p. 625 (1945).

outside area is a function of the fin efficiency as indicated by equation (12) and h_o is related to h_o' as indicated by equation (13).

In general, the value of h_o in equation (2) may be computed by either of two methods, i.e., from equation (3) or by first determining h_o' using equation (8) and then applying equation (13).

$$h_o' = 0.725\, C_N \left(\frac{k_f^3\, \rho_f^2\, g_c\, \lambda}{\mu_f\, \Delta t_{cf}\, N\, D_{eq}'} \right)^{1/4} \quad (8)$$

If equation (8) is used in the trial-and-error calculation of U_{of}, the following outlined procedure may be used. This procedure parallels that used in this study which used equation (3). The calculations based on equation (3) are given in section 2 of Appendix A.

The following is an outline of procedure using equation (8):

(1) Assume h_o'
(2) for $r_o = 0.0005$, find $\left(\frac{1}{h_o'} + r_o\right)$
(3) determine e_f from Fig.5
(4) calculate A_e from equation (12)
(5) calculate $\left(\frac{1}{D_{eq}'}\right)^{1/4}$ from the following equation:

$$\left(\frac{1}{D_{eq}'}\right)^{1/4} = 1.3\, e_f \frac{A_f}{A_e} \left(\frac{1}{L}\right)^{1/4} + \frac{A_r}{A_e} \left(\frac{1}{D_r}\right)^{1/4} \quad (9)$$

(6) calculate h_o from equation (13)
(7) calculate $R_t = \frac{1}{U_{of}}$ from h_o and the other resistance terms determined in equation (2)
(8) find Δt_{cf} from

$$\Delta t_{cf} = \left(\frac{\frac{1}{h_o}}{R_t}\right)\left(\Delta T_{LM}\right) \quad (10)$$

(9) calculate h_o' from equation (8), where N and C_N are determined from the assumed U_{of} and the tube-side arrangement. If the calculated and the assumed values of h_o' check satisfactorily, the corresponding value of h_o is correct.

For design work, the recommended procedure presented in detail in Appendix A is preferable to the above method because for a given tube and outside fouling factor a plot such as Fig. 1 enables one to reduce the various steps involved in the successive approximation of U_{of}.

The following sample calculation illustrated how Fig. 1 was prepared:

Assume h'_o = 500 Btu/hr-°F-sq ft of equivalent area
Mean fin thickness, y = 0.016 inch
Fin height = $\left(\frac{d_o - d_r}{2}\right) = \left(\frac{0.864 - 0.749}{2}\right)$ = 0.0575 inch
k_m = 220 Btu-ft/hr-°F-sq ft for copper
r_o = 0.0005 hr-°F-sq ft/Btu

$$\frac{d_o}{d_r} = \frac{0.864}{0.749} = 1.152$$

The abscissa for fin efficiency curves given in Fig. 5 is

$$H\sqrt{\frac{2}{\left(\frac{1}{h'_o} + r_o\right) k_m Y}} \quad . \tag{11}$$

Therefore, $\left(\frac{1}{h'_o} + r_o\right) = \frac{1}{500} + 0.0005 = 0.0025,$

and the abscissa $= \frac{0.0575}{12}\sqrt{\frac{(2)\ (12)}{(0.0025)\ (220)\ (0.016)}} = 0.250.$

The corresponding fin efficiency, e_f, as read from Fig. 5, is 0.977.

Of the total outside area 80 percent is on the extended surfaces and 20 percent is on the root of the finned tube.

Therefore, A_f = (0.588)(0.80) = 0.470 sq ft/ft,
and A_r = (0.588)(0.20) = 0.118 sq ft/ft.

The equivalent outside area for condensation is defined as

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

Therefore, A_e = (0.977)(0.470) + 0.118 = 0.577 sq ft/ft.

By definition, h_o is related to h'_o as follows:

$$h_o = h'_o \left(\frac{A_e}{A_o}\right) . \tag{13}$$

Therefore, $h_o = (500)\left(\frac{0.577}{0.588}\right) = 491.$

The value of $\left(\frac{1}{D_{eq}}\right)^{1/4}$ is obtained from the following relationship:

$$\left(\frac{1}{D_{eq}}\right)^{1/4} = 1.3\, e_f\, \frac{A_f}{A_o}\left(\frac{1}{L}\right)^{1/4} + \frac{A_r}{A_o}\left(\frac{1}{D_r}\right)^{1/4} , \tag{14}$$

where

$$L = \frac{a_f}{2D_o}$$

and

$$a_f = \frac{A_f}{(12)(N)} = \frac{(0.470)}{(12)(19)} = 0.00206 \ \frac{ft^2}{fin} .$$

$$L = \frac{(0.00206)(12)}{(2)(0.864)} = 0.0143 \text{ ft}$$

$$\left(\frac{1}{D_{eq}}\right)^{1/4} = (1.3)(0.977)\left(\frac{0.470}{0.588}\right)\left(\frac{1}{0.0143}\right)^{1/4} + \left(\frac{0.118}{0.588}\right)\left(\frac{12}{0.749}\right)^{1/4}$$

$$\left(\frac{1}{D_{eq}}\right)^{1/4} = 2.942 + 0.401 = 3.343 \left(\frac{1}{Ft}\right)^{1/4}$$

The computed value of 3.343 for $\left(\frac{1}{D_{eq}}\right)^{1/4}$ which corresponds to the outside condensing coefficient of h_o = 491 was plotted in Fig. 1 as the ordinate against h_o as the abscissa. This procedure was repeated and points were obtained to cover the range of h_o from 0 to 1000. Table III contains a summary of these points. It may be seen that once a curve has been established for a particular tube it has general application, and a series of curves for different sizes of finned tubes made from various metals may be conveniently prepared.

TABLE III

CALCULATED RESULTS OF h_o and $\left(\frac{1}{D_{eq}}\right)^{1/4}$ FOR

WOLVERINE TRUFIN TUBE NO. 196049-01

h_o'	$\frac{1}{h_o' + r_o}$	Abscissa in Fig.5	e_f	A_e	$\frac{A_e}{A_o}$	h_o	$\left(\frac{1}{D_{eq}}\right)^{1/4}$
25	24.7	0.0622	1.000	0.588	1.0	25	3.411
100	95.3	0.122	0.995	0.586	0.996	99.6	3.397
200	182	0.1688	0.990	0.583	0.991	198.2	3.381
300	261	0.202	0.988	0.582	0.990	297	3.376
400	333	0.228	0.982	0.579	0.984	393.5	3.360
500	400	0.250	0.977	0.577	0.981	491	3.343
650	490	0.277	0.974	0.575	0.978	635	3.331
1000	667	0.323	0.965	0.572	0.974	974	3.305

DESIGN OF THE BUTYL-HEADS CONDENSERS

Butyl heads at atmospheric pressure are condensed on the shell side of units using the following different types of tubes:

(a) Wolverine Trufin tube No. 196049-01
(b) Wolverine Trufin tube No. 195065-26
(c) 0.875 in. OD 16 BWG plain copper tube
(d) 0.750 in. OD 14 BWG plain admiralty tube.

Tables IV and V present the summary of the calculated results. All units are of conventional shell-and-tube condenser design; the low-fin tubes having 19 fins per inch were selected because these tubes have plain ends and can be rolled into tube sheets in the conventional manner, and give favorable performance in the condensation of organic vapors.

The composition of butyl heads was estimated from the given specific gravity and latent heat of vaporization data. It was assumed that the mixture contains essentially n-butyl and isobutyl alcohols in equal proportions and water. On this basis the composition of butyl heads was found to be 85.5 weight percent butyl alcohols and 14.5 weight percent water.

Theoretical overall heat-transfer coefficients were calculated for both the finned-tube and plain-tube units to establish the relative performance of the two tubes. Fouling factors recommended by TEMA were used both for the organic vapors (0.0005) and for plant water (0.001).

One set of design calculations based on the recommended procedure is presented in Appendices A and B and summarized in Table IV. A more conservative design is presented in Appendix C and the results are summarized in Table V. These designs are based on the application of the Nusselt equation (3) without the inclusion of the condensate drip factor C_N for the plain-tube units. The design coefficient for the conservative finned-tube units was obtained from the plain-tube overall coefficients by multiplying by $\bar{U}_{of}/U_{op}$ obtained by the recommended procedure. These designs are in accordance with the usual conservative coefficients employed in industrial designs and represent considerable available overcapacity for the four units so designed. Appendix A gives the overall coefficients used in both the recommended and more conservative designs.

DESIGN OF THE ISOPROPYL-ALCOHOL CONDENSERS

Isopropyl alcohol is condensed on the shell side of units using the same tubes specified for the butyl-heads condensation units.

Tables IV and VI give the summary of the calculated results and the shell and tube specifications for finned-tube and plain-tube condensers. The design calculations based on the recommended procedure are presented in Appendices D and E and are summarized in Table IV. The more conservative design is presented in Appendix F and the results are summarized in Table VI. Appendix D summarizes the overall coefficients calculated by the recommended procedure and the more conservative coefficients based on Nusselt's equation.

CONCLUSIONS AND RECOMMENDATIONS

Tables I, IV, V, and VI indicate that finned tubes can be used to definite advantage in the condensation of butyl heads and in the condensation of isopropyl alcohol.

The finned-tube units recommended on the basis of the procedure used in this report are given in Table IV. These designs (numbers 1 and 7) result in substantial economic savings. The finned-tube units recommended for isopropyl-alchohol and butyl-heads condensation cost 31 percent and 29 percent less than the plain-tube units having the same nominal size tubing.

Based on the more conservative design with an overall coefficient of 100 for the plain tube and the corresponding finned-tube coefficient with comparable performance, the recommended designs for finned-tube units are 5 in Table V and 9 in Table VI. The overall coefficients, U_O, for the finned tubes are 80 percent and 70 percent of the overall coefficients for the plain-tube units for isopropyl alcohol and butyl heads respectively. These coefficients can be expressed in terms of unit length of tube and are identified as U_L. On this basis, the overall coefficients per foot of tube length for the finned tube are 105 percent and 80 percent greater than the corresponding plain-tube coefficients for isopropyl alcohol and butyl heads. From Tables V and VI the finned-tube units recommended for isopropyl-alcohol and butyl-heads condensation cost 33 percent and 26 percent less than the plain-tube units having the same nominal size tubings.

The condensation of isopropyl alcohol and butyl heads are good applications for low-fin 19-fins-per-inch tubes.

TABLE IV

SUMMARY OF CONDENSER DESIGNS

Specifications	Service of Unit			
	Butyl Heads		Isopropyl Alcohol	
	1	2	7	8
1. Total heat duty, Btu/hr	6,560,000	6,560,000	8,680,000	8,680,000
2. Lb vapor per hour	18,670	18,670	27,000	27,000
3. Lb water per hour	263,000	263,000	348,000	348,000
4. Shell fluid and pressure, psig	Butyl heads, 0.5	Butyl heads, 0.5	Isopropyl alcohol, 0.5	Isopropyl alcohol, 0.5
5. Tube fluid and pressure, psig	Water, 70	Water, 70	Water, 70	Water, 70
6. Tube characteristics	Trufin No. 196049-01 7/8 in. OD Cu finned tube	7/8 in. OD 16 BWG plain copper tube	Trufin No. 196049-01 7/8 in. OD Cu finned tube	7/8 in. OD 16 BWG plain copper tube
7. Length of tube in bundle, ft	8	10	8	10
8. Number of tubes in bundle	144	232	198	340
9. Total length of tubing, ft	1150	2320	1582	3400
10. Total outside heat-transfer area, ft^2	677	531	931	780
11. No. of exchangers	1	1	1	1
12. Shell inside diameter, in.	18	23	21	27
13. Shell-side pressure drop, psi	0.5	0.5	0.5	0.5
14. Tube-side pressure drop, psi	5.1	9.6	4.8	8.0
15. Cross-sectional area for flow inside tubes, ft^2/pass	0.1662	0.1757	0.2284	0.2575
16. Shell-side passes	1	1	1	1
17. Tube-side passes	2	4	2	4
18. Shell thickness, in.	3/8	3/8	3/8	3/8
19. Shell and tube side nozzles, in. (total number required)	3(4)	4(4)	4(4)	6(4)
20. Tube arrangement	1-1/8 in. square pitch	1-1/8 in. square pitch	1-1/8 in. square pitch	1-1/8 in. square pitch
21. Overall heat-transfer coefficient, Btu/hr-°F-ft^2 outside area	134	180	125	149
22. Logarithmic mean temperature difference, °F	72.4	72.4	75	75
23. Excess heat-transfer area, percent	0	5.4	0.6	0.5
24. Water velocity, ft/sec	7.1	6.72	6.81	6.05
25. Estimated unit cost, dollars	2590	3650	3236	4730

TABLE V

SUMMARY OF BUTYL-HEADS CONDENSER DESIGNS

Specifications	Condenser Number			
	3	4	5	6
1. Total heat duty, Btu/hr	6,560,000	6,560,000	6,560,000	6,560,000
2. Lb butyl heads per hour	18,670	18,670	18,670	18,670
3. Lb water per hour	263,000	263,000	263,000	263,000
4. Shell fluid and pressure	Butyl heads, 0.50 psig	Butyl heads, 0.50 psig	Butyl heads, 0.50 psig	Butyl heads, 0.50 psig
5. Tube fluid and pressure	Water, 70 psig	Water, 70 psig	Water, 70 psig	Water, 70 psig
6. Tube characteristics	Trufin No. 196049-01 7/8 in. OD Cu finned tube	7/8 in. OD 16 BWG plain copper tube	Trufin No. 195065-26 3/4 in. OD admiralty finned tube	3/4 in. OD 14 BWG plain admiralty tube
7. Length of tube in bundle, ft	12	14	12	10
8. Number of tubes in bundle	184	284	264	464
9. Total length of tubing, ft	2208	3980	3162	4640
10. Total outside heat-transfer area, ft^2	1298	911	1298	911
11. Number of exchangers	1	1	1	1
12. Shell inside diameter, in.	20	25	21	29
13. Shell-side pressure drop, psi	0.5	0.5	0.5	0.5
14. Tube-side pressure drop, psi	4.0	8.0	6.2	8.0
15. Cross-sectional area for flow inside tubes, ft^2/pass	0.212	0.215	0.1872	0.216
16. Shell-side passes	1	1	1	1
17. Tube-side passes	2	4	2	4
18. Shell thickness, in.	3/8	3/8	3/8	3/8
19. Shell and tube side nozzles, in. (total number required)	4(4)	6(4)	4(4)	6(4)
20. Tube arrangement	1-1/8 in. square pitch	1-1/8 in. square pitch	1 in. square pitch	1 in. square pitch
21. Overall heat-transfer coefficient, Btu/hr-°F-ft2 outside area	70	100	70	100
22. Logarithmic mean temperature difference, °F	72.4	72.4	72.4	72.4
23. Excess heat-transfer area, percent	0	0.4	0	0.4
24. Water velocity, ft/sec	5.56	5.49	6.30	5.46
25. Estimated unit cost, dollars	3432	4635	4005	5467

Cost of Unit 6 using 3/4 in. OD 16 BWG plain admiralty tubes: 5207 dollars

TABLE VI

SUMMARY OF ISOPROPYL-ALCOHOL CONDENSER DESIGNS

Specifications	Condenser Number			
	9	10	11	12
1. Total heat duty, Btu/hr	8,680,000	8,680,000	8,680,000	8,680,000
2. Lb isopropyl alcohol per hour	27,000	27,000	27,000	27,000
3. Lb water per hour	348,000	348,000	348,000	348,000
4. Shell fluid and pressure	Isopropyl alcohol, 0.50 psig	Isopropyl alcohol, 0.50 psig	Isopropyl alcohol, 0.50 psig	Isopropyl alcohol, 0.50 psig
5. Tube fluid and pressure	Water, 70 psig	Water, 70 psig	Water, 70 psig	Water, 70 psig
6. Tube characteristics	Trufin No. 196049-01 7/8 in. OD Cu finned tube	7/8 in. OD 16 BWG plain copper tube	Trufin No. 195065-26 3/4 in. OD admiralty finned tube	3/4 in. OD 14 BWG plain admiralty tube
7. Length of tube in bundle, ft	12	14	12	10
8. Number of tubes in bundle	206	364	294	592
9. Total length of tubing, ft	2472	5100	3520	5920
10. Total outside heat-transfer area, ft^2	1452	1168	1446	1162
11. Number of exchangers	1	1	1	1
12. Shell inside diameter, in.	22	28	23	31
13. Shell-side pressure drop, psi	0.5	0.5	0.5	0.5
14. Tube-side pressure drop, psi	5.5	8.4	8.3	8.4
15. Cross-sectional area for flow inside tubes, ft^2/pass	0.238	0.2756	0.209	0.275
16. Shell-side passes	1	1	1	1
17. Tube-side passes	2	4	2	4
18. Shell thickness, in.	3/8	3/8	3/8	3/8
19. Shell and tube side nozzles, in. (total number required)	4(4)	6(4)	4(4)	8(4)
20. Tube arrangement	1-1/8 in. square pitch	1-1/8 in. square pitch	1 in. square pitch	1 in. square pitch
21. Overall heat-transfer coefficient, Btu/hr-°F-ft^2 outside area	80	100	80	100
22. Logarithmic mean temperature difference, °F	75	75	75	75
23. Excess heat-transfer area, percent	0.4	0.7	0	0.2
24. Water velocity, ft/sec	6.54	5.65	7.45	5.66
25. Estimated unit cost, dollars	3760	5540	4277	6495

Cost of Unit 12 using 3/4 in. OD 16 BWG plain admiralty tubes: 6160 dollars

APPENDICES

APPENDIX A

CALCULATION OF THEORETICAL OVERALL HEAT-TRANSFER COEFFICIENTS FOR THE CONDENSATION OF BUTYL HEADS ON 7/8 INCH OD FINNED AND PLAIN TUBES

1. Specifications

Average bulk shell-side temp., T_{av} = 172°F
Average bulk tube-side temp., t_w = 97.5°F
Log-mean temp. difference, ΔT_{LM} = 72.4°F
Heat duty, Q = 6,560,000 Btu/hr
Water flow rate = 263,000 lb/hr or 1.18 ft³/sec.
Allowable tube-side ΔP_t = 10 psi

2. Calculation of U_{of} for Wolverine Trufin 196049-01 tubes

The overall heat-transfer coefficient is evaluated by trial-and-error procedure. A value of U_{of} is assumed and a suitable tube layout prepared on the basis of the assumed U_{of}. The individual resistances are computed for the assumed condenser design and the U_{of} is determined. The correct U_{of} is obtained when the computed U_{of} checks with the assumed value.

The following fouling factors are used as recommended by TEMA:

$$r_o = 0.0005 \ \frac{\text{hr-°F-ft}^2}{\text{Btu}}$$

$$r_i = 0.001 \ \frac{\text{hr-°F-ft}^2}{\text{Btu}} \ .$$

Metal resistance, R_m (see tube characteristics for isopropyl-alcohol condenser design, Appendix F):

$$A_p = \frac{(3.14)(0.749)}{12} = 0.196 \text{ ft}^2/\text{ft}$$

$$A_m = \frac{A_p - A_i}{2.303 \log \frac{A_p}{A_i}}$$

$$A_m = \frac{0.196 - 0.171}{2.303 \log \frac{0.196}{0.171}} = 0.1833 \text{ ft}^2/\text{ft}$$

$$R_m = \frac{X_f \; A_o}{k_m \; A_m} = \frac{(0.049)(0.588)}{(12)(220)(0.1833)}$$

$$R_m = 0.00006 \; \frac{\text{hr-°F-ft}^2 \text{ outside}}{\text{Btu}}$$

Assume $U_{of} = 130 \; \frac{\text{Btu}}{\text{hr-°F-ft}^2 \text{ outside}}$

$$A = \frac{6,560,000}{(130)(72.4)} = 697 \text{ ft}^2$$

$$\text{Total tube length} = \frac{697}{0.588} = 1185 \text{ ft}$$

Using 8-ft-long tubes,
number of tubes in bundle, $x = \frac{1185}{8} = 148$

For two tube passes,

$$A_{cst} = \frac{(148)(0.00231)}{(2)} = 0.171 \text{ ft}^2$$

$$\text{Water velocity} = \frac{1.18}{0.171} = 6.9 \text{ ft/sec} .$$

Compared to the results for design No. 3 in Table IV, the tube-side pressure drop corresponding to this water velocity is less than 10 psi. For square-pitch tube arrangement,

$$N = 0.815 \; x^{0.52} \tag{4}$$

$$N = (0.815)(148)^{0.52} = 10.94.$$

The outside condensing coefficient for the average tube in a multi-tube condenser is given by

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

From Fig. 4, $C_N = 1.42$

Substituting,

$$h_o = \frac{(0.725)(1.42)}{(10.94)^{1/4}} \left(\frac{k_f^3 \; \rho_f^2 \; g_c \; \lambda}{\mu_f \; D_{eq} \; \Delta t_{cf}} \right)^{1/4}$$

$$h_o = 0.566 \left(\frac{k_f^3 \; \rho_f^2 \; g_c \; \lambda}{\mu_f \; D_{eq} \Delta t_{cf}} \right)^{1/4} \qquad (15)$$

From equation (7):

$$h_w = 150 \; (1 + 0.011 \times 97.5) \; \frac{(6.9)^{0.8}}{(0.651)^{0.2}}$$

$$h_w = 1588 \quad \text{Btu/hr-°F-ft}^2 \text{ inside}$$

$$\frac{A_o}{A_i \; h_w} = \frac{3.44}{1588} = 0.002164$$

$$\frac{A_o}{A_i} r_i = (3.44)(0.001) = 0.00344$$

$$R_i = 0.002164 + 0.00344 = 0.005604$$

$$\frac{1}{U_{of}} = R_t = R_o + R_m + R_i$$

where

$$R_o = \frac{1}{h_o} + r_o \; .$$

Therefore,

$$\left(R_t - \frac{1}{h_o} \right) = r_o + R_m + R_i$$

$$\left(R_t - \frac{1}{h_o} \right) = 0.0005 + 0.00006 + 0.005604$$

$$\left(R_t - \frac{1}{h_o} \right) = 0.006164 \; \frac{\text{hr-°F-ft}^2 \text{ outside}}{\text{Btu}}$$

The condensing coefficient is determined by trial and error by assuming h_o, finding R_t, Δt_{cf}, $(1/D_{eq})^{1/4}$, and calculating h_o from equation (15).

$$\text{Assume } h_o = 940, \; \frac{1}{h_o} = \frac{1}{940} = 0.001064$$

$$R_t = 0.006164 + 0.001064 = 0.007228$$

$$\Delta t_{cf} = \frac{\left(\frac{1}{h_o}\right)}{(R_t)} \Delta T_{LM} = \frac{(0.001064)}{(0.007228)} (72.4) = 10.66\text{°F}$$

$$(\Delta t_{cf})^{1/4} = (10.66)^{1/4} = 1.808 .$$

From Fig. 1

$$\left(\frac{1}{D_{eq}}\right)^{1/4} = 3.308 .$$

Condensate film temperature

$$T_f = T_{sv} - 1/2\ \Delta t_{cf} \tag{6}$$

$$T_f = 172 - 1/2\ (10.66) = 166.7°F .$$

From Fig. 2

$$\left(\frac{k_f^3\ \rho_f^2\ g_c\ \lambda}{\mu_f}\right)^{1/4} = 780 .$$

From equation (15)

$$h_o = (0.566)\ \frac{(780)(3.308)}{(1.808)} = 809 .$$

Since the calculated h_o(809) is considerably less than the assumed h_o(940), for the second trial select

$$h_o = 770, \frac{1}{h_o} = 0.00130$$

$$R_t = 0.006164 + 0.00130 = 0.007464$$

$$\Delta t_{cf} = \frac{(0.00130)(72.4)}{(0.007464)} = 12.60°F .$$

$$(\Delta t_{cf})^{1/4} = (12.60)^{1/4} = 1.884$$

$$\left(\frac{1}{D_{eq}}\right)^{1/4} \quad 3.320 \quad \text{(Fig. 1)}$$

$$\left(\frac{k_f^3\ \rho_f^2\ g_c\ \lambda}{\mu_f}\right)^{1/4} = 778 \quad \text{(Fig. 2)}$$

$$h_o = (0.566)\ \frac{(778)(3.320)}{(1.884)} = 775.$$

The agreement between assumed and calculated h_o is satisfactory.

Therefore,

$$U_{of} = \frac{1}{R_t} = \frac{1}{0.007464} = 134 \frac{\text{Btu}}{\text{hr-°F-ft}^2 \text{ outside}}$$

The calculated value agrees closely with the assumed value of 130.

3. Calculation of U_{op} for 7/8-inch OD 16 BWG Plain Copper Tubes

Metal resistance, R_m (see tube characteristics for isopropyl-alcohol condenser design, Appendix F):

$$A_m = \frac{0.229 - 0.195}{2.303 \log \frac{0.229}{0.195}} = 0.210 \text{ ft}^2/\text{ft}$$

$$R_m = \frac{(0.065)(0.229)}{(12)(220)(0.210)} = 0.000027 .$$

Assume

$$U_{op} = 175$$

$$A = \frac{6,560,000}{(175)(72.4)} = 519 \text{ ft}^2 .$$

$$\text{Total tube length} = \frac{519}{0.229} = 2260 \text{ ft} .$$

Using 10-ft long tubes, number of tubes in bundle $= \frac{2260}{10} = 226$.

For 228 tubes and four tube passes,

$$A_{cst} = \frac{(228)}{(4)} (0.003025) = 0.1726 \text{ ft}^2 .$$

$$\text{Water velocity} = \frac{1.18}{0.1726} = 6.85 \text{ ft/sec.}$$

The tube side pressure drop is computed for this unit:

$$\text{Mass velocity, } G_i = \frac{263,000}{0.1726} = 1,523,000 \frac{\text{lb}}{\text{hr-ft}^2}$$

$$Re = \frac{(0.745)(1,523,000)}{(12)(1.79)} = 52,900 .$$

From p. 836, Kern, Process Heat Transfer

$$f_t = 0.000174$$

$$\Delta P \text{ tubes} = \frac{f_t\, G_i^2 \text{ (tube length) } n_2}{(5.22 \times 10^{10})\, D_i\, S\, \mu_g} \text{ psi}$$

where

$$\mu_g = 1.0226$$

(See Appendix F)

$$S = 0.995$$

$$\Delta P\text{tubes} = \frac{(0.000174)(1.523 \times 10^6)^2 (10)(4)(12)}{(5.22 \times 10^{10})(0.745)(0.995)(1.0226)}$$

$$\Delta P \text{ tubes} = 4.90 \text{ psi}$$

$$\Delta P \text{ headers} = \frac{4(V_t^2)}{(S)(2g_c)}\, n_2$$

From Kern, p. 837

$$\frac{V_t^2}{2g_c} = 0.32$$

$$\Delta P \text{ headers} = \frac{(4)(0.32)(4)}{(0.995)} = 5.15 \text{ psi}$$

$$\Delta P_t = 4.90 + 5.06 = 9.96 \text{ or } 10.0 \text{ psi approx.}$$

This pressure drop agrees with the allowable.

For square-pitch tube arrangement,

$$N = 0.815(228)^{0.52} = 13.6 \text{ (by equation 4)}$$

and

$$C_N = 1.5 \text{ (Fig. 4)} .$$

Therefore,

$$h_o = (0.725)(1.5)\left(\frac{12}{0.875 \times 13.6}\right)^{1/4} \left(\frac{k_f^3\, \rho_f^2\, g_c\, \lambda}{\mu_f\, \Delta t_{cf}}\right)^{1/4}$$

$$h_o = 1.09 \left(\frac{k_f^3\, \rho_f^2\, g_c\, \lambda}{\mu_f\, \Delta t_{cf}}\right)^{1/4} \qquad (16)$$

$$h_w = 150\,(1 + 0.011 \times 97.5)\,\frac{(6.85)^{0.8}}{(0.745)^{0.2}} = 1538$$

$$\frac{A_o}{A_i\ h_w} = \frac{1.173}{1538} = 0.000764$$

$$\frac{A_o}{A_i} r_i = (1.173)(0.001) = 0.001173$$

$$R_i = 0.000764 + 0.001173 = 0.001937$$

$$\left(R_t - \frac{1}{h_o}\right) = 0.0005 + 0.00003 + 0.001937 = 0.002467 .$$

Assume $h_o = 315, \ \frac{1}{h_o} = \frac{1}{315} = 0.00318$

$$R_t = 0.002467 + 0.00318 = 0.00564$$

$$\Delta t_{cf} = \frac{(0.00318)}{(0.00564)} (72.4) = 40.8°F$$

$$(\Delta t_{cf})^{1/4} = (40.8)^{1/4} = 2.526$$

$$T_f = 172 - 1/2(40.8) = 151.6°F .$$

From Fig. 2,

$$\left(\frac{k_f^3\ \rho_f^2\ g_c\ \lambda}{\mu_f}\right)^{1/4} = 750 .$$

From equation (16),

$$h_o = (1.09)\frac{(750)}{(2.526)} = 323 .$$

A value of h_o = 324 would be the correct one on the basis of the second trial.

$$h_o = 324, \ \frac{1}{h_o} = \frac{1}{324} = 0.00309$$

$$R_t = 0.002467 + 0.00309 = 0.005557$$

$$U_{op} = \frac{1}{0.005557} = 180 \ \frac{\text{Btu}}{\text{hr-°F-ft}^2 \text{ outside area}}$$

This value of U_{op} is in close agreement with the assumed value of 175. A more conservative design is obtained by using a value of 100 for U_{op} which corresponds to a condensing coefficient based on the Nusselt equation (3) without the recommended condensate drip correction factor of C_N. The theoretically calculated overall coefficients based on the recommended procedure outlined in sections 2 and 3 of this appendix are used for design calculations in Appendix B, and the results are summarized in Table IV. In addition, the overall coefficients calculated for plain-and finned-tube units are used to find the relative performance U_{of}/U_{op} and this ratio is used to obtain the overall design coefficient for finned-tube units on the same basis as the more conservative value of 100 for the plain-tube units.

Thus,

$$\frac{U_{of}}{U_{op}} = \frac{134}{180} = 0.745 .$$

An average value of this ratio for butyl heads is computed in section 4 of this appendix. A set of condenser designs based on these conservative overall coefficients is shown in Appendix C for butyl heads and in Appendix F for isopropyl alcohol. The results are summarized in Tables V and VI for butyl heads and isopropyl alcohol respectively.

4. Short-cut Method for Approximating U_{of} and U_{op}

The following method substitutes for the $C_N/N^{1/4}$ factor in equation (3) a constant factor of 1/1.1 or 0.91. Thus, the overall coefficients U_{of} and U_{op} can be estimated for a desirable water velocity without specifying the condenser tube arrangement.

For Wolverine Trufin 196049-01:

$$h_o = \frac{0.725}{1.1}\left(\frac{k_f^3\ \rho_f^2\ g_c\ \lambda}{\mu_f\ D_{eq}\ \Delta t_{cf}}\right)^{1/4}$$

$$h_o = 0.66\left(\frac{k_f^3\ \rho_f^2\ g_c\ \lambda}{\mu_f\ D_{eq}\ \Delta t_{cf}}\right)^{1/4} . \qquad (17)$$

For a water velocity V_t = 6 ft/sec,

$$h_w = 150\ (1 + 0.011 \times 97.5)\ \frac{(6)^{0.8}}{(0.651)^{0.2}} = 1418$$

$$\frac{A_o}{A_i\ h_w} = \frac{3.44}{1418} = 0.00243$$

$$R_t - \frac{1}{h_o} = 0.0005 + 0.00006 + 0.00243 + 0.00344 = 0.00643 \ .$$

Assume

$$h_o = 940, \ \frac{1}{h_o} = \frac{1}{940} = 0.001064$$

$$R_t = 0.00643 + 0.001064 = 0.007494$$

$$\Delta t_{cf} = \frac{(0.001064)}{(0.007494)} (72.4) = 10.28°F$$

$$(\Delta t_{cf})^{1/4} = (10.28)^{1/4} = 1.791 \ .$$

From Fig. 1,

$$\left(\frac{1}{D_{eq}}\right)^{1/4} = 3.308$$

$$T_f = 172 - 1/2(10.28) = 166.9°F \ .$$

From Fig. 2,

$$\left(\frac{k_f^3 \ \rho_f^2 \ g_c \ \lambda}{\mu_f}\right)^{1/4} = 780.5 \ .$$

From equation (17),

$$h_o = (0.66) \frac{(780.5)(3.308)}{(1.791)} = 952 \ .$$

The calculated h_o checks closely the assumed h_o.

$$U_{of} = \frac{1}{0.007494} = 133.6 \ .$$

For 7/8-inch OD 16 BWG plain copper tube:

$$h_o = \frac{(0.725)}{(1.1)} \left(\frac{12}{0.875}\right)^{1/4} \left(\frac{k_f^3 \ \rho_f^2 \ g_c \ \lambda}{\mu_f \ \ \Delta t_{cf}}\right)^{1/4}$$

$$h_o = 1.27 \left(\frac{k_f^3 \ \rho_f^2 \ g_c \ \lambda}{\mu_f \ \Delta t_{cf}}\right)^{1/4} \qquad (18)$$

For a water velocity $V_t = 6$ ft/sec,

$$h_w = 150\ (1 + 0.011 \times 97.5)\ \frac{(6)^{0.8}}{(0.745)^{0.2}} = 1380$$

$$\frac{A_o}{A_i\ h_w} = \frac{1.173}{1380} = 0.00085$$

$$\left(R_t - \frac{1}{h_o}\right) = 0.0005 + 0.00003 + 0.00085 + 0.001173 = 0.00255.$$

Assume $h_o = 380,\ \frac{1}{h_o} = \frac{1}{380} = 0.002635$.

$$R_t = 0.00255 + 0.002635 = 0.005185$$

$$\Delta t_{cf} = \frac{(0.002635)}{(0.005185)}\ (72.4) = 36.8°F$$

$$(\Delta t_{cf})^{1/4} = (36.8)^{1/4} = 2.462$$

$$T_f = 172 - 1/2(36.8) = 153.6°F\ .$$

From Fig. 2,

$$\left(\frac{k_f^3\ \rho_f^2\ g_c\ \lambda}{\mu_f}\right)^{1/4} = 754\ .$$

From equation (18),

$$h_o = \frac{(1.27)(754)}{(2.462)} = 388\ .$$

A value of $h_o = 390$ would be the correct one on the basis of the second trial.

$$h_o = 390,\ \frac{1}{h_o} = \frac{1}{390} = 0.002564$$

$$R_t = 0.00255 + 0.002564 = 0.005114$$

$$U_{op} = \frac{1}{0.005114} = 195.6$$

The corresponding ratio U_{of}/U_{op} obtained by this method is

$$\frac{U_{of}}{U_{op}} = \frac{133.6}{195.6} = 0.683\ .$$

The average value of this ratio from the two methods given in sections 2, 3, and 4 is conservative and has the following value:

$$\frac{U_{of}}{U_{op}} = \frac{0.745 + 0.683}{2} = 0.714.$$

Thus, for the design of the butyl-heads condensers the design overall coefficient for the finned-tube units is conservatively taken as 70 percent of that of the plain-tube units.

Hence,

$$U_{op} = 100 \text{ Btu/hr-°F-ft}^2 \text{ outside}$$

$$U_{of} = (100)(0.70) = 70 \text{ Btu/hr-°F-ft}^2 \text{ outside} .$$

Corresponding values of the overall coefficient per foot of tube length are:

Trufin No. 196049-01,

$$U_{Lf} = U_o A_o = (70)(0.588) = 41.1 .$$

For 7/8-inch OD 16 BWG plain tube,

$$U_{Lp} = (100)(0.229) = 22.9$$

$$\frac{U_{Lf}}{U_{Lp}} = \frac{41.1}{22.9} = 1.795 .$$

APPENDIX B

DESIGN CALCULATIONS FOR BUTYL-HEADS CONDENSERS
BASED ON CALCULATED OVERALL COEFFICIENTS

Design No. 1. Design of Condenser with Trufin No. 196049-01 Tubes

1. Tube Specifications

See isopropyl-alcohol condenser design (Appendix F).

2. Tube Arrangement and Tube-side ΔP_t

From Appendix A, section 2 for Trufin No. 196049-01 tubes,
at V_t = 6.9 ft/sec
U_{of} = 134 Btu/hr-°F-ft^2 outside area
Required heat-transfer area:

$$A = \frac{(650,000)}{(72.4)(134)} = 677 \text{ ft}^2$$

$$\text{Required tube length} = \frac{(677)}{(0.588)} = 1150 \text{ ft}$$

Length per tube = 8 ft

$$\text{Number of tubes in bundle} = \frac{1150}{8} = 144$$

$$\text{For two tube passes, } A_{cst} = \left(\frac{144}{2}\right)(0.00231) = 0.1662 \text{ ft}^2$$

$$\text{Water velocity} = \frac{1.18}{0.1662} = 7.1 \text{ ft/sec}$$

$$\text{Mass velocity} = \frac{263,000}{0.1662} = 1,580,000 \frac{\text{lb}}{\text{hr-ft}^2}$$

$$Re = \frac{(0.651)(1,580,000)}{(12)(1.79)} = 47,900, \quad f_t = 0.00018$$

$$\Delta P = \frac{(0.00018)(1.58 \times 10^6)^2(8)(2)(12)}{(5.22 \times 10^{10})(0.995)(0.651)(1.0623)} = 2.4 \text{ psi}$$

$$\Delta P \text{ header losses} = \frac{(4)(0.34)(2)}{0.995} = 2.74 \text{ psi}$$

$\Delta P_t = 2.4 + 2.74 = 5.1$ psi

For two tube passes, 144 tubes placed on 1-1/8-inch-square pitch require a shell with an inside diameter of 18 inches.

Excess heat-transfer area = none

Design No. 2. Design of condenser with 0.875-inch OD 16 BWG Plain Copper Tube

1. Tube Specifications

See isopropyl-alcohol condenser design (Appendix F).

2. Tube Arrangement and Tube-side ΔP_t

From Appendix A, section 3 for 7/8-inch OD plain tubes

at $V_t = 6.85$ ft/sec

$U_{op} = 180$ Btu/hr-°F-ft² outside area

$$A = \frac{(6,560,000)}{(72.4)(180)} = 504 \text{ ft}^2$$

$$\text{Required tube length} = \frac{504}{0.229} = 2200 \text{ ft}$$

Length of tubes = 10 ft

$$\text{Number of tubes in bundle} = \frac{2200}{10} = 220$$

Use 232 tubes to meet the allowable ΔP_t

Actual A = (232)(10)(0.229) = 531 ft²

For four tube passes,

$$V_t = \frac{(4)(1.18)}{(232)(0.003025)} = 6.72 \text{ ft/sec}$$

$$\text{Mass velocity} = \frac{(263,000)}{0.1757} = 1,496,000 \frac{\text{lb}}{\text{hr-ft}^2}$$

$$Re = \frac{(0.745)(1,496,000)}{(12)(1.79)} = 52,000 \quad f_t = 0.000175$$

$$\Delta P = \frac{(0.000175)(1.496 \times 10^6)^2(10)(4)(12)}{(5.22 \times 10^{10})(0.995)(0.745)(1.0226)} = 4.75 \text{ psi}$$

$$\Delta P \text{ header losses} = \frac{(4)(0.30)(4)}{(0.995)} = 4.82 \text{ psi}$$

Total ΔP_t = 4.75 + 4.82 = 9.57 or 9.6 psi

Shell inside diameter = 23 inches

$$\text{Excess heat-transfer area} = \left(\frac{531-504}{504}\right)(100) = 5.4 \text{ percent}$$

Cost Estimation of Units

The estimated unit costs are based on the same conditions as those for the isopropyl-alcohol units (Appendix F).

APPENDIX C

DESIGN CALCULATIONS FOR BUTYL-HEADS CONDENSERS

For the computation of condensing and overall coefficients, the composition of the butyl heads was assumed to be n-butyl alcohol, isobutyl alcohol, and water. The two butyl alcohols were assumed to be present in equal proportions. On this basis, two equations, one utilizing additive densities and the other additive heats of vaporization, were solved simultaneously to give the following butyl-heads composition:

Component	Wt. Fraction
Butyl alcohol	0.855
Water	0.145
Total	1.000

The design overall coefficients are taken as follows (see section 4, Appendix A):

$$U_{op} = 100 \text{ Btu/hr-°F-ft}^2 \text{ outside}$$

$$U_{of} = (0.70)(100) = 70 \text{ Btu/hr-°F-ft}^2 \text{ outside}$$

Two sets of calculations were made for the following tube applications:

Design No. 3. Wolverine Trufin No. 196049-01
0.875-in. 16 BWG plain-end copper tube

Design No. 4. 0.875-in. OD 16 BWG plain copper tube

Design No. 5. Wolverine Trufin No. 195065-26
0.750-in. 14 BWG plain-end admiralty tube

Design No. 6. 0.750-in. OD 14 BWG plain admiralty tube

Design No. 3. Design of Condenser with Trufin No. 196049-01 Tubes

1. Tube Specifications

See isopropyl-alcohol condenser design (Appendix F).

2. Tube Arrangement and Tube-side ΔP_t

Allowable tube-side pressure drop = 10 psi
Allowable shell-side pressure drop = 0.5 psi

$$\Delta T_{LM} = \frac{(160-85)-(184-110)}{2.303 \log \frac{75}{74}} = 72.4°F$$

Heat duty, Q = (18,670)(352) = 6,560,000 Btu/hr

Average water temperature $= \frac{85 + 110}{2} = 97.5°F$

$$\mu = 1.79 \text{ lb/ft-hr}$$
$$\rho = 62.0 \text{ lb/ft}^3$$

Water flow rate = 263,000 lb/hr or $\frac{(263,000)}{(62.0)(3600)} = 1.18 \text{ ft}^3\text{/sec}$

Required heat-transfer area,

$$A = \frac{(6,560,000)}{(72.4)(70)} = 1298 \text{ ft}^2$$

Required tube length $= \frac{(1298)}{(0.588)} = 2208$ ft

Length per tube = 12 ft

Number of tubes in bundle $= \frac{2208}{12} = 184$ tubes

For two tube passes, $A_{cst} = \frac{(184)}{(2)}(0.00231) = 0.212 \text{ ft}^2$

Water velocity $= \frac{1.18}{0.212} = 5.56$ ft/sec

Mass velocity $= \frac{(263,000)}{(0.212)} = 1,240,000 \frac{\text{lb}}{\text{hr-ft}^2}$

$$Re = \frac{(0.651)(1,240,000)}{(12)(1.79)} = 37,600$$

$f_t = 0.00019$

$$\Delta P = \frac{(0.00019)(1.24 \times 10^6)^2(12)(2)(12)}{(5.22 \times 10^{10})(0.995)(0.651)(1.0623)} = 2.34 \text{ psi}$$

ΔP header losses $= \frac{(4)(0.21)(2)}{(0.995)} = 1.69$ psi

Total tube-side ΔP_t = 2.34 + 1.69 = 4.03 or 4.0 psi

For two tube passes, 184 tubes placed on 1-1/8-inch-square pitch require a shell with an inside diameter of 20 inches.

Excess heat-transfer area = 0 percent.

Design No. 4. Design of Condenser with 0.875 in. OD 16 BWG Plain Copper Tube

1. Tube Specifications

See isopropyl-alcohol condenser design (Appendix F).

2. Tube Arrangement and Tube-side ΔP_t

Required heat-transfer area,

$$A = \frac{(6,560,000)}{(72.4)(100)} = 907 \text{ ft}^2$$

$$\text{Required tube length} = \frac{(907)}{(0.229)} = 3960 \text{ ft}$$

Length per tube = 14 ft

$$\text{Number of tubes in bundle} = \frac{3960}{14} = 283$$

Use 284 tubes, four tube passes

$$\text{Number of tubes per pass} = \frac{284}{4} = 71$$

Total tube length = (284)(14) = 3980 ft

A = (3980)(0.229) = 911 ft²

A_{cst} = (71)(0.003025) = 0.215 ft²

$$\text{Water velocity} = \frac{1.18}{0.215} = 5.49 \text{ ft/sec}$$

$$\text{Mass velocity} = \frac{(263,000)}{(0.215)} = 1,222,000 \ \frac{\text{lb}}{\text{hr-ft}^2}$$

$$Re = \frac{(0.745)(1,222,000)}{(12)(1.79)} = 42,400$$

f_t = 0.000185

$$\Delta P = \frac{(0.000185)(1.222 \times 10^6)^2(14)(4)(12)}{(5.22 \times 10^{10})(0.995)(.745)(1.0226)} = 4.66 \text{ psi}$$

$$\Delta P \text{ header losses} = \frac{(4)(0.21)(4)}{(0.995)} = 3.38$$

Total ΔP_t = 4.66 + 3.38 = 8.04 or 8.0 psi

Shell inside diameter = 25 inches

Excess heat-transfer area $= \frac{(911-907)}{(907)} (100) = 0.44$ percent

Design No. 5. Design of Condenser with Trufin No. 195065-26 Tubes

1. Tube Specifications

See isopropyl-alcohol condenser design (Appendix F).

2. Tube Arrangement and Tube-side ΔP_t

Required A = 1298 ft^2

Required tube length $= \frac{1298}{0.410} = 3162$ ft

Length per tube = 12 ft

Number of tubes in bundle $= \frac{3162}{12} = 264$

For two tube passes, $A_{cst} = \frac{(264)}{(2)} (0.00142) = 0.1872$ ft^2

Water velocity $= \frac{1.18}{0.1872} = 6.3$ ft/sec

Mass velocity $= \frac{(263,000)}{(0.1872)} = 1,402,000 \ \frac{\text{lb}}{\text{hr-ft}^2}$

$$Re = \frac{(0.510)(1,402,000)}{(12)(1.79)} = 33,300$$

$f_t = 0.000198$

$$\Delta P = \frac{(0.000198)(1.402 \times 10^6)^2(12)(2)(12)}{(5.22 \times 10^{10})(0.995)(0.510)(1.0623)} = 3.98 \text{ psi}$$

$$\Delta P \text{ header losses} = \frac{(4)(0.272)(2)}{(0.995)} = 2.18 \text{ psi}$$

Total ΔP_t = 3.98 + 2.18 = 6.16 or 6.2 psi

For two tube passes, 264 tubes placed on 1-inch-square pitch require a shell with an inside diameter of 21 inches.

Excess heat transfer area = none

Design No. 6. Design of Condenser with 0.750-in. OD 14 BWG Plain Admiralty Tube

1. Tube Specifications

   See isopropyl-alcohol condenser design (Appendix F).

2. Tube Arrangement and Tube-side ΔP_t

   Required heat-transfer area, A = 907 ft^2

$$\text{Required tube length} = \frac{907}{0.1963} = 4610 \text{ ft}$$

Length per tube = 10 ft

$$\text{Number of tubes in bundle} = \frac{4610}{10} = 461$$

Use 464 tubes, four tube passes

$$\text{Number of tubes per pass} = \frac{464}{4} = 116$$

Total tube length = (464)(10) = 4640

$A = (4640)(0.1963) = 911 \text{ ft}^2$

$A_{cst} = (116)(0.00186) = 0.216 \text{ ft}^2$

$$\text{Water velocity} = \frac{1.18}{0.216} = 5.46 \text{ ft/sec}$$

$$\text{Mass velocity} = \frac{(263{,}000)}{(0.216)} = 1{,}218{,}000 \ \frac{\text{lb}}{\text{hr-ft}^2}$$

$$Re = \frac{(0.584)(1{,}218{,}000)}{(12)(1.79)} = 33{,}100$$

$f_t = 0.0002$

$$\Delta P = \frac{(0.0002)(1.218 \times 10^6)^2(10)(4)(12)}{(5.22 \times 10^{10})(0.995)(0.584)(1.0226)} = 4.58 \text{ psi}$$

$$\Delta P \text{ header losses} = \frac{(4)(0.21)(4)}{(0.995)} = 3.38 \text{ psi}$$

Total ΔP_t = 7.96 or 8.0 psi

For four tube passes, 464 tubes placed on 1-inch-square pitch require a shell with an inside diameter of 29 inches.

$$\text{Excess heat-transfer area} = \left(\frac{911-907}{907}\right)(100) = 0.44 \text{ percent.}$$

Cost Estimation of Units

The estimated unit costs are based on the same conditions as those for isopropyl-alcohol units.

APPENDIX D

CALCULATION OF THEORETICAL OVERALL HEAT-TRANSFER COEFFICIENTS FOR THE CONDENSATION OF ISOPROPYL ALCOHOL ON 7/8-INCH OD FINNED AND PLAIN TUBES

A set of calculations similar to those for the butyl heads were completed to determine the ratio U_{of}/U_{op} from which the design U_{of} could be determined. The following are the values obtained by the two methods outlined previously.

Method	V_t	U_{of}	V_t	U_{op}	U_{of}/U_{op}
$C_N/N^{1/4}$	6.78	125.2	5.85	149	0.84
Short-cut method $\left(\frac{1}{1.1}\right)$	6	126.8	6	166.8	0.76

$$\text{average } \frac{U_{of}}{U_{op}} = \frac{0.84 + 0.76}{2} = 0.80$$

Hence,

$$U_{op} = 100 \text{ Btu/hr-°F-ft}^2 \text{ outside}$$

$$U_{of} = (100)(0.80) = 80 \text{ Btu/hr-°F-ft}^2 \text{ outside}$$

Corresponding values of the overall coefficient per foot of tube length are:

Trufin No. 196049-01,

$$U_{Lf} = U_oA_o = (80)(0.588) = 47.0$$

For 7/8-inch OD 16 BWG plain tube,

$$U_{Lp} = (100)(0.229) = 22.9$$

$$\frac{U_{Lf}}{U_{Lp}} = \frac{47.0}{22.9} = 2.05$$

APPENDIX E

DESIGN CALCULATIONS FOR ISOPROPYL-ALCOHOL CONDENSERS BASED ON CALCULATED OVERALL COEFFICIENTS

Design No. 7. Design of Condenser with Trufin No. 196049-01 Tubes

1. Tube Specifications

See isopropyl-alcohol condenser design (Appendix F).

2. Tube Arrangement and Tube-side ΔP_t

From Appendix D, for Trufin No. 196049-01 tubes

at V_t = 6.78 ft/sec

U_{of} = 125.2 or 125 Btu/hr-°F-ft² outside area

$$A = \frac{8{,}680{,}000}{(75)(125)} = 925 \text{ ft}^2$$

$$\text{Required tube length} = \frac{925}{0.588} = 1572 \text{ ft}$$

Length per tube = 8 ft

$$\text{Number of tubes in bundle} = \frac{1572}{8} = 196.5$$

Use 198 tubes,

actual total tube length = (198)(8) = 1582 ft

actual A = (1582)(0.588) = 931 ft²

For two tube passes

A_{cst} = (99)(0.00231) = 0.2284 ft²

$$V_t = \frac{(348{,}000)}{(3600)(62.0)(0.2284)} = \frac{(1.56)}{(0.2284)} = 6.81 \text{ ft/sec}$$

$$\text{Mass velocity} = \frac{348{,}000}{0.2284} = 1{,}520{,}000 \ \frac{\text{lb}}{\text{hr-ft}^2}$$

$$Re = \frac{(0.651)(1{,}520{,}000)}{(12)(1.79)} = 46{,}000, f_t = 0.00018$$

$$\Delta P = \frac{(0.00018)(1.52 \times 10^6)^2(8)(2)(12)}{(5.22 \times 10^{10})(0.995)(0.651)(1.0623)} = 2.21 \text{ psi}$$

$$\Delta P \text{ header losses} = \frac{(4)(0.32)(2)}{(0.995)} = 2.57 \text{ psi}$$

$\Delta P_t = 2.21 + 2.57 = 4.78$ or 4.8 psi

Shell inside diameter = 21 inches

$$\text{Excess heat-transfer area} = \frac{(931-925)}{(925)}(100) = 0.65 \text{ percent}$$

Design No. 8. Design of Condenser with 0.875-inch OD 16 BWG Plain Copper Tubes

1. Tube Specifications

See isopropyl-alcohol condenser design (Appendix F).

2. Tube Arrangement and Tube-side ΔP_t

From Appendix D, for 7/8-inch OD plain tubes

at $V_t = 5.85$ ft/sec

$U_{op} = 149$ Btu/hr-°F-ft² outside area

$$A = \frac{8,680,000}{(75)(149)} = 776 \text{ ft}^2$$

$$\text{Required tube length} = \frac{776}{0.229} = 3384 \text{ ft}$$

Length per tube = 10 ft

Using 3400 ft of tubing

A actual = (3400)(0.229) = 780 ft²

$$\text{Number of tubes in bundle} = \frac{3400}{10} = 340$$

For four tube passes,

$$A_{cst} = \left(\frac{340}{4}\right)(0.003025) = 0.2575 \text{ ft}^2$$

$$V_t = \frac{1.56}{0.2575} = 6.05 \text{ ft/sec}$$

$$\text{Mass velocity} = \frac{348,000}{0.2575} = 1,350,000 \frac{\text{lb}}{\text{hr-ft}^2}$$

$$Re = \frac{(0.745)(1,350,000)}{(12)(1.79)} = 46,800, \quad f_t = 0.00018$$

$$\Delta P = \frac{(0.00018)(1.35 \times 10^6)^2 (10)(4)(12)}{(5.22 \times 10^{10})(0.995)(0.745)(1.0226)} = 3.97 \text{ psi}$$

$$\Delta P \text{ header losses} = \frac{(4)(0.25)(4)}{(0.995)} = 4.02 \text{ psi}$$

$$\Delta P_t = 3.97 + 4.02 = 7.99 \text{ or } 8.0 \text{ psi}$$

Shell inside diameter = 27 inches

$$\text{Excess heat-transfer area} = \frac{(780-776)}{(776)} (100) = 0.52 \text{ percent}$$

Cost Estimation of Units

The estimated unit costs are based on the same conditions as those for isopropyl-alcohol units (Appendix F).

APPENDIX F

DESIGN CALCULATIONS FOR ISOPROPYL-ALCOHOL CONDENSERS

The overall design coefficients are taken as follows (from Appendix D):

U_{op} = 100 Btu/hr-°F-ft² outside

U_{of} = (0.80)(100) = 80 Btu/hr-°F-ft² outside

Two sets of calculations were made for the following tube applications:

Design No. 9. Wolverine Trufin No. 196049-01
0.875-in. OD 16 BWG plain end copper tube

Design No. 10. 0.875-in. OD 16 BWG plain copper tube

Design No. 11. Wolverine Trufin No. 195065-26
0.750-in. 14 BWG plain-end admiralty tube

Design No. 12. 0.750-in. OD 14 BWG plain admiralty tube

Design No. 9. Design of Condenser with Trufin No. 196049-01 Tube

1. Tube Specifications

Trufin No. 196049-01 19 fins/in.

d_o = 0.864 in.

d_r = 0.749 in.

Wall thickness = 0.049 in.

d_i = 0.749 - (2)(0.049) = 0.651 in.

A_o = 0.588 ft2/ft

A_i = 0.171 ft2/ft

A_o/A_i = 3.44

$$A_{cs} = \frac{\pi(0.651)^2}{(576)} = 0.00231 \text{ ft}^2\text{/tube}$$

2. Tube Arrangement and Tube-side ΔP_t

Allowable tube-side pressure drop = 10 psi

Allowable shell-side pressure drop = 0.5 psi

$$\Delta T_{LM} = \frac{(165-85) - (180-110)}{2.303 \log \frac{80}{70}} = 75°F$$

$$\text{Average water temperature} = \frac{110 + 85}{2} = 97.5°F$$

At 97.5°F, μ = 0.74 cp. or 1.79 $\frac{lb}{ft\text{-}hr}$ (Kern, Process Heat Transfer p. 823

ρ = 62.0 lb/ft³

From previous calculations about 60 percent of the overall temperature drop occurs between the bulk water and the inside tube wall. Therefore average inside tube wall temperature,

$$T_m = 97.5 + (0.60)(75) = 97.5 + 45 = 142.5°F$$

μ_w at tube wall temperature = 0.48 cp. (Kern, p.823)

$$\mu_g = \left(\frac{\mu}{\mu_w}\right)^{0.14} = \left(\frac{0.74}{0.48}\right)^{0.14} = 1.0623$$

$$\text{Heat duty, } Q = (27{,}000)(320) = 8.68 \times 10^6 \ \frac{Btu}{hr}$$

$$\text{Required heat-transfer area, } A = \frac{(8.68 \times 10^6)}{(75)(80)} = 1446 \ ft^2$$

$$\text{Required tube length} = \frac{(1446)}{(0.588)} = 2460 \ ft$$

Length per tube = 12 ft

$$\text{Number of tubes in bundle} = \frac{2460}{12} = 205$$

Use 206 tubes, two tubes passes

$$\text{Number of tubes per pass} = \frac{206}{2} = 103$$

Total flow cross-sectional area, A_{cst} = (103)(0.00231) = 0.238 ft²

Water flow rate, W_t = 348,000 lb/hr

$$\text{Mass velocity, } G_i = \frac{(348{,}000)}{(0.238)} = 1{,}460{,}000 \ \frac{lb}{hr\text{-}ft^2}$$

$$\text{Water velocity} = \frac{1{,}460{,}000}{(3600)(62.0)} = 6.54 \ ft/sec$$

$$Re = \frac{D_i G_i}{\mu} = \frac{(0.651)(1{,}460{,}000)}{(12)(1.79)} = 44{,}300$$

f_t = 0.00183 (p. 836, Kern)

$$\Delta P \text{ inside tubes} = \frac{f_t\, G_i^2\, n_2 \text{ (tube length)}}{(5.22 \times 10^{10})\, D_i\, S\, \mu_g}$$

$$= \frac{(0.000183)(1.46 \times 10^6)^2(12)(2)(12)}{(5.22 \times 10^{10})(0.995)(0.651)(1.0623)} = 3.12 \text{ psi}$$

where specific gravity, $S = \frac{62.0}{62.4} = 0.995$

$$\Delta P \text{ tube-side return pressure loss} = \frac{4\, V_t^2}{2g_c S}\, n^2$$

$$= \frac{(4)(0.29)(2)}{(0.995)} = 2.33 \text{ psi}$$

(Kern, p. 837)

Total tube-side pressure drop, ΔP_t = 3.12 + 2.33 = 5.45 or 5.5 psi The pressure drop is satisfactory.

For two tube passes, 176 Trufin No. 196049-01 tubes placed on 1-1/8-inch-square pitch require a shell with an inside diameter of 22 inches (Kern, p. 841).

actual A = (206)(12)(0.588) = 1452 ft^2

$$\text{Excess heat-transfer area} = \frac{1452-1446}{1446}(100) = 0.42 \text{ percent}$$

Design No. 10. Design of Condenser with 0.875-in. OD 16 BWG Plain Copper Tube

1. Tube Specifications

d_o = 0.875 in.
Wall thickness = 0.065 in.
d_i = 0.875-(2)(0.065) = 0.745 in.

$$A_o = \frac{(\pi)(0.875)}{(12)} = 0.229 \text{ ft}^2/\text{ft}$$

$$A_i = \frac{(\pi)(0.745)}{(12)} = 0.195 \text{ ft}^2/\text{ft}$$

$$A_o/A_i = \frac{0.229}{0.195} = 1.173$$

$$A_{cs} = \frac{(\pi)(0.745)^2}{(576)} = 0.003025 \text{ ft}^2/\text{tube}$$

2. Tube Arrangement and Tube-side ΔP_t

Allowable tube-side pressure drop = 10 psi.

From previous calculations about 21 percent of the overall temperature drop occurs between the bulk water and the inside tube wall.

Average inside tube wall temperature,

$T_m = 97.5 + (0.21)(75) = 113.2°F$

μ_w at 113.2°F = 0.63 cp.

$$\mu_g = \left(\frac{\mu}{\mu_w}\right)^{0.14} = \left(\frac{0.74}{0.63}\right)^{0.14} = 1.0226$$

Required heat-transfer area, A = 1160 ft^2

$$\text{Required tube length} = \frac{1160}{0.229} = 5060 \text{ ft}$$

Length per tube = 14 ft

$$\text{Number of tubes in bundle} = \frac{5060}{14} = 362$$

Use 364 tubes, 4 tube passes

$$\text{Number of tubes per pass} = \frac{364}{4} = 91$$

Total flow cross-sectional area,

$$A_{cst} = (91)(0.003025) = 0.2756 \text{ ft}^2$$

$$\text{Mass velocity } G_i = \frac{(348,000)}{(0.2756)} = 1,262,000 \ \frac{\text{lb}}{\text{hr-ft}^2}$$

$$\text{Water velocity} = \frac{(1,262,000)}{(3600)(62.0)} = 5.65 \text{ ft/sec}$$

$$Re = \frac{D_i G_i}{\mu} = \frac{(0.745)(1,262,000)}{(12)(1.79)} = 43,800$$

f_t = 0.000183 (Kern, p. 836)

$$\Delta P \text{ inside tubes} = \frac{(0.000183)(1.262 \times 10^6)^2(14)(4)(12)}{(5.22 \times 10^{10})(0.995)(0.745)(1.0226)}$$

$$= 4.95 \text{ psi}$$

ΔP tube-side return pressure loss =

$$\frac{(4)(0.215)(4)}{(0.995)} = 3.46 \text{ psi}$$

Total tube-side pressure drop,

ΔP_t = 4.95 + 3.46 = 8.41 or 8.4 psi.

The pressure drop is satisfactory.

For four tube passes, 364 tubes placed on 1-1/8-inch-square pitch require a shell with an inside diameter of 28 inches.

$$\text{Actual A} = (364)(14)(0.229) = 1168 \text{ ft}^2$$

$$\text{Excess area} = \frac{(1168-1160)}{(1160)} (100) = 0.69 \text{ percent}$$

Design No. 11. Design of Condenser with Trufin No. 195065-26 Tubes

1. Tube Specifications

Trufin No. 195065-26 19 fins/in.

d_o = 0.737 in.
d_r = 0.640 in.
Wall thickness = 0.065 in.
d_i = 0.640 - (2)(0.065) = 0.510 in.
A_o = 0.410 ft2/ft
A_i = 0.134 ft²/ft
A_o/A_i = 3.06

$$A_{cs} = \frac{(\pi)(0.510)^2}{(576)} = 0.00142 \text{ ft}^2$$

2. Tube Arrangement and Tube-side ΔP_t

Allowable tube-side pressure drop = 10 psi

$$\text{Required tube length} = \frac{(1446)}{(0.410)} = 3520 \text{ ft}$$

Length per tube = 12 ft

$$\text{Number of tubes in bundle} = \frac{(3520)}{(12)} = 294$$

Use 294 tubes, two tube passes
Number of tubes per pass = $\frac{294}{2}$ = 147
Total flow cross-sectional area,

$$A_{cst} = (147)(0.00142) = 0.209 \text{ ft}^2$$

$$\text{Mass velocity } G_i = \frac{(348,000)}{(0.209)} = 1,662,000 \ \frac{\text{lb}}{\text{hr-ft}^2}$$

$$\text{Water velocity} = \frac{(1,662,000)}{(3600)(62.0)} = 7.45 \text{ ft/sec}$$

$$Re = \frac{(0.510)(1,662,000)}{(12)(1.79)} = 39,500$$

$$f_t = 0.000187$$

$$\Delta P \text{ inside tubes} = \frac{(0.000187)(1.662 \times 10^6)^2(12)(2)(12)}{(5.22 \times 10^{10})(0.995)(0.510)(1.0623)} = 5.27 \text{ psi}$$

$$\Delta P \text{ tube-side return pressure loss} = \frac{(4)(0.38)(2)}{0.995} = 3.05 \text{ psi}$$

Total tube-side pressure drop,

$$\Delta P_t = 5.27 + 3.05 = 8.32 \text{ or } 8.3 \text{ psi}$$

The pressure drop is satisfactory.

For two tube passes, 294 Trufin No. 195(65-26 tubes placed on 1-inch-square pitch require a shell with an inside diameter of 23 inches.

Excess heat-transfer area = none

Design No. 12. Design of Condenser with 0.750-inch OD 14 BWG Plain Admiralty Tubes

1. Tube Specifications

d_o = 0.750 in.
Wall thickness = 0.083 in.
d_i = 0.584 in.
A_o = 0.1963 ft²/ft
A_i = 0.1529 ft²/ft

$$A_o/A_i = \frac{0.1963}{0.1529} = 1.285$$

A_{cs} = 0.268 in.² or 0.00186 ft²

2. Tube Arrangement and Tube-side ΔP

Allowable tube-side pressure drop = 10 psi

$$\text{Required tube length} = \frac{(1160)}{(0.1968)} = 5900 \text{ ft}$$

Length per tube = 10 ft

$$\text{Number of tubes in bundle} = \frac{5900}{10} = 590$$

Use 592 tubes, four tube passes

$$\text{Number of tubes per pass} = \frac{592}{4} = 148$$

Total flow cross-sectional area,

$$A_{cst} = (148)(0.00186) = 0.275 \text{ ft}^2$$

$$\text{Mass velocity } G_i = \frac{348{,}000}{0.275} = 1{,}265{,}000 \ \frac{\text{lb}}{\text{hr-ft}^2}$$

$$\text{Water velocity} = \frac{(1{,}265{,}000)}{(3600)(62.0)} = 5.66 \text{ ft/sec}$$

$$Re = \frac{(0.584)(1,265,000)}{(12)(1.79)} = 34,400$$

$$f_t = 0.000197$$

$$\Delta P \text{ inside tubes} = \frac{(0.000197)(1.265 \times 10^6)^2(10)(4)(12)}{(5.22 \times 10^{10})(0.995)(0.584)(1.0226)}$$

$$= 4.87 \text{ psi}$$

$$\Delta P \text{ tube-side return pressure loss} = \frac{(4)(0.22)(4)}{(0.995)} = 3.54 \text{ psi}$$

Total tube-side pressure drop,

$$\Delta P_t = 4.87 + 3.54 = 8.41 \text{ or } 8.4 \text{ psi}$$

The pressure drop is satisfactory.

For four tube passes, 592 tubes placed on 1-inch-square pitch require a shell with an inside diameter of 31 inches.

$$\text{Actual A} = (592)(10)(0.1963) = 1162 \text{ ft}^2$$

$$\text{Excess area} = \frac{(1162-1160)}{(1160)}(100) = 0.17 \text{ percent}$$

Cost Estimation of Units

The estimated unit costs are based on the following conditions:

1. Condenser costs except the tube are based on September, 1950, values corrected to 1954 by cost indices of Marshall and Stevens.
2. Cost of finned tubes is based on March, 1954, price list.
3. Cost of plain tubes is based on June, 1953, price list, corrected to March, 1954.

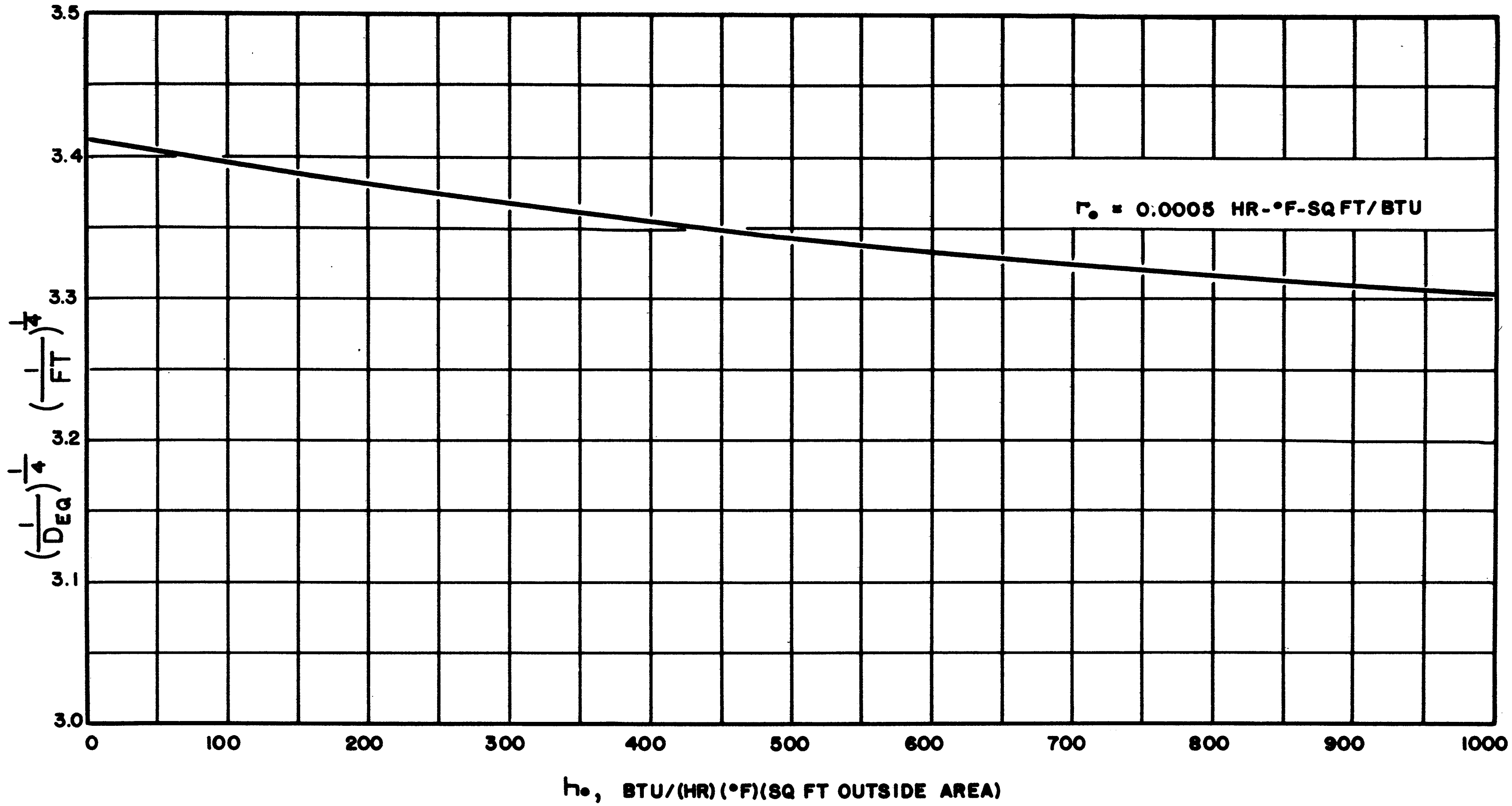

FIGURE 1 VARIATION OF $(1/D_{EQ})^{1/4}$ WITH OUTSIDE FILM COEFFICIENT FOR WOLVERINE TRUFIN 196049-01

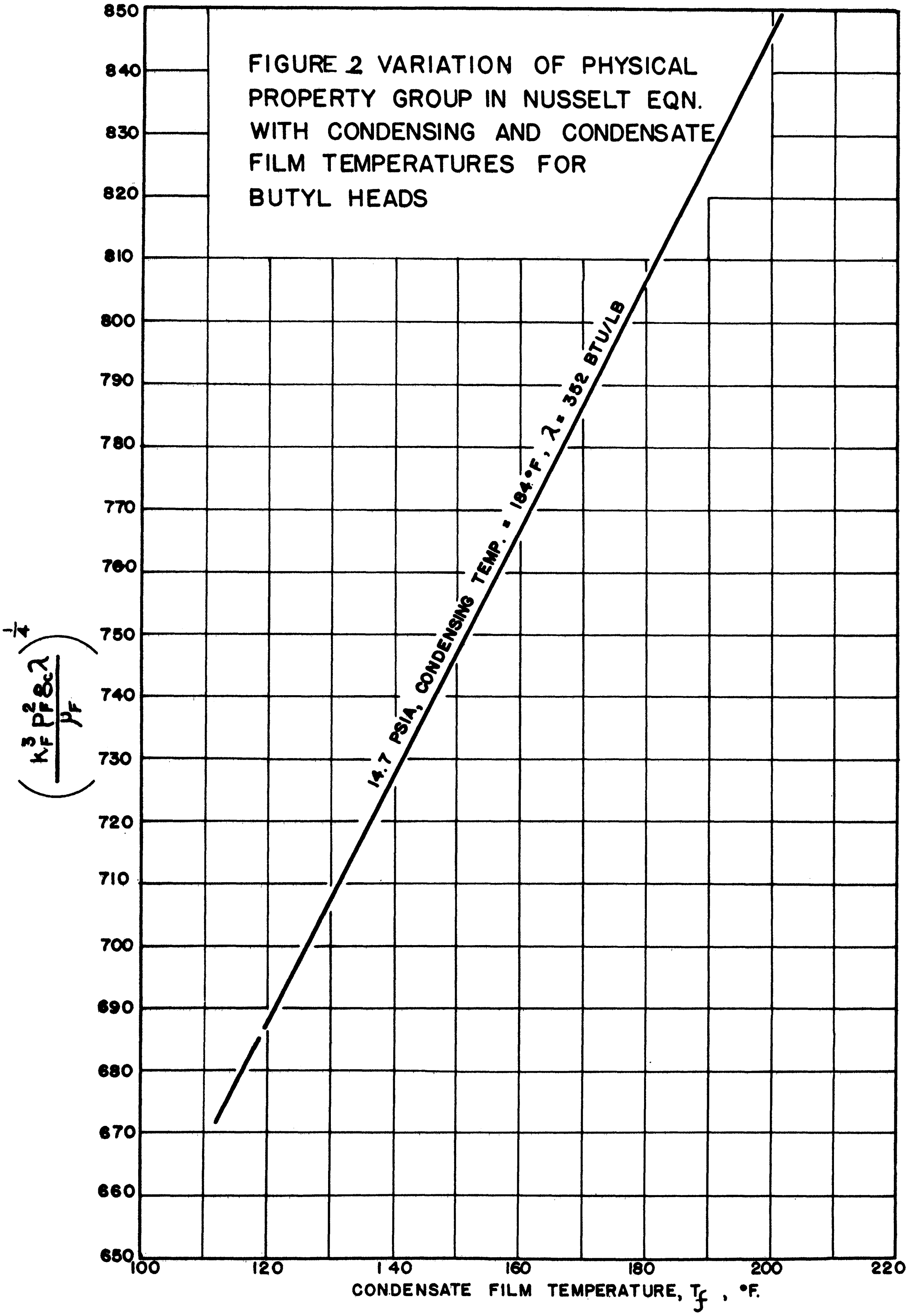

FIGURE 2 VARIATION OF PHYSICAL PROPERTY GROUP IN NUSSELT EQN. WITH CONDENSING AND CONDENSATE FILM TEMPERATURES FOR BUTYL HEADS

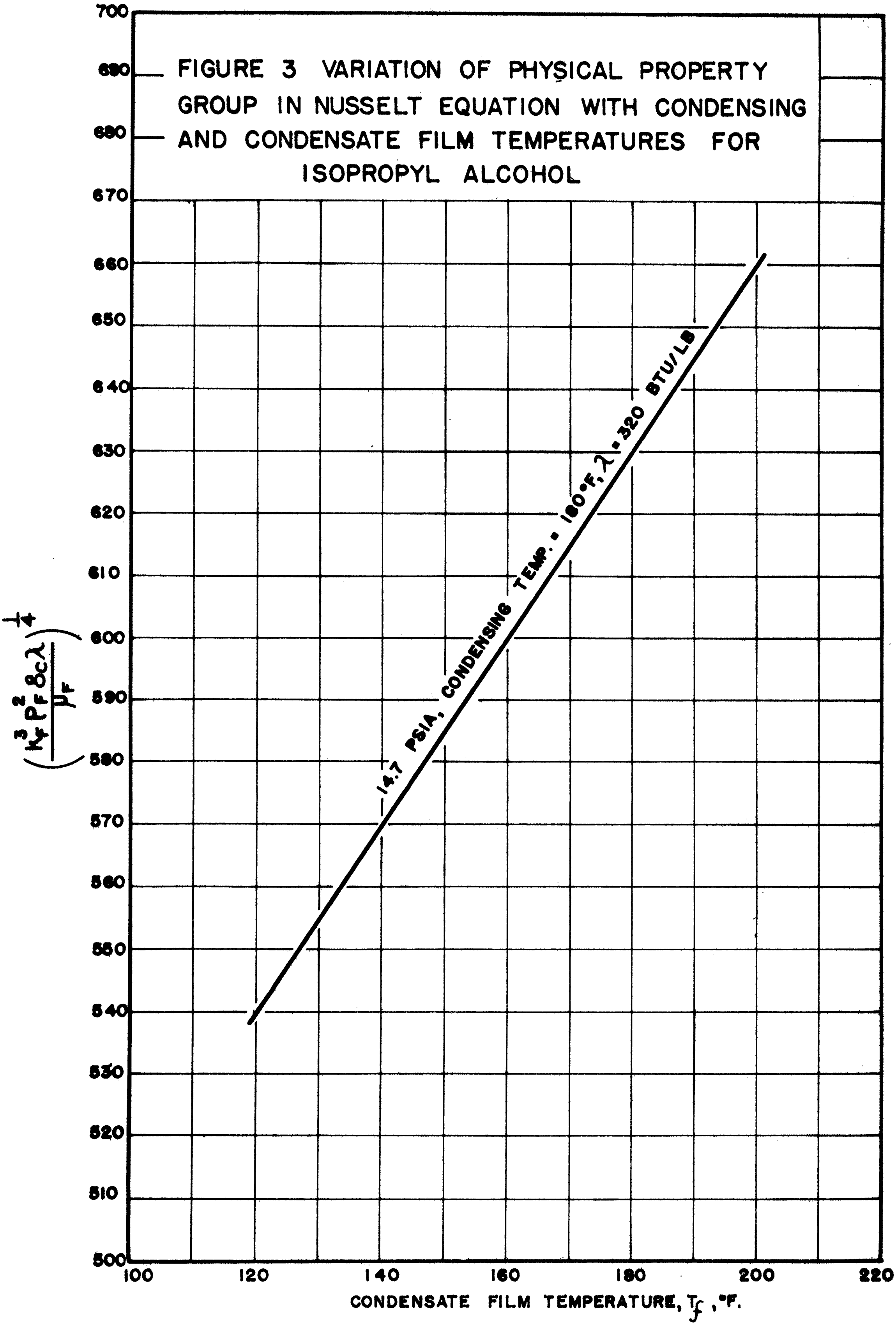

FIGURE 3 VARIATION OF PHYSICAL PROPERTY GROUP IN NUSSELT EQUATION WITH CONDENSING AND CONDENSATE FILM TEMPERATURES FOR ISOPROPYL ALCOHOL

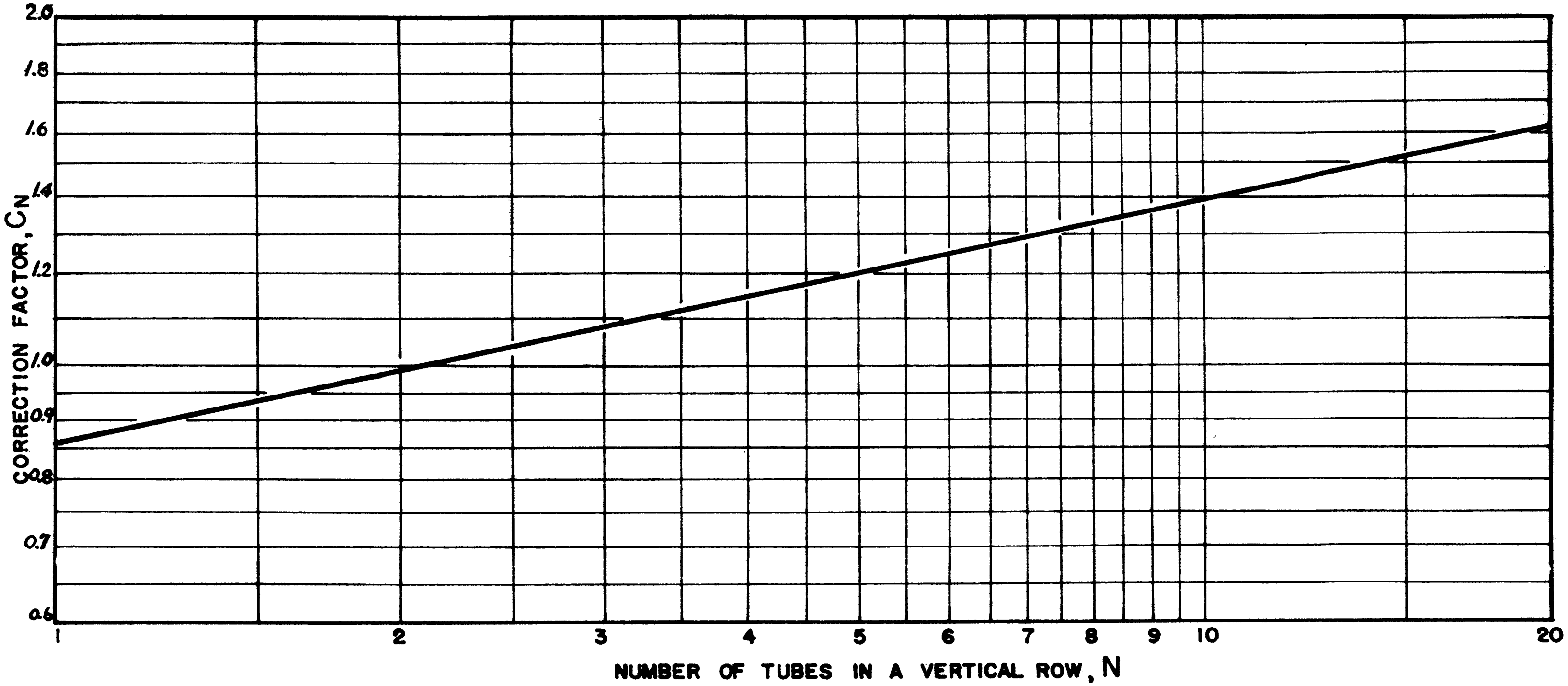

FIGURE 4 RATIO OF EXPERIMENTAL TO THEORETICAL CONDENSATION COEFFICIENTS FOR FREON-12

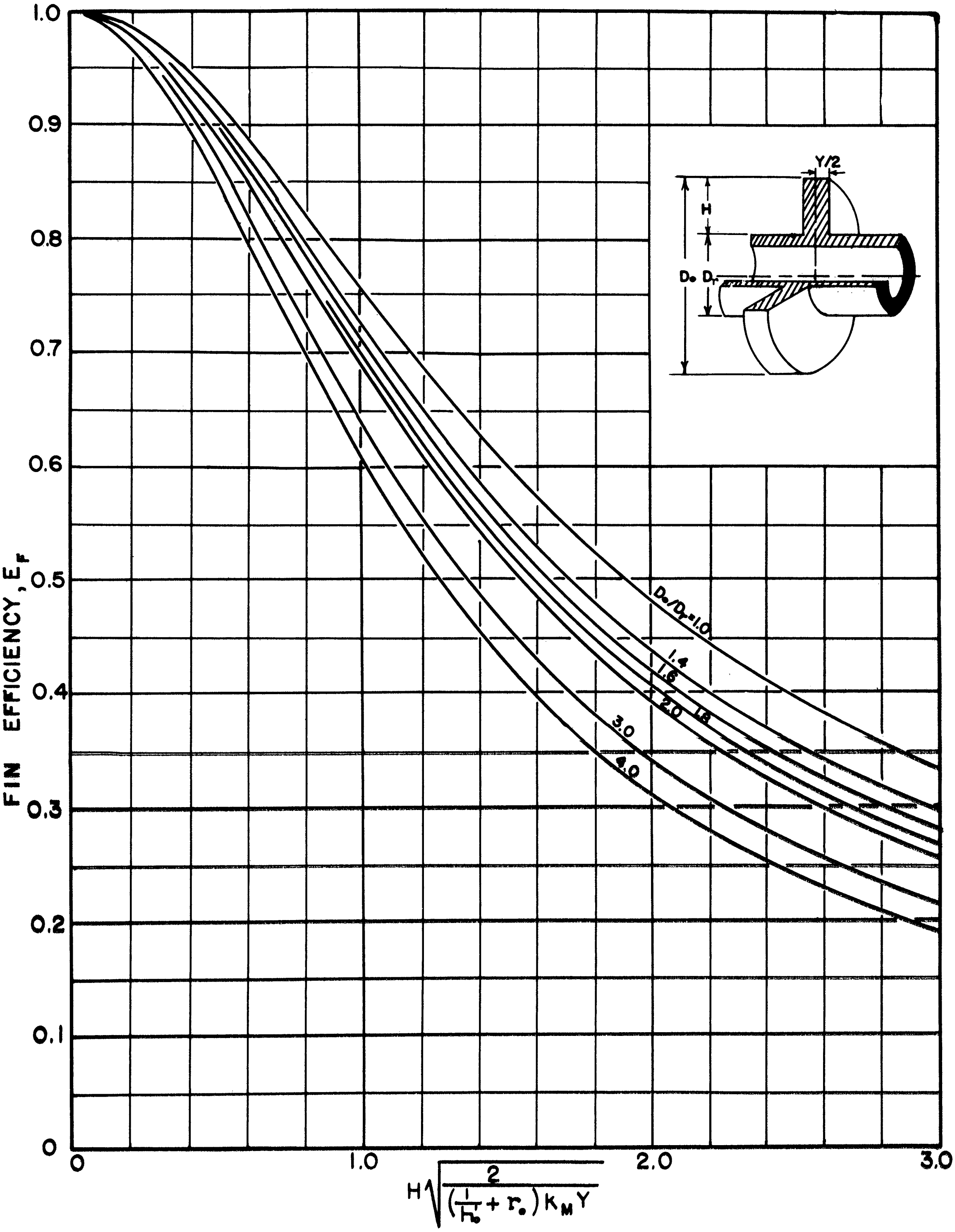

FIGURE 5 EFFICIENCY OF ANNULAR FINS OF CONSTANT THICKNESS

NOMENCLATURE

A	Total outside tube area in heat exchanger, ft^2
A_{cs}	Inside cross-sectional flow area of tube, ft^2
A_{cst}	Total inside cross-sectional flow area of tubes, per pass, ft^2
A_e	Equivalent outside area, ft^2/ft, $A_e = e_f A_f + A_r$
A_f	Outside finned-tube fin area ft^2/ft
a_f	Area of one fin (both sides), ft^2
A_i	Inside tube area, ft^2/ft of tube length
A_m	Logarithmic mean metal area between D_i and D_r, ft^2/ft
A_o	Outside tube-surface area, ft^2/ft of tube length
A_o/A_i	Ratio of outside to inside surface areas
A_p	Outside surface area of plain tube with same outside diameter as root diameter of finned tube, ft^2/ft length of tube
A_r	Outside finned-tube root area, ft^2/ft
C_N	Correction factor for condensing coefficient with N number of tubes in a vertical row
D_{eq}	Equivalent outside diameter for calculating condensing coefficients defined as follows: $$\left(\frac{1}{D_{eq}}\right)^{1/4} = 1.3\, e_f \frac{A_f}{A_o} \left(\frac{1}{L}\right)^{1/4} + \frac{A_r}{A_o} \left(\frac{1}{D_r}\right)^{1/4}$$
D'_{eq}	Equivalent outside diameter for calculating condensing coefficient based on outside equivalent area, ft $$\left(\frac{1}{D'_{eq}}\right)^{1/4} = 1.3\, e_f \frac{A_f}{A_e} \left(\frac{1}{L}\right)^{1/4} + \frac{A_r}{A_e} \left(\frac{1}{D_r}\right)^{1/4}$$
D_i	Inside tube diameter, ft
d_i	Inside tube diameter, in.
D_o	Diameter over the fins, ft
d_o	Diameter over the fins, in.
D_r	Finned-tube root diameter, ft
d_r	Finned-tube root diameter, in.
e_f	Fin efficiency factor, decimal equivalent
f_t	Friction factor, tube side
$---_f$	Subscript f attached to density, viscosity, and thermal conductivity of condensate represents fluid properties at the film temperature of condensing fluid, T_f
g_c	Gravitational constant, 4.17×10^8 ft/hr^2 or 32.2 ft/sec^2
G_i	Mass velocity inside tubes, $lb/(hr)(ft^2)$
H	Fin height, ft
h_o	Outside film coefficient corrected to base of fin, $Btu/(hr)(°F)(ft^2)$
h'_o	Outside film coefficient, $Btu/(hr)(°F)(ft^2)$ based on equivalent outside area
h_w	Inside film coefficient for water, $Btu/(hr)(°F)(ft^2)$
k	Thermal conductivity, $(Btu)(ft)/(hr)(°F)(ft^2)$
k_m	Thermal conductivity of tube wall, $(Btu)(ft)/(hr)(°F)(ft^2)$
L	Mean effective fin height, ft, $L = a_f/2D_o$

Nomenclature (continued)

N	Number of fins per inch, or average number of tubes in a vertical row
n_2	Number of tube passes
ΔP_t	Total pressure drop through tube side, psi
Q	Total heat load, Btu/hr
R_i	Total inside resistance, $(1/h_i + r_i)(A_o/A_i)$
r_i	Inside fouling resistance, (hr)(°F)(ft²)/Btu
R_m	Tube metal resistance, (hr)(°F)(ft²)/Btu
R_o	Total outside resistance, $(1/h_o + r_o)$
r_o	Outside fouling resistance, (hr)(°F)(ft²)/Btu
R_t	Total resistance to heat transfer, $(R_i + R_m + R_o) = 1/U_o$ or $1/U_d$
Re	Reynolds number
S	Specific gravity of fluid
T_{av}	Average bulk shell-side temperature, °F
Δt_{cf}	Temperature drop across the condensate film, °F
T_f	Mean condensate film temperature, °F, $T_f = T_{sv} - 1/2\ \Delta t_{cf}$
ΔT_{LM}	Logarithmic mean temperature difference, °F
T_m	Average tube metal wall temperature, °F
T_{sv}	Saturation temperature of a condensing vapor, °F
t_w	Average water temperature in tubes, °F
U_{of}	Overall coefficient of heat transfer for finned tube, Btu/(hr)(°F)(ft²) outside surface
U_{op}	Overall coefficient of heat transfer for plain tube, Btu/(hr)(°F)(ft²) outside surface
V_t	Tube-side velocity, ft/sec
W_t	Total flow through tube side, lb/hr
x	Number of tubes required for condensing
X_f	Tube wall thickness, ft
x_f	Tube wall thickness, in.
Y	Mean fin thickness, ft
y	Mean fin thickness, in.

Greek Symbols:

μ	Viscosity of fluid, lb/(ft)(hr)
μ_w	Viscosity at tube wall temperature, lb/(ft)(hr)
μ_g	Viscosity gradient, $\mu_g = (\mu/\mu_w)^{0.14}$
ρ	Density, lb/ft³
λ	Latent heat of condensation, Btu/lb