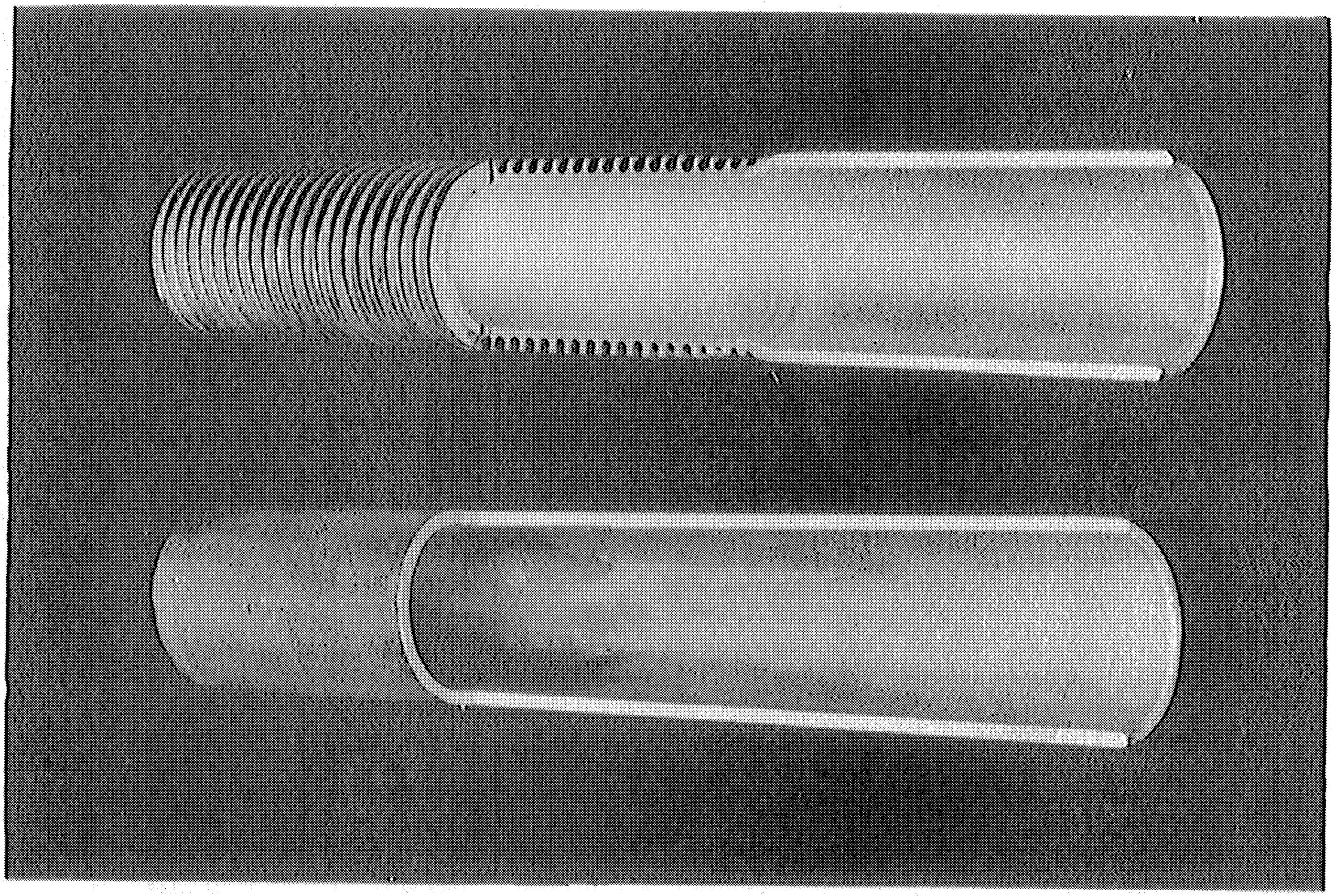
WHICH TUBE

GIVES THE MORE ECONOMICAL

HEAT TRANSFER ?



THIS REPORT PROVIDES THE BASIC DATA TO PERMIT THE EVALUATION OF FINNED TUBES FOR USE IN CONVENTIONAL SHELL AND TUBE EXCHANGERS.

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PERFORMANCE OF FINNED TUBES IN SHELL AND TUBE HEAT EXCHANGERS

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ABSTRACT

Heat transfer coefficients have been determined on three pairs of tube bundles, all 48 inches long. The bundles of a pair are identical except that plain tubes are used in one bundle and finned tubes in the other. One pair is 6 inches in diameter with 5/8-inch tubes. The other two pairs are 8 inches in diameter and have 1/2-inch and 3/4-inch tubes, respectively. The finned tubes have 19 nominal fins per inch about 1/20 inch high, with the diameter over the fins slightly less than the diameter of the plain ends. These finned tubes have from 2.07 to 2.76 times as much outside surface as plain tubes.

Heat transfer measurements were made for water, lubricating oil, and glycerine on the shell side. Several temperature levels and temperature differences were used to give a variety of viscosities and other fluid properties. Shell-side coefficients were determined by extrapolating to infinite water velocity the overall coefficients for a series of water velocities inside the tubes. These shell-side coefficients are correlated by the following equation:

$$\frac{h_o D}{k} = C \left(\frac{D G_m}{\mu}\right)^{.65} \left(\frac{C_p \mu}{k}\right)^{.375} \left(\frac{\mu}{\mu_w}\right)^{.14}$$

The values of C depend upon the bundle and vary from 0.225 to 0.115. The pressure drop data are correlated by the methods presented by Donohue.

The heat transferred per degree of temperature difference for the clean finned-tube bundles varied from 110 per cent of that for the corresponding plain-tube bundles for water to 200 per cent for the lubricating oil. For the same mass velocity, the shell-side coefficients for the finned tubes based on the outside area are approximately 80 $\pm$ 20 per cent of the plain tube coefficients. In all cases, at the same mass velocities the pressure drop is less for the finned-tube bundles.

A study of the economics of plain versus finned tubes shows that finned tubes are more economical than plain tubes for viscous fluids with low shell-side coefficients. Example calculations are given to make a comparison of shell and tube exchanger costs for cooling lubricating oil, absorber oil, and corn sirup with water. The metal requirements for these

exchangers are computed. For the example problems, selected computations indicate savings in cost from 20 to 25 per cent and savings in metal requirements from 29 to 36 per cent when exchangers equipped with finned tubes are used.

A comparison is made of plain and finned-tube exchangers designed for the same service:

	Heat Transfer Btu/hr	U_o, Btu / hr °F sq ft	Standard Exchanger Required: Outside Area sq ft	Size	Total Cost	Total Weight lbs
Case I Lube Oil						
Plain	840,000	80.	768	24"x8'	$3852.54	5,758
Finned	840,000	57.7	978	18"x8'	$2899.35	3,673
Case II Absorber Oil						
Plain	14,400,000	116.	4330	42"x16'	$14,116	26,663
Finned	14,400,000	87.5	5780	33"x16'	$10,530	17,762
Case III Corn Sirup						
Plain	1,500,000	67.1	425	20"x8'	$ 3,288	4,308
Finned	1,500,000	49.0	565	16"x8'	$ 2,627	3,049

The following constants are recommended for use in Equations (4a) and (4b)

$$\frac{hD}{k} = C' \left(\frac{DG_m}{\mu}\right)^{.60} \left(\frac{C_p \mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{.14} \qquad (4b)$$

Type of Exchanger	C for Eq (4a): Plain	C for Eq (4a): Finned	C' for Eq (4b): Plain	C' for Eq (4b): Finned
Shell Circle Design Bored Shell	0.19	0.13	0.34	0.23
Standard Design Bored Shell	0.14	0.098	0.25	0.175
Standard Design Unbored Shell	0.125	0.087	0.22	0.155

ACKNOWLEDGEMENTS

The courtesy of Mr. Sigmund Kopp of the Alco Products Division of the American Locomotive Company is acknowledged for supplying the Alco Heat Exchanger price book. Mr. Townsend Tinker of the Ross Heater and Manufacturing Company gave advice during the initial stages of the program. Mr. Earl L. Tyner, Mr. Ralph F. Johnson, and Mr. J. M. Gibbs assisted in the construction of the experimental equipment, in obtaining the data, and in the calculations.

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SHELL AND TUBE HEAT EXCHANGERS

INTRODUCTION

Finned tubes are used to increase the rate of heat transfer over that obtained by plain tubes. A major requirement for the effective use of fins on the outside of tubes is that the heat-transfer coefficient on the outside must be low relative to the coefficient on the inside. Heating of air on the outside of tubes by steam inside the tubes is an example of an effective use of fins, since the heat-transfer coefficient between air and the outside of the tubes is very low relative to the steam coefficient.[1] For this service, when the outside coefficient is about 1/100 of the inside coefficient, high fins are used to give up to 20 times as much outside surface as a plain tube. Subcooling of a refrigerant liquid by cold vapors also employs finned tubes advantageously.[2]

Many commercial processes employing shell and tube exchangers result in a low coefficient on the outside of tubes as compared to the coefficient inside the tubes. The introduction of tubes with plain ends and low fins, about 1/20 inch high, made it feasible to use finned tubes in standard shell and tube exchangers. These tubes provide about 2.5 times as much outside surface as plain tubes. The services in which these tubes may be used economically will have outside coefficients of the order of 1/5 of the inside coefficient, and hence the surface ratio of 2.5 is sufficient. The condensing of refrigerants such as Freon 12 is an example of the effective use of finned tubes in shell and tube units accepted in the industry.[3-7] The relatively low condensing coefficients for organic substances as compared to water-convection coefficients inside the tubes provides the necessary conditions for the advantageous use of fins. Boiling of organic liquids outside of tubes makes effective use of fins when the temperature difference is low.[6,8]

The cooling or heating of viscous materials such as lubricating oils provides the necessary ratio of coefficients for the advantageous use of finned tubes in shell and tube exchangers.[9] Armstrong[10] reported test data on a baffled shell and tube exchanger employing finned tubes for cooling a viscous oil and concluded that tubes with low fins were advantageous for this service. These data were encouraging but seemed insufficient to predict the increase in heat transfer which one would expect for various fluids in shell and tube exchangers. No direct comparison was made with plain tube exchangers of the same dimensions.

An experimental program was developed to compare heat-transfer coefficients between plain tubes and finned tubes for fluids on the shell side of shell and tube exchangers. Exchangers identical in all details except for the tubes were obtained, using the shell-circle type of design known to give efficient heat exchange.[11] The exchangers were tested by circulating water to standardize them, and then measurements were obtained using 40 SAE lubricating oil and glycerine as viscous fluids. The data were taken in a manner which made it possible to obtain the shell-side coefficients in order that they might be correlated by the usual dimensionless groups. Pressure-drop data for the shell-side fluids were measured, since pressure drop may provide a limitation on the fluid velocity in the exchangers.

This report also includes methods of predicting the services for which these helical finned tubes are economical in standard shell and tube exchangers.

EXPERIMENTAL INSTALLATION

Equipment was installed in the Chemical Engineering Laboratory to circulate oil through two shell and tube exchangers. The exchangers are respectively 8 inches and 6 inches in diameter and the removable tube bundles are 48 inches long. The oil is cooled in the exchanger under test and is heated in the other exchanger.

A flow diagram of the experimental unit is shown in Fig. 1, while Fig. 2 is a photograph of the installation. One pumping system circulates the shell-side fluid, which passes through the shell of one exchanger to cool it and through the shell of the exchanger not under test to heat it in a continuous steady-state experiment. A second system circulates water through the tubes of the cooler under test. Steam on the inside of the tubes was used to heat the fluid in the exchanger not under test. The piping was arranged so that the cooling water could be circulated through either exchanger and the steam enter the tubes of either exchanger. This arrangement made it possible to test the tube bundle in each exchanger by operating valves.

The significant specifications for the 6-inch and 8-inch bundles are given in Table I, page 109. The plain-tube exchangers, shells and bundles, were commercial units built to the manufacturer's specifications.

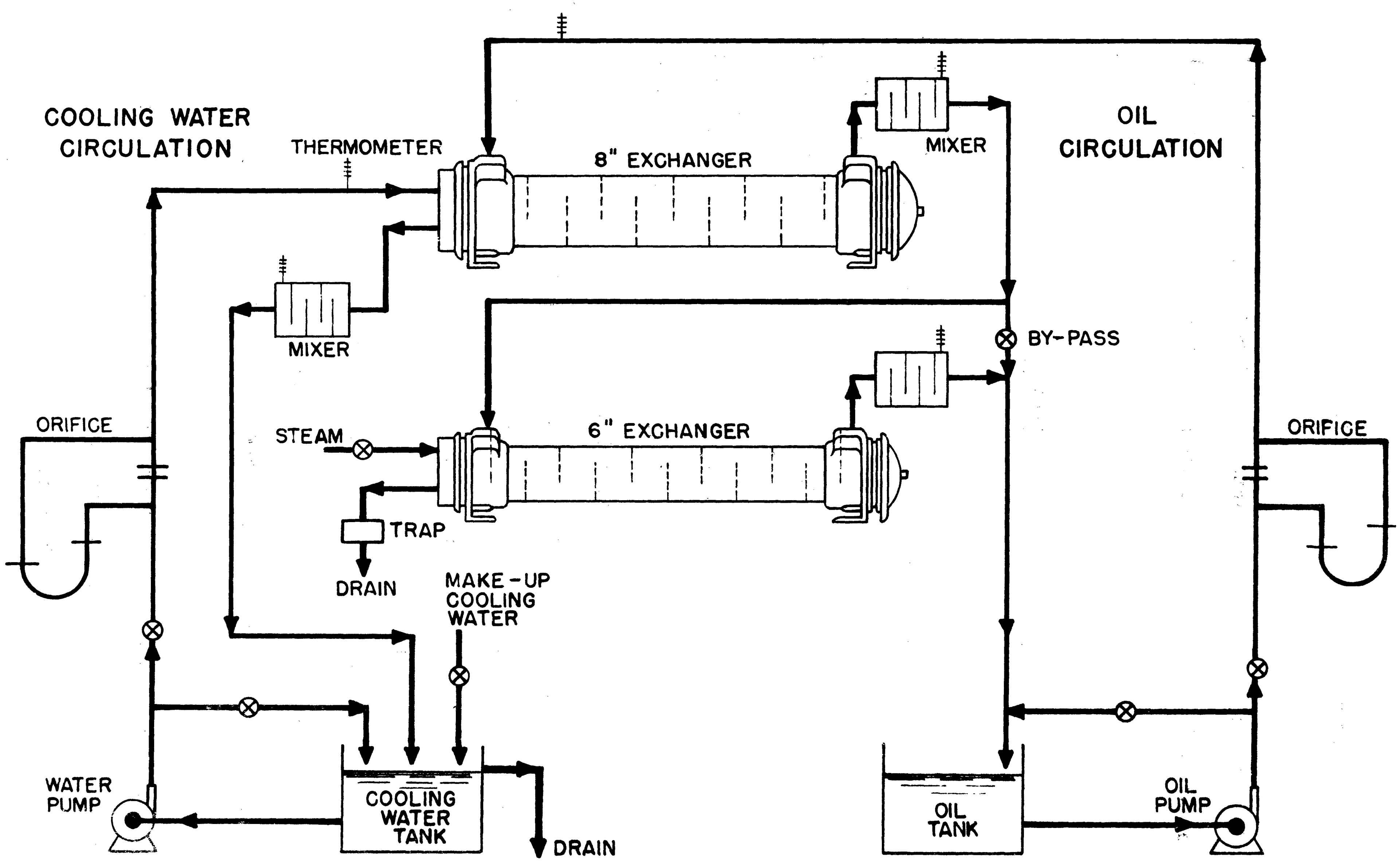


FIG. I FLOW DIAGRAM OF EXCHANGER TEST UNIT WHEN TESTING 8" EXCHANGER

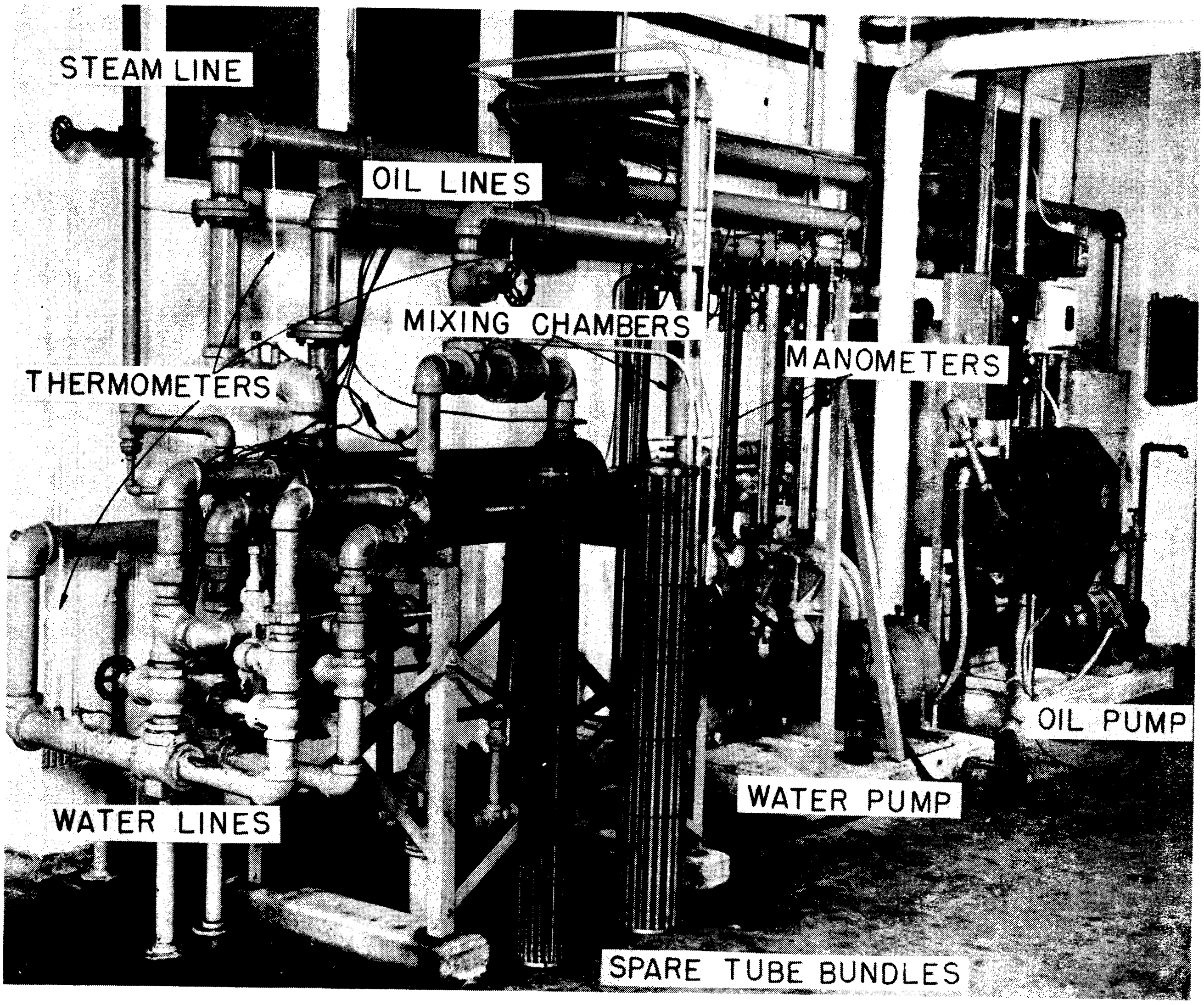
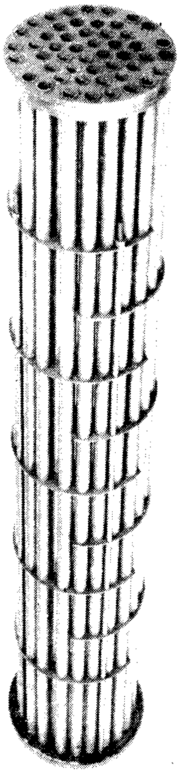


FIG. 2 PHOTOGRAPH OF EXPERIMENTAL INSTALLATION

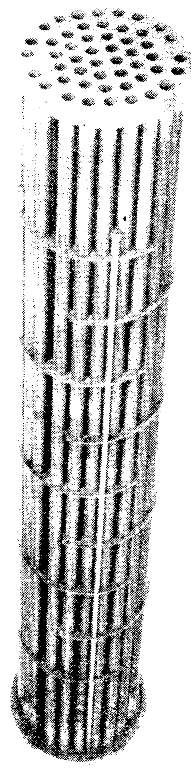
The finned-tube bundles were constructed to the same specifications as the plain-tube bundles but were tubed with finned tubes supplied by the Wolverine Tube Division. One pair of bundles was tested in the 6-inch exchanger with 5/8-inch outside diameter plain Admiralty tubes in one bundle and 5/8-inch outside diameter finned tubes in the other bundle. Four bundles were used in the 8-inch exchanger; 1/2-inch copper plain and finned tubes and 3/4-inch Admiralty plain and finned tubes were tested. In all cases the bundles were identical for each pair with respect to the tube-sheet layout, number of baffles, baffle spacing, and all other dimensions. A baffle spacing of 4 inches was used in all bundles, with 9 baffles in the 8-inch bundles and 11 baffles in the 6-inch bundles. The layout of the baffles in relation to nozzles is indicated in Fig. 1.

Tube-sheet layouts for the three tube sizes are shown in Figs. 3, 4, 5, pages 94-96. It may be seen that the tubes fill the shell circle completely and are spaced inside this circle on a triangular pitch, though they do not necessarily fill this area completely.

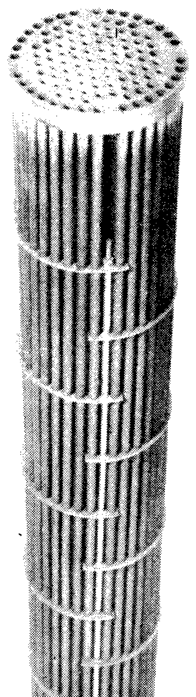
Figs. 6 and 7 are photographs of the finned and plain-tube bundles. It may be seen that the finned tubes have plain ends similar to plain tubes. The fins have a diameter slightly less than the plain end, so that the tubes may be inserted in the bundle in a manner identical with that used for plain tubes. The finned tubes have 19 nominal fins per inch and the fins are about 1/20 inch high. Fig. 8 is a photograph of tube cross sections, showing the contour of the fins. The tubes were sectioned, mounted in bakelite, and the sections polished. The section of the tube indicates the inside of the tube to be smooth. However, when a length of tube is held to the light, small markings are observed on the inside surface corresponding to the fins. The dimensions of all tubes in a given bundle are uniform within 0.001 inch.



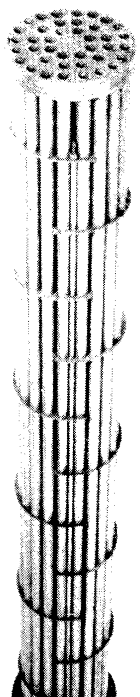
BUNDLE NO. 1



BUNDLE NO. 2

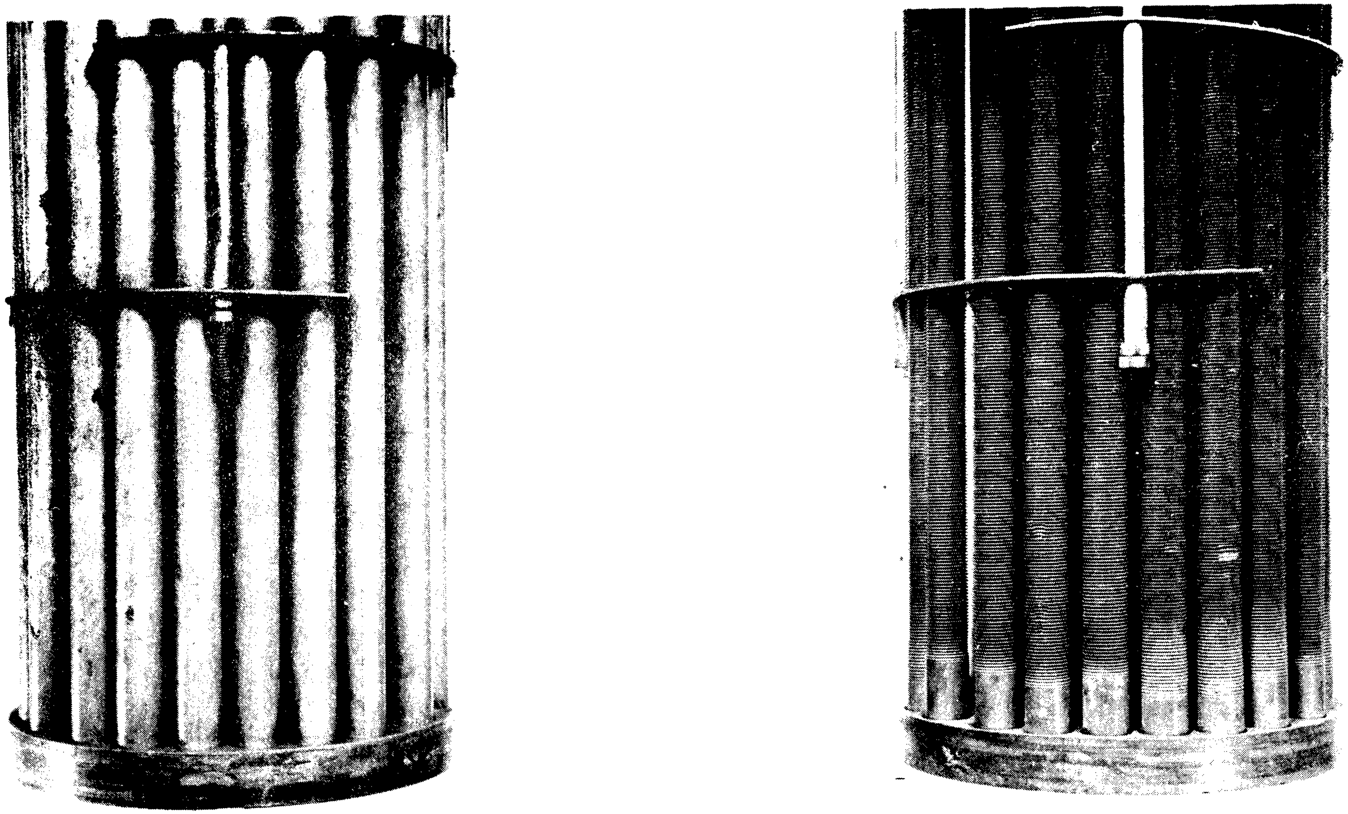


BUNDLE NO.4



BUNDLE NO. 5

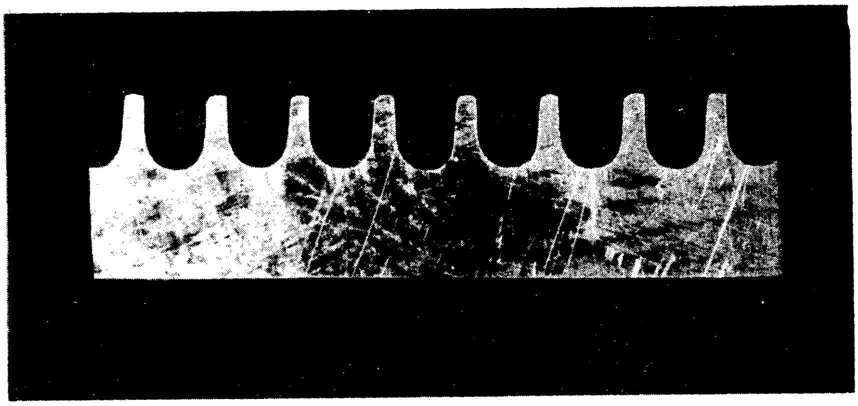
FIG. 6 PHOTOGRAPHS OF BUNDLES REMOVED FROM EXCHANGERS



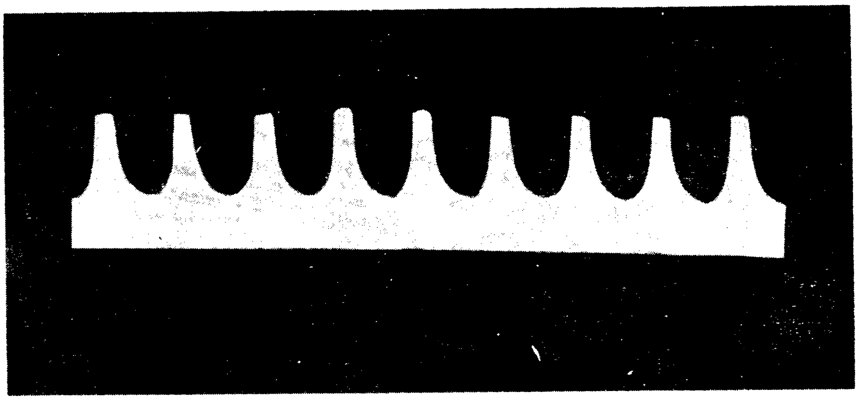
BUNDLE I

BUNDLE 2

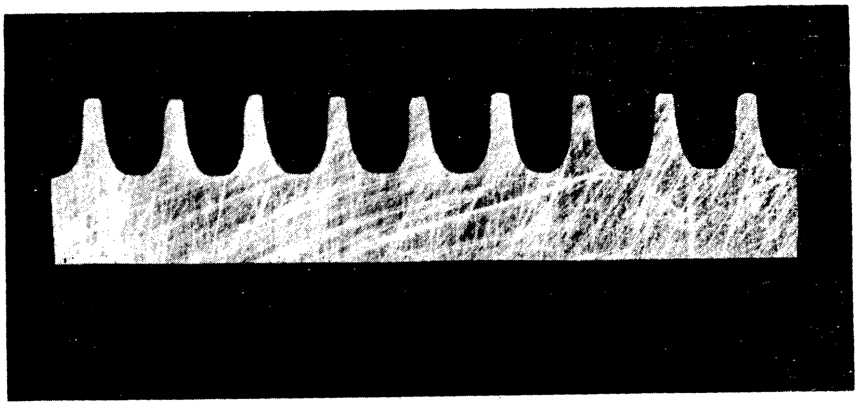
FIG. 7 PHOTOGRAPHS SHOWING INTERCHANGEABILITY OF FINNED AND PLAIN TUBES IN BUNDLES.



BUNDLE 2



BUNDLE 4



BUNDLE 6

FIG. 8 LONGITUDINAL SECTION OF FINNED TUBES except for fin height, which varies up to 0.003 inch. The outside areas for the finned tubes are computed by assuming an idealized shape. The fin is assumed to have a rectangular section with square ends, and the root is assumed to be a semicircle, as shown in Fig. 9. The area so computed is within 2 to 3 per cent of the integrated area given by the actual contours. The tube dimensions are identified in Fig. 9, and the calculated areas for the bundles are given in Table I, page 109.

These finned tubes are manufactured from plain tubes by extruding the metal wall into a continuous helical fin. In addition to having plain ends, the tubes may also have plain sections at any desired points along their lengths.

Measurements in Heat-Transfer Tests

Measurements were made of the temperatures of the two liquid streams entering and leaving the test exchanger, the flow rates of the two streams, and the pressure drop across the shell side of the test exchanger. This information is sufficient to permit computation of overall coefficients (U) in Equation (1):

$$q = U\,A\,\Delta T\ , \tag{1}$$

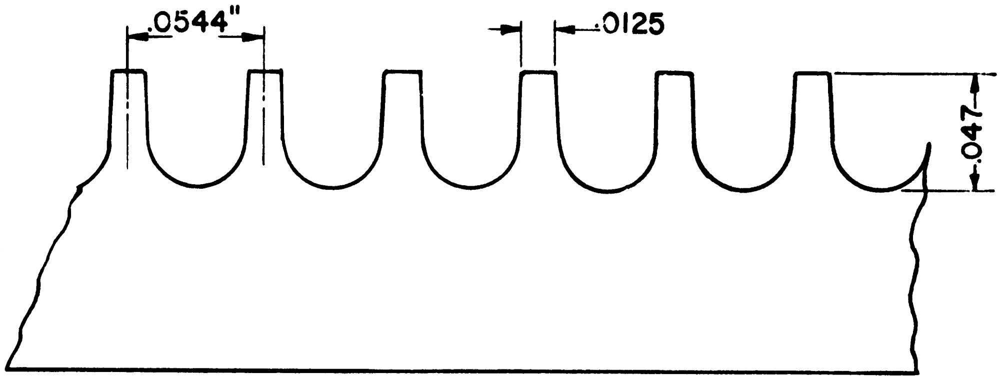
in which q = heat transferred, Btu per hour

U = overall coefficient, Btu per (hr)(°F)(sq ft)

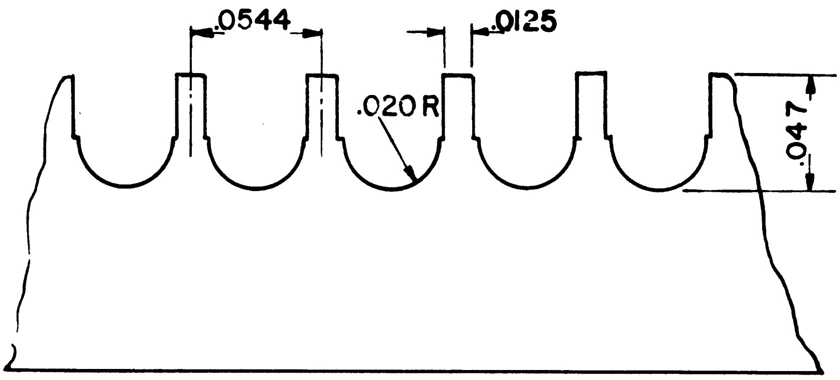
A = heat-transfer area, sq ft

ΔT = temperature difference, °F.

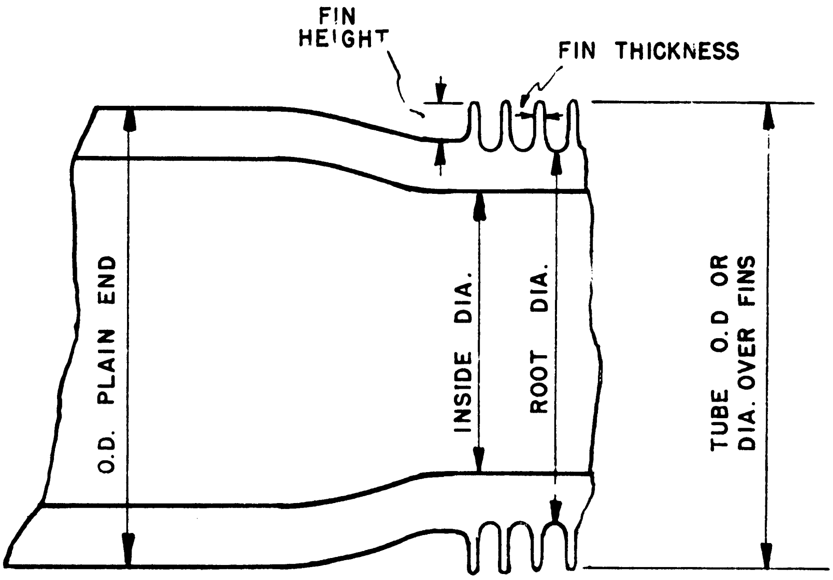
Temperature measurements were made with mercury-in-glass thermometers installed in thermometer wells in mixing chambers equipped with disc and doughnut baffles, as shown in Fig. 10, page 97. Shell-side fluids were



ACTUAL FIN PROFILE BUNDLE 2



IDEALIZED FIN FOR AREA CALCULATION BUNDLE 2



NOMENCLATURE OF TUBE DIMENSIONS

FIG. 9 DIMENSIONS OF FINNED TUBES

measured with thermometers graduated to 0.1°C, while the water on the tube side was measured either with these thermometers or with Beckmann thermometers graduated to 0.01°C. All thermometers are calibrated against Bureau of Standards thermometers. The mixing device was considered necessary to make sure that the fluid leaving the heat exchanger had been sufficiently mixed so that the temperature was a true average or mean bulk temperature.

Flow rates in both circulation systems were measured by sharp-edged orifices installed in the 3-inch circulation lines. They were calibrated with water in both systems and with oil in the shell-side system. Bundles of 3/4-inch tubing were used as straightening vanes some 50 diameters ahead of the orifice installation in the 3-inch pipe. Mercury manometers were used to indicate the orifice differential. All the orifice coefficients for water were within the range of 0.601 to 0.608 for the four orifices used. For oil, the coefficient was plotted as a function of the Reynolds number through the orifice in the range from 1000 to 20,000 and was found to lie 1.5 per cent above the curve reported from standard orifice installations.[12] The use of orifice coefficient as a function of Reynolds number resulted in individual rate curves as a function of pressure drop across the orifice for oil and for glycerine at each temperature level. For water, a single curve was used with corrections for density made by multiplying the rate by the square root of the ratio of the density at 60°F to the density at the flowing temperature. Fig. 11, page 98, gives typical calibration curves for water flowing through two different plates.

Pressure drops were obtained for the shell-side fluid by the use of mercury manometers attached to outlets on the circulation line approximately 1 inch from the exchanger nozzle. Similar measurements were made for the water on the tube side during the cooling tests with the exchanger.

PROPERTIES OF FLUIDS

Water was selected as one of the shell-side fluids because its properties are so well known that it serves as a standardizing fluid for the exchangers. It was not expected that much advantage would be observed for finned-tube bundles as compared to plain-tube bundles when cooling the water because of the high coefficient between the water and the outside of the tube. Lubricating oil, 40 SAE, was chosen as a typical viscous oil, and glycerine was selected as a second viscous fluid with properties distinct from those of mineral oil.

The viscosity and thermal conductivity of water were taken from McAdams,[13] while the heat capacity and density were taken from steam tables.[14]

The densities of the oil and of the glycerine were determined in the laboratory at room temperature. The changes in density with temperature were taken from the National Standard Petroleum Oil Tables[15] for the oil and from the International Critical Tables[16] for glycerine. Fig. 12, page 99, is a plot of the densities as a function of temperature.

Viscosities of the oil and glycerine were determined in the laboratory by Fenske pipettes at five temperatures, as plotted in Fig. 13, page 100.

The thermal conductivity of the lubricating oil was determined in the laboratory at 86°F. The value fell in the area expected for oils. A curve drawn through the experimental point with a slope equal to that of similar oils[17] is plotted in Fig. 14, page 101. The thermal conductivities for glycerine were taken from Smith[17] and are plotted on Fig. 14.

The specific heat of the lubricating oil was taken from TEMA[18], while the values for glycerine were taken from the literature.[16] These are plotted in Fig. 15, page 102.

To insure that the oil and glycerine were constant in properties during the heat-transfer tests, samples were taken at intervals to determine the density and viscosity. The oil showed no change in density from the initial value. The glycerine showed a small change in viscosity and density between the initial sample and the sample after it had been heated in operation. The change in density corresponded to a change in water content from 1.6 per cent at the beginning to 0.5 per cent during steady operation. The viscosity of the glycerine during steady operation also corresponded to 99.5 per cent glycerine from data in the literature,[19] as shown in Fig. 13.

TEST PROCEDURES AND OPERATION OF EQUIPMENT

The six tube bundles were tested using in turn water, oil, and glycerine as the shell-side fluid, with cooling water inside the tubes. Four temperature levels for the fluids were used for some bundles and fluids, while three levels were used on others. Most of the tests used a temperature difference between the shell-side fluid and the cooling water of around 25°F. For about 10 per cent of the data, temperature differences of 50 to 55°F were employed. Fig. 16 illustrates the conditions of the tests for bundle 4, while Table II gives a similar summary of tests for all bundles.

All tests consisted of obtaining data for an overall coefficient of heat transfer under steady state. Two types of data were obtained. The first consisted of a series of consecutive measurements in which the cooling-water velocity was varied between individual tests, while the shell-side inlet and outlet temperatures as well as flow rate remained constant. This set of data was required for Wilson plots. The other type of data was individual

BUNDLE NO. 4
1/2" FINNED TUBES
IN 8" SHELL

WATER

TEMP. LEVELS, INLET °F	ΔT's °F	FLOW RATES LBS / HR.
193	11	70,600 TO 15,100
158	12	71,800 TO 15,600
122	14	77,000 TO 15,200

SAE 40 LUBE OIL

TEMP. LEVELS, INLET °F	ΔT's °F	FLOW RATES LBS / HR.
227	18	56,000 TO 13,800
194	22	55,800 TO 13,900
158	23	56,500 TO 14,800
122	26	44,300 TO 14,800

GLYCERINE

TEMP. LEVELS, INLET °F	ΔT's °F	FLOW RATES LBS / HR.
226	20	75,000 TO 16,300
195	20	75,500 TO 13,800
158	23	74,200 TO 16,200
122	27	48,800 TO 18,300

FIG. 16 CONDITIONS OF TESTS FOR BUNDLE 4

TABLE II

SUMMARY OF TEST CONDITIONS

Tube Bundle	Shell Side	Water	SAE 40 Lube Oil	Glycerine
Bundle 1 3/4-in. plain tubes in 8-in. shell	Inlet Temp., °F	177	196, 113	223, 193, 158, 122
	ΔT, °F	17.8 - 11.8	63.9 - 30.6	38.4 - 24.8
	Flow Rate, lbs/hr	61,000 - 17,000	45,500 - 13,100	73,000 - 15,400
Bundle 2 3/4-in. finned tubes in 8-in. shell	Inlet Temp., °F	177	196, 113	227, 195, 157, 131
	ΔT, °F	15.8 - 9.2	56.5 - 25.8	54.0 - 18.1
	Flow Rate, lbs/hr	61,000 - 17,100	51,000 - 13,000	78,300 - 13,400
Bundle 3 1/2-in. plain tubes in 8-in. shell	Inlet Temp., °F	193, 158, 122	228, 193, 158, 122	227, 194, 158, 122
	ΔT, °F	22.0 - 11.2	30.3 - 18.1	29.3 - 16.4
	Flow Rate, lbs/hr	72,400 - 13,300	60,000 - 13,500	70,500 - 13,600
Bundle 4 1/2-in. finned tubes in 8-in. shell	Inlet Temp., °F	193, 158, 122	227, 194, 158, 122	226, 195, 158, 122
	ΔT, °F	16.5 - 8.1	27.2 - 16.8	27.2 - 16.9
	Flow Rate, lbs/hr	77,000 - 15,200	56,000 - 13,800	75,500 - 13,800
Bundle 5 5/8-in. plain tubes in 6-in. shell	Inlet Temp., °F	177	195, 113	227, 193, 158, 122
	ΔT, °F	32.3 - 10.4	63.5 - 33.5	38.4 - 23.5
	Flow Rate, lbs/hr	40,000 - 11,300	42,000 - 8,600	52,000 - 14,800
Bundle 6 5/8-in. finned tubes in 6-in. shell	Inlet Temp., °F	177	196, 114	227, 194, 131
	ΔT, °F	15.5 - 8.8	55.4 - 23.6	53.2 - 16.1
	Flow Rate, lbs/hr	49,600 - 12,100	39,300 - 10,800	60,100 - 13,500

tests to give single values of the overall coefficients at selected conditions. The latter type of data was taken only after sufficient data of the first type had been accumulated to give reliable coefficients for the cooling water inside the tubes.

All measurements of an overall heat-transfer coefficient were made in similar manner once the conditions of the test had been selected. The test unit was operated for a sufficient period of time to bring it to a steady state before taking measurements. Once the flow rates for the shell-side fluid and cooling water were fixed, the controls consisted of the steam rate to the exchanger not under test and the make-up cooling-water rate to the tank in the cooling-water system. Thermoregulators were installed and were used between tests, but final control was usually manual.

The data tabulated for a single overall coefficient consisted of the four temperatures for the shell-side fluid and cooling water entering and leaving the test exchanger and of the manometer readings to give the flow rates and pressure drops for the shell and tube-side fluids. Table III, page 110, gives actual data recorded for run 26, which consisted of four overall coefficients, since it was of the first type described. Four recordings of the temperatures and manometer readings were made at about one-minute intervals. The flow rates were such that the contents of the exchanger were changed several times in a minute. Also, the heat transfer was large as compared to the heat capacity of the exchanger. Therefore, the short test period was considered as satisfactory when the inlet temperatures and flow rates were essentially constant.

The recordings for runs of the second kind, in which individual overall coefficients were determined, varied from those for Wilson plots;

ten consecutive readings were taken at one-minute intervals instead of the four readings.

A summary of the experimental data and calculated results is given in Table IV, page 115. All the pertinent data used in the calculations are given, including the heat transfer on both the tube side and the shell side. There are 208 runs, which represent approximately 490 determinations of overall heat-transfer coefficient, since Wilson-plot data with four overall coefficients each were taken on about half of the runs reported. The dimensionless groups used in correlating the data are also included in Table IV.

Bundles 1, 2, 5, and 6 were obtained at the time the installation was made and were tested with water, oil, and glycerine in turn. Bundles 3 and 4 were procured later and were tested with glycerine, water, and oil in turn. The run numbers in Table IV indicate the exact order of the tests.

The tests may be considered as applying to clean tubes. Water deposited a thin film removable by touch or washing. The inside of the tubes was cleaned with a stiff brush and dilute hydrochloric acid at the beginning of a series of tests for each bundle with each fluid. The outside of the tubes was cleaned after tests with water on the shell side. The tubes were rinsed by pumping dilute hydrochloric acid solution containing a detergent through the shell side.

When changing fluids, it was necessary to clean the circulating system. Water was removed by draining and filling with oil or glycerine and operating the shell-side system at temperatures above 212°F. Glycerine was removed by circulating water. Oil was removed by circulating a kerosene-water-detergent emulsion at elevated temperatures.

CALCULATION OF SHELL-SIDE COEFFICIENTS

There are two methods of computing the shell-side convection coefficients, corresponding to the two types of data. For the runs in which a series of overall coefficients was determined at constant conditions for the shell-side fluid, the shell-side coefficient is found from a Wilson plot.[20] For the individual determinations of the overall coefficient, the cooling-water film and metal resistances were subtracted from the overall resistance by calculation to give the shell film resistance.

Overall Coefficients

Overall coefficients are computed by Equation (1). The quantity of heat transferred was measured for both the shell-side and the tube-side fluids. The hot shell-side fluid lost heat to the surroundings and therefore the computed shell-side heat transfer might be expected to be greater than the actual transfer. Likewise, the cooling water lost heat to the air between the points of temperature measurement. It was found that a difference in heat transfer between the two streams was of the order of 100 Btu per degree temperature difference between the shell fluid and the room. This difference represented from 1 to 10 per cent of the total heat transfer. It appeared logical to average the heat transferred on the two sides to obtain q, unless other runs in the series indicated that the shell-side value was in error.

The actual outside area of the exchanger is taken from Table I.

The temperature difference is the logarithmic mean difference corrected for the two passes on the water side by Fig. T-4A in TEMA[18], or else the temperature difference was computed by Equation 10, page 145 of McAdams.[13]

An example calculation of the overall coefficient is given in Table III. All coefficients for each determination are recorded in Table IV.

Wilson Plots

A plot of the reciprocal of the overall coefficient as a function of the reciprocal of the water velocity to the 0.8 power is termed a Wilson plot.[20] An extrapolation of the line to infinite water velocity gives the resistance of the remainder of the heat-transfer path. An example calculation of the convection coefficient between the shell fluid and the outside of the tubes is given for run 26 in Table III, page 110. The Wilson plot for these data is shown by Fig. 17. The intercept in Fig. 17 gives the resistance to heat transfer for the shell-side fluid and the metal, since the coefficient has been extrapolated to infinite water velocity. After subtracting the metal resistance from the intercept, the shell-side resistance, or its reciprocal, the shell-side coefficient, is obtained.

The use of a temperature correction for the Wilson plot may merit brief discussion. The Wilson plot should have as its ordinate the reciprocal of the overall coefficient and as its abscissa the reciprocal of the water-film coefficient. If all mean water temperatures in one run were the same, then the reciprocal of the 0.8 power of the water velocity would be directly proportional to the reciprocal of the water-film coefficient, when assuming that the water-film coefficient is a function of the Reynolds number to the 0.8 power. However, when the water temperature for one velocity differs from that for another, one should plot the convection coefficient with the proper variation in properties. Since the convection coefficient for water has been simplified to be a function of temperature

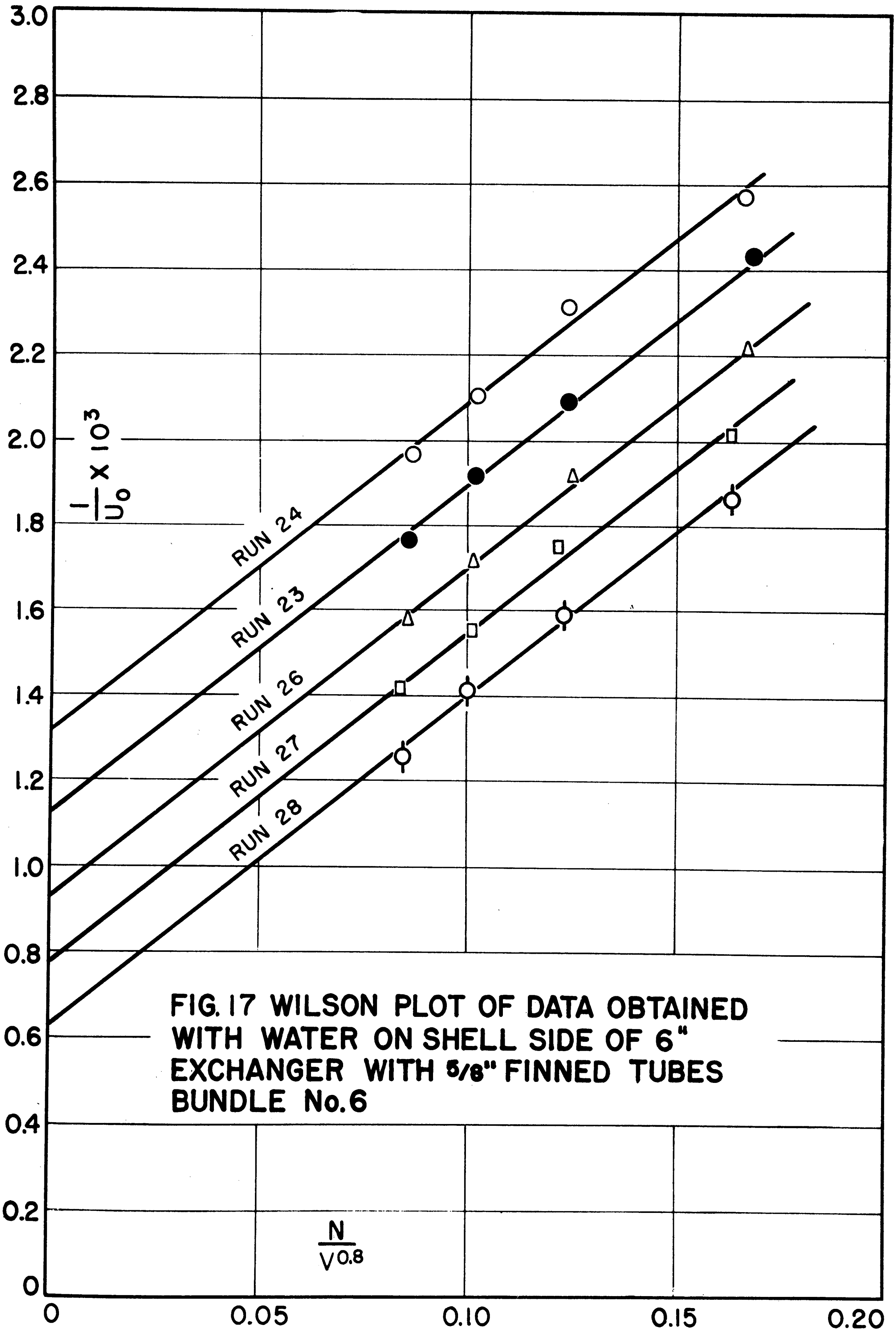


FIG. 17 WILSON PLOT OF DATA OBTAINED WITH WATER ON SHELL SIDE OF 6" EXCHANGER WITH 5/8" FINNED TUBES BUNDLE No. 6

and velocity,[13] the use of $N/V^{0.8}$ for the reciprocal of the convection coefficient gives Wilson plots with straight lines of constant slope for a given tube bundle. If this temperature correction had not been made, the lines through points of a different mean water temperature would have different slopes. It was a valuable correlation factor to know that all Wilson-plot lines for a given bundle were of the same slope.

Fin Efficiency

During heat transfer in the finned-tube exchangers, the temperature along the outside surface of the fin is higher than at the base of the fin. The procedure which has been found satisfactory for evaluating the effect of this temperature distribution is the use of a fin efficiency.[21] The fin efficiency is defined as follows:

$$\phi = \frac{\int_0^{a_f} \Delta T' \, d(a_f)}{\Delta T_B \, a_f} \qquad (2)$$

where $\Delta T'$ = the variable temperature difference between the bulk-fluid temperature and the point fin temperature,

ΔT_B = the temperature difference at the base of the fin or at the root of the fin,

a_f = area of the fin,

ϕ = fin efficiency.

An effective area (A_e) is defined as the sum of the root area and the fin area times the fin efficiency. This effective area may be used in heat transfer equations along with the temperature differences which apply for outside surface temperatures at the root of the fin. Gardner[21] has computed the fin efficiency for several shapes; Fig. 6 of his paper was used with a fin of constant cross section for heat flow. These efficiencies depend upon the coefficient of heat transfer adjacent to the fin surface as well as the conductivity of the fin metal and fin dimensions. To solve problems involving fin efficiency, Fig. 18, page 103, has been prepared, which gives the ratio of the total outside area (A_o) to the effective outside area (A_e) as a function of both the outside coefficient based on the actual area and of the outside coefficient based on the effective area. The dashed curves in Fig. 18, which relate A_o/A_e to the convection coefficient (h_o') based on the actual area, are required to compute the experimental data on a basis of effective fin area, while solid curves, which relate A_o/A_e to the convection coefficient (h_o) based on the effective area, are convenient to find actual exchanger sizes from computed effective areas. These curves for the low-fin tubes with 19 nominal fins per inch are the same for several sizes of tube, but are different for metals of different thermal conductivity. They do not apply to finned tubes when the fin profile is different from that of Fig. 8.

In using Gardner's procedure, it was decided that 80 per cent of the surface is fin surface and 20 per cent is root surface and represents prime surface. Reference to the sections in Fig. 8 illustrates that the entire surface could be considered as fin surface. In once case, with 80 per cent of the area considered as fin and 20 per cent as prime surface, the use of Fig. 6 of Gardner gave the same effective surface as the computation of the fin efficiency by numerical methods for the actual cross section shown in Fig. 8.

Film Coefficients from Single Overall Coefficients

After several Wilson plots had been determined for each tube bundle to make sure that the slope was determined correctly, individual overall coefficients were used to determine shell-side coefficients. Rather than drawing a line through a single point on a Wilson plot, the equation for the water-film coefficient was determined for each bundle as listed in Table V. The constants in these equations were determined from the slopes of the Wilson-plot lines.

TABLE V

EQUATIONS FOR INSIDE COEFFICIENTS

Bundle No.	Shell and Tube	Equation
1	8" 3/4" Plain	$h_i' = 138\ (1 + .011\ T)\ V^{0.8}$
2	8" 3/4" Finned	$h_i' = 58.9(1 + .011\ T)\ V^{0.8}$
3	8" 1/2" Plain	$h_i' = 137\ (1 + .011\ T)\ V^{0.8}$
4	8" 1/2" Finned	$h_i' = 45.0(1 + .011\ T)\ V^{0.8}$
5	6" 5/8" Plain	$h_i' = 129\ (1 + .011\ T)\ V^{0.8}$
6	6" 5/8" Finned	$h_i' = 55.5(1 + .011\ T)\ V^{0.8}$

where h_i' = inside coefficient for water based on the actual <u>outside area</u>, Btu per (hr)(°F)(sq ft outside),

T = mean bulk water temperature, °F,

V = water velocity, ft per sec.

The equations in Table V permitted computation of the shell)side coefficient by the following fomrula:

$$\frac{1}{h_o'} = \frac{1}{U_o} - \left(\frac{L\ A_o}{kA_{av}} + \frac{1}{h_i'}\right) \tag{3}$$

In Table IV all single determinations of overall coefficients were converted to shell-side coefficients by this procedure. It was necessary to convert the h_o' based on the actual area to h_o based on the effective area for these runs in the same manner as for runs having Wilson plots.

CORRELATION OF HEAT-TRANSFER DATA

The data on the plain-tube bundles were correlated first since they might be expected to follow correlations previously established by Donohue,[22] Short,[23] or Tinker.[11] The data obtained did not permit a study of baffle spacing, baffle height, or tube arrangement. The data did permit a study of Reynolds number at constant Prandtl number and of Prandtl number at constant Reynolds number. The correlations for the finned-tube data closely paralleled the correlation of the plain-tube data.

Plain Tubes

The shell-side coefficients were assumed to follow an equation of the following form:

$$\frac{h_o D}{k} = C \left(\frac{DG_m}{\mu}\right)^m \left(\frac{C_p \mu}{k}\right)^o \left(\frac{\mu}{\mu_w}\right)^{.14} , \qquad (4)$$

in which h_o = the film heat-transfer coefficient,

D = outside diameter of the tube,

k = the thermal conductivity of the fluid at the mean bulk temperature,

G_m = mean mass velocity, lbs per (sq ft)(hr),

μ = viscosity of the fluid at the mean bulk temperature, lbs per (ft)(hr),

μ_w = viscosity of the fluid at the wall temperature, lbs per (ft)(hr),

C_p = specific heat of the fluid at the mean bulk temperature,

C, o, m = constants

The physical and thermal properties of the shell-side fluid are taken at the mean bulk temperature, with the exception of the viscosity at the tube wall.

The velocity of the fluid as it passes through the bundle between the tubes and baffles will vary. Several procedures are available for this computation.[11,22,23] After due consideration, it was decided to use a procedure recommended by Donohue, as follows:

$$G_m = \frac{w}{A_m} \tag{5}$$

in which w = pounds of fluid flowing per hr through shell side

G_m = pounds flowing per (hr)(sq ft)

A_m = the mean area for the shell side of the exchanger defined by Equation (6).

The mean flow area is defined as follows:

$$A_m = \sqrt{A_w \times A_c}\ , \tag{6}$$

in which A_w = area of the window opening in baffle minus the cross section of the tubes in the window

A_c = minimum cross-flow area through the row of tubes nearest the center line of the exchanger and normal to the direction of the fluid flow.

An example calculation of the mean flow area is given in Table VI, page 131. The mean flow areas for all exchangers are listed in Table I. For the eight-inch exchanger a slight modification of the calculation procedure was necessary since the two ends of the exchanger did not contain baffles spaced the same as in the center portion of the exchanger. The area of cross flow when the fluid was flowing between the baffles was different from that when it flowed across the tubes on either end. For this exchanger two values of A_c were obtained, resulting in two values of A_m

for the center portion and for the end portion of the exchanger. These two values of A_m then were averaged, based on the respective length which each represented to give the final values of A_m used.

For each experimental determination of the shell-side coefficient listed in Table IV, the Nusselt number, hD/k, the Reynolds number, DG_m/μ, the Prandtl number, $C_p\mu/k$, and the viscosity ratio, μ/μ_w, were computed and listed in the table.

A series of successive approximations was required to obtain the final correlation between these dimensionless groups. The graphs shown are the final result after much study and several approximations.

The Nusselt number was plotted against the Reynolds number for several series of runs in which the temperature level, and hence the Prandtl number, were essentially constant for the series. Fig. 19 for bundle 1 is an example of the plots obtained. It may be observed that there is a lower slope for these curves at low Reynolds numbers than at high Reynolds numbers. This means that Equation (4), with constant exponents, will not give the best correlation of the data. However, a practical correlation is required for design procedures, and an average slope of the lines on figures similar to Fig. 19 was used as the exponent for the Reynolds number. To determine the exponent for the Prandtl number, it was plotted against the product of the Nusselt number, the Reynolds number to the 0.65 power, and the viscosity ratio to 0.14 power, with Fig. 20 for bundle 1 shown as an example of the final result. The slope of this curve was used as the exponent on the Prandtl number for the final correlation by Equation (4) and plotted for bundle 1 on Fig. 21.

In the first trial of this procedure, the slope found on the graph corresponding to Fig. 21 did not agree with the average slope

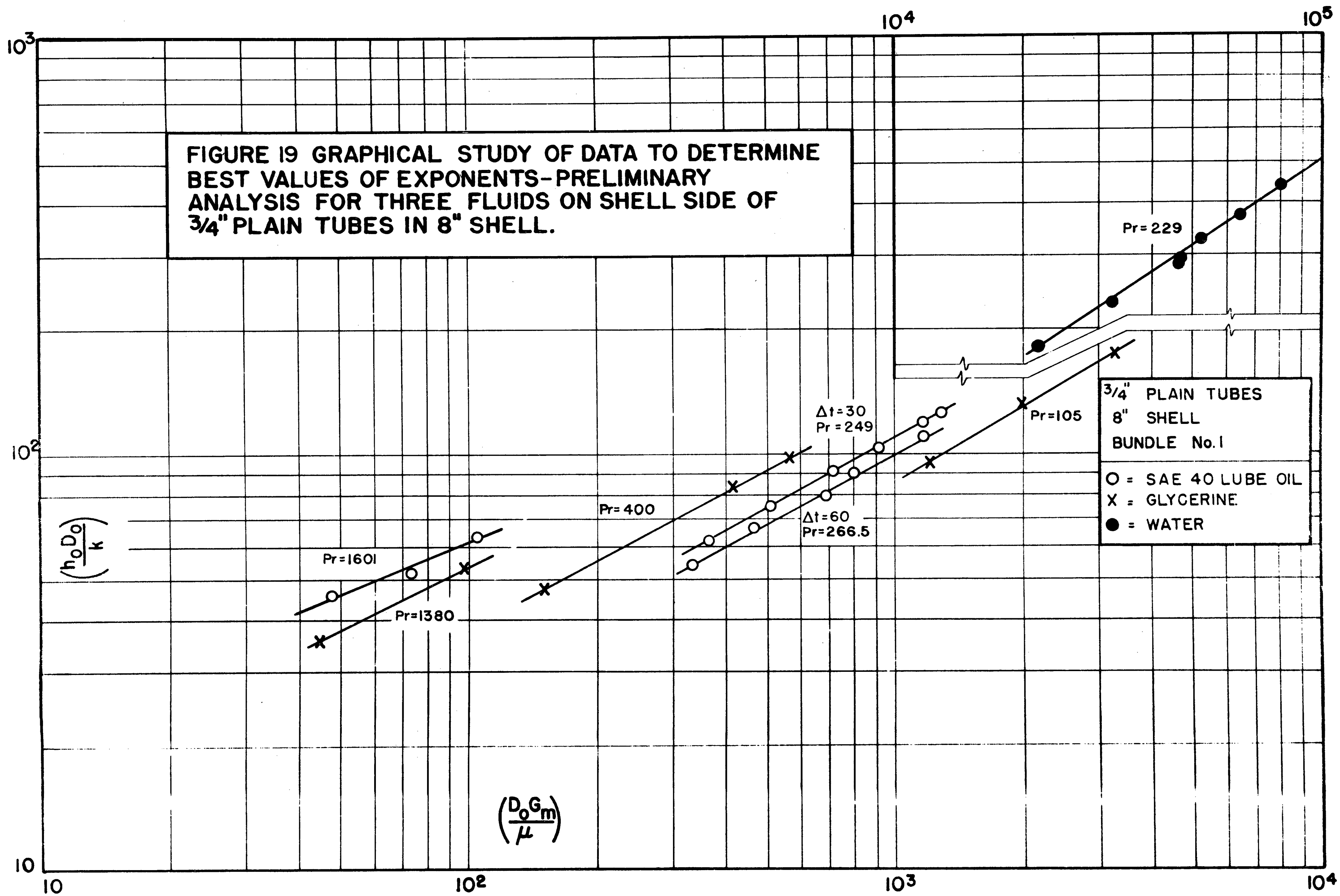


FIGURE 19 GRAPHICAL STUDY OF DATA TO DETERMINE BEST VALUES OF EXPONENTS-PRELIMINARY ANALYSIS FOR THREE FLUIDS ON SHELL SIDE OF 3/4" PLAIN TUBES IN 8" SHELL.

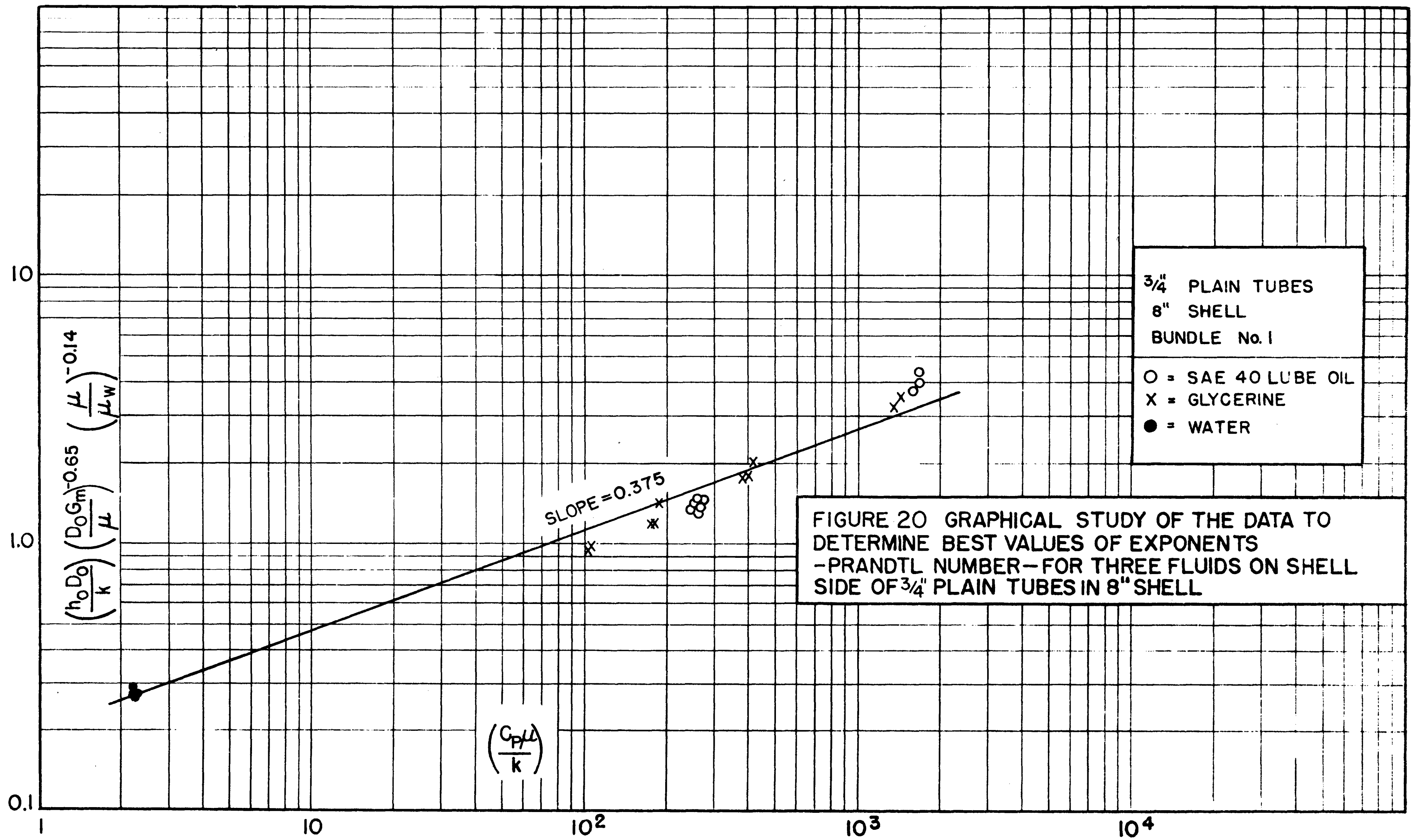


FIGURE 20 GRAPHICAL STUDY OF THE DATA TO DETERMINE BEST VALUES OF EXPONENTS -PRANDTL NUMBER- FOR THREE FLUIDS ON SHELL SIDE OF 3/4" PLAIN TUBES IN 8" SHELL

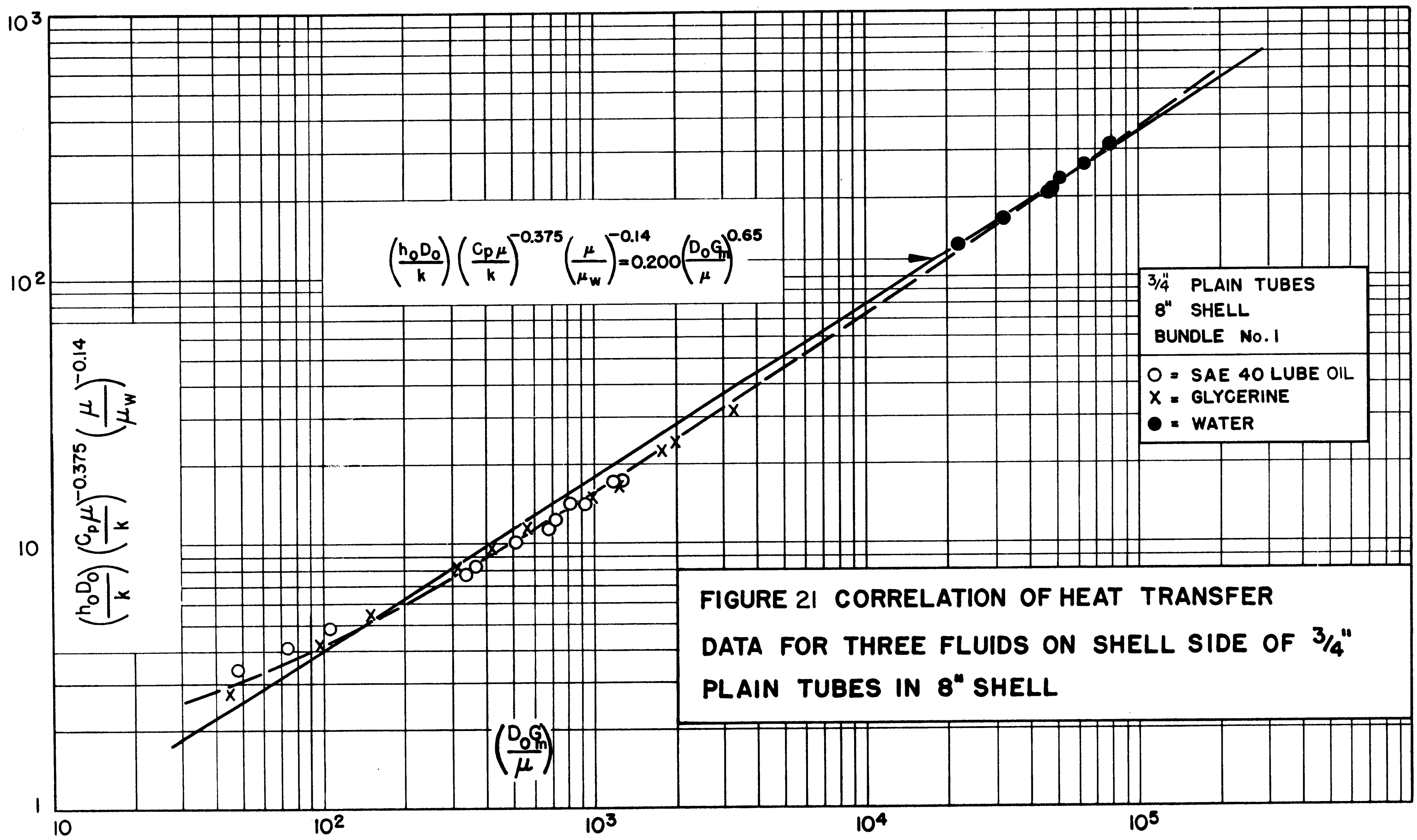


FIGURE 21 CORRELATION OF HEAT TRANSFER DATA FOR THREE FLUIDS ON SHELL SIDE OF 3/4" PLAIN TUBES IN 8" SHELL

selected from Fig. 19 and used as the exponent for the Reynolds number in the graph corresponding to Fig. 20. This required a second trial for the graphs similar to Figs. 20 and 21. In addition, it seemed appropriate to arrive at the same values of the exponents of each dimensionless group for all bundles if the data would permit. The final values for the Reynolds number exponent, 0.65, and the Prandtl number exponent, 0.375, were used for Figs. 20 and 21 rather than the exponents used in the first trials.

The exponent for the viscosity ratio was selected as 0.14, based on its acceptance for previous correlations[11,22,23] and the data supporting it.[24] However, the data of this research verify that the best correlation requires a variable exponent for this dimensionless group as well as for the Reynolds and Prandtl numbers.

The final correlations for plain-tube bundles 3 and 5 are given by Figs. 22 and 23, pages 104 and 105. They were obtained in a manner similar to that described for Fig. 21.

Finned Tubes

The data and correlations for finned tubes paralleled those for plain tubes.

In Equation (4), D, the diameter of the tube, becomes D_e, the equivalent outside diameter. It is defined as the outside diameter of a plain tube having the same inside diameter and the same weight of metal. The values of D_e are given in Table I, page 109. In computing cross-flow area, A_c, D_e is used for tubes, but in computing the window area, A_w, the diameter over the fins is used.

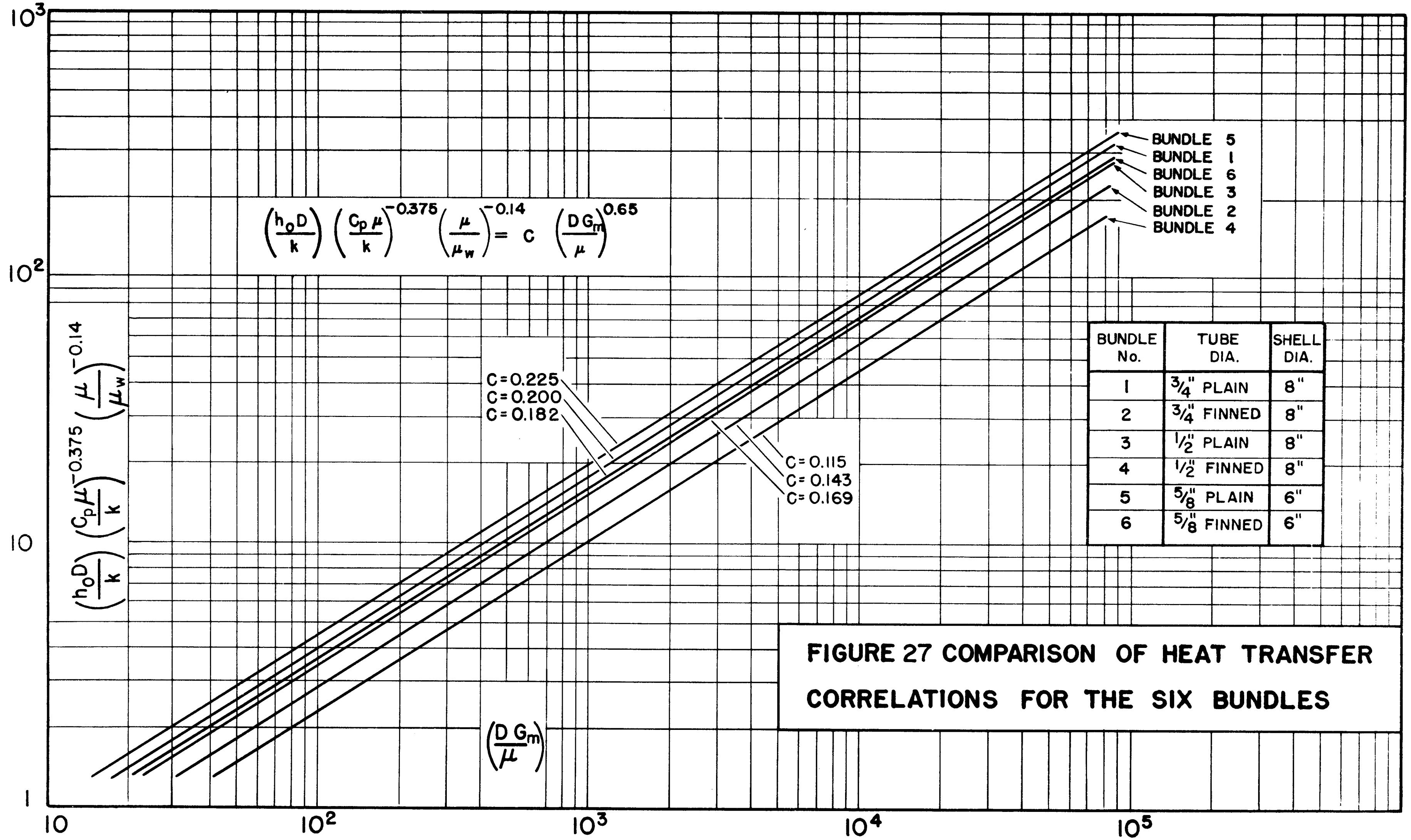
The final correlations of the heat-transfer data with finned tubes are given by Figs. 24, 25, and 26, pages 106-108. The exponents for the Prindtl number and Reynolds number have been taken as 0.375 and 0.65, the same as for the plain tubes. These values represent an average value for the several bundles, but the average values are no more restrictive in obtaining a fit between the data points and a single curve than is the assumption that the exponents are constant.

A comparison of all the data is made in Fig. 27. They are represented by a single equation (4a), except that the constant, C, in the equation varies with the exchanger bundle.

$$\frac{h_o D}{k} = C\left(\frac{DG_m}{\mu}\right)^{.65}\left(\frac{C_p\mu}{k}\right)^{.375}\left(\frac{\mu}{\mu_w}\right)^{.14} \tag{4a}$$

The three plain tube bundles could be represented by a single equation in which C = 0.197, with the data falling within +50, -26 per cent of the curve. Similarly, an equation with C = 0.147 represents all the data on finned-tube bundles within +50, -55 per cent.

In both plain and finned-tube runs for which the cooling-water temperatures were in the vicinity of 200°F, certain discrepancies were observed. The shell-side coefficients were often high as compared to any correlation, the heat balances were more erratic, and some Wilson plots appeared to be of different slope. Two possible explanations were considered, namely, incipient boiling of the water in the tubes and vaporization in the water orifice ahead of the exchanger. Calculations of tube-wall temperature indicated that it could not have reached the boiling point of water. The orifice readings reached some 24 inches of mercury for the high flow rate of a Wilson plot. These points, at high rates, scattered more than usual, but no definite trend could be found which would prove that erroneous flow rates for the water side were obtained at these high orifice differentials.



BUNDLE No.	TUBE DIA.	SHELL DIA.
1	3/4" PLAIN	8"
2	3/4" FINNED	8"
3	1/2" PLAIN	8"
4	1/2" FINNED	8"
5	5/8" PLAIN	6"
6	5/8" FINNED	6"

FIGURE 27 COMPARISON OF HEAT TRANSFER CORRELATIONS FOR THE SIX BUNDLES

As shown under the discussion of pressure drop, bundles 1, 3, and 2 gave the lower tube-side pressure drop and hence lower pressures at the orifice. It is quite possible that vaporization occurred in the orifice for some of the high-velocity—high-temperature conditions for these bundles.

Effect of Clearance

The effects of clearance between the tubes and baffle and between baffle and shell are likely to have a significant effect on the constants in Equation (4a) for the individual exchanger. A consideration of clearances is of no assistance in explaining the relative behavior of the plain-tube bundles but it does help to explain the differences between plain and finned-tube bundles.

The clearance between the tube and the baffle, for the assembled bundles, could not be measured readily, but the difference between the tube diameter and baffle hole for the plain-tube bundles is known to be less than 0.005 inch. Measurements of shell diameters, baffle diameters, and tube diameters are listed in Table I, page 109. Since each pair of bundles was made at the same time, it may be assumed that the baffle holes for the plain and corresponding finned-tube bundles are the same diameter. From tube diameter measurements, the difference in the clearances between finned and plain tubes then becomes the difference in the outside diameter of the tubes.

Tinker[22] stated that for a particular exchanger an additional 1/64 inch in the clearance between the tube and the baffle over the minimum mechanically feasible would reduce the shell-side coefficient by 10 per cent.

It happened that one finned-tube bundle (No. 6) had the same diameter over the fins as the duplicate plain-tube bundle (No. 5). In this case the ratio of the constants in Equation (4a) was .225/.182 = 1.24. This ratio is representative of the difference between plain and finned tubes for the same clearance. Part of this ratio is due to the fact that D_e is smaller for the finned tube than D_o for the plain tube, and hence the Nusselt numbers should not be the same at a given Reynolds number if the coefficients were the same. Part of the ratio represents leakage between the fins and might be related to the baffle thickness (.065 inch for all bundles) and fin spacing (.0525 inch).

For bundles 3 and 4 the ratio of the constants is 1.47. The clearance of the finned tube is .018 inch greater than that for the plain tube in this case. In addition, the peripheral length for leakage is 130 per cent greater for this pair of exchangers than for bundles 5 and 6 because of the increased number of tubes.

Bundles 1 and 2 have .016 inch extra clearance for the finned tube compared with the plain tube and 80 per cent more peripheral length than bundles 5 and 6. Bundles 1 and 2 have a ratio of constants of 1.40.

It appears that the concepts concerning the effect of clearance or leakage are substantiated by the differences in the pairs of bundles and that the major differences in convection coefficients between plain and finned tubes found are due to differences in leakage. For finned tubes there may be two causes for the leakage: the fins may not have the same outside diameter as the plain tube and there is an inherent flow of liquid in the helical space between the fin. It would appear that if plain and finned tubes had the same clearance of the first kind, the ratio of the constants in Equation (4a) might be the same for all pairs of bundles.

Effect of Tube and Shell Diameter

A comparison of the performance of the plain-tube exchangers gives no clue as to effects of tube and shell diameter beyond those observed in Equation (4) except that, over the range of dimensions used, they are not critical.

Bundle 5 gave the best performance but had a higher clearance between baffle and shell, and the tube diameter was undersize rather than oversize. The space between the shell-circle tubes and the shell was the lowest. The 5/8-inch tubes are intermediate between the 1/2-inch tubes of bundle 3 and 3/4-inch tubes of bundle 1. A difference between the 6-inch bundles and the 8-inch bundles which may be significant is that two more baffles were used for the 6-inch bundles. The effect of these baffles was noted in computing the flow area, A_m, as shown in Table VI, page 131. However, there is no assurance that the weighting of the flow areas for the baffled section and unbaffled section compensated for the difference.

Are Exponents for Dimensionless Groups Constants?

Equation (4) was derived by dimensional analysis which specifies the dimensionless groups but does not require that the exponents are constant. The value of using the equation as compared to a graph depends upon the assumption that the exponents are constant. As a practical matter, these exponents were used as constants in the correlation presented above. However, the exponents for the Reynolds number, the Prandtl number, and the viscosity ratio could well have been variables.

Curves have been drawn through the data for the plain-tube bundles in Figs. 21, 22, and 23, and these curves represent the data better than the straight lines. Between Reynolds numbers of 50 and 50,000, the slope of the curve varies from 0.41 to 0.70. For the finned-tube bundles, the curvature at low Reynolds numbers is not observed, but the water data, especially in Fig. 25, show a definitely higher slope than 0.65. These observations would indicate that extrapolation of these results to higher or lower Reynolds numbers might employ the curve on the graph rather than use the constant exponent. Also, it may be expedient to use one exponent for a range of Reynolds numbers and another for a different range.

The viscosity ratio was studied by Gardner and Siller[24], who observed that the exponent increased with increasing Reynolds number. The two lines for water in Fig. 25 were run at different temperatures. In Fig. 22 also, a scattering of the water data results from runs at temperature differences. These differences in the water data at high Reynolds numbers could be minimized by employing an exponent for the viscosity ratio of about 0.8, as suggested by Gardner and Siller.[24]

To bring the water data to a single curve, similar arguments could be advanced for a variable exponent on the Prandtl number instead of on the viscosity ratio.

Correlation with Exponents from Literature

In order that comparisons might be made on the basis of the exponents used by Donohue, Tinker, and Short, the experimental data were plotted with these exponents according to Equation (4b).

$$\frac{hD}{k} = C'\left(\frac{DG_m}{\mu}\right)^{.60}\left(\frac{C_p\mu}{k}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{.14} \qquad (4b)$$

This changes the exponent for the Prandtl number from 0.375 to 0.333 and the exponent for the Reynolds number from 0.65 to 0.60. Fig. 28 is an example of the correlation with these exponents from the literature for

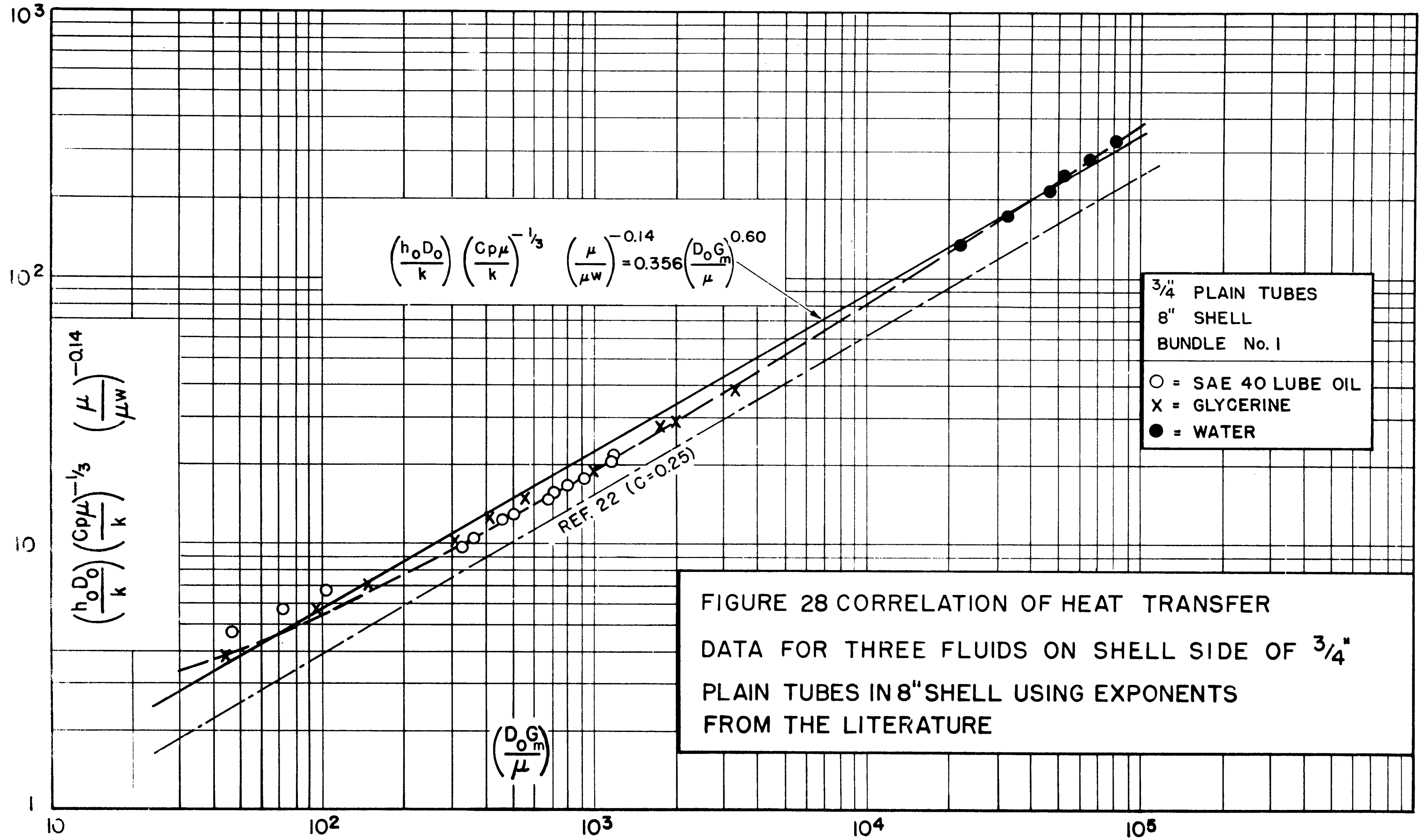


FIGURE 28 CORRELATION OF HEAT TRANSFER DATA FOR THREE FLUIDS ON SHELL SIDE OF 3/4" PLAIN TUBES IN 8" SHELL USING EXPONENTS FROM THE LITERATURE

bundle 1. It may be observed that the exponents derived in this study and used in Fig. 21 yield a slightly better fit of the data to the line than do the exponents from the literature.

A similar correlation for the other bundles gave the constants C'(shown in Table VII for Equation (4b)) to facilitate comparisons between these results and others based on this form of the equation.

TABLE VII

CONSTANTS IN CONVECTION COEFFICIENT EQUATION

Bundle No.	C' in Eq. (4b)	C in Eq. (4a)
1 (Plain)	.356	.200
2 (Finned)	.255	.143
3 (Plain)	.302	.169
4 (Finned)	.205	.115
5 (Plain)	.400	.225
6 (Finned)	.324	.182

The convection coefficient equation recommended by Donohue for commercial exchangers with bored shells is of the form (4b) with C' equal to 0.25. Donohue's equation is plotted in Fig. 28. It may be observed that the poorest plain-tube performance for the test exchangers gave 20 per cent higher shell-side coefficients than does the recommended literature value, while the best plain-tube exchanger in these tests gave coefficients 60 per cent higher than recommended by Donohue.

RECOMMENDED SHELL-SIDE COEFFICIENTS FOR FINNED TUBES

The data are not complete in the sense that they do not provide coefficients for other exchanger designs for finned tubes. The basic data and calculations are presented in detail so that engineers in the heat-exchanger industry can arrive at their estimate of the best coefficients to use in the light of the data obtained. However, there are those who would like to take recommended coefficients and proceed with the design.

The mechanical design of the exchangers, including clearances, baffle arrangement, etc., is important to the extent that no accurate prediction can be made without knowledge of these factors. In the case of exchangers of standard design for which the performance with plain tubes is known, these mechanical features are evaluated. The finned-tube performance can be given in terms of the plain-tube performance equation.

Finned Tubes When Plain-Tube Performance is Known

For plain-tube exchangers with known performance, the value of C' is known for Equation (4b) and the equation may be used to compute convection coefficients

$$\frac{hD_o}{k} = C' \left(\frac{D_o G_m}{k}\right)^{.6} \left(\frac{C_p \mu}{k}\right)^{.33} \left(\frac{\mu}{\mu_w}\right)^{.14} \qquad (4b)$$

For finned-tube exchangers of the same design and clearances, this same equation may be used with D_e, the equivalent diameter, replacing D_o, and with a new value for C'. It is recommended that:

$$C' \text{ (finned tube)} = C' \text{ (plain tube)} \times 0.7$$

This factor, 0.7, is the average of the values observed for bundles 1 and 2 and bundles 3 and 4. It assumes that the height of fins on the tubes are near the lower limit of the specifications. If assurance could be given that the fin height is close to the plain-tube diameter this factor could well be as high as 0.8. Due to velocity and diameter changes with finned tubes as compared to plain tubes it is necessary to compute the convection coefficient; one cannot apply the above factor directly to the coefficient for satisfactory results.

Plain Tubes with Shell-Circle Design

For exchangers similar to those of this study, in which the shell circle was filled with tubes and relatively close tolerances were used, the absolute values of C or C' in Equations (4a) or (4b) can be used. For plain tubes from 1/2 to 1 inch in diameter, Equation (4c) may be used, as follows:

$$\frac{hD_o}{k} = .19\left(\frac{D_o G_m}{\mu}\right)^{.65}\left(\frac{C_p\mu}{k}\right)^{.375}\left(\frac{\mu}{\mu_w}\right)^{.14} \qquad (4c)$$

If desired, the corresponding form of Equation (4b) may be used:

$$\frac{hD_o}{k} = .34\left(\frac{D_o G_m}{\mu}\right)^{.60}\left(\frac{C_p\mu}{k}\right)^{.33}\left(\frac{\mu}{\mu_w}\right)^{.14} \qquad (4d)$$

Finned Tubes for Shell-Circle Design

For finned tubes in similar exchangers, the coefficients are given by the following forms of Equation (4):

$$\frac{hD_e}{k} = .13\left(\frac{D_e G_m}{\mu}\right)^{.65}\left(\frac{C_p\mu}{k}\right)^{.375}\left(\frac{\mu}{\mu_w}\right)^{.14} \qquad (4e)$$

$$\frac{hD_e}{k} = .23\left(\frac{D_e G_m}{\mu}\right)^{.60}\left(\frac{C_p\mu}{k}\right)^{.33}\left(\frac{\mu}{\mu_w}\right)^{.14} \qquad (4f)$$

These equations apply for tubes described in this report with 19 nominal fins per inch and from 1/2 to 1 inch in diameter.

The constants in Equations (4e) and (4f) are based on the assumption that the height of the fins on the tubes are near the lower limit of the specifications and will have large clearances. In case the fin height is such that the diameter over the fin equals the diameter of the plain end, the values for C and C' may rise to 0.15 for Equation (4e) and 0.27 for Equation (4f) for the finned tubes.

It is appreciated that no data are available for 1-inch tubes, but there is no evidence to indicate that a significant difference may be expected.

COMPARISON OF PLAIN AND FINNED-TUBE PERFORMANCE

The correlations of heat-transfer coefficients permit quantitative calculations to compare finned tubes with plain tubes, but the relative amounts of heat transfer are not readily discernible. Comparisons will be made between the coefficients and between the heat transfer for the bundles identical except for the tubes.

Heat Transfer Per Degree Temperature Difference

The heat transferred by the plain-tube bundle could be compared directly with the heat transferred by the finned-tube bundle if the temperature level and temperature differences were the same. Since data are not available which have exactly the same temperature difference for the two exchangers, the heat transfer per degree temperature differance may be compared, provided the temperature levels and hence the physical properties of the fluids are essentially the same. The comparisons with water would show the least increase in heat transfer for finned tubes due to the high coefficients on the shell side, while the comparisons for oil will show the greatest benefit from the finned tubes. Bundles 3 and 4 will show the least improvement for the finned tubes with oil, and bundles 5 and 6 the most improvement due to the nature of the clearances between the tube and the baffle, as explained in connection with Fig. 27.

The tests with water on the shell side gave the highest rates of heat transfer. From the clearances discussed previously, bundles 1 and 2 or 3 and 4 would be expected to show the least improvement for finned tubes. However, due to the high heat-transfer rates, the conductivity of the tube metal becomes significant. Therefore, the copper tubes in bundles 3 and 4 show the greatest improvement for the finned tubes as compared to the plain tubes. Fig. 29 is a comparison of the performance of bundles 3 and 4, which show from 57 to 60 per cent more heat transfer for the finned bundle at the same mass rates for water on the tube side.

The least improvement for finned tubes is with water on the shell side of bundles 1 and 2, having Admiralty tubes. Fig. 30 gives the performance under specified conditions, with an increase in heat transfer from 11 to 18 per cent for the finned-tube bundle as compared to the plain-tube bundle.

Fig. 31, for oil on the shell side of exchanger bundles 5 and 6, shows the maximum benefit found for the finned tubes. The finned tubes more than doubled the heat transfer at the higher water velocities. Figs. 32 and 33 show typical increases in heat transfer of from 60 to 70 per cent when oil is the fluid and the clearances between the tubes and the baffle are the maximum to be encountered.

Overall and Film Coefficients

The overall coefficients of heat transfer may be compared for a given temperature level of the fluid. If the actual outside area is used, it should be remembered that the finned-tube bundles have from 2.06 to 2.77 times as much surface as the plain-tube bundles. On the other hand, if the inside areas are used the finned-tube bundles have only from 0.78

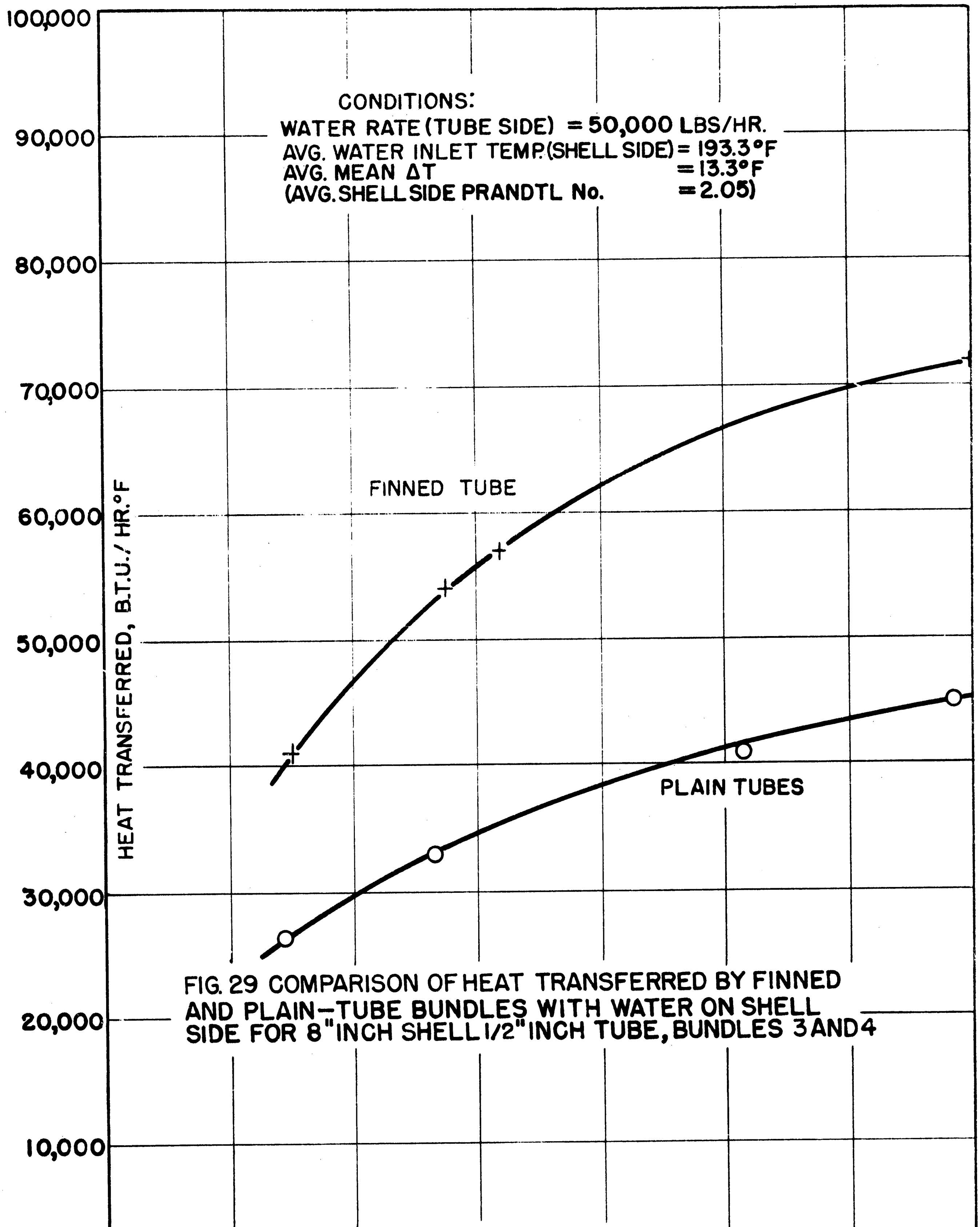


FIG. 29 COMPARISON OF HEAT TRANSFERRED BY FINNED AND PLAIN-TUBE BUNDLES WITH WATER ON SHELL SIDE FOR 8" INCH SHELL 1/2" INCH TUBE, BUNDLES 3 AND 4

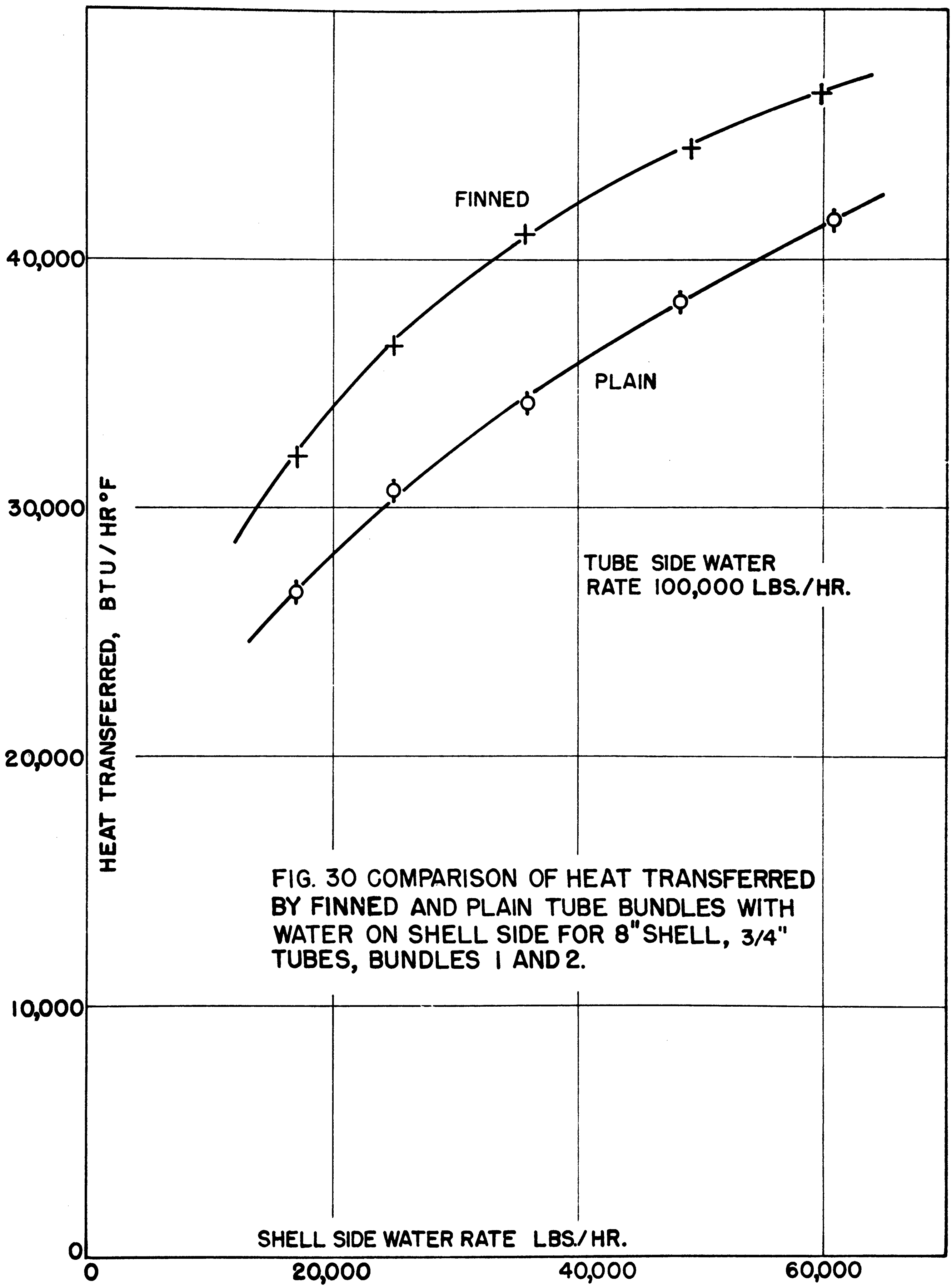


FIG. 30 COMPARISON OF HEAT TRANSFERRED BY FINNED AND PLAIN TUBE BUNDLES WITH WATER ON SHELL SIDE FOR 8" SHELL, 3/4" TUBES, BUNDLES 1 AND 2.

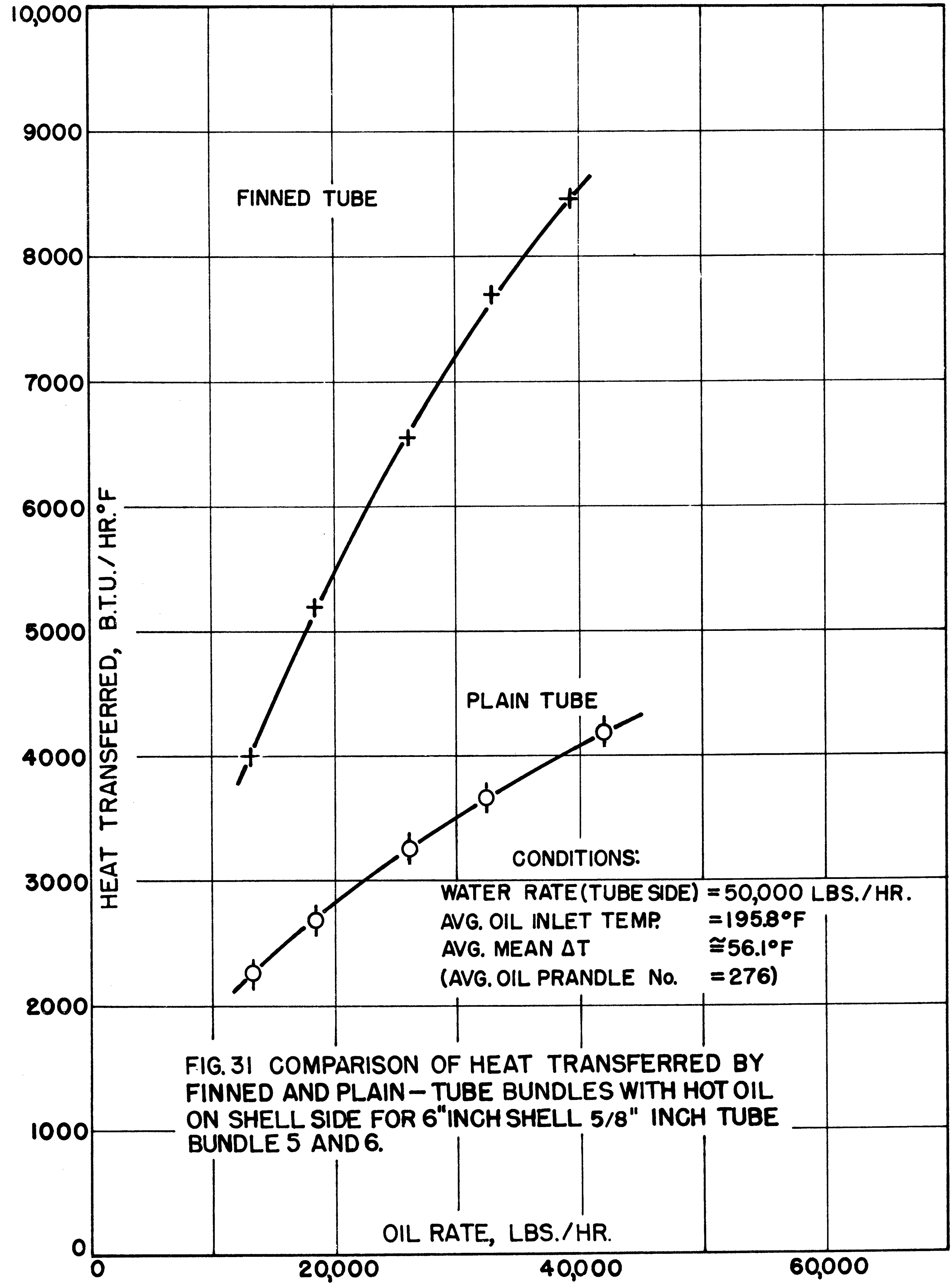


FIG. 31 COMPARISON OF HEAT TRANSFERRED BY FINNED AND PLAIN–TUBE BUNDLES WITH HOT OIL ON SHELL SIDE FOR 6" INCH SHELL 5/8" INCH TUBE BUNDLE 5 AND 6.

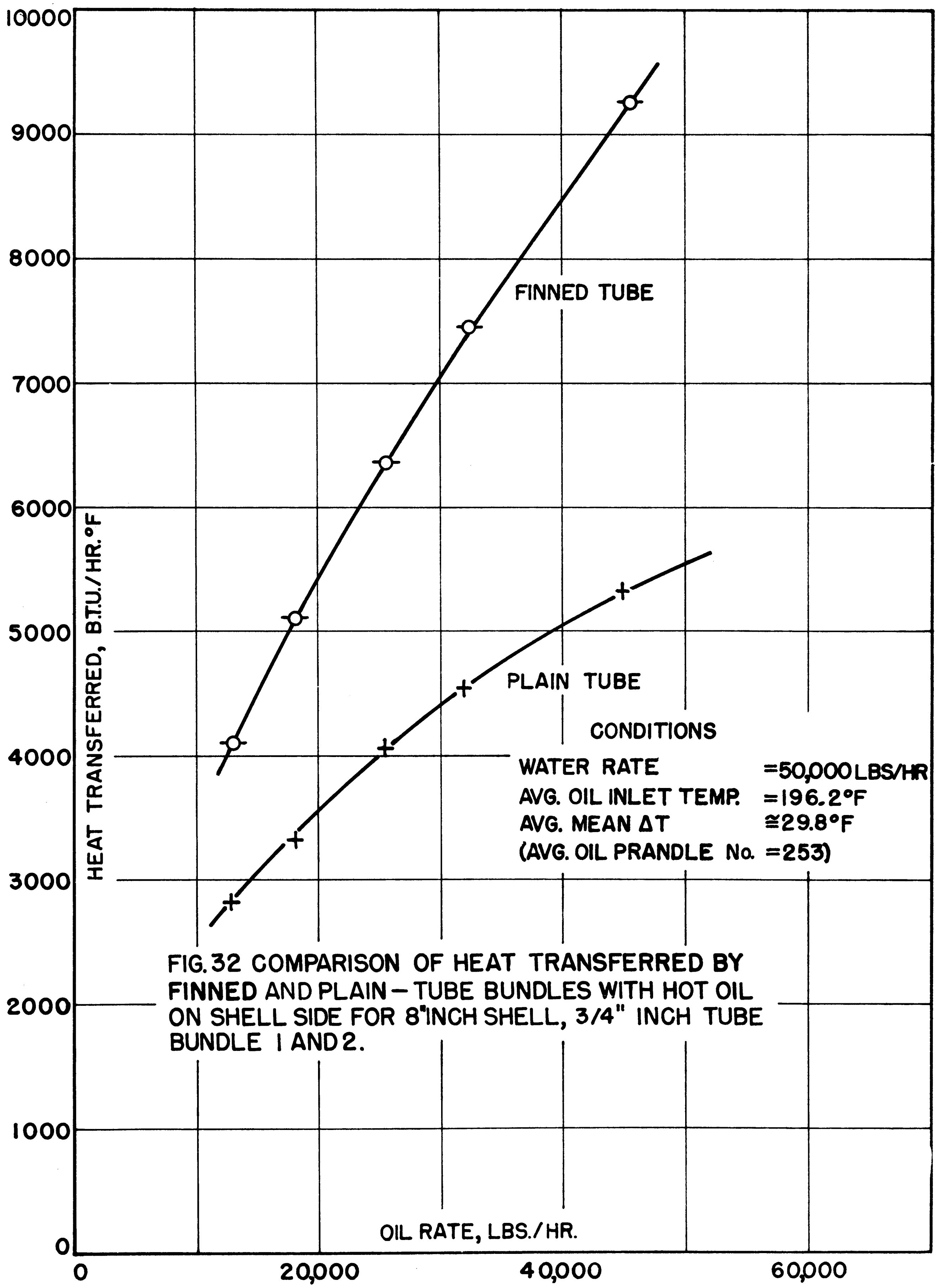


FIG. 32 COMPARISON OF HEAT TRANSFERRED BY FINNED AND PLAIN – TUBE BUNDLES WITH HOT OIL ON SHELL SIDE FOR 8" INCH SHELL, 3/4" INCH TUBE BUNDLE 1 AND 2.

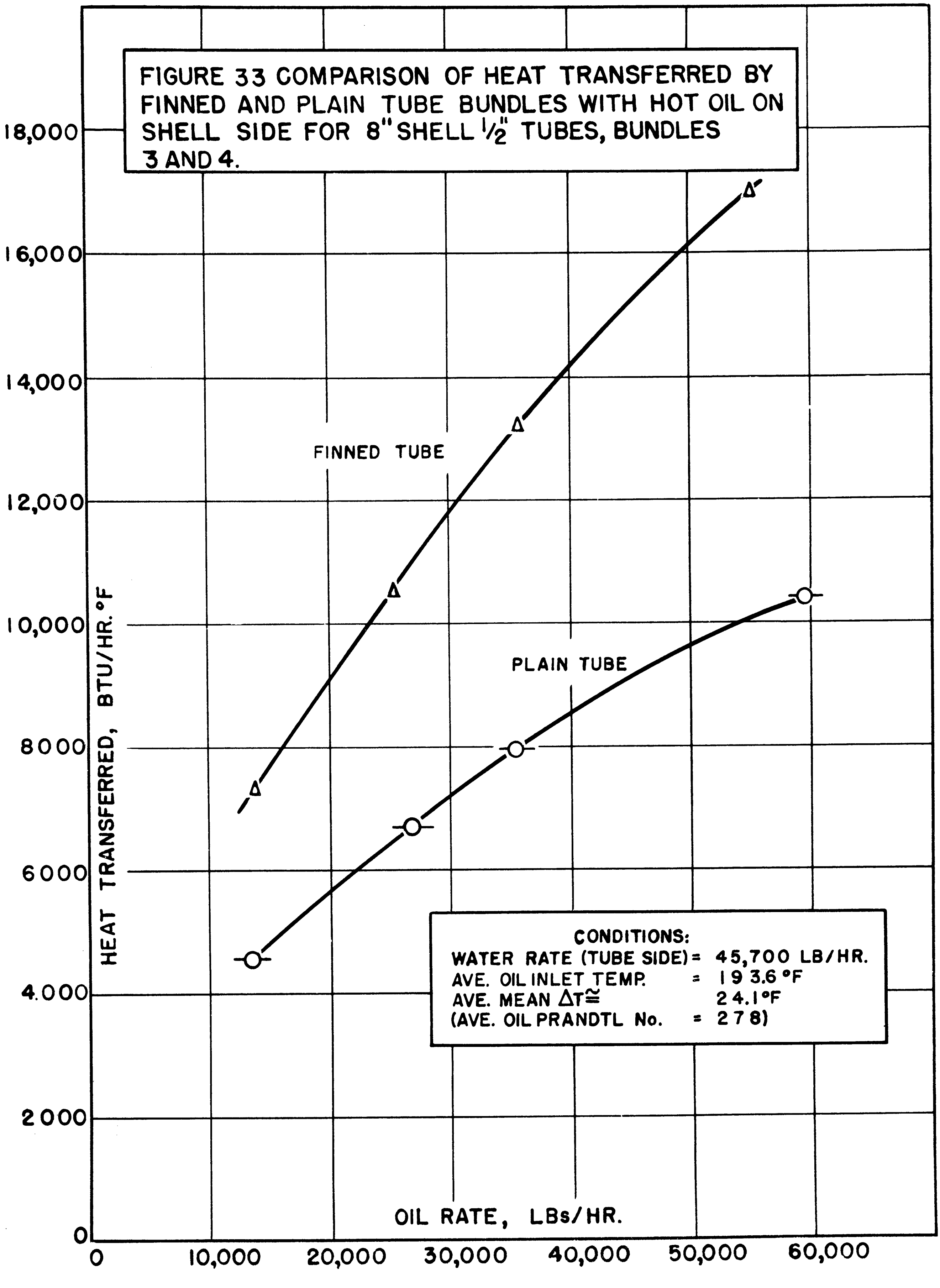


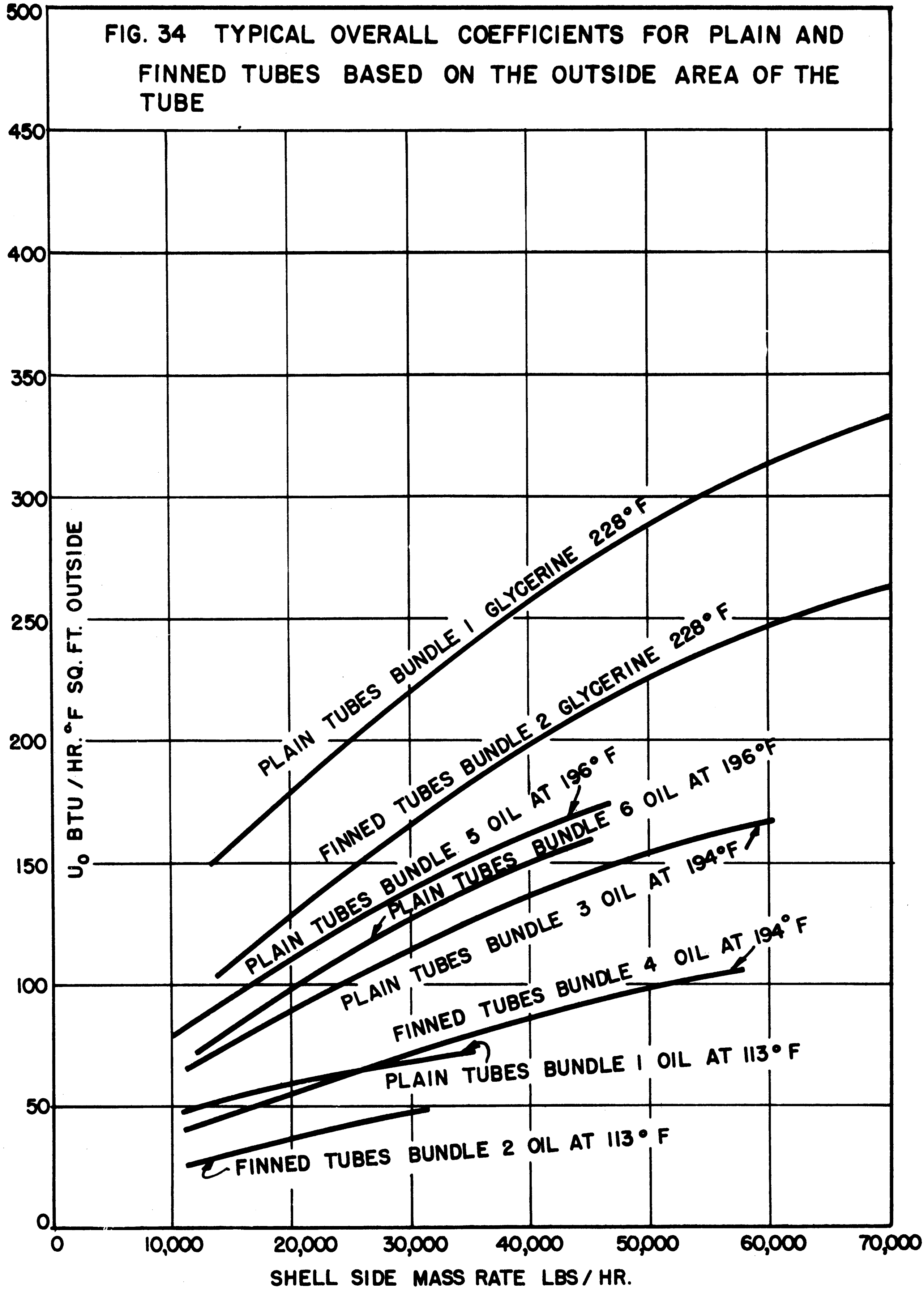
FIGURE 33 COMPARISON OF HEAT TRANSFERRED BY FINNED AND PLAIN TUBE BUNDLES WITH HOT OIL ON SHELL SIDE FOR 8" SHELL 1/2" TUBES, BUNDLES 3 AND 4.
FINNED TUBE
PLAIN TUBE
CONDITIONS:
WATER RATE (TUBE SIDE) = 45,700 LB/HR.
AVE. OIL INLET TEMP. = 193.6 °F
AVE. MEAN ΔT ≅ 24.1 °F
(AVE. OIL PRANDTL No. = 278)
HEAT TRANSFERRED, BTU/HR. °F
OIL RATE, LBs/HR.
0
2000
4.000
6000
8000
10,000
12,000
14,000
16,000
18,000
0
10,000
20,000
30,000
40,000
50,000
60,000

to 0.65 times as much inside surface as the plain-tube bundles. Figs. 34 and 35 show typical overall coefficients for glycerine and oil, based on the outside and on the inside areas of the tubes.

The magnitude of the overall coefficients observed is worthy of consideration. For example, with water on the shell side of bundle 5, run 16, an overall coefficient of 1222 Btu per (Hr)(°F)(sq ft) was observed. This coefficient requires film coefficients of 2500 or more and indicates that fouling was extremely small or entirely absent. For finned tubes, run 28 gives an overall coefficient of 792 Btu per (hr)(°F)(sq ft outside). This coefficient, based on the outside fin area, requires film coefficient of about 3000.

The convection coefficients on the shell side are of interest. These coefficients are compared on the basis of the outside area. For the finned tubes, the effective outside area is used rather than the actual outside area since any inefficiency in the fin should not be allowed to detract from the convection coefficient. For water on the shell side of Admiralty finned tubes, the fin efficiency was as low as 70 per cent, and the effective area is this fraction of the actual area for the fins, which constitute 80 per cent of the surface, i.e., the effective area may be only 76 per cent of the actual area. For oil and glycerine, the fin efficiencies seldom dropped below 95 per cent, and the actual outside area approximates the effective area. Fig. 36 compares typical convection coefficients plotted as a function of mass flow rate on the shell side. When convection coefficients for finned tubes are the same as for plain tubes, it follows that the fluid between the fins must be interchanged with fluid in the main stream as rapidly as the fluid adjacent to a plain-tube wall.

FIG. 34 TYPICAL OVERALL COEFFICIENTS FOR PLAIN AND FINNED TUBES BASED ON THE OUTSIDE AREA OF THE TUBE



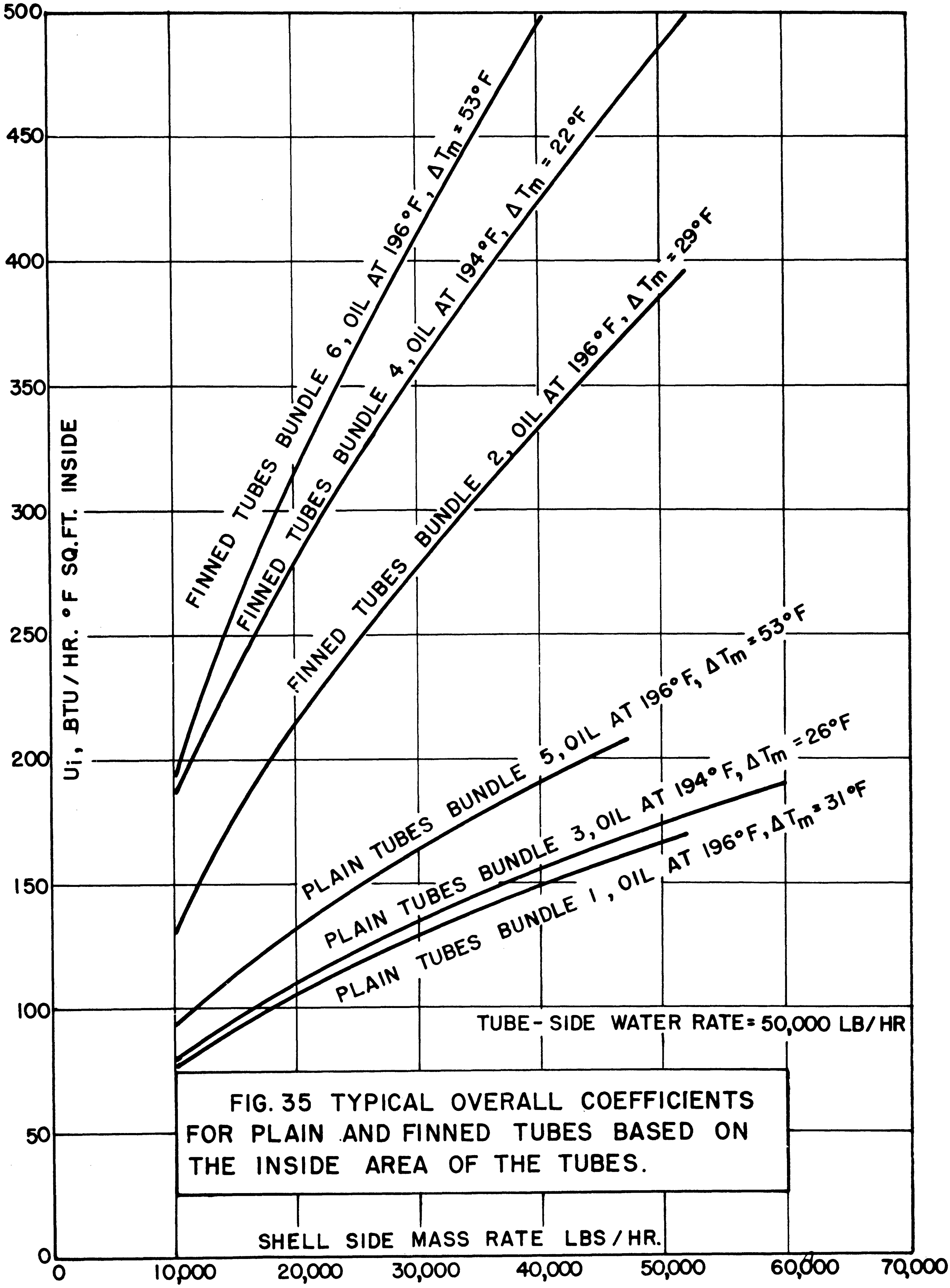


FIG. 35 TYPICAL OVERALL COEFFICIENTS FOR PLAIN AND FINNED TUBES BASED ON THE INSIDE AREA OF THE TUBES.

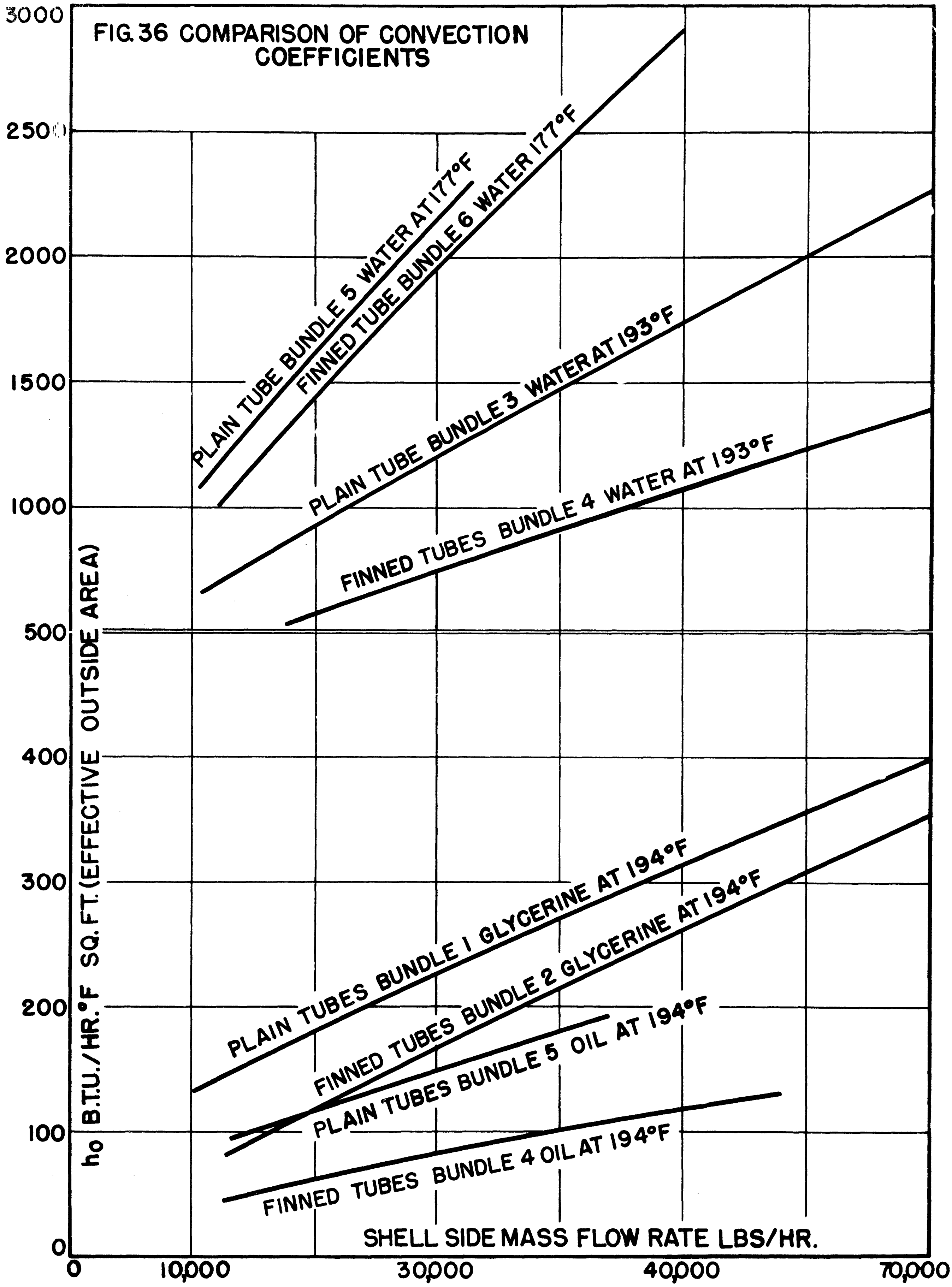


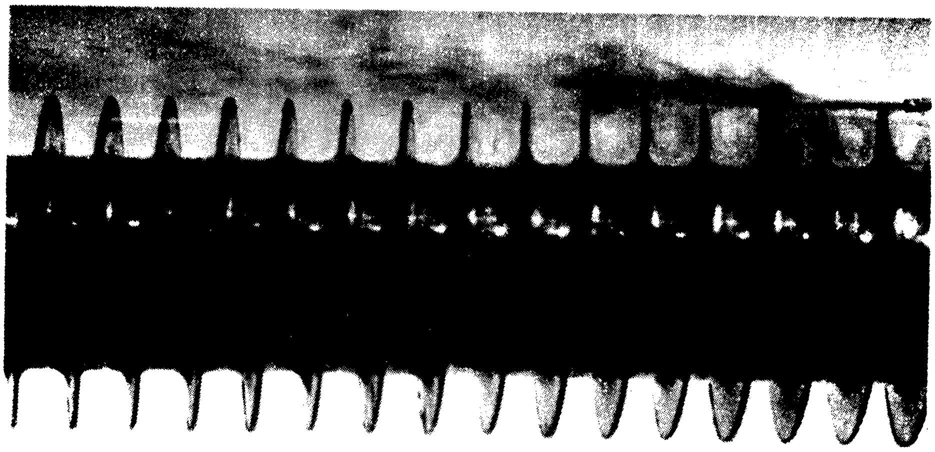
FIG. 36 COMPARISON OF CONVECTION COEFFICIENTS
PLAIN TUBE BUNDLE 5 WATER AT 177°F
FINNED TUBE BUNDLE 6 WATER 177°F
PLAIN TUBE BUNDLE 3 WATER AT 193°F
FINNED TUBES BUNDLE 4 WATER AT 193°F
PLAIN TUBES BUNDLE 1 GLYCERINE AT 194°F
FINNED TUBES BUNDLE 2 GLYCERINE AT 194°F
PLAIN TUBES BUNDLE 5 OIL AT 194°F
FINNED TUBES BUNDLE 4 OIL AT 194°F
ho B.T.U./HR. °F SQ. FT. (EFFECTIVE OUTSIDE AREA)
SHELL SIDE MASS FLOW RATE LBS/HR.
3000
2500
2000
1500
1000
500
400
300
200
100
0
0
10,000
30,000
40,000
70,000

An explanation for the similarity between the convection coefficients for finned and plain tubes may be found in the paper by Knudsen and Katz.[25] Knudsen studied the heat transfer and fluid flow in annuli containing finned tubes. He observed that the fluid enters the space between the fins and forms eddies, as shown in Fig. 37. These eddies appear to be responsible for replacing the fluid between the fins when flow is turbulent and parallel to the tube. In cross flow, it is understandable that the fluid enters the space between the fins.

The results of this study indicate that at very low Reynolds numbers the plain-tube coefficients do not decrease as rapidly as the finned-tube coefficients because the fluid probably does not enter the space between the fins at these low velocities. On the other hand, at very high Reynolds numbers, the coefficients for the finned tubes may exceed those for plain tubes because of the extra turbulence caused by the fins.

CORRELATION OF PRESSURE-DROP DATA

Pressure-drop data were observed for the shell-side fluid on all runs and recorded in Table IV, page 115. For the same fluid rate and temperatures, the pressure drop for the finned-tube bundle was less than for the plain-tube bundle. This statement applies for all three fluids and the three pairs of bundles. Typical pressure-drop data for the shell-side fluid are plotted in Fig. 38. It is appreciated that the flow rates in the exchangers under test exceeded flow rates normally used in commercial operation. Mass rates up to 1.5 million pounds per sq ft per hour were obtained with pressure losses up to 12 psi. Flow rates in large exchangers with a pressure drop limit of 12 psi would attain mass rates of 200,000 to 500,000 lbs per sq ft per hour. The



THIS PHOTOGRAPH IS OF DYE INJECTED IN WATER FLOWING IN THE ANNULUS BETWEEN A HELICAL FINNED TUBE AND A PLASTIC OUTER TUBE

FIG. 37 NATURE OF FLOW BETWEEN FINS FROM KNUDSEN

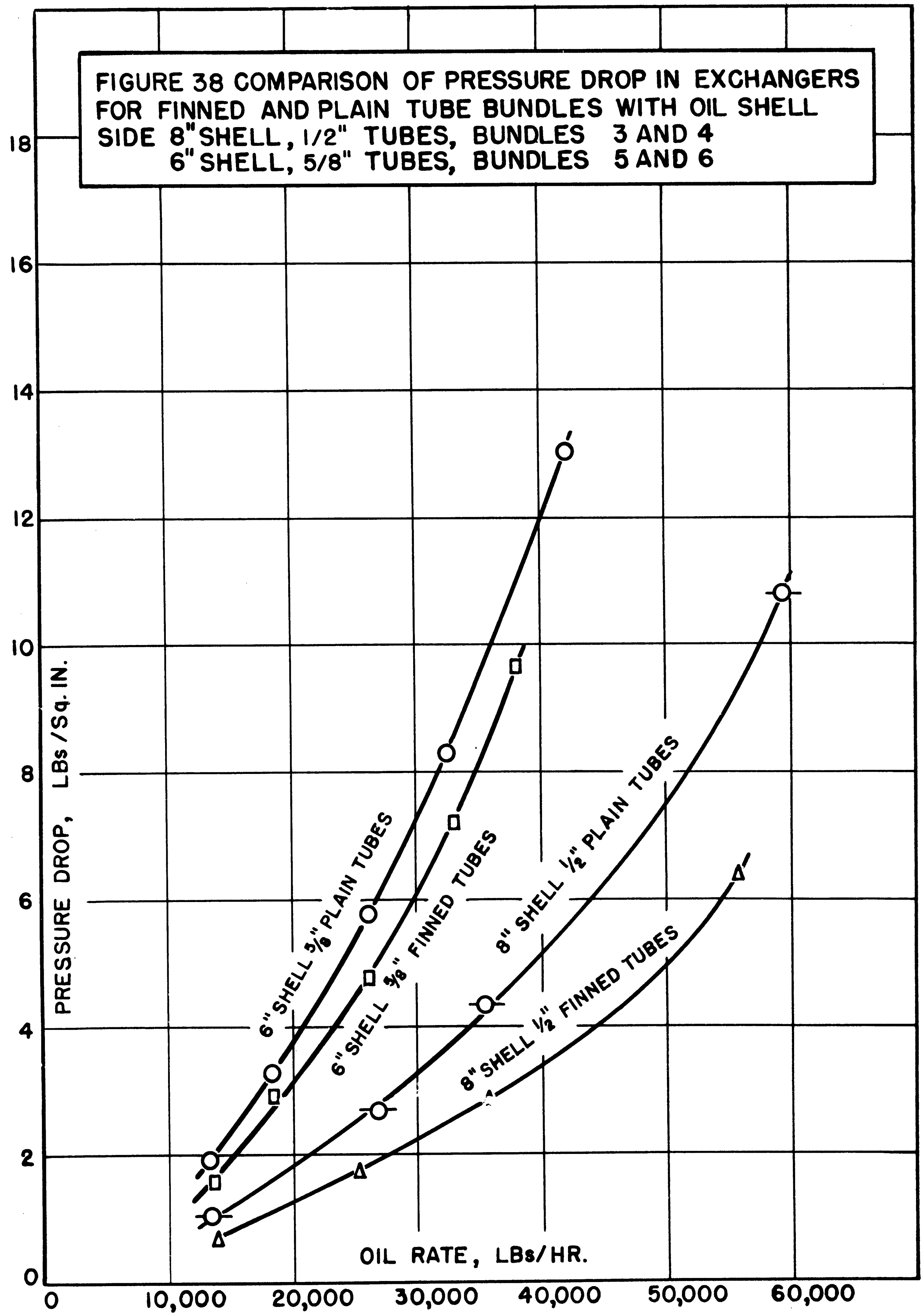


FIGURE 38 COMPARISON OF PRESSURE DROP IN EXCHANGERS FOR FINNED AND PLAIN TUBE BUNDLES WITH OIL SHELL SIDE 8" SHELL, 1/2" TUBES, BUNDLES 3 AND 4
6" SHELL, 5/8" TUBES, BUNDLES 5 AND 6
PRESSURE DROP, LBs/Sq. IN.
OIL RATE, LBs/HR.
6" SHELL 5/8" PLAIN TUBES
6" SHELL 5/8" FINNED TUBES
8" SHELL 1/2" PLAIN TUBES
8" SHELL 1/2" FINNED TUBES
0
2
4
6
8
10
12
14
16
18
0
10,000
20,000
30,000
40,000
50,000
60,000

greatest difference for finned and plain tubes occurs between bundles 3 and 4 because these two differed the most in their clearances, while bundles 5 and 6 have the least difference in pressure drop, corresponding to the small difference in clearance.

Reference to Figs. 6 and 7 will show that there is more space for the shell-side fluid to flow in cross flow in the case of the finned tubes as compared to the plain tubes. Also, the inherent leakage through the helical space between the fins at the baffle will reduce the pressure drop for the finned-tube bundle.

An analysis of pressure-drop data divides the total pressure drop into the following items[22,23]: (1) enlargement and contraction loss at the nozzles, (2) loss during flow through baffle windows, and (3) friction loss during flow across tubes. The enlargement and contraction loss for the nozzles was taken as the kinetic energy of the fluid in the exit nozzle; the inlet kinetic energy was assumed to be dissipated. The loss during flow through a window was assumed to follow the equation of Donohue[22]:

$$(P_1 - P_2)_w = \frac{2.9\ G_w^2}{10^{13} \text{sp gr}} \tag{7}$$

where $(P_1 - P_2)_W$ = pressure drop per baffle window, lbs/sq in.

G_W = mass velocity at A_W, lbs/(hr)(sq ft)

sp gr = specific gravity referred to water at 60°F or the density in g/cc

Short[23] related pressure loss at baffle windows to the velocity squared and a function of the product of the Reynolds number through the window and the square root of the Prandtl number. Even for this more complex relationship, considerable scattering occurred for the data obtained by Short. Equation (7) is adopted because of its simplicity and the demonstration by Donohue that it was essentially as good as the more complex relationship. The total baffle loss is the pressure drop per baffle times the number of baffle windows.

The friction loss of flow across tubes may be expected to follow a friction-factor curve such as was used by Donohue[22]. The friction factor is related to pressure drop as follows:

$$(P_1 - P_2)_c = \frac{1.07 \; f \; n \; G_c^{\;2}}{10^9 \; g_c \rho} \left(\frac{\mu}{\mu_w}\right)^{-0.14} \tag{8}$$

where f = friction factor

n = the minimum number of rows of tubes the fluid passes in flowing from one window to the next

G_c = mass velocity at A_c in cross flow, lbs/(hr)(sq ft)

ρ = fluid density, lbs/cu ft

g_c = 32.2, conversion factor

$(P_1 - P_2)_c$ = pressure drop per baffle space due to friction of cross flow in lbs/sq in.

μ/μ_w = viscosity ratio between bulk temperature and wall temperature

The friction factor is a function of the Reynolds number on the shell side.

The experimental data were used to evaluate f, the friction factor, in the following manner. The outlet-nozzle kinetic energy and the total baffle loss were computed as described above. The sum of these two pressure drops was subtracted from the experimental pressure drop to obtain the pressure loss due to cross flow, $(P_1 - P_2)_c$. This value was substituted in Equation (8) for $(P_1 - P_2)_c$ and the friction factor, f, was computed for each run. These friction factors are plotted in Figs. 39, 40, and 41 for the three pairs of exchangers.

The friction factors are above those reported by Donohue[22], indicating the effect of the clearance between the tubes on the shell-circle and the shell for the exchangers in these tests.

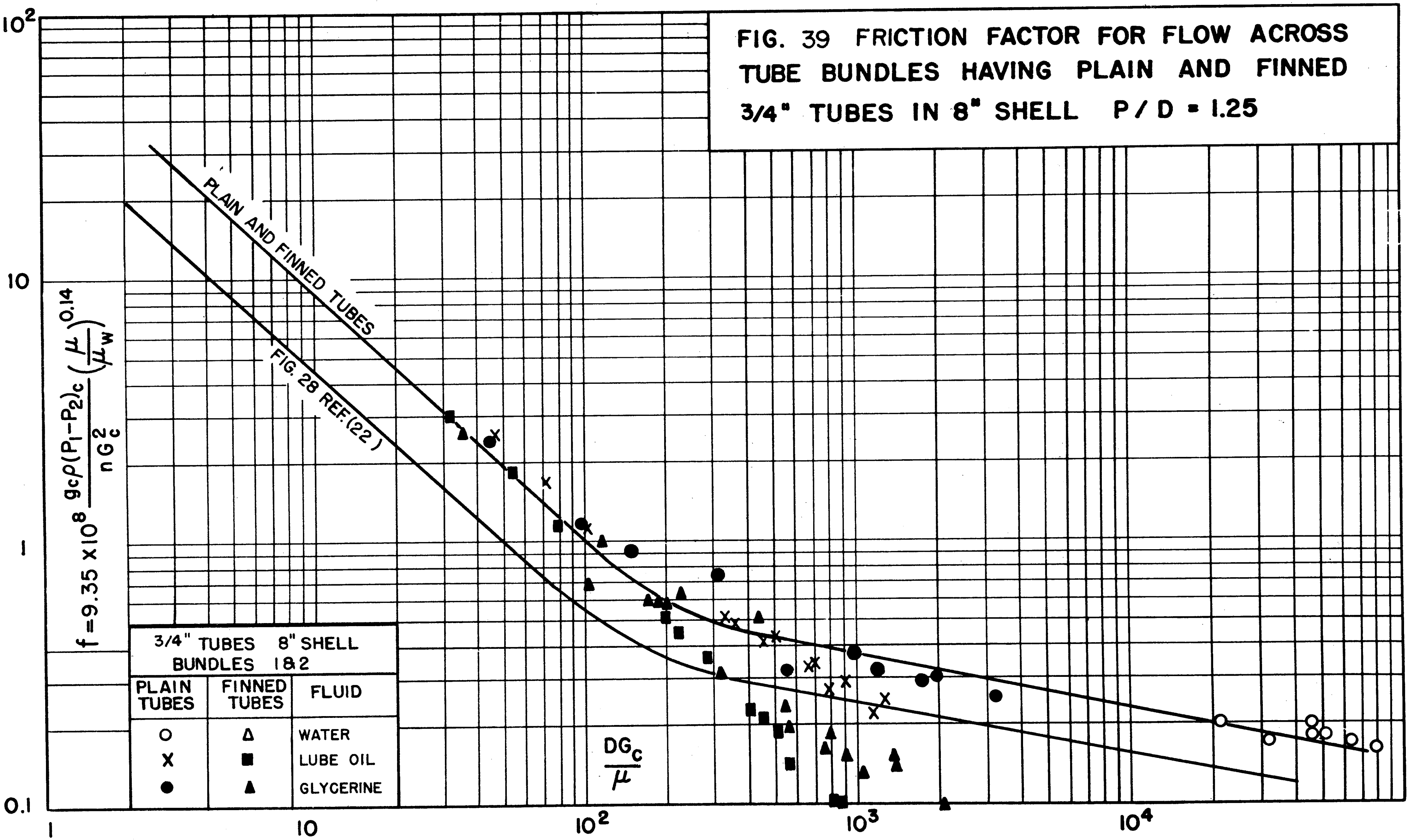


FIG. 39 FRICTION FACTOR FOR FLOW ACROSS TUBE BUNDLES HAVING PLAIN AND FINNED 3/4" TUBES IN 8" SHELL P/D = 1.25

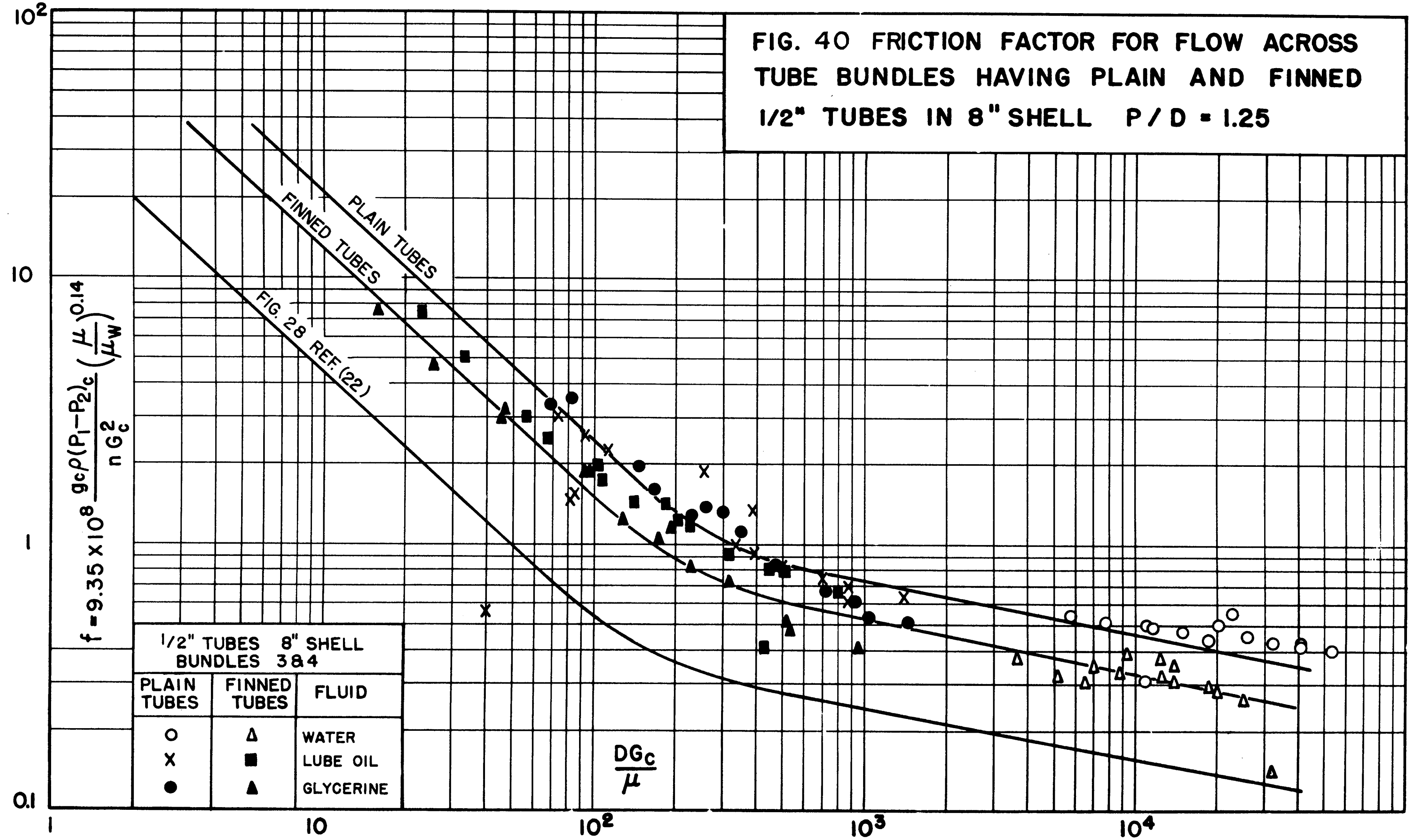


FIG. 40 FRICTION FACTOR FOR FLOW ACROSS TUBE BUNDLES HAVING PLAIN AND FINNED 1/2" TUBES IN 8" SHELL P / D = 1.25

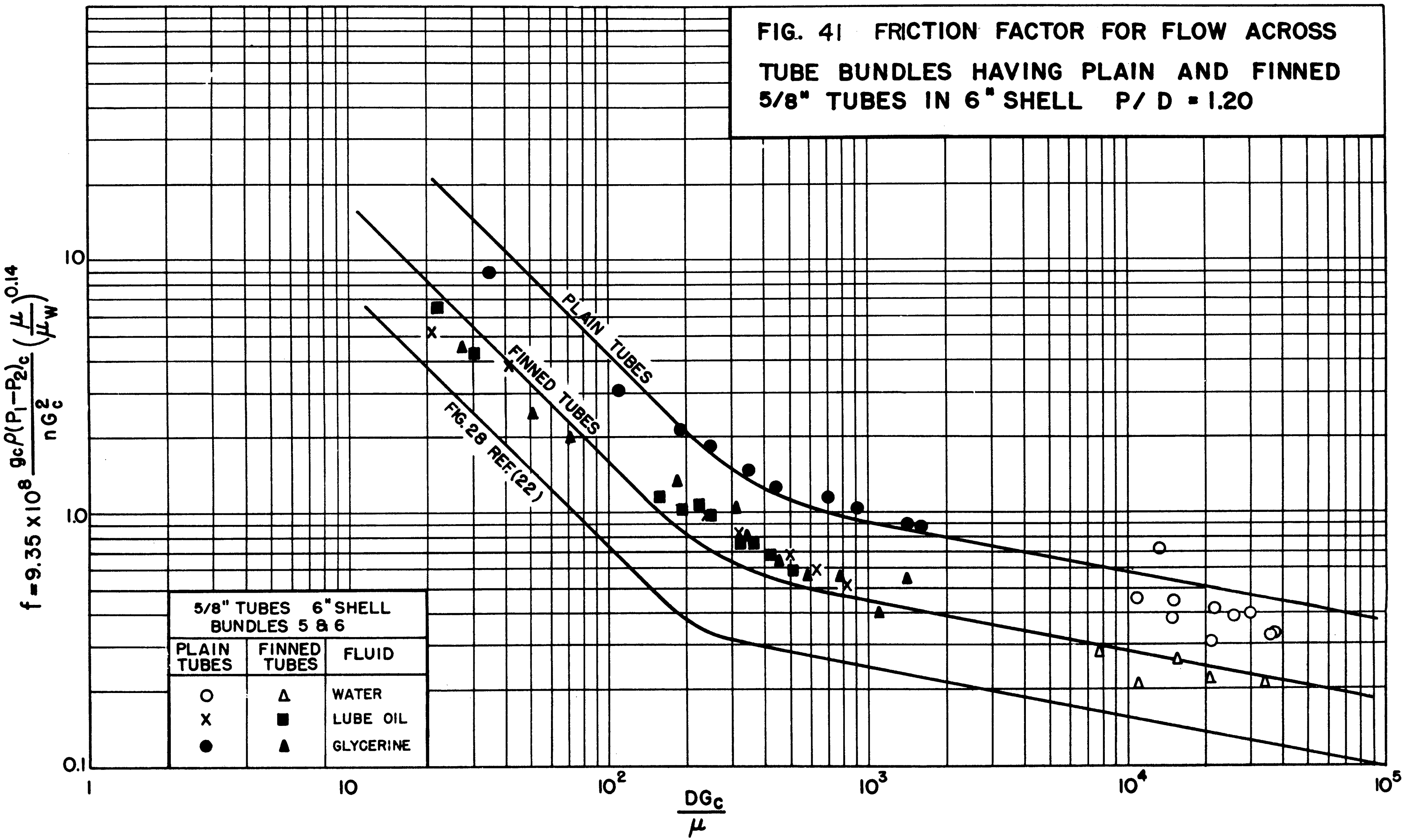


FIG. 41 FRICTION FACTOR FOR FLOW ACROSS TUBE BUNDLES HAVING PLAIN AND FINNED 5/8" TUBES IN 6" SHELL P/D = 1.20

The symmetry of the curves from this study and that of Donohue would indicate that the equation for the window loss was suitable for the test exchangers. Fig. 39 is an exception in that the finned-tube exchanger has a lower pressure drop than the curve of Donohue,[22] and in some cases the experimental pressure drop was less than the sum of the kinetic energy of outlet nozzle and the computed window loss. No explanation for this can be found, but it should be noted that these bundles, 1 and 2, had a smaller window than the other bundles.

These studies indicate that when a friction factor curve is available for an exchanger with plain tubes, this curve would be conservative for computing the pressure drop for the same exchanger equipped with finned tubes.

On the tube side, the pressure drop for the cooling water was observed, but it was not recorded in Table IV. Fig. 42 gives typical pressure-drop values for the water passing through the tubes at several velocities at mean water temperatures in the range from 155 to 165°F.

ECONOMICS OF FINNED TUBES FOR SHELL AND TUBE EXCHANGERS

The only reliable procedure for evaluating the economics of finned tubes is to make designs and cost estimates of exchangers with plain tubes and with finned tubes for a given heat-transfer duty. The information required to make this comparison is (1) methods of sizing exchangers for a given duty and pressure-drop limitation and (2) methods of determining the costs of the components of exchangers. The data and correlations reported in this study serve as a basis for sizing exchangers. The Exchangers Price Section of the book, Alco Heat Exchangers,

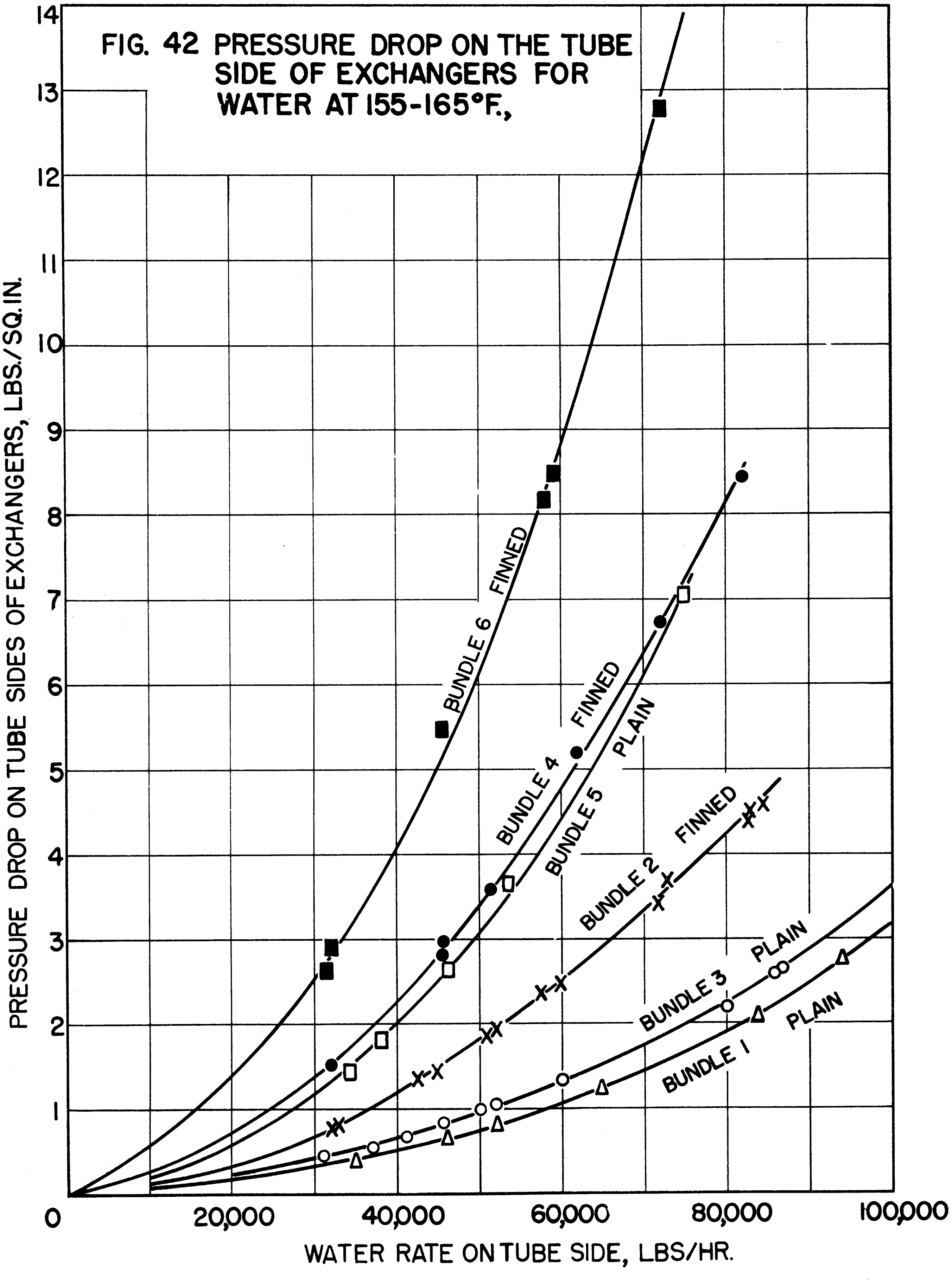


FIG. 42 PRESSURE DROP ON THE TUBE SIDE OF EXCHANGERS FOR WATER AT 155-165°F.,
PRESSURE DROP ON TUBE SIDES OF EXCHANGERS, LBS./SQ.IN.
WATER RATE ON TUBE SIDE, LBS/HR.
0
20,000
40,000
60,000
80,000
100,000
1
2
3
4
5
6
7
8
9
10
11
12
13
14
BUNDLE 6 FINNED
BUNDLE 4 FINNED
BUNDLE 5 PLAIN
BUNDLE 2 FINNED
BUNDLE 3 PLAIN
BUNDLE 1 PLAIN

September 28, 1950,[26] will be used to determine the prices of the components of the exchangers, except for the tubes. The cost data are for shells with a working pressure of 150 lbs per sq in. The prices for plain and finned tubes were obtained from the Wolverine Tube Division of the Calumet and Hecla Consolidated Copper Company, as of December 4, 1950.

Typical problems for which finned tubes might be economical are selected to illustrate the use of the data in sizing and pricing exchangers. Some exchangers will be sized using the shell-circle design of the test exchangers. Comparisons more favorable for finned tubes could have been made if the shell-side coefficients had been obtained from the equation recommended for standard exchangers.[22]

Fouling

One of the important items in heat-exchanger design is the fouling factor, which is used to provide the proper size of exchanger for a given duty when the exchanger has accumulated a specified amount of foreign material on the heat-transfer surface. The accepted procedure in this regard is set forth in the TEMA standards.[18] It should be obvious that, if a tube is expected to foul severely on the inside, extended surface on the outside can be of little value. Finned-tube exchangers have been reported to perform on relatively dirty oil on the shell-side without undue fouling as compared to plain tubes.[10] Even though deposits may settle at the root of the fin, the end of the fin may be considerably cleaner than the surface of plain tubes. The eddies between the fins may prevent fouling in some cases. The heat-transfer services selected for design in this report require moderate fouling factors, which are used according to TEMA standards for plain tubes.

Procedures for Shell-Circle Design

The procedures for designing a shell and tube exchanger of the type tested in this study will be outlined. An example calculation will follow to illustrate the method.

In general, the length of the exchanger is specified or fixed at the maximum practical length for the service. The mass flow rate, G_m, is normally selected as a starting point in the calculations. From this flow rate and the fluid properties, the shell-side coefficient is found from Equation (4e). An overall coefficient is computed from the shell-side coefficient and an assumed inside coefficient. The heat-transfer area for the exchanger is computed in the first trial, which area indicates the diameter of the exchanger for a given length and tube. The number of tubes and the water requirement determines the water passes for a given water velocity, normally chosen in the range of 3 to 6 feet per second.

At this point, a second trial is made. The overall coefficient is computed from the inside coefficient and fouling factor to find a second trial area and diameter. It is necessary to go through the design again with a viscosity ratio in the shell-side film coefficient. After this part of the design is complete, the pressure drop is computed. This requires a baffle spacing and window opening based on the mass velocity relation of Equations (5) and (6) and exchanger dimensions. If the pressure drop is excessive, a lower mass-flow rate is required, and, if the pressure drop is low, a higher mass-flow rate is used.

Engineers with experience in the correlation of heat-transfer coefficients as influenced by tube size and shell diameter over and above the effects reflected in Equation (4) may wish to employ their procedures for obtaining shell-side coefficients for plain tubes and performance of finned tubes based on the results presented in Figs. 21 through 26.

Comparison of Costs for Finned and Plain-Tube Exchangers

Five cases have been selected to illustrate the comparative performances which may be expected for finned and plain-tube exchangers when cooling viscous fluids. The first three cases represent designs for a given heat duty. The fourth case is included to illustrate that finned tubes may be used in exchangers previously equipped with plain tubes when a reduction in the temperature difference is desired. Case V is a plain-tube design to match the performance of the finned-tube unit in Case IV. The five cases are:

Case I	Lube Oil Cooler, oil 25-centipoise viscosity
Case II	Absorption Oil Cooler, oil 2.6-centipoise viscosity
Case III	Corn Sirup Cooler, sirup 61-centipoise viscosity
Case IV	Corn Sirup Cooler, Sirup 73-centipoise viscosity
Case V	Corn Sirup Cooler, sirup 73-centipoise viscosity

The calculations for Case I are given completely, while the results are tabulated for the other four cases.

Case I, Example Design of Lube Oil Cooler

Given. Design of lubricating oil cooler for a diesel engine. The tubes are to be 8 feet long. TEMA fouling factors are required.

Shell-Side Fluid. 40 SAE lubricating oil, 170 gallons per minute, in at 190°F and out at 167°F.

Tube-Side Fluid. Treated circulating water, 600 gallons per minute, in at 160°F. Pressure loss on shell side should not exceed 12 psi.

Design of Plain-Tube Exchanger. 40 SAE oil is assumed to have properties of Figs. 12-15.

Average oil temperature = 178.5°F

Oil Properties.

C_p = 0.5 Btu/lb °F from Fig. 15

ρ = .860 x 62.4 = 53.6 lbs/cu ft from Fig. 12

k = .081 Btu/(hr)(sq ft)(°F/ft) from Fig. 14

μ = 23 centipoises from Fig. 13

Oil rate in lbs/hr = 170 x 60 x 8.34 x .86 = 73,100 lbs/hr

Heat load = 73,100 x (190 - 167) x .50 = 840,000 Btu/hr

Water temperature rise = $\frac{840{,}000}{600 \times 60 \times 8.2}$ = 2.84°F. Water out at 162.84°F

Log mean temperature difference = $\frac{(190 - 162.8) - (167 - 160)}{\ln (190-162.8)/(167 - 160)}$ = 14.9°F

Correction factor, F, from TEMA, Fig. T-4A = 0.955 for two tube passes.

Mean temperature difference (ΔT_m) = 14.9 x 0.955 = 14.2

Equation (4c) with recommended value of 0.19 for C, with plain tubes.

$$\frac{h_o D_o}{k} = .19 \left(\frac{D_o G}{\mu}\right)^{.65} \left(\frac{C_p \mu}{k}\right)^{.375} \left(\frac{\mu}{\mu_w}\right)^{.14} \qquad (4c)$$

A 5/8-inch (.0521-ft) O.D. Admiralty tube, 18-gauge, 0.049-inch wall on a 13/16-inch triangular pitch, will be selected for this design. A mass velocity G_m of 350,000 lbs per (hr)(sq ft) is assumed for the exchanger.

First trial solution for shell-side coefficient:

$$h_o \cong \frac{0.19 \times .081}{.0521} \left(\frac{.0521 \times 350{,}000}{23 \times 2.42}\right)^{.65} \left(\frac{0.5 \times 23 \times 2.42}{.081}\right)^{.375} \left(\frac{\mu}{\mu_w}\right)^{.14}$$

$$h_o = .295\ (327)^{.65}\ (344)^{.375}\ \left(\frac{\mu}{\mu_w}\right)^{.14}$$

$$h_o = 113 \left(\frac{\mu}{\mu_w}\right)^{.14} \text{ Btu/(hr)(°F)(sq ft)}$$

Assume an overall coefficient (U_o) of 90 Btu per (hr)(°F)(sq ft). The heat-transfer area by Equation (1):

$$A = \frac{q}{U_o \, \Delta T_m} = \frac{840{,}000}{90 \times 14.2} = 657 \text{ sq ft outside}$$

From the Alco book,[26] the smallest exchanger, 8 ft long, with more than 657 sq ft outside is a 24-inch diameter shell which was 768 sq ft with 586 tubes for a two-pass tube side. It is assumed that there would be approximately the same number of tubes in a 24-inch bundle with the shell-circle design as with regular triangle pitch.

Flow area tube side: inside flow area, tubes 0.2181 sq in.

$$\frac{0.2181 \times 293}{144} = 0.444 \text{ sq ft}$$

$$\text{water velocity} = \frac{600 \times 8.2}{60 \times 0.444 \times 61.0} = 3.03 \text{ ft/sec}$$

Water convection coefficient, McAdams,[13] page 183.

$$h_i = 150(1 + 0.011\ T)\ \frac{V^{0.8}}{d_i^{\ 0.2}}$$

$$h_i = 150(1 + 0.011 \times 161)\ \frac{3.03^{\ 0.8}}{0.527^{0.2}} = 1150 \text{ Btu per (hr)(°F)(sq ft inside)}$$

Computed overall coefficient:

$$U_o = \frac{1}{\frac{1}{113} + 0.001 + \frac{.625 \times 0.001}{.527} + \frac{.625}{.527 \times 1150}} = 83.0 \text{ Btu per (hr)(°F)(sq ft)}$$

$$\text{Tube wall temperature} = 178.5 - 14.2 \times \frac{83.0}{113} = 168.0°\text{F}$$

$$\mu_w = 27 \text{ centipoises}$$

$$\left(\frac{\mu}{\mu_w}\right)^{.14} = (23/27)^{.14} = .978$$

$$h_o = 113 \times .978 = 111 \text{ Btu per (hr)(°F)(sq ft)}$$

$$U_o = \frac{1}{\frac{1}{111} + .001 + \frac{.625}{.527} \times .001 + \frac{.625}{.527 \times 1280}} = 82.0 \text{ Btu per (hr)(°F)(sq ft outside)}$$

$$\text{Heat-transfer area} = \frac{840,000}{82.0 \times 14.2} = 721 \text{ sq ft outside}$$

The 24-inch bundle with 768 sq ft is sufficient for this second trial area and will be used if the pressure drop calculations are satisfactory

Baffle-Spacing Calculation. Minimum space between tubes along a diameter perpendicular to flow, 28 tubes per row:

$$\frac{24 - (28 \times 0.625)}{12} = 0.541 \text{ ft}$$

Cross-flow area equals 0.541 x baffle spacing. Baffle cut and spacing will be chosen to give

$$G_c = 210,000 \text{ and a } G_w = 590,000$$

or

$$G_m = \sqrt{210,000 \times 590,000} = 350,000$$

Area cross flow of 210,000 lbs per (sq ft)(hr) is $\frac{73,000}{210,000} = 0.348$ sq ft

$$\text{Baffle spacing} = \frac{0.348}{0.541} = 0.644 \text{ ft}$$

Number of baffles $= \frac{8}{0.644} - 1 = 11.4$; 10 baffles will be used to give appropriate space at ends or 11 spaces for cross flow.

Pressure Drop. Cross flow passes about 10 rows of tubes from edge of one window to edge of next.

$$(P_1 - P_2)_c = \frac{1.07}{9} \frac{f\, n\, G_c^2}{g c \rho} \left(\frac{\mu}{\mu_w}\right)^{-.14} \tag{8}$$

$$\left(\frac{D_o G_c}{\mu}\right)_c = \frac{.0521 \times 210{,}000}{2.42 \times 23} = 197, \text{ f from Fig. 40} = 2.0$$

$$(P_1 - P_2)_c = \frac{1.07 \times 1.9 \times 10 \times (210{,}000)^2}{10^9 \times 32.2 \times 53.6 \times .978} \times 11 = 6.15 \text{ psi cross flow pressure drop.}$$

Window pressure drop:

$$(P_1 - P_2)_w = \frac{2.9}{10^{13}} \times \frac{G_w^2}{\text{sp gr}} \qquad (7)$$

$$(P_1 - P_2)_w = \frac{2.9}{10^{13}} \times \frac{(590{,}000)^2}{0.86} = 0.118 \text{ psi/baffle.}$$

Total window loss:

$$0.118 \times 10 = 1.18 \text{ psi}$$

$$\text{Total pressure drop} = 6.15 + 1.18 = 7.33 \text{ psi}$$

Design of Finned-Tubing Exchanger. Since this unit will be smaller in diameter, a mass velocity (G_m) of 500,000 will be used with a G_w of 830,000 and a G_c of 300,000. Convection coefficient shell side, using recommended value for C of 0.13:

$$\frac{h_o D_e}{k} = 0.13 \left(\frac{D_e G_m}{\mu}\right)^{.65} \left(\frac{C_p \mu}{k}\right)^{.375} \left(\frac{\mu}{\mu_w}\right)^{.14} \qquad (4e)$$

D_e for 5/8" end finned tube from Table I = 0.554 inches or .0461 feet

$$h_o = \frac{0.13 \times .081}{.0461} \left(\frac{.0461 \times 500{,}000}{23 \times 2.42}\right)^{.65} (234)^{.375} \left(\frac{\mu}{\mu_w}\right)^{.14}$$

$$h_o = 0.228 \times 414^{.65} \times 7.75 \times \left(\frac{\mu}{\mu_w}\right)^{.14} = 88.9 \left(\frac{\mu}{\mu_w}\right)^{.14}$$

Assume a U_o of 65.

$$\text{First trial heat-transfer area} = \frac{840{,}000}{14.2 \times 65} = 910 \text{ sq ft outside.}$$

$$\text{Area of finned tube} = 0.361 \text{ sq ft per ft, Table I.}$$

$$\text{No. of tubes, 8 ft long} = \frac{910}{0.361 \times 8} = 315 \text{ tubes.}$$

$$\frac{A_o}{A_i} = \frac{54.5}{16.7} = 3.26 \text{ from Table I.}$$

For one-pass water side, an 18-inch diameter exchanger will have 336 tubes. Tubes have 0.133 sq in. flow area based on I.D. of 0.411 inch.

$$\text{Flow area for water} = \frac{336 \times 0.133}{144} = 0.310 \text{ sq ft}$$

$$\text{Water velocity} = \frac{600 \times 8.2}{60 \times 0.310 \times 61.0} = 4.34 \text{ ft/sec}$$

Inside coefficient:

$$h_i = 150(1 + 0.011\ T)\ \frac{V^{0.8}}{D_i^{\ 0.2}} = \frac{416 \times 3.24}{0.837} = 1610 \text{ Btu per(hr)(°F)(sq ft)}$$

Computed overall coefficient

$$U_o = \frac{1}{\frac{1}{88.9} + .001 + .001(3.26) + \frac{3.26}{1610}} = 60.5$$

Wall temperature (100 per cent fin efficiency) $= 178.5 - 14.2 \times 60.5/88.9$

$$= 168.8$$

$$\text{Viscosity ratio} = \left(\frac{\mu}{\mu_w}\right)^{.14} = \left(\frac{23}{26}\right)^{.14} = 0.983$$

$$h_o = 88.9 \times 0.983 = 87.4$$

This h_o is based on the effective outside area. To convert to h_o', based on the actual outside area, use Fig. 18, which gives A_o/A_e. In this figure the total outside conductance, including the outside fouling, should be used. This equals $\frac{1}{1/87.4 + .001} = 80.5$; from Fig. 18, $A_o/A_e \cong 1.0$. When A_o/A_e is significantly greater than one, the h_o is divided by A_o/A_e to get h_o'. In this case $h_o' \cong h_o$.

$$U_o = \frac{1}{1/87.4 + .001 + .00326 + 3.26/1610} = 59.8 \text{ Btu per (hr)(°F)(sq ft)}$$

Second trial heat-transfer area, $\Delta T = 14.9$ for single pass.

$$A_o = \frac{840,000}{59.8 \times 14.9} = 943 \text{ sq ft}$$

$$\text{No. of tubes} = \frac{943}{0.361 \times 8} = 326 \text{ tubes}$$

The 18-inch exchanger is satisfactory.

Baffle Spacing

No. of tubes on a diameter = 20 tubes.

$$\text{Minimum space between tubes} = \frac{18 - 20 \times .554}{12} = 0.577 \text{ ft}$$

$$\text{Area for cross flow of 300,000 is } \frac{73,100}{300,000} = 0.244 \text{ sq ft}$$

$$\text{Baffle spacing} = \frac{0.244}{0.577} = 0.422 \text{ ft}$$

$$\text{No. of baffles} = \frac{8}{0.422} - 1 = 18$$ Allow 2 baffle spaces for nozzles and use 16.

Pressure Drop

No. of rows of tubes in cross flow between baffle windows is about 6.

$$\left(\frac{D_e G_c}{\mu}\right)_c = \frac{.0461 \times 300,000}{23 \times 2.42} = 248, \quad f = 0.75$$

$$(P_1 - P_2)_c = \frac{1.07 \times 6 \times 0.75 \times (300,000)^2}{10^9 \times 32.2 \times 53.6 \times .983} = 0.255 \text{ psi}$$

$$\text{Total cross flow } (P_1 - P_2) = 17 \times 0.255 = 4.33 \text{ psi}$$

$$\text{Window loss} = (P_1 - P_2)_w = \frac{2.9 \times (830,000)^2}{10^{13} \times 0.86} \times 16 = 3.72 \text{ psi}$$

$$\text{Total pressure drop} = 3.72 + 4.33 = 8.05 \text{ psi}$$

Pricing of Exchangers. The Alco Heat Exchanger book[26] is used for prices of exchanger components. Tube costs were furnished by Wolverine Tube Division, as follows: plain Admiralty \$0.172 per ft, finned Admiralty \$0.275 per ft, both in large quantities. A summary of the results of these calculations and the costs for the components of the exchangers are given in Table VIII.

The saving with finned tubes is computed:

$$\begin{array}{r} \$4159.64 \\ \underline{3206.35} \\ \$\ 953.29 \end{array}$$

$$\text{Per cent saving} = \frac{953.29 \times 100}{4159.64} = 22.9\%.$$

Case II, Absorption Oil Cooler

Given: Cool 2,000,000 gallons per day of absorption oil from 140°F to 90°F. Treated cooling water is available at 70°F and will be permitted to rise to 85°F. The oil has a molecular weight of 210 and a density of 0.875 g/cc at 60°F. TEMA fouling factors should be used, 0.002 outside and 0.001 inside. Admiralty tubes should be used.

The designs were computed in a manner similar to that of Case I for the lubricating oil cooler. The results of the calculations are given in Table IX. The shell diameters computed are 42 inches and 33 inches for the plain and finned-tube units, respectively. The cost of the plain-tube exchanger is \$14,166 - \$10,530 = \$3,586 more than the finned-tube unit. This represents a decrease of 25.3 per cent in the cost of this type of absorption oil cooler when using the finned tubes described in Table I.

Case III, Corn Sirup Cooler

Given: Cool 50 gallons per minute of 38° Be corn sirup from 200°F to 130°F. River water is available at 90°F and a 20°F rise may be used. The exchanger tubes are to be 8 feet long, 3/4-inch copper. The fouling factors are 0.002 on both sides.

A comparison of the designs and cost is given in Table X.

TABLE VIII

DESIGN AND COSTS OF LUBRICATING OIL COOLERS -- CASE I

Properties of Oil at Mean Bulk Temperature

= 23 centipoises C_p = 0.5 ρ = 53.6 k = 0.081

	Units	Plain Tubes	Finned Tubes
Heat duty	Btu/(hr)	840,000	840,000
Mean temperature difference, ΔT_m	°F	14.2	14.9
Shell side velocities	lbs/(hr)(sq ft)		
G_m*	"	350,000	500,000
G_c	"	210,000	300,000
G_w	"	590,000	830,000
Shell side coefficient	Btu/(hr)(°F)(sq ft) outside	111	87.4
Water velocity, tube side	ft/sec	3.03	4.34
Water coefficient	Btu/(hr)(°F)(sq ft) inside	1150	1610
Overall coefficient	Btu/(hr)(°F)(sq ft) outside	82.0	59.8
Heat transfer area required	sq ft outside	721	943
Heat transfer area standard exchanger	sq ft outside	768	972
Exchanger dimensions			
Length of tubes*	ft	8	8
Diameter of tubes* (0.049" wall)	inch	5/8	5/8
Spacing of tubes*	inch	13/16 triangular	13/16 triangular
Number of tubes		586	336
Diameter of shell	inch	24	18
Passes water side		2	1
Number of baffles		10	16
Shell side pressure drop	lbs/sq in.	7.33	8.05
Costs	Dollars		
Shell	"	$1158.20	$ 893.68
Tube side	"	508.28	398.19
Nozzles	"	307.00	307.00
Tube sheets	"	911.00	480.00
Baffles	"	467.16	388.48
Tubes	"	808.00	739.00
TOTAL COST	"	$4159.64	$3206.35

*Assumed in design procedure

TABLE IX

DESIGN AND COSTS OF ABSORPTION OIL COOLERS -- CASE II

Properties of Oil at Mean Bulk Temperature

μ = 2.6 centipoises ρ = 53.1 k = 0.0815 C_p = 0.485

	Units	Plain Tubes	Finned Tubes
Heat duty	Btu/(hr)	14,400,000	14,400,000
Mean temperature difference, ΔT_m	°F	30.3	30.3
Shell side velocities	lbs/(hr)(sq ft)		
G_m*	"	500,000	600,000
G_c	"	400,000	500,000
G_w	"	625,000	720,000
Shell side coefficient	Btu/(hr)(°F)(sq ft) outside	237	240
Water velocity, tube side	ft/sec	5.21	6.30
Water coefficient	Btu/(hr)(°F)(sq ft) inside	1140	1390
Overall coefficient	Btu/(hr)(°F)(sq ft) outside	116	87.5
Heat transfer area required	sq ft outside	4100	5430
Heat transfer area standard exchanger	sq ft outside	4430	5780
Exchanger dimensions			
Length of tubes*	ft	16	16
Diameter of tubes* (0.65" wall)	inch	3/4	3/4
Spacing of tubes*	inch	15/16 triangular	15/16 triangula
Number of tubes		1378	882
Diameter of shell	inch	42	33
Passes water side		4	2
Number of baffles		8	11
Shell side pressure drop	lbs/sq in.	4.1	5.3
Costs			
Shell	Dollars	$ 2684	$ 1645
Tube side	"	1278	751
Nozzles	"	507	507
Tube sheets	"	3093	1665
Baffles	"	859	717
Tubes	"	5700	5250
TOTAL COST		$14116	$10530

*Assumed in design procedure

TABLE X

DESIGN AND COST OF CORN SIRUP COOLERS --CASE III

Properties of Sirup at Mean Bulk Temperature

μ = 61 centipoises ρ = 83.5 k = 0.28 C_p = 0.64

	Units	Plain Tubes	Finned Tubes
Heat duty	Btu/(hr)	1,500,000	1,500,000
Mean temperature difference, ΔT_m	°F	57.0	57.0
Shell side velocities	lbs/(hr)(sq ft)		
G_m*	"	200,000	250,000
G_c	"	200,000	250,000
G_w	"	200,000	250,000
Shell side coefficient	Btu/(hr)(°F)(sq ft) outside	121	100
Water velocity, tube side	ft/sec	3.57	3.08
Water coefficient	Btu/(hr)(°F)(sq ft) inside	1410	1320
Overall coefficient	Btu/(hr)(°F)(sq ft) outside	67.1	48.3
Heat transfer area required	sq ft outside	392	545
Heat transfer area standard exchanger	sq ft outside	425	563.0
Exchanger dimensions			
Length of tubes*	ft	8	8
Diameter of tubes* (0.065" wall)	inch	3/4	3/4
Spacing of tubes*	inch	15/16 triangular	15/16 triangular
Number of tubes		270	172
Diameter of shell	inch	20	16
Passes water side		6	2
Number of baffles		17	17
Shell side pressure drop	lbs/sq in.	4.0	3.8
Costs			
Shell	Dollars	$1011	$ 875
Tube side	"	521	408
Nozzles	"	221	221
Tube sheets	"	596	366
Baffles	"	359	281
Tubes	"	580	476
TOTAL COST		$3288	$2627

*Assumed in design procedure

All prices for corn sirup coolers are based on exchangers having plain steel shells. The use of other metals for the shells would change the price of both units in a manner that would not affect the comparison adversely.

Case IV, Replacement of Plain Tubes by Finned Tubes in Corn Sirup Cooler

Given: In Case III, a plain-tube exchanger was designed for cooling 50 gallons of sirup per minute. The sirup left at 130°F and the water at 110°F. Compute the performance of this exchanger when it is equipped with finned tubes.

The temperature of the sirup is now reduced to 117°F instead of 130°F, while the cooling water is maintained at an outlet temperature of 110°F. The finned tubes transfer an added 275,000 Btu per hr, as shown in Table XI.

This example cannot be used to evaluate the merit of finned tubes since it is impossible to evaluate the cost of the added cooling without further comparisons. Case V is a design of a plain-tube exchanger to match the performance of the exchanger in Case IV, i.e., the plain-tube unit of Case III when it contains finned tubes.

Case V, Closer Temperature Approach with Finned and Plain Tubes

Design a plain-tube exchanger to cool 50 gallons per minute of 38° Be corn sirup from 200°F to 117°F, with water in at 90°F and out at 110°F. Fouling factors of 0.002 to be used on both sides. Exchanger tubes are 8 feet long, 3/4-inch copper. Note that this is the performance of the exchanger with finned tubes in Case IV.

The results of the design and cost calculations are given in Table XI. It may be seen that the additional heat transfer to cool the sirup from 130°F to 117°F costs $116.00 when employing finned tubes and $930.00 when obtaining a larger plain-tube exchanger.

TABLE XI

DESIGN AND COSTS FOR CORN SIRUP -- COVERS CASES IV AND V

Properties of Sirup at Mean Bulk Temperature of Cases IV and V

μ = 73 centipoises ρ = 84.0 k = 0.28 C_p = 0.64

	Units	Case IV Plain Tube Shell of Case III, with Finned Tubes	Case V Plain Tube Unit to Match Case IV
Heat duty	Btu/(hr)	1,775,000	1,775,000
Mean temperature difference, ΔT_m	°F	47.1	47.1
Shell side velocities	lbs/(hr)(sq ft)		
G_m*	"	172,000	172,000
G_c	"	147,000	172,000
G_w	"	200,000	172,000
Shell side coefficient	Btu/(hr)(°F)(sq ft) outside	78.3	109
Water velocity, tube side	ft/sec	5.6	2.68
Water Coefficient	Btu/(hr)(°F)(sq ft) inside	2120	1120
Overall coefficient	Btu/(hr)(°F)(sq ft) outside	44.2	68.2
Heat transfer area required	sq ft outside	885	626
Exchanger dimensions			
Length of tubes*	ft	8	8
Diameter of tubes* (0.065" wall)	inch	3/4	3/4
Spacing of tubes	inch	15/16 triangular	15/16 triangular
Number of tubes		270	398
Diameter of shell	inch	20	24
Passes water side		6	6
Number of baffles		17	17
Shell side pressure drop	lbs/sq in.	3.1	4.5
Costs			
Shell	Dollars	\$1011	\$1158
Tube side	"	521	581
Nozzles	"	221	221
Tube sheets	"	596	819
Baffles	"	359	475
Tubes	"	746	869
TOTAL COST		\$3454	\$4123

*Assumed in design procedure

It might be added that the extra cooling below 130°F for 50 gallons per minute costs \$1.23 per °F for equipping with finned tubes and \$7.15 per °F when obtaining the requisite plain-tube exchanger.

WHEN ARE FINNED TUBES ECONOMICAL IN SHELL AND TUBE EXCHANGERS?

The above cases have been chosen to show typical advantages of finned tubes by selecting services where the shell-side coefficients are relatively low. A comparison of costs for services with high shell-side coefficients could result in little or no saving due to fouling factors, low fin efficiencies for high coefficients, and the slightly lower coefficients for the finned surface as compared to the plain surface, Fig. 36. A complete calculation, such as is given in Case I, is required to arrive at a definite comparison of costs.

One element of the economics is the working pressure of the exchanger. High pressures require heavy shells which are relatively expensive, and a given size reduction will save a higher per cent of the cost at higher pressures. These studies are based on working pressures of 150 pounds per sq in.

A general statement concerning the probability that finned-tube exchangers will or will not be economical can be made when the inside resistances to heat transfer and the outside resistances are known. Equation (9) gives the relationship between the clean and fouled coefficients, individual coefficients, and fouling factors.

$$U_o \text{ (fouled)} = \frac{1}{\dfrac{1}{h_o'} + f_o + \dfrac{L\,A_o}{k\,A_{av}} + \dfrac{f_i\,A_o}{A_i} + \dfrac{A_o}{h_i A_i}} \qquad (9)$$

$$= \frac{1}{\dfrac{1}{U_o \text{ (clean)}} + f_o + f_i\,\dfrac{A_o}{A_i}} \qquad (9a)$$

For cases in which water flows inside of tubes at a given velocity and the metal resistance is known, the relationship between the inside resistance and the outside resistance is given by the two fouling factors and the overall coefficient on either a clean basis or on a fouled basis. Thus it is possible, at a fixed water-film coefficient and specified tube sizes, to make cost studies for plain and finned-tube exchangers as a function of the fouling factors and the overall coefficient.

Since situations arise in which either clean or fouled coefficients are known, two charts have been prepared for predicting whether finned or plain-tube exchangers are economical. Fig. 43 is based on overall coefficients for the fouled exchanger, and Fig. 44 plots overall coefficients for the clean exchanger. These charts have curves for given fouling factors on the outside and plot the fouling factor on the inside as ordinates. The curves represent the condition for equal costs of plain and finned-tube exchangers. The area above each curve represents overall coefficients for plain-tube exchangers for which cost calculations will show that plain tube exchangers are cheaper, while the area below the curve represents coefficients for which cost calculations will show that finned-tube exchangers are cheaper.

For example, Case I has a fouled overall coefficient of 87.1 for the plain-tube exchanger with $f_i = 0.001$ and $f_o = 0.001$. Reference to Fig. 43 shows that an overall coefficient of 87.1 is well below the curve for $f_o = 0.001$ at $f_i = 0.001$. In fact, all exchangers having overall coefficients in the fouled condition below 175, are more economical with finned tubes for fouling factors of 0.001. The clean overall coefficient for Case I is 110 and is well below the curve for $f_o = 0.001$ in Fig. 43 at $f_i = 0.001$.

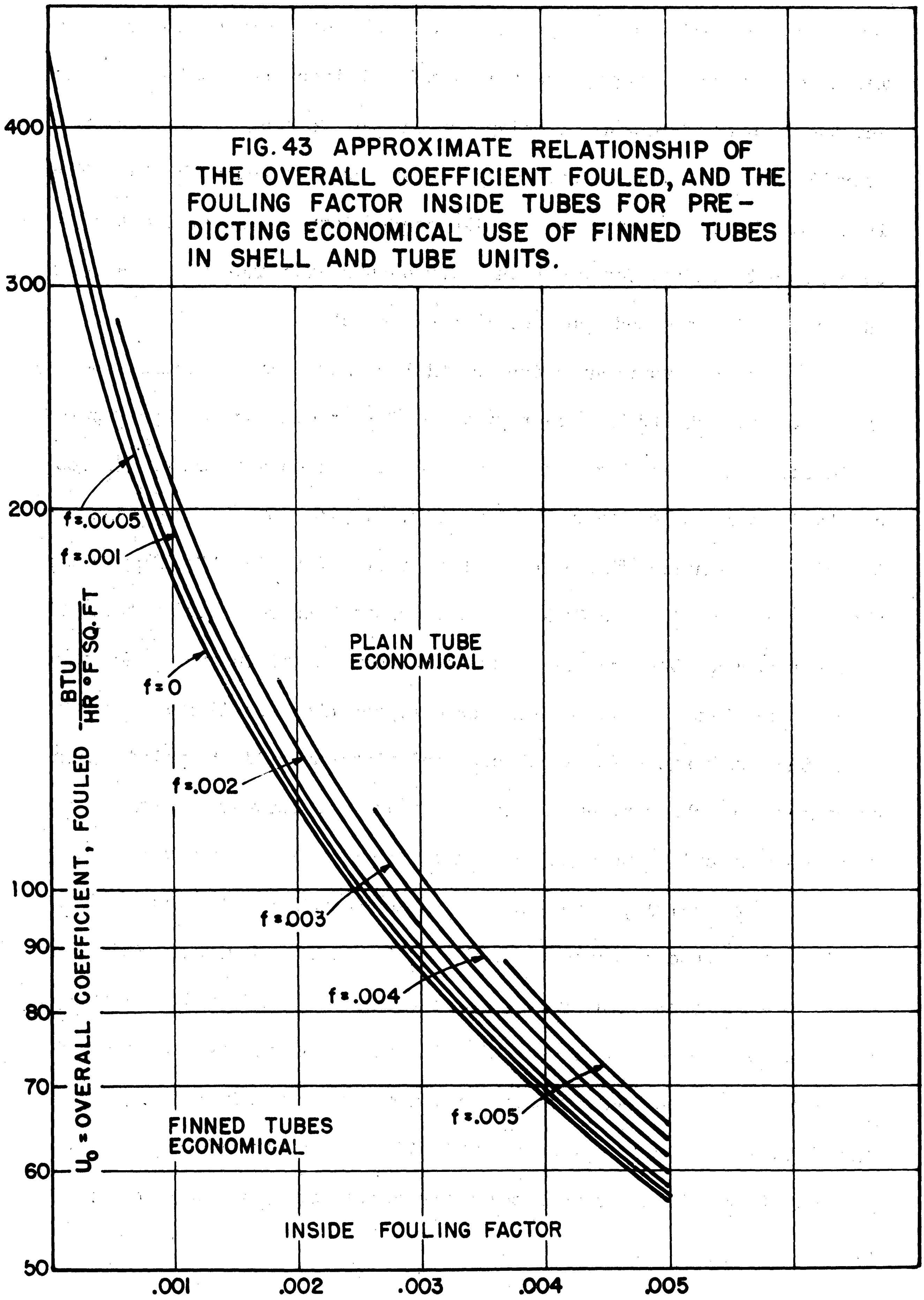


FIG. 43 APPROXIMATE RELATIONSHIP OF THE OVERALL COEFFICIENT FOULED, AND THE FOULING FACTOR INSIDE TUBES FOR PREDICTING ECONOMICAL USE OF FINNED TUBES IN SHELL AND TUBE UNITS.
400
300
200
100
90
80
70
60
50
f = .0005
f = .001
f = 0
f = .002
f = .003
f = .004
f = .005
PLAIN TUBE ECONOMICAL
FINNED TUBES ECONOMICAL
U0 = OVERALL COEFFICIENT, FOULED BTU / HR °F SQ. FT
INSIDE FOULING FACTOR
.001
.002
.003
.004
.005

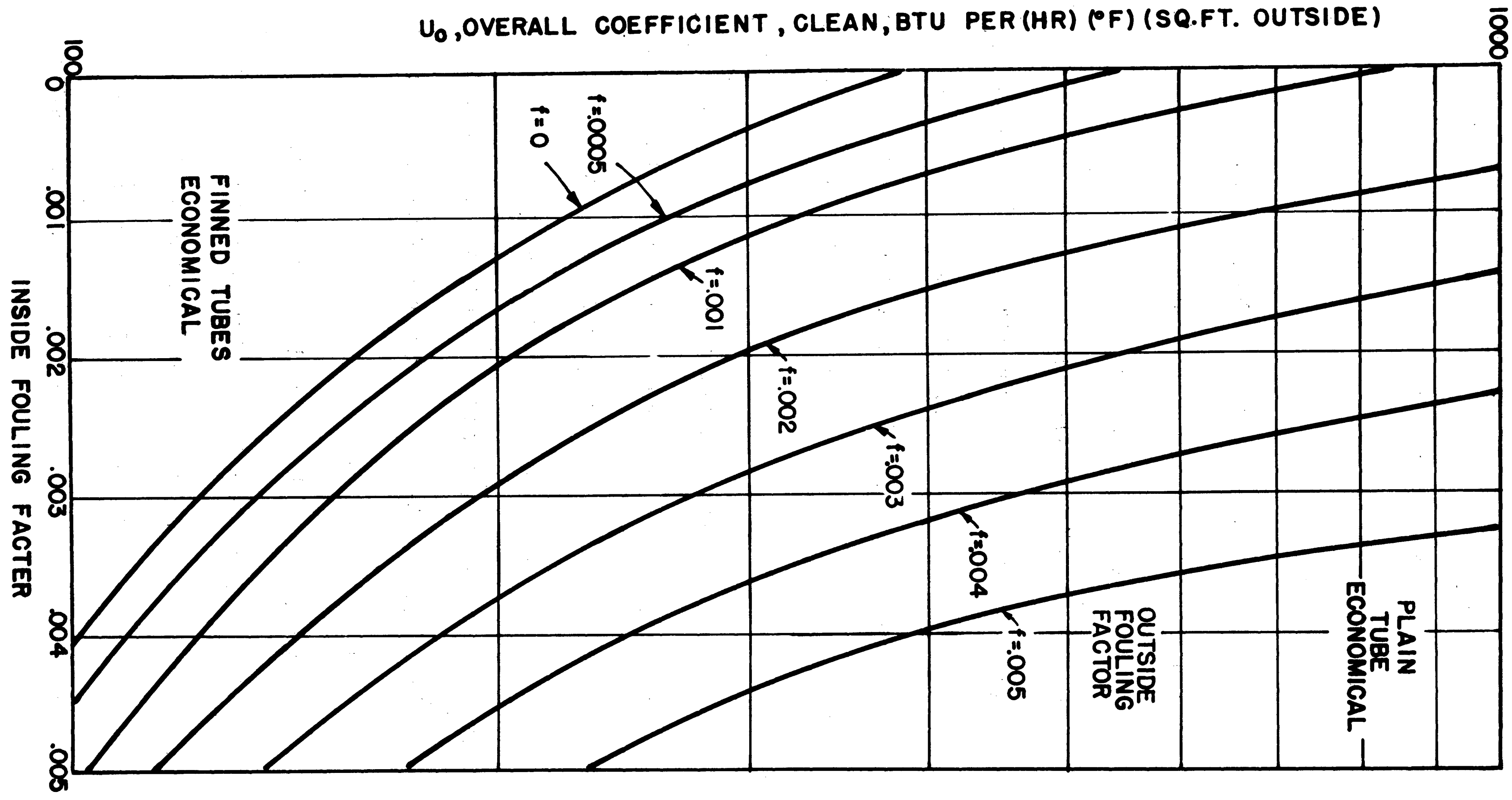


FIG. 44 APPROXIMATE RELATIONSHIP OF THE OVERALL COEFFICIENT CLEAN AND THE FOULING FACTOR INSIDE TUBES FOR PREDICTING ECONOMICAL USE OF FINNED TUBES IN SHELL AND TUBE UNITS

These charts are based on specific calculations with the following data and assumptions:

(1) Inside water coefficient = 1500.

(2) 3/4-inch plain and finned Admiralty tubes of dimensions shown in Table I.

(3) 24-inch diameter shell, 12-foot tubes, 4-pass plain-tube exchanger and 22-inch diameter shell, 12-foot tubes, 2-pass finned-tube exchanger.

(4) It is assumed that the outside coefficient for finned tubes equals 80 per cent of the outside plain-tube coefficient.

(5) Costs are those used in this study, which are \$4.65 per sq ft outside for the plain-tube exchanger(\$4495 total) and \$2.61 per sq ft outside for the finned exchanger (\$4578 total).

The curves represent the overall coefficients for the plain-tube exchanger and given fouling factors at which the exchangers of this size and equal costs transfer the same amount of heat per degree temperature difference. For Fig. 43 the computations simply involve finding the points at which U_o(Plain fouled) divided by U_o(Finned fouled) becomesequal to 4.65/2.61 or 1.78. The curves in Fig. 43 are terminated when the outside film coefficients begin to rise rapidly.

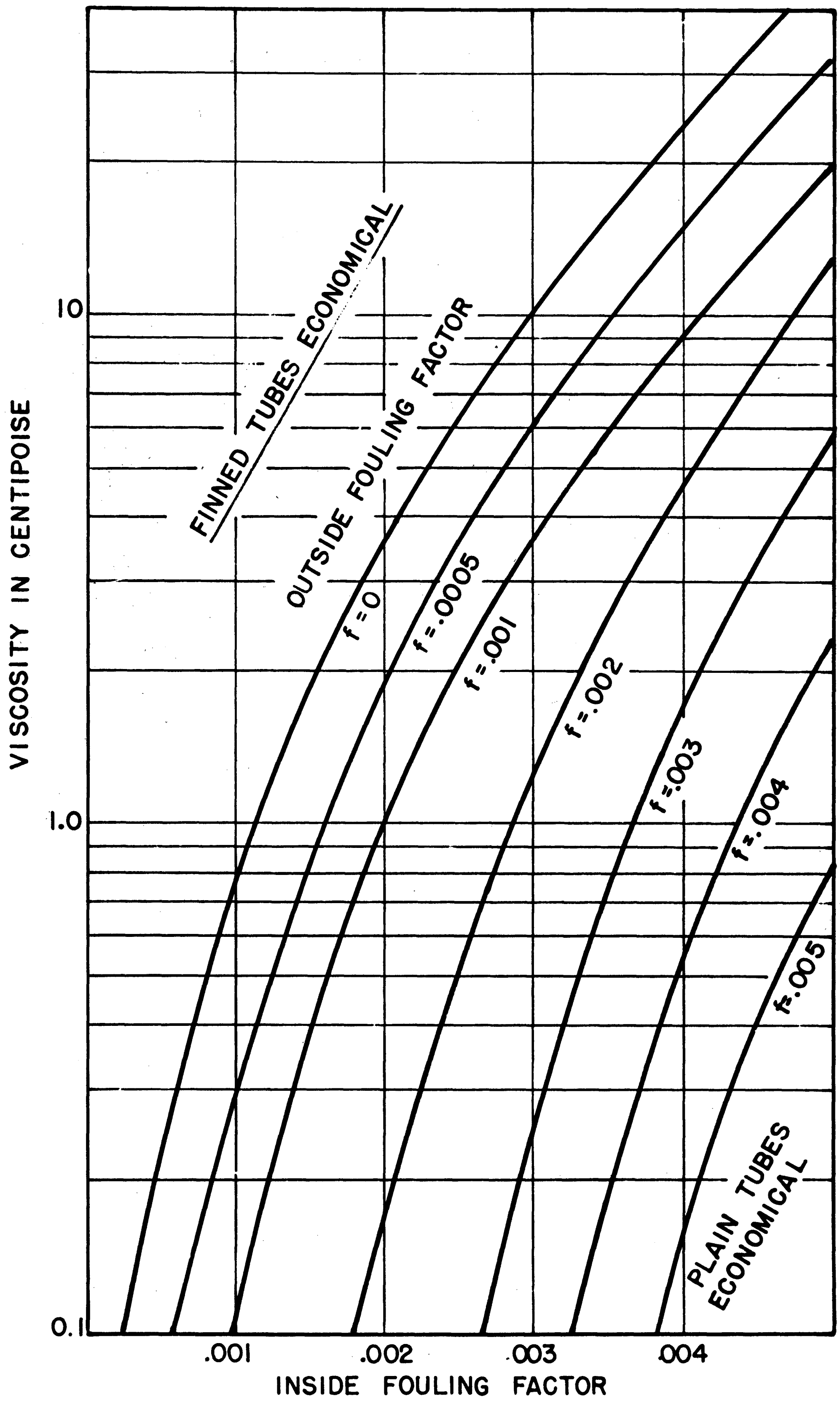
It is appreciated that these charts do not apply accurately for conditions other than those specified. For cases in which the exchanger diameters are lower than those given (22-24 inches), the break-even point for finned-tube exchangers will occur at lower overall coefficients. For larger sizes, the coefficients for the break-even point may be slightly higher.

The use of large fouling factors for the outside may be questioned in that such fouling might plug the spaces between the fins. Although such fouling could occur, it should not be overlooked that there is turbulence between the fins. The charts assume a uniform fouling on the outside, such as might occur for paraffin accumulation in contrast to solid accumulation, such as scale formation.

For petroleum oils used with the exchangers and conditions employed for Figs. 43 and 44, the overall coefficients are essentially a function of the oil viscosity. For a thermal conductivity of 0.078 Btu per (hr)(°F) (ft) and a specific heat of 0.48 Btu per (lb)(°F), the clean overall coefficient can be computed from the oil viscosity , an assumed velocity of 400,000 lbs/(hr)(sq ft) and the conditions specified. Thus, a chart can be drawn to indicate whether, as a function of the oil viscosity and the two fouling factors, Fig. 45, finned-tube coolers are cheaper than plain-tube coolers. This chart is based on the same assumptions and conditions as Figs. 43 and 44. For oils of viscosity higher than a given curve, finned-tube exchangers (water inside tubes) are more economical. The viscosity is taken at the mean bulk temperature.

Taking Case I as an example, the lube oil has a viscosity of 23.0 centipoises, with 0.001 fouling factor outside and 0.001 fouling factor inside. The viscosity of 23 centipoises is considerably above the 0.1 centipoise read from the chart as the viscosity at the break-even point for these fouling factors for mineral oils. Again, this chart is based on the cost ratio of 1.78 for plain surface to outside finned surface and applies to exchangers 22 to 24 inches in diameter. Small exchangers cannot tolerate the low viscosities indicated for economical use of finned tubes, while larger exchangers would break even at even

FIG. 45 ROUGH RELATIONSHIP FOR PREDICTING OIL VISCOSITIES AT WHICH FINNED TUBES BECOME ECONOMICAL IN SHELL AND TUBE UNITS.



lower viscosities than those shown.

It should be emphasized that these charts give only approximate values and are intended as "rules of thumb" for evaluating finned tubes. They are not offered as a substitute for the design calculations.

METAL REQUIREMENTS OF PLAIN AND FINNED TUBE EXCHANGERS

The weights of heat exchangers for a given duty are of interest for three reasons:

(1) metal requirement in times of metal shortage

(2) weight as a factor in handling, in design of supporting structures, and for mobile equipment

(3) shipping costs.

Any reduction in cost due to the use of finned tubes will be primarily the result of reduction in shell size, and therefore a reduction in the amount of metal used.

A comparison of the weights of the exchangers in Cases I, II, and III is given in Table XII. For these cases, approximately one third of the tube and shell metal is saved when finned tubes are employed. These weights are for operating pressures of 150 lbs per sq in.

CONCLUSION

The heat-transfer and pressure-drop experiments provide adequate data for designing exchangers to determine the relative costs for shell and tube exchangers equipped with plain tubes and with finned tubes. For uses in which the shell-side resistance, including fouling, is somewhat

TABLE XII

COMPARISON OF METAL REQUIREMENTS

	Case I Lube Oil Cooler	Case II Absorption Oil Cooler	Case III Sirup Cooler
Plain Tubes			
Weight of exchanger* less tubes, lbs	4,268	14,633	3,148
Weight of tubes, lbs	1,490	12,000	1,160
Total weight, lbs	5,758	26,633	4,308
Finned Tubes			
Weight of exchanger* less tubes, lbs	2,688	9,612	2,256
Weight of finned tubes, lbs	985	8,150	793
Total weight, lbs	3,673	17,762	3,049
Per cent saving in tube metal	33.9	32.1	31.6
Per cent saving in total exchanger	36.2	33.3	29.1

*Taken as Alco shipping weights less 25 lbs per ft exchanger length allowance for skids.

higher than the inside resistance, a design calculation should be made to determine the cheaper exchanger. Cooling of mineral oils with water is a typical example in which a saving of 20 per cent in exchanger cost and of 30 per cent in metal may be realized by the use of finned tubes.

It is appreciated that the industry may desire further assurances that the fouling resistances on the shelled side of finned tubes will be similar to those on plain tubes. Also, such problems as corrosion of the fins and erosion due to the higher water volocities on the inside of the tubes may require some study under actual service conditions.

NOMENCLATURE

A	=	Heat-transfer area, sq ft
A_{av}	=	Average heat-transfer area through metal wall of tube, sq ft per ft of length
A_c	=	Flow area across the tube bundle, sq ft
A_e	=	Effective outside area of finned tube, sq ft per ft of tube
A_f	=	Area of fins on finned tube, sq ft per ft of tube
A_m	=	Mean area for fluid flow on shell side of tube bundle, $\sqrt{A_c A_w}$, sq ft
A_o	=	Outside area of tube, sq ft per ft of length
A_w	=	Flow area through baffle window, sq ft
a_f	=	Area of a fin
C	=	Constant in heat transfer equation (4, 4a)
C'	=	Constant in heat transfer equation (4b)
C_p	=	Heat capacity, Btu per (lb)(°F)
D	=	Diameter of tube, ft
D_e	=	Equivalent outside diameter of finned tube, ft = outside diameter of plain tube having same inside diameter and same weight of metal
D_o	=	Outside diameter of plain tube, ft
d_i	=	Inside diameter of tube, in. (See Equation (9).)
F	=	Correction factor for ΔT in multipass exchangers from TEMA
G	=	Mass-flow rate, lbs per (sq ft)(hr)
G_c	=	Cross-flow mass-flow rate at A_c, lbs per (sq ft)(hr)
G_m	=	Mean flow rate, shell side, at A_m lbs per (sq ft)(hr)
G_w	=	Mass-flow rate at A_w, lbs per (sq ft)(hr)
g_c	=	Conversion factor = 32.17 ft per sec^2
h_i	=	Inside coefficient based on inside area, Btu per (sq ft)(°F)(hr)

h_i' = Inside coefficient based on outside area, Btu per (sq ft)(°F)(hr)

h_o = Outside coefficient based on outside area for plain tubes and effective outside area for finned tubes, Btu per (sq ft()°F)(hr)

$h_{o'}$ = Outside film coefficient based on the actual outside area for finned tubes, Btu per (sq ft)(°F)(hr)

k = Thermal conductivity, Btu per (ft)(°F)(hr)

L = Length of heat-transfer path through metal wall of tube, ft

m = Exponent of Reynolds Number, Equation (4)

N = 2.31/(1 + 0.o11T) for use in Wilson plots, where T = mean bulk temperature of water inside tubes

n = Number of rows of tubes crossed in cross flow between baffle windows

o = Exponent of Prandtl Number, Equation (4)

$(P_1-P_2)c$ = Pressure drop per baffle space due to friction of cross flow, lbs per sq in.

$(P_1 - P_2)w$ = Pressure drop per baffle window, lbs per sq in.

q = Heat transferred, Btu per hr

T = Bulk water temperature, °F

ΔT = Temperature difference, °F

$\Delta T'$ = Variable temperature difference between bulk fluid temperature and the point fin temperature, °F

ΔT_B = Temperature difference between the bluk fluid temperature and that of the base or root of a fin, °F

ΔT_m = Log mean temperature difference times the correction factor F for a multipass tube side, from TEMA, °F

U = Overall heat-transfer coefficient, Btu per (sq ft)(°F)(hr)

U_o = Overall heat-transfer coefficient based on the actual outside area (plain and finned), Btu per (sq ft)(°F)(hr)

V = Water velocity inside tube, ft per sec

w = Shell-side mass-flow rate, lbs per hr

μ = Fluid viscosity at bulk temperature

μ_w = Fluid viscosity at tube wall temperature

μ/μ_w = Viscosity ratio

ϕ = Fin efficiency (See Equation (2).)

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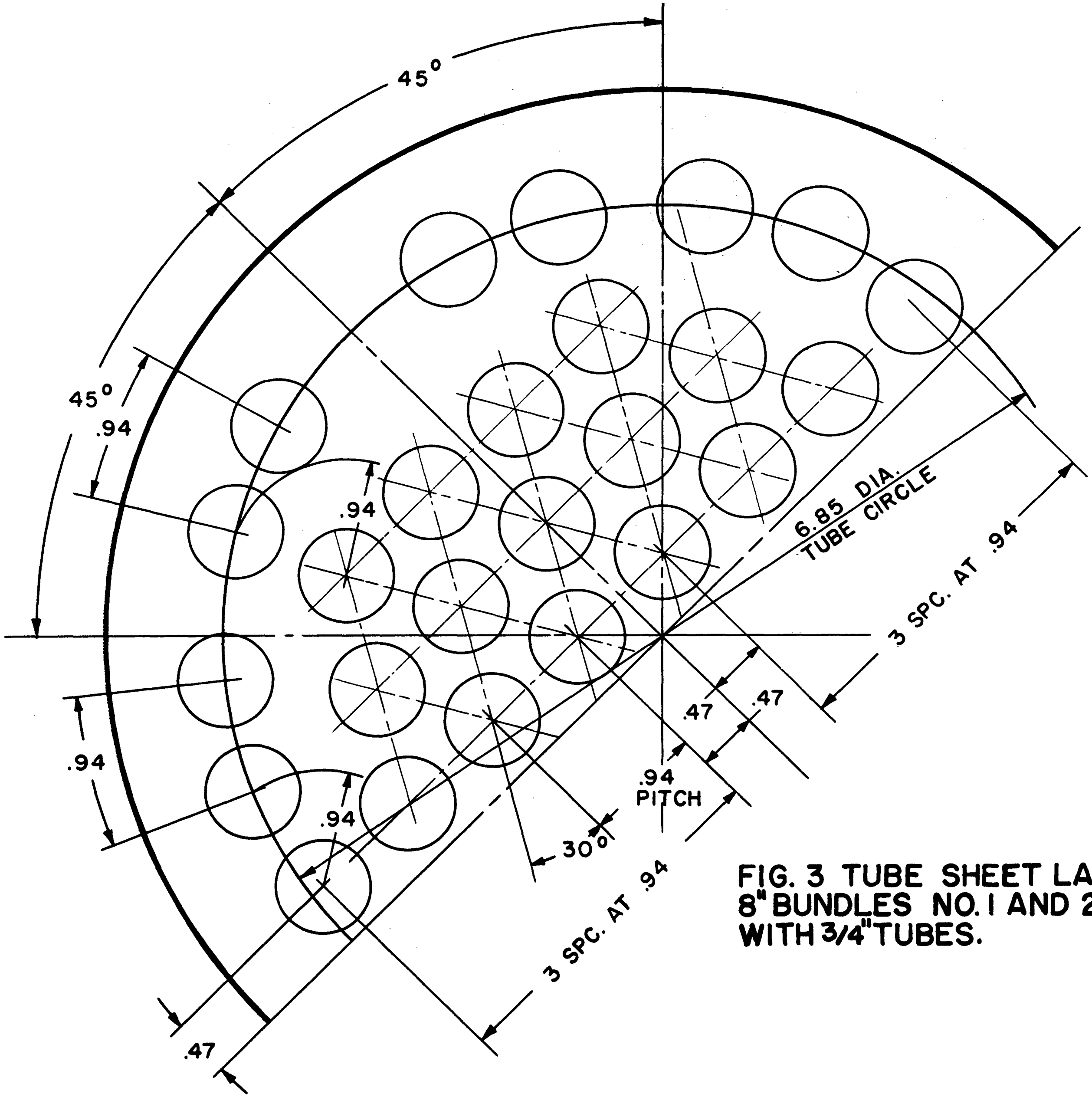
APPENDIX



50 TUBE HOLES
$.755 {}^{+.003}_{-.002}$ DIA.

LAYOUT SYMMETRICAL
ABOUT 45° ℄
DIMENSIONS IN INCHES

FIG. 3 TUBE SHEET LAYOUT FOR
8" BUNDLES NO. 1 AND 2
WITH 3/4" TUBES.

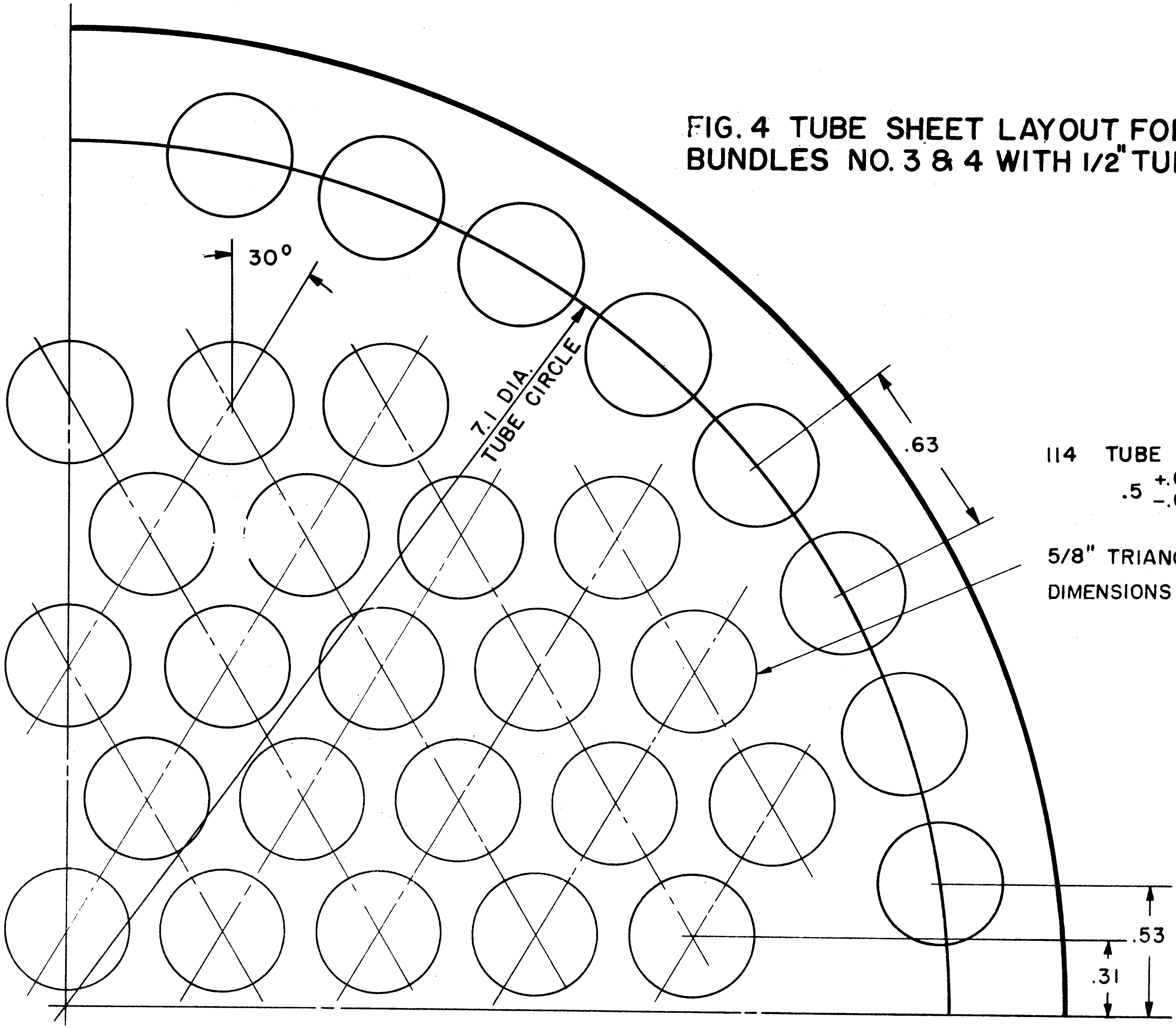
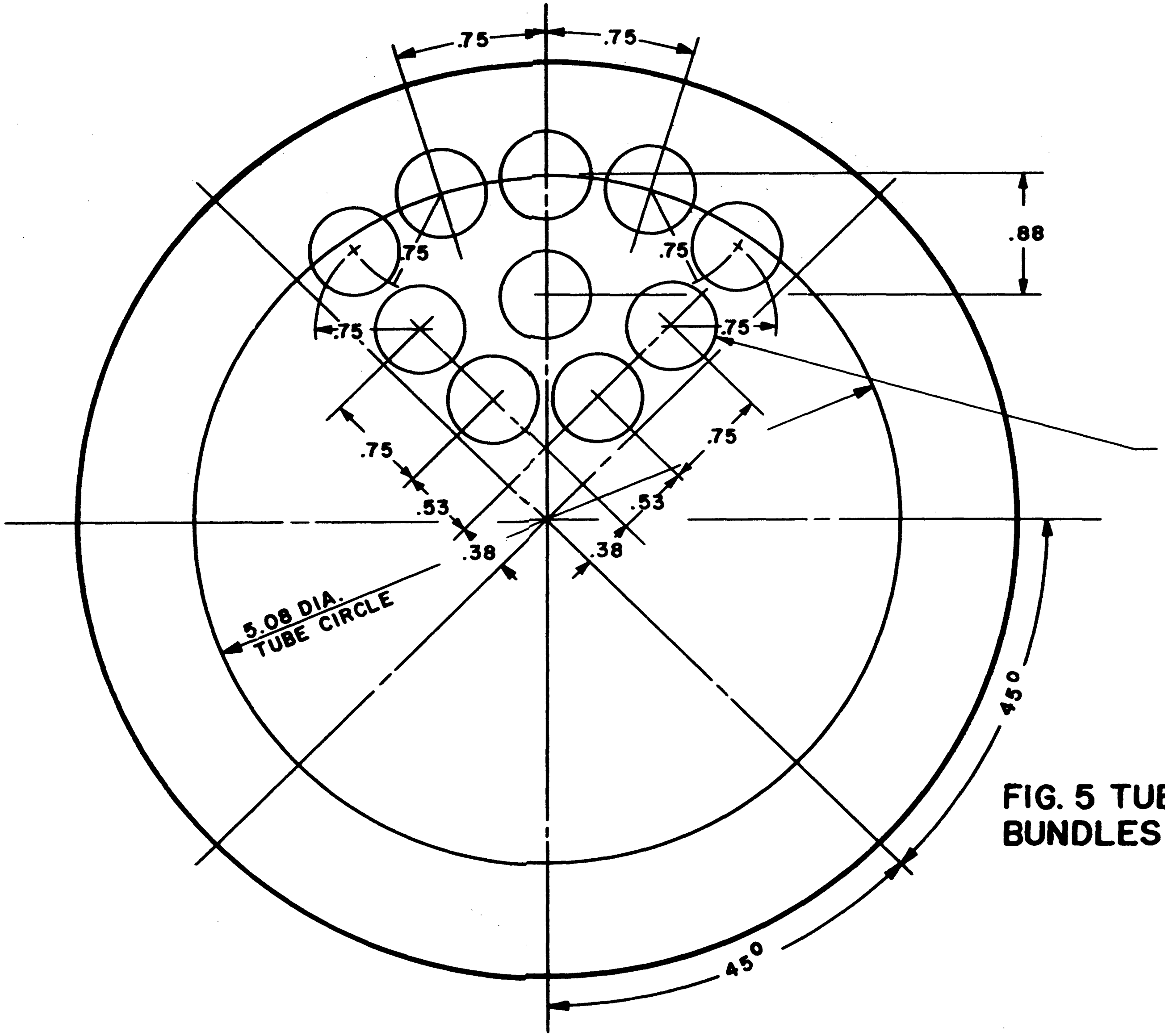


FIG. 4 TUBE SHEET LAYOUT FOR 8" BUNDLES NO. 3 & 4 WITH 1/2" TUBES

114 TUBE HOLES $.5^{+.003}_{-.002}$

5/8" TRIANGULAR PITCH

DIMENSIONS IN INCHES



40 TUBE HOLE LAYOUT
.630 $^{+.003}_{-.002}$ DIA.

LAYOUT SYMMETRICAL ABOUT 45° ℄

DIMENSION IN INCHES

FIG. 5 TUBE SHEET LAYOUT FOR 6" BUNDLES NO. 5 & 6 WITH 5/8" TUBES

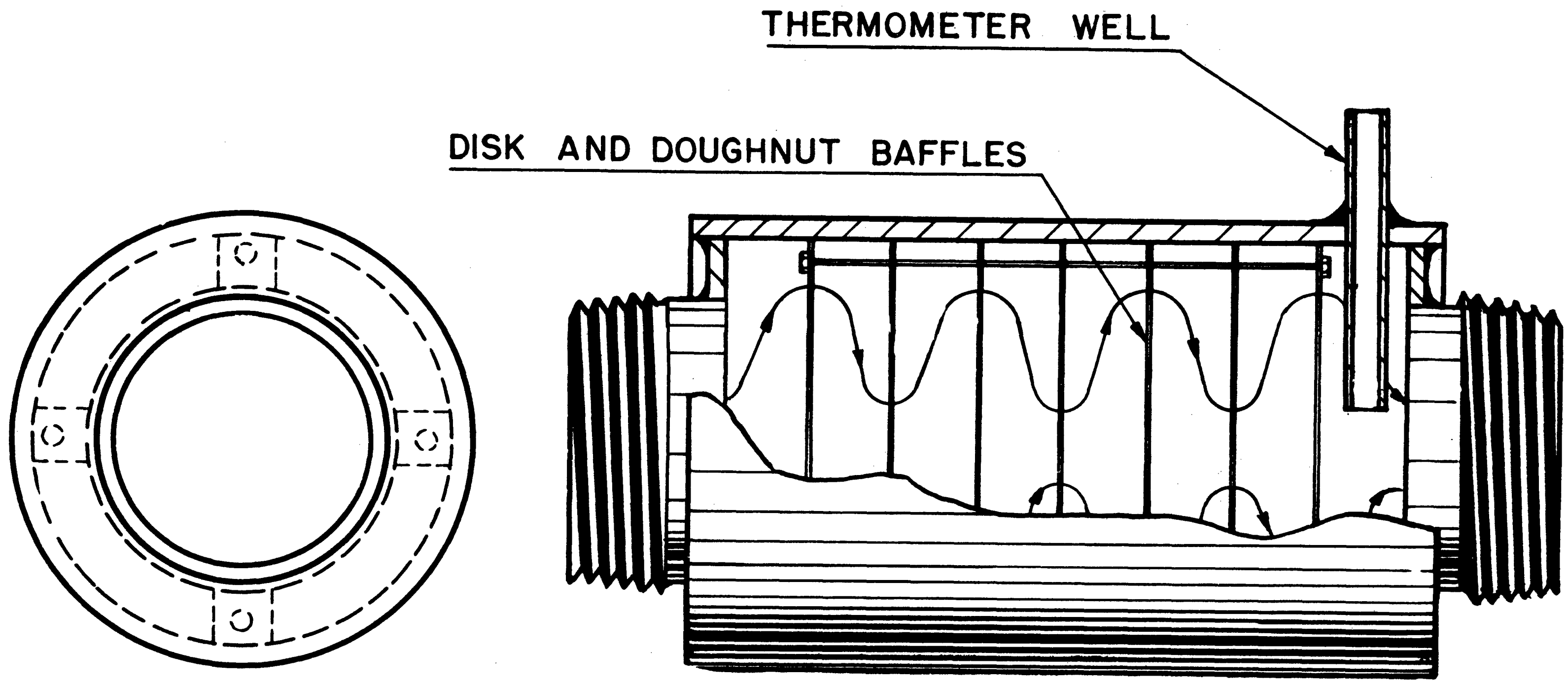


FIG. 10 SKETCH SHOWING INSIDE DETAIL OF MIXING CHAMBERS

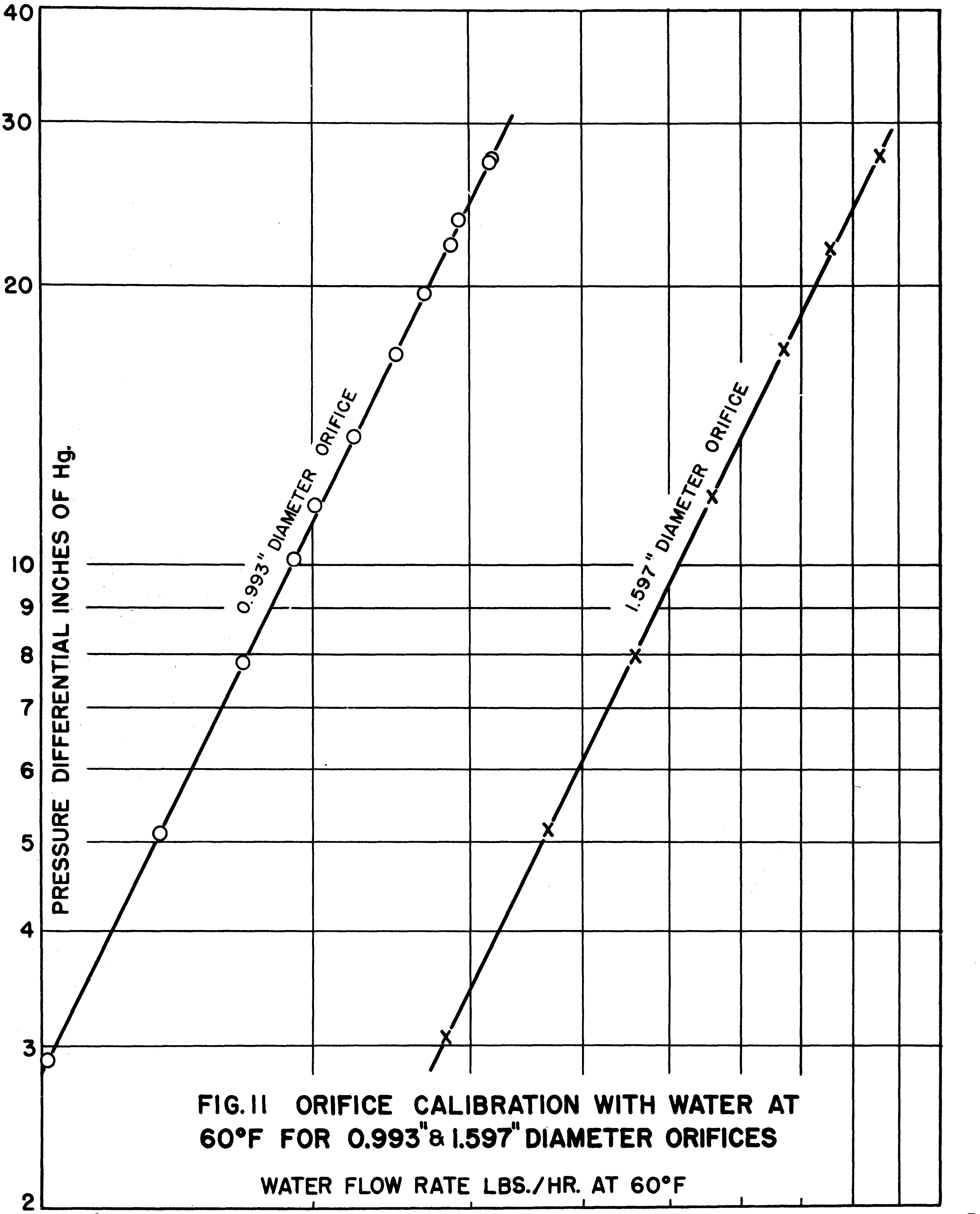


FIG. 11 ORIFICE CALIBRATION WITH WATER AT 60°F FOR 0.993" & 1.597" DIAMETER ORIFICES

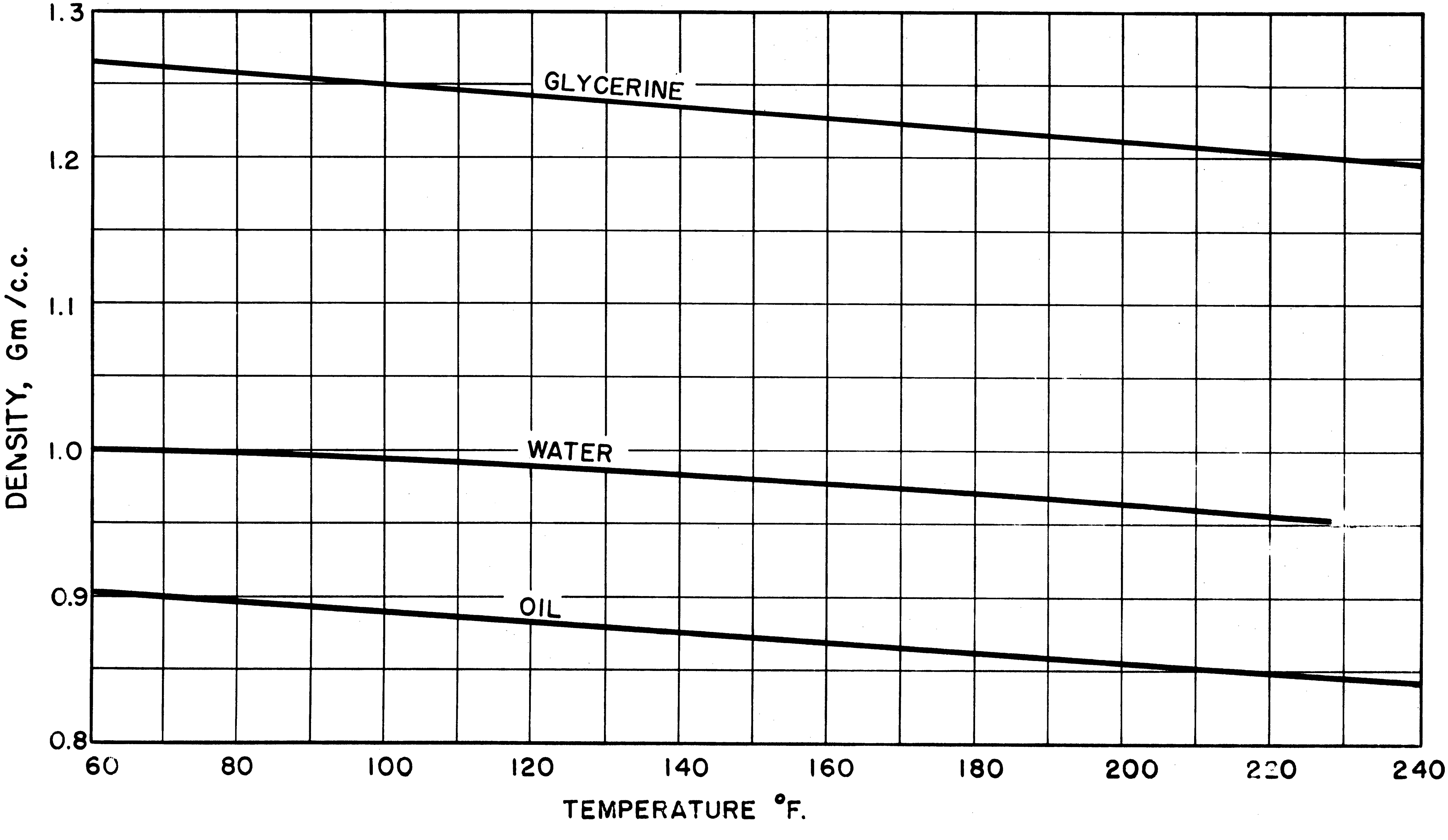


FIG.12 DENSITY OF FLUIDS

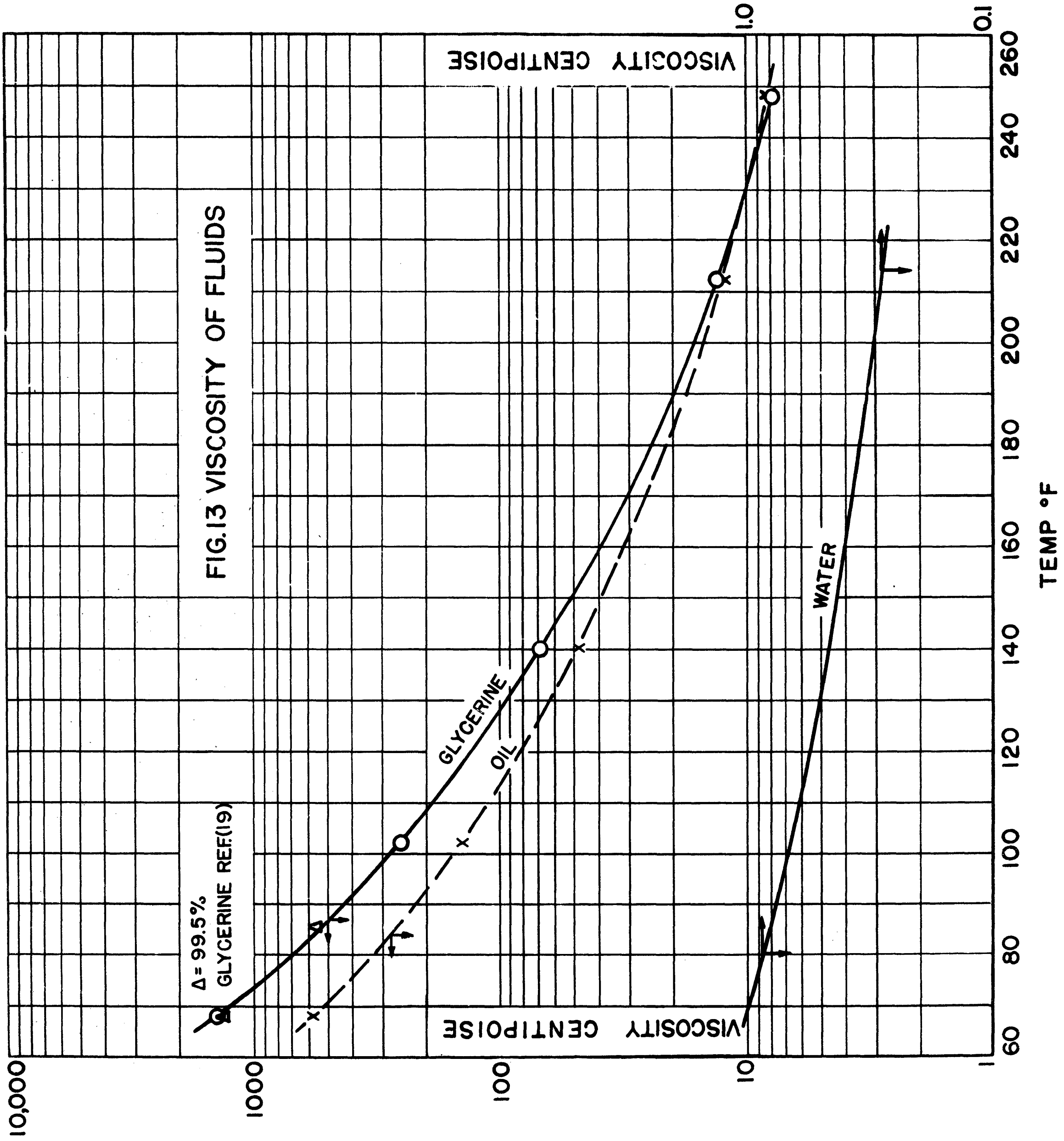


FIG.13 VISCOSITY OF FLUIDS

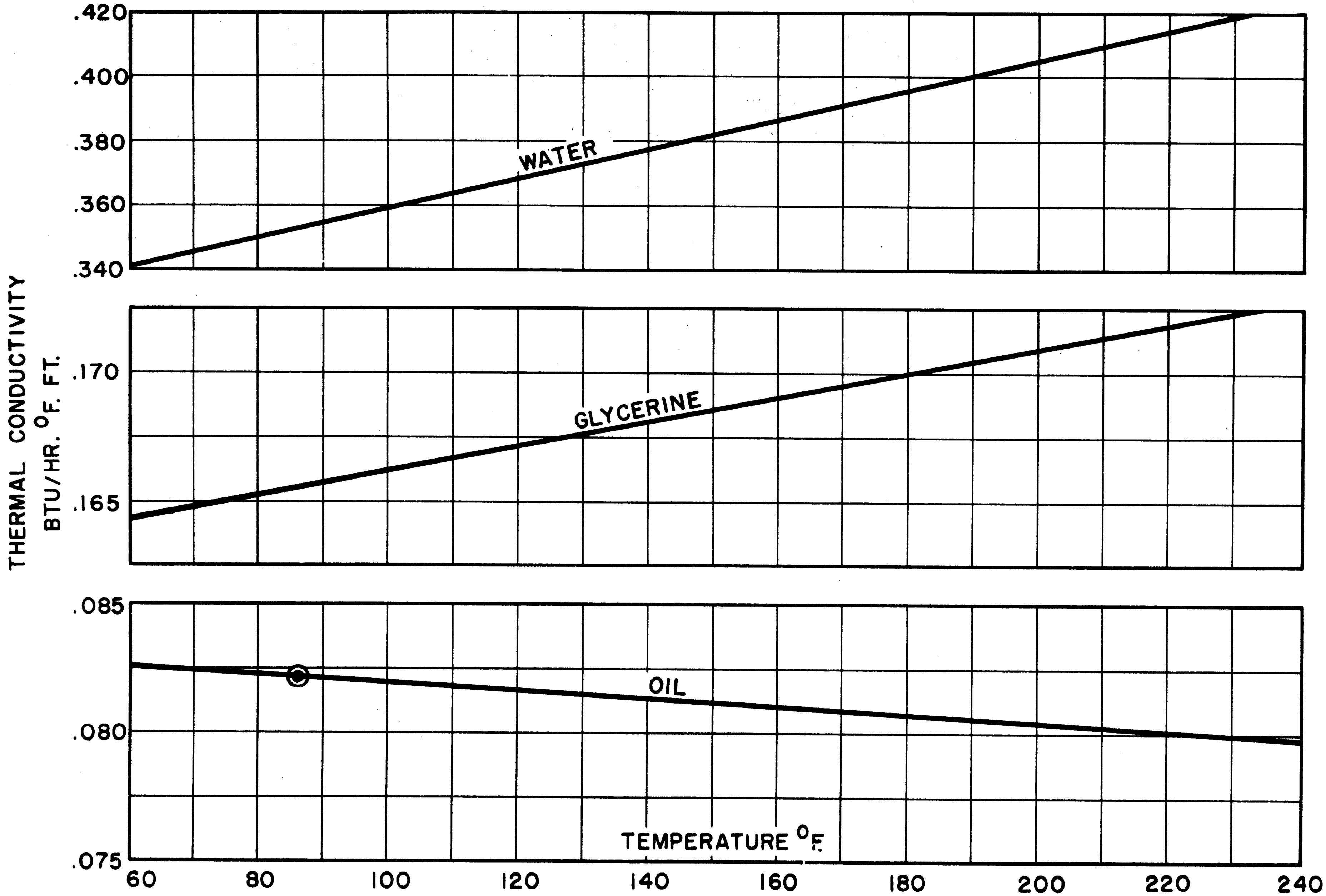


FIG. 14 THERMAL CONDUCTIVITY OF FLUIDS

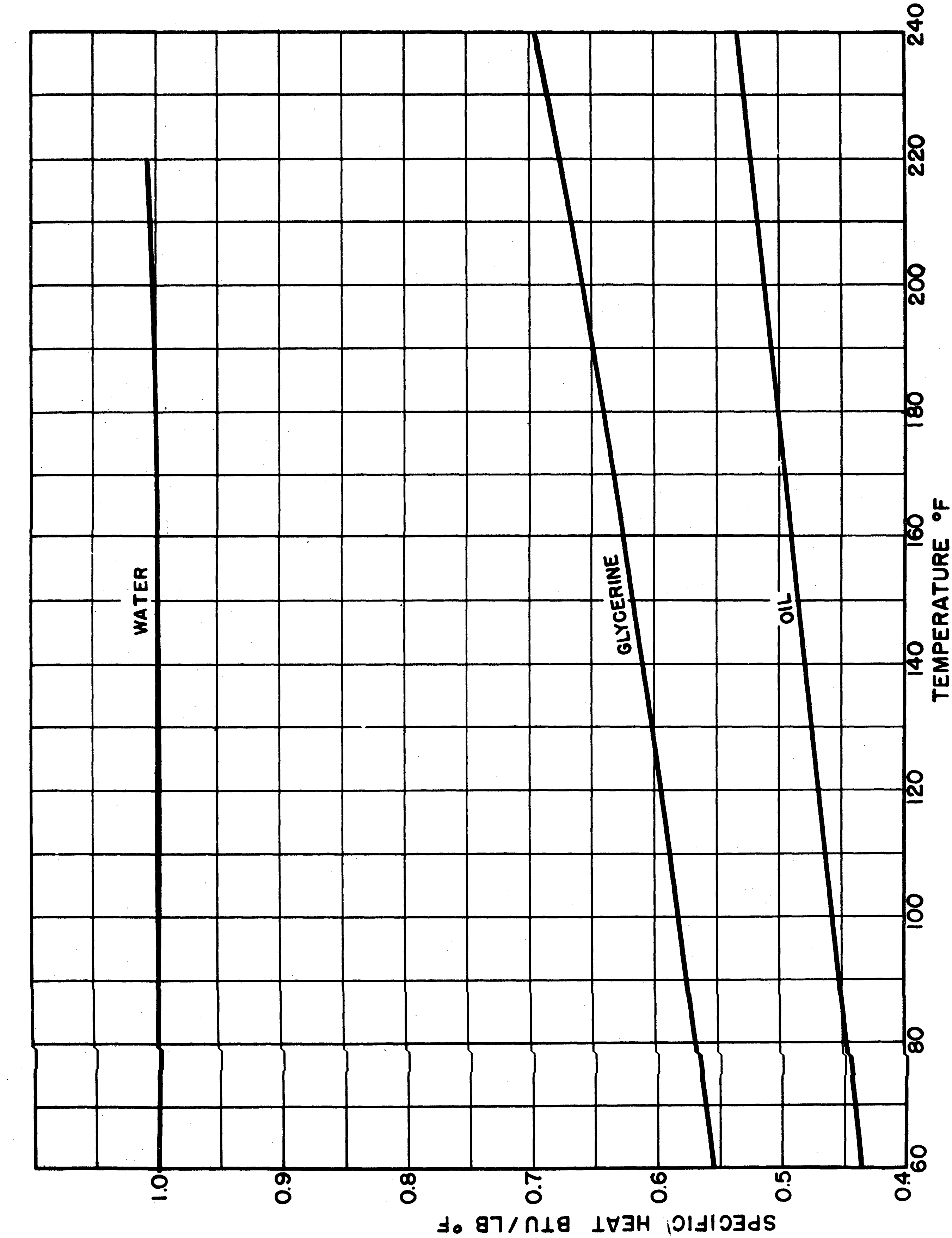


FIG.15 SPECIFIC HEAT OF FLUIDS

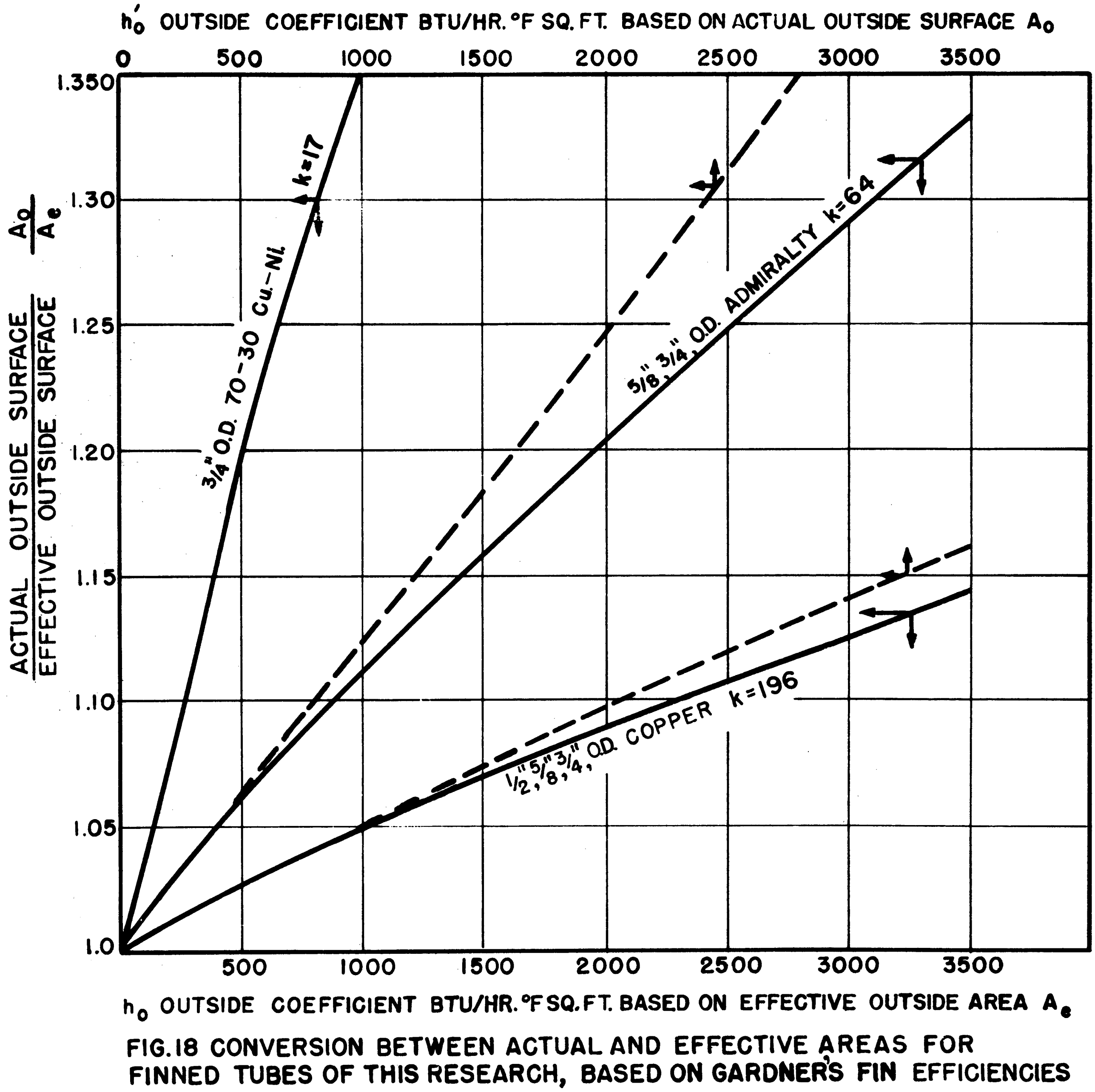


FIG. 18 CONVERSION BETWEEN ACTUAL AND EFFECTIVE AREAS FOR FINNED TUBES OF THIS RESEARCH, BASED ON GARDNER'S FIN EFFICIENCIES

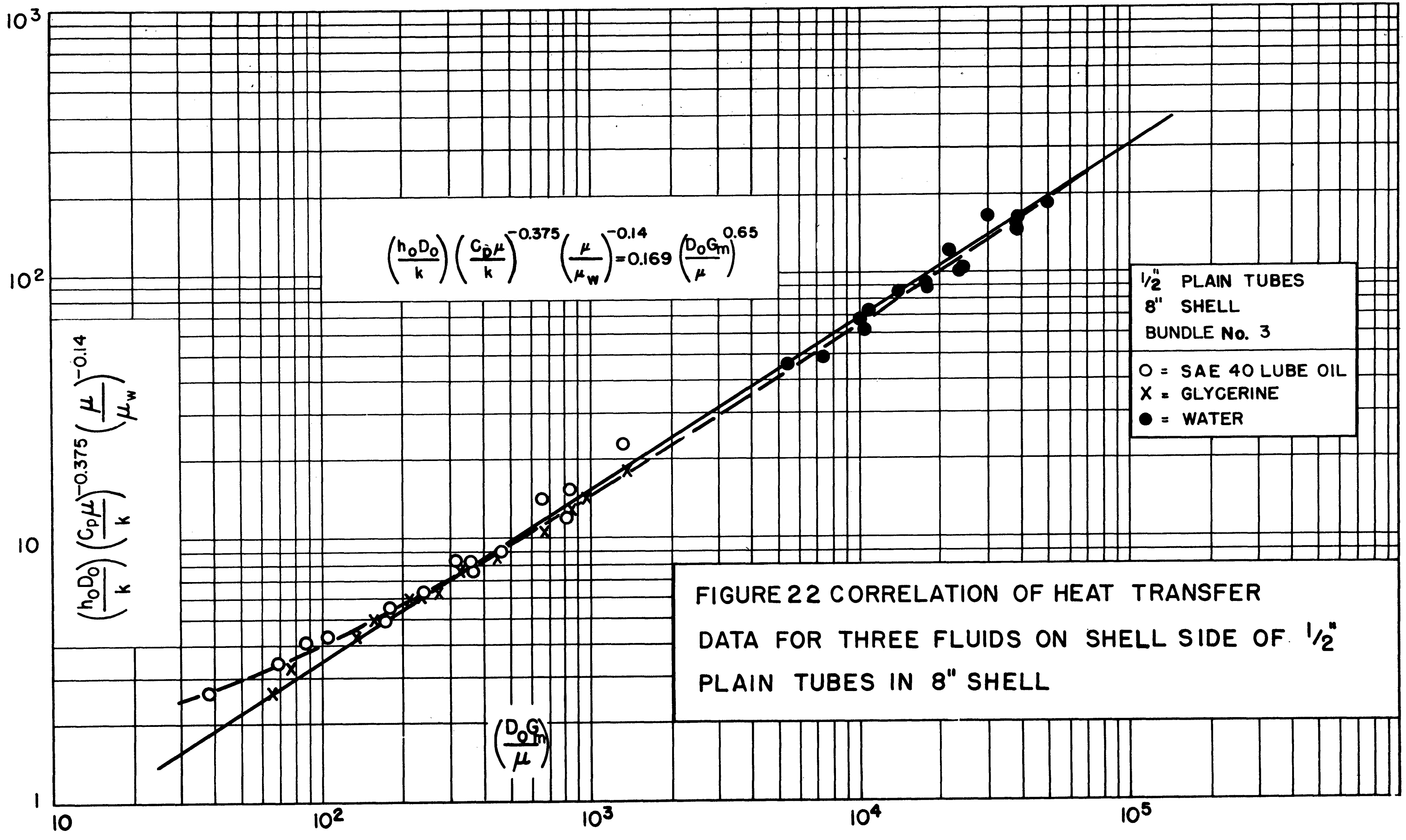


FIGURE 22 CORRELATION OF HEAT TRANSFER DATA FOR THREE FLUIDS ON SHELL SIDE OF 1/2" PLAIN TUBES IN 8" SHELL

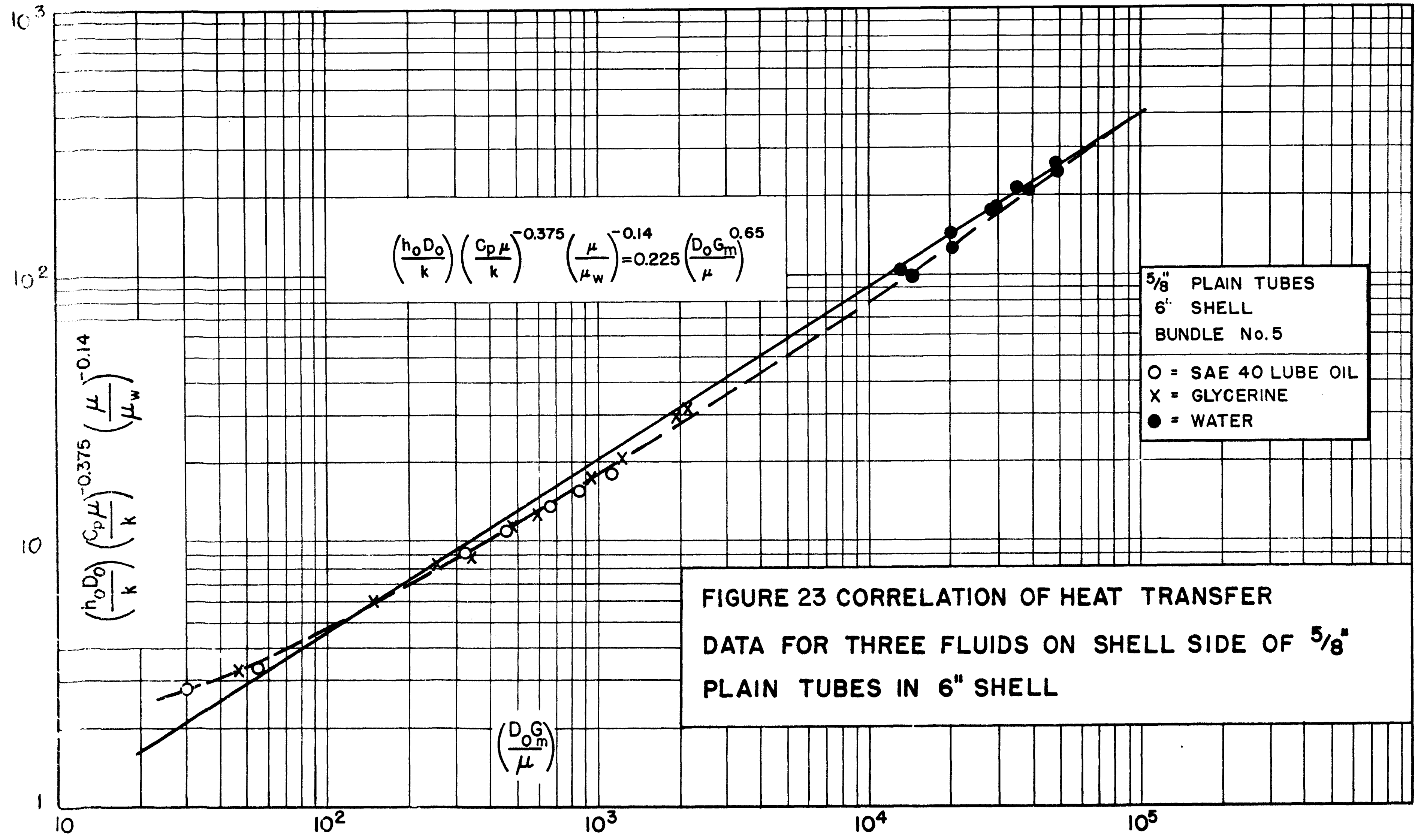


FIGURE 23 CORRELATION OF HEAT TRANSFER DATA FOR THREE FLUIDS ON SHELL SIDE OF 5/8" PLAIN TUBES IN 6" SHELL

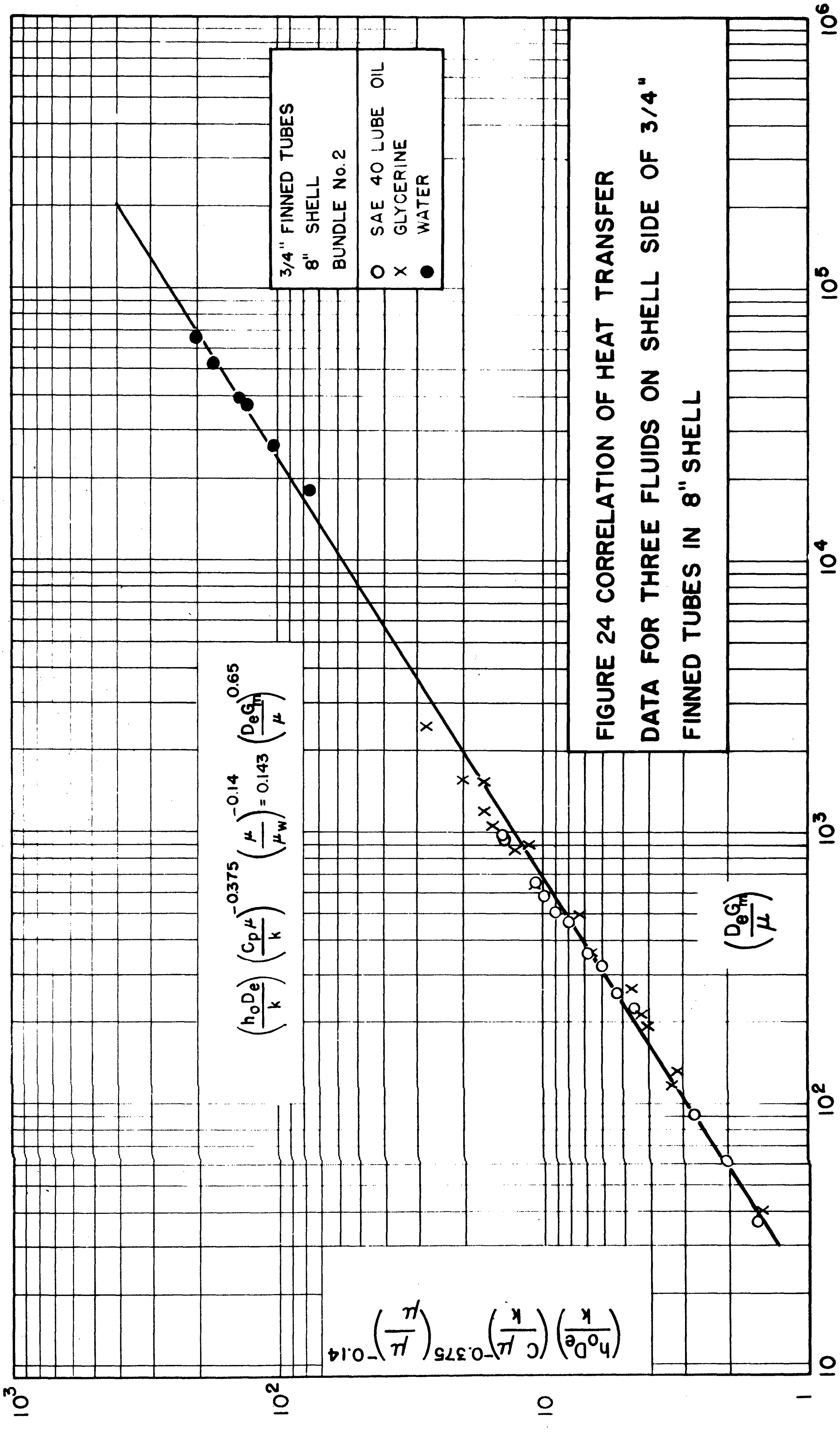


FIGURE 24 CORRELATION OF HEAT TRANSFER DATA FOR THREE FLUIDS ON SHELL SIDE OF 3/4" FINNED TUBES IN 8" SHELL

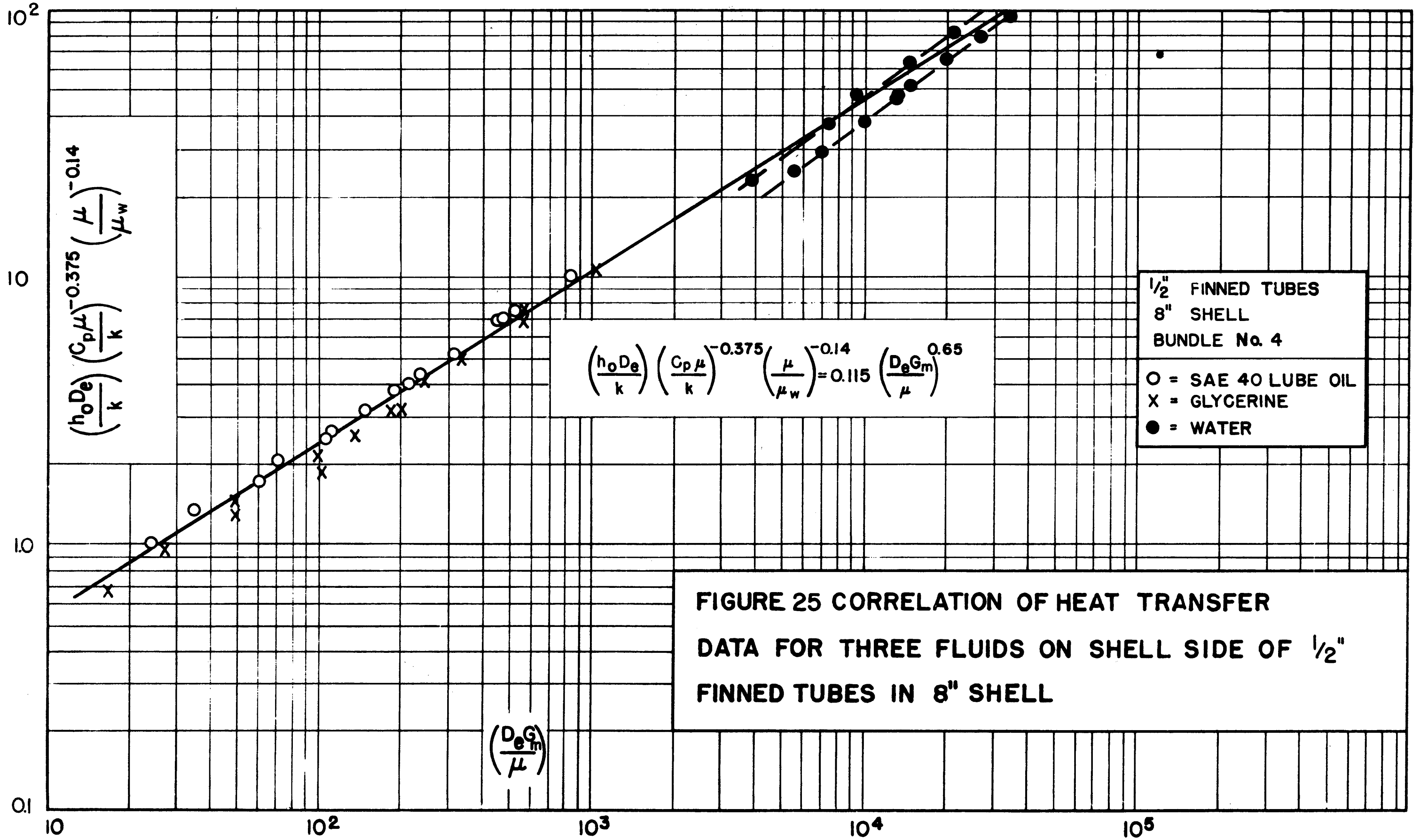


FIGURE 25 CORRELATION OF HEAT TRANSFER DATA FOR THREE FLUIDS ON SHELL SIDE OF 1/2" FINNED TUBES IN 8" SHELL

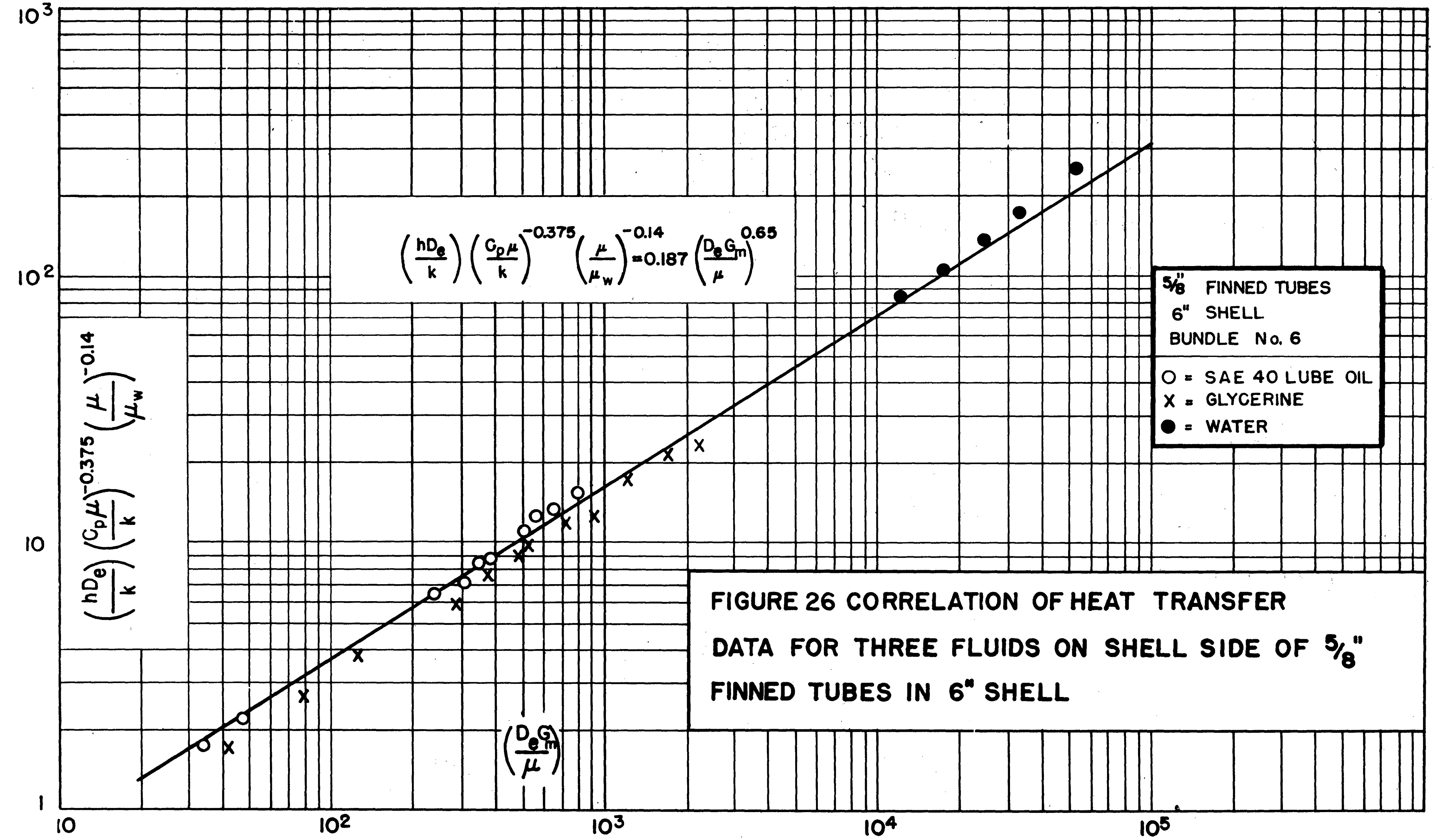


FIGURE 26 CORRELATION OF HEAT TRANSFER DATA FOR THREE FLUIDS ON SHELL SIDE OF 5/8" FINNED TUBES IN 6" SHELL

TABLE I

DIMENSIONS OF EXCHANGER SHELLS, BUNDLES, AND TUBES

Quantity	8-Inch Exchanger				6-Inch Exchanger	
Shell Inside Diameter, in.	7.972				6.008	
Number of Tube Passes in Exchanger	2				2	
Tube-Side Connection	2-1/2-inch I.P.T.				2-inch I.P.T.	
Shell-Side Connection	3-inch I.P.T.				2-inch I.P.T.	
Tube Bundle No.	1	2	3	4	5	6
Length of Tube Bundle, in.	48	48	48	48	48	48
Length of Tubes in Bundle, in.	46.64	46.64	46.88	46.88	46.64	46.64
Type of Tube	Admiralty Plain	Admiralty Finned	Copper Plain	Copper Finned	Admiralty Plain	Admiralty Finned
Tube Outside Diameter, in.	0.751	0.735	0.504	0.486	0.621	0.620
Tube Inside Diameter, in.	0.646	0.495	0.430	0.305	0.517	0.411
Tube Root Diameter, in.		0.639		0.378		0.520
Tube Equivalent Outside Diameter, in.		0.660		0.416		0.554
Fins per Inch		18.42		19.33		18.10
Height of Fins, in.		0.048		0.054		0.050
Fin Thickness at Midpoint, in.		0.0150		0.0155		0.0160
Tube Outside Area, sq ft per ft	0.196	0.410	0.132	0.304	0.163	0.361
Number of Tubes in Bundle	50	50	114	114	40	40
Total Outside Area of Tubes, sq ft	38.25	79.00	59.08	163.53	25.32	54.50
Total Inside Area of Tubes, sq ft	32.92	25.00	51.31	33.13	21.04	16.70
Baffle Outside Diameter, in.	7.933	7.933	7.930	7.930	5.956	5.956
Height of Baffle Cut, in.	1.94	1.94	3.07	3.07	2.03	2.03
Length of Baffled Section, in.	32	32	32	32	40	40
Number of Baffles	9	9	9	9	11	11
Cross-Sectional Area for Flow Inside Tubes per Pass, sq ft	0.0568	0.0335	0.0575	0.0286	0.02916	0.0184
Cross-Sectional Area for Flow Outside Tubes, sq ft	0.0540	0.0643	0.0722	0.0886	0.0465	0.0503

TABLE III

EXAMPLE DATA AND CALCULATIONS OF COEFFICIENTS

Run No. 26: 6-in. exchanger, Bundle No. 6
5/8-in. O.D. finned Admiralty tubes

Water shell side: 0.993-in. orifice
Water tube side: 1.597 in. orifice

ORIGINAL DATA

	Temperature Readings, °C				Pressure Drop Manometer Readings, Inches Hg							
	Shell Side		Tube Side		Shell Side				Tube Side			
	In No. 7	Out No. 8	In No. 5	Out No. 6	Orifice		Exchanger		Orifice		Exchanger	
					L	R	L	R	L	R	L	R
(a)	79.85	72.10	65.05	70.50	7.90	8.05	2.50	2.15	1.85	2.00	2.70	2.95
	79.90	72.25	65.15	70.60	7.90	8.05	2.50	2.15	1.85	2.00	2.70	2.95
	79.85	72.20	65.05	70.50	7.90	8.05	2.50	2.20	1.85	2.00	2.70	2.90
	79.90	72.20	65.05	70.55	7.90	8.10	2.50	2.20	1.85	2.00	2.70	2.90
Av.	79.88	72.19	65.08	70.54	7.90	8.06	2.50	2.18	1.85	2.00	2.70	2.93
(b)												
	79.90	72.00	66.85	70.65	7.85	8.05	2.50	2.20	3.80	3.90	5.25	5.50
	79.95	72.25	67.15	70.90	7.85	8.05	2.50	2.20	3.85	3.90	5.25	5.50
	80.00	72.30	66.95	70.70	7.95	8.05	2.50	2.20	3.85	3.90	5.30	5.55
	80.25	72.20	67.00	70.75	7.90	8.10	2.50	2.20	3.85	3.95	5.25	5.55
Av.	80.03	72.19	66.99	70.75	7.89	8.06	2.50	2.20	3.84	3.91	5.26	5.53
(c)												
	79.95	72.20	68.30	71.00	7.90	8.10	2.50	2.20	6.50	6.55	8.60	8.95
	79.95	72.10	68.00	70.75	7.90	8.05	2.50	2.20	6.50	6.60	8.60	9.00
	80.00	72.20	68.25	71.00	7.95	8.10	2.50	2.20	6.50	6.60	8.60	9.00
	79.90	72.15	68.05	70.85	7.95	8.10	2.50	2.20	6.45	6.60	8.60	9.00
Av.	79.95	72.16	68.15	70.90	7.93	8.09	2.50	2.20	6.49	6.59	8.60	8.99
(d)												
	79.95	72.10	68.85	71.00	7.90	8.05	2.50	2.20	9.95	10.05	12.90	13.20
	79.80	72.20	69.05	71.20	7.90	8.05	2.50	2.20	9.95	10.10	12.85	13.20
	80.05	72.35	68.80	70.95	7.85	8.00	2.50	2.20	9.90	10.10	12.85	13.20
	80.15	72.15	68.80	71.00	7.90	8.05	2.50	2.20	9.90	10.10	12.80	13.20
Av.	79.99	72.20	68.87	71.04	7.89	8.04	2.50	2.20	9.92	10.09	12.85	13.20

TABLE III, continued

CALCULATIONS FOR RUN 26a

Tube Side

Temperatures:

Water in	65.08°C	Water out	70.54°C
Correction	+0.02°C	Correction	+0.37°C
	65.10°C		70.91°C
	or 149.18°F (Col.2)*		or 159.58°F (Col.3)

Flow Rates:

Left	1.85 in. Hg
Right	2.00 in. Hg
Manometer Reading	3.85

From Fig. 11, flow rate at 60°F = 31,700 lbs/hr

Flow rate corrected for temperature $= 31{,}700 \times \sqrt{\frac{\rho_{60}}{\rho_{155}}} = 31{,}700 \times 0.99$

$= 31{,}400$ lbs/hr (Col.5)

Shell Side

Temperatures:

Water in	79.88°C	Water out	72.19°C
Correction	+0.68°C	Correction	+0.40°C
	80.56°C		72.59°C
	or 177.01°F (Col.8)		or 162.66°F (Col.9)

Flow Rates:

Left	7.90 in. Hg
Right	8.06 in. Hg
Manometer Reading	15.96 in. Hg

From Fig. 11, flow rate at 60°F = 24,000 lbs/hr

Flow rate corrected for temperature = 24,000 x 0.99 = 23,800 lbs/hr (Col.11)

*These column numbers refer to Table IV.

TABLE III, continued

Heat Transfer

Tube Side:

$31{,}400 \times (159.58 - 149.18) = 326{,}000$ Btu/hr (Col.6)

Shell Side:

$23{,}800 \times (177.01 - 162.66) = 341{,}000$ Btu/hr (Col.12)

Average 334,000

Mean Temperature Difference

$$\text{L.M.T.D.} = \frac{(162.66 - 149.18) - (177.01 - 159.58)}{\ln \dfrac{162.66 - 149.18}{177.01 - 159.58}}$$

$$= 15.30°F$$

Correction for two-pass tube side, Fig. T-4A TEMA, F = 0.887*

Mean temperature difference = 0.887 x 15.30 - 13.58 (Col.14)

Overall Coefficient based on outside area of 54.5 sq ft (Table I)

$$U_o = \frac{q}{A\,\Delta T_m} = \frac{334{,}000}{54.5 \times 13.58} = 451 \text{ Btu/(hr)(°F)(sq ft)} \quad \text{(Col.15)}$$

Similar calculations give the following overall coefficients for 26b, 26c, and 26d:

Run	U_o	$1/U_o$	V	$V^{0.8}$	$1/V^{0.8}$	N	$N/V^{0.8}$
26a	451	.00222	7.75	5.14	.194	.860	.167
26b	524	.00191	11.03	6.83	.146	.854	.125
26c	584	.00171	14.31	8.40	.119	.850	.101
26d	636	.00157	17.73	9.99	.101	.847	.0855

*When the F of TEMA was less than about 0.9, Equation (19), p. 145 of McAdams, was used to obtain the mean temperature difference directly, permitting the reporting of F to the third place.

TABLE III, continued

Linear Velocity of Water in the tubes of 0.0184 sq ft cross section = V

$$= \frac{31{,}400\ (\text{lbs/hr}) \times 0.01636\ (\text{cu ft/lb})}{3600 \times 0.0184\ (\text{sq ft})} = 7.75\ \text{ft/sec} \qquad (\text{Col.}7)$$

This velocity is taken to the 0.8 power and the reciprocal obtained as shown. A correction to bring all runs to the same equivalent water temperature of 120°F is obtained from Equation (9c), page 183 of McAdams:[13]

$$N = \frac{2.32}{1 + 0.011\ T}$$

where T is the mean water temperature in °F. Then

$$N = \frac{2.32}{1 + (0.011 \times 154)} = 0.860$$

and

$$\frac{N}{V^{0.8}} = 0.167 \qquad (\text{Col.}17)$$

The Wilson plot of the four points gives an intercept on Fig. 17 of 0.000930. Then

$$\frac{1}{h_o'} + \frac{L}{K}\,\frac{A_o}{A_m} = 0.000930$$

$$\frac{1}{h_o'} = 0.000930 - \frac{0.0545 \times 0.361}{12 \times 64^{*} \times 0.122}$$

$$= 0.000720$$

$$h_o' = \frac{1}{0.000720} = 1390$$ outside coefficient based on actual outside area.

*Standards Copper and Brass Research Association.

TABLE III, concluded

The desired shell-side coefficients are those based on the effective area in order that the usual temperature difference may be employed. Fig. 18 gives the conversion from actual to effective area for this tube, based on the outside coefficient computed for the actual area:

$$h_o = \frac{1}{0.000720} \frac{A_o}{A_e} = \frac{1.165}{0.000720}$$

$$= 1620 \text{ Btu/(hr)(°F)(sq ft effective outside area)*}$$

*Column 19 in Table IV

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

3/4" PLAIN TUBES IN 8" SHELL BUNDLE I

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_i'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-.14}$
							Water on Shell Side															
2	148.75	157.68	8.93	46530	416000	3.73	177.60	165.51	12.09	35343	426500	3.15	17.2	637	.00157	0.302						
	152.46	158.86	6.40	65043	417000	5.21	177.69	165.56	12.13	35491	431000	3.22	14.8	749	.00134	0.228		1770	282	46500	2.28	208
	153.70	158.85	5.15	83340	428000	6.66	177.30	165.11	12.19	35739	435000	3.24	13.9	813	.00123	0.188						
	155.05	159.33	4.28	103455	442000	8.29	177.29	164.98	12.31	35780	440500	3.24	12.9	896	.00112	0.156						
3	148.40	157.59	9.19	46000	422000	3.68	177.35	165.49	11.86	35800	425000	3.14	17.5	631	.00159	0.304						
	152.08	158.59	6.51	64400	419000	5.15	177.58	165.56	12.02	35800	431000	3.14	15.2	731	.00137	0.231		1825	290	46500	2.29	213
	154.00	159.10	5.10	83000	423000	6.64	177.28	165.40	11.88	35800	425000	3.14	13.7	807	.00124	0.187						
	156.09	160.20	4.11	103000	423000	8.24	177.57	165.68	11.89	35800	425000	3.14	12.4	891	.00112	0.157						
4	149.20	159.22	10.02	46100	462000	3.69	177.51	167.97	9.54	49100	468000	5.81	17.5	689	.00145	0.303						
	153.72	160.78	7.06	65200	461000	5.21	177.31	167.95	9.36	49000	459000	5.83	14.5	828	.00121	0.227		2310	367	64000	2.28	271
	155.21	160.93	5.72	83100	476000	6.65	177.30	167.61	9.69	48800	473000	5.80	13.6	915	.00109	0.187						
	157.84	162.27	4.43	105000	465000	8.40	177.26	167.95	9.31	48600	452500	5.73	11.8	1011	.00099	0.153						
5	148.86	159.64	10.78	45800	500000	3.67	177.17	168.85	8.32	60600	504000	8.85	17.8	725	.00138	0.305						
	152.80	160.77	7.97	64900	516000	5.18	177.40	168.91	8.49	61000	516000	8.85	15.6	865	.00116	0.229		2740	435	80000	2.26	322
	155.61	161.87	6.26	83200	522000	6.65	177.49	169.00	8.49	61000	516000	8.82	13.8	977	.00102	0.186						
	157.46	162.25	4.79	103000	493000	8.23	177.17	168.69	8.48	61000	518000	8.60	12.5	1058	.000945	0.156						
6	149.29	156.70	7.41	46100	342000	3.69	177.08	162.88	14.20	25000	355000	1.54	15.6	582	.00172	0.304						
	151.86	157.17	5.31	65200	346000	5.22	177.04	162.46	14.58	25100	365000	1.52	13.8	671	.00149	0.228		1435	229	32400	2.30	168
	154.60	158.72	4.12	83300	343000	6.66	178.03	163.54	14.49	25100	363000	1.52	12.6	732	.00137	0.187						
	155.30	158.79	3.49	103500	361000	8.27	178.36	163.38	14.98	25100	375000	1.52	12.2	786	.00127	0.157						
7	148.60	154.63	6.03	46100	278000	3.69	176.67	159.21	17.46	17000	297000	0.74	14.3	522	.00191	0.306						
	151.54	155.79	4.25	65400	278000	5.23	177.17	159.66	17.51	17000	298000	0.74	12.8	590	.00169	0.230		1130	181	21700	2.35	133
	151.61	155.12	3.51	82200	288000	6.57	177.08	158.86	18.22	16900	308000	0.74	12.3	631	.00159	0.192						
	152.42	155.43	3.01	102000	307000	8.15	177.40	158.90	18.50	17000	314000	0.74	11.9	685	.00146	0.161						
18	148.87	158.43	9.56	45500	435000	3.64	177.58	166.42	11.16	39700	433000	3.84	17.4	659	.00152	0.306						
	153.39	159.75	6.36	65000	414000	5.20	176.86	166.14	10.72	39700	426000	3.86	14.0	785	.00127	0.228		2050	326	52000	2.27	241
	155.71	160.77	5.06	83400	422000	6.66	177.62	166.50	11.12	39700	422000	3.86	12.8	880	.00114	0.186						
	157.19	161.17	3.98	101500	404000	8.12	176.85	166.33	10.52	39700	418000	3.86	11.5	939	.00107	0.157						
							Oil on Shell Side															
49	157.105	160.884	3.779	46600	176000	3.73	196.23	188.65	7.58	45200	174000	6.81	33.6	136.2	.00734	0.294						
	158.014	160.943	2.929	60500	177000	4.84	196.09	188.47	7.62	45200	175000	6.79	32.0	143.5	.00697	0.240		161.0	125	1285	246	17.2
	159.184	161.334	2.150	83200	179000	6.66	196.03	188.42	7.61	45200	175000	6.79	31.9	144.0	.00695	0.185						
	160.032	161.811	1.779	102000	181000	8.16	196.16	188.58	7.58	45200	174000	6.79	31.4	146.1	.00685	0.157						
50	157.098	160.265	3.167	46200	146000	3.70	196.82	187.47	9.35	32000	152000	3.74	33.5	116.0	.00862	0.296						
	157.996	160.421	2.425	60300	146000	4.82	196.05	187.16	8.89	32000	144500	3.74	32.4	117.1	.00854	0.240		133.5	103.5	915	246	14.2
	158.802	160.578	1.776	83600	148000	6.69	196.14	187.03	9.11	32000	148000	3.74	31.9	121.5	.00822	0.185						
	159.942	161.401	1.459	103000	150000	8.25	195.93	187.16	8.77	32000	143000	3.74	30.6	125.5	.00796	0.155						
51	177.280	160.128	2.848	46200	132000	3.70	196.48	186.87	9.61	25500	124500	2.60	33.0	101.5	.00985	0.296						
	158.187	160.317	2.139	61300	131000	4.90	196.23	186.51	9.72	25400	125500	2.60	32.0	104.8	.00954	0.237		116.5	90.5	717	250	12.4
	158.943	160.519	1.576	83300	131000	6.66	195.80	186.15	9.65	25600	125500	2.63	31.1	107.8	.00928	0.185						
	159.890	161.190	1.300	102000	133000	8.16	196.00	186.51	9.49	25500	123000	2.66	30.7	109.1	.00916	0.157						
52	157.447	159.795	2.348	46200	108000	3.70	196.88	185.49	11.39	18100	104400	1.52	32.2	86.2	.0116	0.296						
	158.149	159.946	1.797	61300	110000	4.90	196.88	185.52	11.36	18100	104200	1.52	31.6	88.5	.0113	0.237						
	158.858	160.144	1.286	84000	108000	6.72	196.23	184.96	11.27	18100	103500	1.52	30.8	89.3	.0112	0.184		96.0	74.5	508	250	10.3
	159.659	160.810	1.151	98200	113000	7.86	196.93	185.74	11.19	18100	102700	1.52	30.9	91.3	.01095	0.161						

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

3/4" PLAIN TUBES IN 8" SHELL BUNDLE I

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_i'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
							Oil on Shell Side															
53	157.825	159.719	1.894	46200	87500	3.70	196.05	183.58	12.47	13200	83500	0.89	30.5	73.6	.01360	0.296						
	158.282	159.721	1.439	61300	88100	4.90	195.98	183.49	12.49	13000	83400	0.89	30.4	73.8	.01355	0.238		79.2	61.5	304	254	8.35
	158.655	159.752	1.097	83500	91600	6.68	195.98	183.38	12.60	13000	83800	0.86	30.1	74.5	.01342	0.184						
	158.862	159.831	0.969	94100	91100	7.53	196.41	183.65	12.76	13100	85000	0.86	30.2	74.2	.01348	0.167						
54	122.110	125.447	3.337	46500	155000	3.70	196.52	175.30	21.22	13200	142000	0.93	61.9	62.6	.01595	0.344						
	124.585	127.044	2.459	61400	151000	4.88	196.21	175.04	21.17	13100	140000	0.91	59.6	63.8	.01571	0.273		69.8	54.1	334	273	7.70
	126.169	127.976	1.807	83300	149000	6.63	195.54	174.61	20.93	12900	136000	0.93	57.7	64.6	.01550	0.213						
	127.483	128.972	1.489	103000	153000	8.18	196.34	175.39	20.95	13000	137500	0.93	57.6	65.8	.01520	0.179						
55	122.207	126.167	3.960	45800	181000	3.64	195.39	176.41	18.98	18000	172500	1.55	61.4	75.1	.01331	0.349						
	123.964	126.941	2.977	61500	183000	4.89	196.23	177.15	19.08	18000	173500	1.55	61.0	76.3	.01310	0.274		85.0	66.0	462	273	9.03
	125.440	127.665	2.225	84000	186000	5.52	196.57	177.40	19.17	18000	174000	1.55	60.4	78.0	.01286	0.218						
	126.489	128.266	1.777	103000	183000	8.18	195.63	176.76	18.87	18100	172500	1.55	58.8	79.2	.01265	0.180						
56	122.479	127.199	4.720	46200	218000	3.67	195.78	179.28	16.50	25600	214000	2.69	62.5	90.3	.01111	0.346						
	124.295	127.895	3.600	60800	219000	4.84	195.75	179.19	16.56	25600	215000	2.68	61.1	92.7	.01078	0.276		103.0	79.8	685	264	11.0
	125.848	128.520	2.672	83300	219000	6.63	195.54	179.05	16.49	25500	213000	2.69	60.0	94.3	.01060	0.213						
	127.641	129.789	2.148	102000	219000	8.11	195.58	179.35	16.23	25500	210000	2.67	58.7	95.5	.01046	0.180						
62	122.315	128.655	6.340	46100	292000	3.66	195.78	182.94	12.84	45500	296500	6.76	63.9	120.5	.00830	0.345						
	124.480	129.317	4.837	61000	295000	4.85	195.48	182.68	12.80	45500	296000	6.76	62.2	124.4	.00804	0.273		142.5	110	1170	262	16.0
	126.061	129.584	3.523	83200	299000	6.62	195.19	182.32	12.87	45300	296000	6.76	60.9	127.5	.00785	0.212						
	127.468	130.304	2.836	105000	298000	8.30	195.63	182.71	12.92	45300	297000	6.80	60.3	129.0	.00775	0.176						
63	122.608	127.850	5.242	46100	242000	3.66	195.91	180.75	15.14	32100	246500	3.75	63.1	101.0	.00990	0.346						
	124.450	128.417	3.967	61100	242000	4.86	195.66	180.55	15.11	32000	245000	3.75	61.7	103.0	.00970	0.274		116.5	90.5	800	261	13.2
	126.378	129.299	2.921	83200	240000	6.63	195.75	180.64	15.11	32000	245000	3.76	60.4	105.2	.00951	0.212						
	127.620	129.976	2.356	104500	246000	8.31	195.40	180.43	14.97	32000	243000	3.75	59.1	108.1	.00925	0.177						
64	71.139	73.269	2.130	46600	99400	3.66	112.60	105.44	7.16	28000	93500	7.60	36.8	68.6	.01458	0.458						
	71.854	73.494	1.640	61900	101500	4.80	113.04	105.73	7.31	28000	95000	7.57	36.7	70.0	.01429	0.365		77.8	63.6	105	153[illegible]	4.86
	70.729	72.000	1.271	84200	107000	6.63	113.27	105.80	7.47	27000	93000	7.64	38.2	68.8	.01452	0.286						
	70.367	71.442	1.075	105500	113500	8.31	113.74	105.91	7.83	27700	100300	7.69	38.9	72.6	.01380	0.240						
65	70.288	71.922	1.634	46600	76100	3.66	112.77	102.24	10.53	12900	63000	3.55	36.3	50.0	.0200	0.461						
	71.521	72.779	1.258	61200	77000	4.80	113.00	102.56	10.44	13300	64800	3.63	35.6	52.0	.0192	0.369		56.3	46.0	47.8	1655	3.41
	72.230	73.200	0.970	84200	81600	6.63	113.40	103.01	10.39	13500	65000	3.72	35.5	54.0	.0185	0.284						
	71.631	72.410	0.779	106000	82500	8.31	113.18	102.52	10.66	13100	64600	3.56	35.8	53.7	.0186	0.238						
66	69.514	71.461	1.947	46600	90800	3.66	113.45	104.47	8.98	19300	80400	5.20	38.5	58.2	.01720	0.463						
	69.917	71.460	1.543	61600	95100	4.84	114.15	104.86	9.29	19100	82100	5.01	38.5	60.1	.01665	0.370		65.8	53.8	72.8	1586	4.13
	70.551	71.665	1.114	84000	93600	6.60	113.04	103.96	9.08	19200	80700	5.20	37.6	60.8	.01645	0.288						
	71.067	72.001	0.934	106000	99000	8.33	113.85	104.67	9.18	19500	82900	5.24	37.6	63.1	.01584	0.238						

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

3/4" PLAIN TUBES IN 8" SHELL BUNDLE I

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_i'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-.014}$
							Glycerine on Shell Side															
104	195.30	200.44	5.14	38800	199000	3.16	228.33	218.05	10.28	27700	195000	2.06	24.9	207	.00484	0.291	1100	260	94.5	1210	106	17.2
105	195.108	200.98	5.90	45100	264000	3.67	228.47	219.60	8.87	45000	273000	5.24	25.6	274	.00365	0.258	1240	361	131	1990	106	23.9
106	195.30	202.80	7.50	45100	338000	3.67	228.38	221.77	6.61	73000	332000	12.50	26.1	335	.00298	0.258	1245	476	173	3270	104	31.7
113	158.88	164.93	6.05	52100	315000	4.18	193.75	186.84	6.91	70700	318000	12.62	28.1	295	.00339	0.266	1205	402	147	1776	175	22.6
114	195.58	163.87	4.28	52100	223000	4.18	193.17	184.55	8.62	40700	228000	4.85	26.8	218	.00459	0.266	1205	271	99.3	991	179	15.1
115	195.48	162.72	3.24	34700	112000	2.78	193.39	178.95	14.44	13500	126000	0.86	24.8	125.5	.00796	0.368	835	149	54.8	313	188	8.25
116	121.98	125.20	3.22	35500	114000	2.82	158.27	145.72	12.55	15400	120500	1.46	28.2	108.3	.00923	0.430	747	128	47.4	150	420	5.47
117	112.13	127.11	4.98	36000	180000	2.86	157.42	149.88	7.54	42800	200000	5.67	28.8	172	.00581	0.421	760	226	83.5	418	403	9.68
118	123.44	128.98	5.54	39000	216000	3.10	158.85	152.11	6.73	52300	221000	8.91	29.2	196	.00510	0.393	815	263	97.5	565	378	11.5
119	76.86	81.18	4.32	40000	173000	3.15	122.07	113.16	8.91	32300	171000	8.11	38.4	117.0	.00855	0.497	645	144	53.6	97.5	1340	4.26
120	77.09	80.85	3.51	33100	116000	2.62	121.68	110.80	10.88	16400	107000	3.88	37.1	80.9	.01238	0.575	558	95.1	35.4	45.1	1425	2.78

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

3/4" FINNED TUBES IN 8" SHELL BUNDLE 2

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_i'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
										Water on Shell Side												
19	148.9	161.13	12.15	31600	384000	4.29	177.22	166.14	11.08	36200	401000	1.61	15.2	328	.00305	0.268						
	154.76	162.72	7.96	45400	361000	6.16	176.83	166.35	10.48	36100	378000	1.61	11.8	398	.00252	0.198		1350	189	39800	2.27	139.5
	157.86	163.90	6.04	57500	348000	7.81	177.19	166.98	10.21	35900	367000	1.62	10.2	445	.00226	0.162						
	159.29	163.83	4.54	82700	375000	11.22	177.19	166.63	10.56	35800	378000	1.62	9.20	519	.00197	0.121						
20	148.50	162.14	13.64	32600	445000	4.42	177.04	167.97	9.07	49400	447000	3.14	15.8	356	.00281	0.261						
	152.87	162.88	10.01	45700	457000	6.21	177.28	168.04	9.24	49600	459000	3.14	13.7	424	.00236	0.197		1680	234	53300	2.28	172.5
	155.41	163.29	7.88	57900	455000	7.86	177.33	167.92	9.41	49500	466000	3.14	12.2	475	.00211	0.162						
	158.18	163.51	5.33	83100	433000	11.30	176.94	167.54	9.40	49500	465000	3.14	10.5	549	.00182	0.121						
21	149.56	163.22	13.66	32400	442000	4.40	177.11	169.56	7.55	61400	463000	4.71	15.8	361	.00277	0.262						
	154.72	164.65	9.93	45100	448000	6.12	177.08	169.71	7.37	61300	452000	4.73	12.8	444	.00226	0.197		1920	272	66000	2.25	201.0
	157.03	164.75	7.72	57400	443000	7.80	177.17	169.59	7.58	60900	461000	4.67	11.7	488	.00205	0.163						
	159.71	165.33	5.42	82700	448000	11.22	177.24	169.56	7.68	60600	465000	4.66	10.1	572	.00175	0.121						
22	149.86	159.71	9.85	32600	321000	4.42	177.24	163.44	13.80	25200	348000	0.81	13.9	304	.00329	0.262		990	139	26600	2.31	102.0
	152.69	160.02	7.33	45500	334000	6.18	177.26	163.31	13.95	25300	353000	0.81	12.3	352	.00284	0.198						
	154.81	160.59	5.78	57800	334000	7.85	177.44	163.33	14.11	25200	356000	0.81	10.9	400	.00250	0.163						
	156.22	160.20	3.98	83200	331000	11.30	176.97	162.75	14.22	25200	358000	0.81	9.75	446	.00224	0.121						
23	148.75	157.42	8.67	32100	278000	4.36	177.40	160.07	17.33	17100	296500	0.37	13.5	270	.00370	0.266						
	151.12	157.33	6.21	45200	281000	6.14	176.79	159.53	17.26	17100	295000	0.37	11.7	312	.00321	0.201		724	102.	18400	2.33	74.7
	151.79	156.86	5.07	57500	291000	7.81	177.15	158.95	18.20	17100	311000	0.37	11.2	340	.00296	0.166						
	154.35	157.87	3.52	83800	295000	11.38	177.60	159.69	17.91	17100	306000	0.37	9.90	385	.00260	0.122						
29	148.59	160.82	12.23	32100	393000	4.35	177.15	165.96	11.19	36100	404000	1.65	15.4	328	.00305	0.270						
	152.38	161.24	8.86	45300	402000	6.14	177.38	165.87	11.51	36100	416000	1.65	13.5	381	.00262	0.199		1245	175	37500	2.28	129.1
	155.43	162.18	6.75	58000	392000	7.86	177.38	166.07	11.31	36100	409000	1.65	11.7	432	.00231	0.162						
	156.04	161.56	5.52	71900	396000	9.75	176.81	165.38	11.43	36100	413000	1.65	11.1	460	.00217	0.137						
										Oil on Shell Side												
39	122.34	136.90	14.56	31700	461000	4.30	195.67	177.02	18.65	50300	474000	5.16	58.1	101.7	.00983	0.297						
	127.35	137.91	10.56	45500	481000	6.17	195.78	177.13	18.65	52000	490000	5.62	54.3	112.5	.00889	0.250		151.0	103.1	975	270	14.2
	129.87	137.98	8.11	59800	485000	8.11	195.57	176.86	18.71	50800	480000	5.53	51.9	117.6	.00850	0.176						
	133.52	139.19	5.67	84800	481000	11.51	195.67	176.79	18.88	50800	484000	5.34	49.6	123.0	.00812	0.132						
40	122.25	133.07	10.82	32900	356000	4.46	195.99	173.93	22.06	32200	358000	2.51	56.5	79.9	.01251	0.291						
	125.78	133.61	7.83	45600	357000	6.19	195.69	173.52	22.17	32000	357000	2.50	54.1	83.4	.01200	0.223		105.2	71.5	585	277	9.88
	128.87	134.80	5.93	60100	356000	8.15	195.87	173.75	22.12	32000	356000	2.50	51.8	87.0	.01150	0.177						
	131.30	135.56	4.26	85100	362000	11.57	195.67	173.68	21.99	32000	354000	2.50	48.6	92.3	.01082	0.133						
41	122.59	132.19	9.60	31700	305000	4.30	195.57	172.54	23.03	25700	298000	1.72	55.6	67.8	.01475	0.300						
	125.24	132.04	6.80	45700	311000	6.20	195.94	172.47	23.47	25700	304000	1.72	54.8	71.0	.01410	0.223		85.2	58.0	469	283	7.96
	128.44	133.46	5.02	60000	301000	8.14	195.96	172.63	23.33	25700	302000	1.72	52.3	73.1	.01365	0.178						
	130.41	134.08	3.67	84800	311000	11.51	195.75	172.45	23.30	25700	302000	1.72	50.6	76.5	.01310	0.134						
42	122.36	129.63	7.27	32700	238000	4.43	196.14	170.58	25.56	18100	232500	1.03	56.3	52.9	.01890	0.296						
	122.22	130.50	5.28	45700	241000	6.20	196.05	170.58	25.47	18100	232000	1.03	54.1	55.4	.01808	0.224		63.0	43.0	324	288	5.95
	126.66	130.66	4.00	59800	239000	8.11	196.10	170.52	25.58	18100	232500	1.03	53.4	56.0	.01785	0.180						
	128.43	131.32	2.89	84300	244000	11.50	196.17	170.58	25.59	18100	233000	1.03	52.8	57.5	.01740	0.135						
43	122.39	128.19	5.80	32100	186000	4.35	196.38	168.48	27.90	13000	183000	0.64	55.9	41.8	.0239	0.300						
	125.00	129.14	4.14	45500	188000	6.17	196.11	168.51	27.60	13000	181000	0.64	54.0	43.2	.0232	0.266		47.8	32.7	227	294	4.50
	125.92	129.02	3.10	60100	187000	8.15	195.93	168.24	27.69	13000	181000	0.64	53.1	43.8	.0228	0.180						
	127.45	129.72	2.27	85000	192000	11.54	196.02	168.35	27.67	13000	181000	0.64	52.0	44.0	.0223	0.136						

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

3/4" FINNED TUBES IN 8" SHELL BUNDLE 2

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_i'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
							Oil on Shell Side															
44	156.86	160.76	3.90	32100	125000	4.35	196.38	180.21	16.17	13000	106000	0.64	28.0	51.1	.01955	0.201		57.6	39.4	256	263	5.25
	158.20	160.91	2.71	45500	123000	6.17	195.69	180.12	15.57	13000	102000	0.65	27.5	51.7	.01934	0.197						
	158.89	160.82	1.93	60000	116000	8.12	195.90	179.70	16.20	13000	107000	0.57	27.1	52.0	.01922	0.157						
	159.22	160.61	1.39	84700	118000	11.50	196.41	179.79	16.62	13000	109400	0.57	27.2	53.0	.01885	0.119						
45	156.82	161.11	4.29	33400	143000	4.53	196.08	180.91	15.17	18100	139000	0.93	28.8	61.9	.01615	0.254		74.1	50.5	360	263	6.72
	157.99	161.10	3.11	45500	142000	6.17	195.51	180.46	15.05	18100	138000	0.93	27.0	64.1	.01560	0.197						
	159.29	161.75	2.46	59400	146000	8.06	196.48	181.20	15.28	18100	140000	0.93	27.4	65.9	.01515	0.158						
46	156.84	162.39	5.55	31900	177000	4.33	195.85	182.34	13.51	25700	176000	1.00	28.8	77.5	.01290	0.201		97.1	66.5	511	258	8.88
	158.64	162.56	3.92	45500	178000	6.17	196.02	182.40	13.62	25700	177000	1.02	28.1	80.0	.01250	0.190						
	159.92	162.92	3.00	60100	180000	8.15	196.20	182.61	13.59	25700	177000	1.02	27.4	82.3	.01214	0.150						
	159.99	162.19	2.20	85200	188000	11.58	195.81	181.88	13.93	25700	181000	1.02	26.9	86.8	.01152	0.116						
47	156.858	163.271	6.413	32100	206000	4.35	196.12	183.51	12.61	32300	206500	2.30	29.4	86.7	.01150	0.259		115.2	78.7	655	254	10.6
	159.105	163.544	4.439	45500	202000	6.17	196.38	183.58	12.80	32100	208500	2.27	28.2	92.1	.01085	0.195						
	160.581	164.012	3.431	60100	206000	8.15	196.38	183.65	12.73	32100	207500	2.27	27.2	96.3	.01040	0.155						
	161.868	164.329	2.461	84700	208000	11.50	196.93	183.90	13.03	31600	209000	2.10	26.8	98.4	.01015	0.118						
48	156.567	164.225	7.658	32500	249000	4.40	196.47	185.14	11.32	44600	257000	4.03	29.6	108.2	.00924	0.250		151	103	932	250	13.8
	159.598	165.098	5.500	45500	250000	6.17	196.14	185.05	11.09	45800	258000	4.13	27.7	115.6	.00865	0.195						
	161.117	165.292	4.175	60800	254000	8.25	196.03	184.94	11.09	45400	256000	4.13	26.9	119.1	.00840	0.154						
	162.667	165.697	3.030	85100	257000	11.57	196.20	185.02	11.18	45200	257000	4.22	25.8	124.0	.00806	0.116						
69	70.277	73.272	2.995	46600	139800	6.31	113.22	103.75	9.47	29500	129300	5.59	37.1	46.0	.02178	0.297		54.2	36.4	92.0	1611	2.69
	72.041	74.266	2.225	61800	137500	8.36	113.07	103.75	9.32	29500	127300	5.64	35.3	47.5	.02102	0.235						
	71.921	73.719	1.798	80200	144000	10.78	113.29	103.77	9.52	30000	132500	5.69	35.6	49.1	.02030	0.192						
	73.137	74.590	1.453	102000	148200	13.58	113.40	103.96	9.44	30600	134000	5.77	34.8	51.4	.01946	0.159						
70	69.768	72.293	2.525	46600	117800	6.31	113.18	102.29	10.89	20100	101000	3.78	36.4	38.0	.0263	0.298		42.6	27.6	62.0	1652	2.04
	70.067	71.998	1.931	61800	119400	8.36	113.58	102.52	11.06	20600	105500	3.86	36.7	38.8	.0258	0.238						
	70.257	71.746	1.489	80700	120000	10.79	113.25	102.13	11.12	20700	107000	3.86	36.7	39.2	.0255	0.194						
	71.506	72.707	1.201	101000	121200	13.55	113.27	102.29	10.98	20700	105000	3.89	35.3	40.6	.0246	0.162						
71	70.905	72.752	1.847	46600	86100	6.31	112.84	100.00	12.84	13100	77800	2.37	34.4	30.9	.0332	0.297	460	33.0	21.6	36.8	1697	15.7
							Glycerine on Shell Side															
74	121.60	127.85	6.25	50500	316000	6.85	195.12	162.66	32.51	15200	316000	0.66	52.8	70.0	.0132	0.210	653	87.5	28.2	226	217	4.48
75	122.49	130.69	8.20	51300	420000	6.96	193.46	166.23	27.23	24100	420000	1.20	52.0	102.5	.00975	0.206	666	125	40.5	357	214	6.36
76	122.32	134.51	12.19	51300	626000	6.96	193.06	169.47	23.59	42100	640000	3.06	51.7	155.0	.00645	0.204	670	213	68.8	641	208	10.7
77	122.79	143.02	20.23	32500	658000	4.40	194.70	174.65	20.05	51000	663000	4.18	50.4	166	.00602	0.288		248	79.8	872	197	12.6
	127.18	143.17	15.99	42800	684000	5.80	195.51	175.28	20.23	52400	687000	4.39	49.0	177	.00565	0.228						
	131.77	143.00	11.23	58400	656000	7.91	195.03	174.78	20.25	52500	689000	4.47	46.6	184	.00543	0.177						
	134.[illegible]	144.10	9.32	72600	677000	9.86	194.95	174.85	20.10	52400	683000	4.48	44.6	193	.00519	0.148						
78	122.61	138.51	15.90	51300	817000	6.96	195.46	175.39	20.07	63300	826000	6.31	53.8	194	.00516	0.202	677	295	95.0	1050	192	15.2
79	122.14	138.95	16.81	51300	865000	6.96	194.97	175.23	18.74	72300	881000	8.04	54.0	204	.00490	0.202	677	320	103	1205	191	16.5
89	193.39	202.71	9.32	50500	470000	6.85	226.92	216.90	10.02	71500	488000	7.29	22.9	269	.00372	0.156	872	485	155	2510	107	27.7

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

3/4" FINNED TUBES IN 8" SHELL BUNDLE 2

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_i'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
							Glycerine on Shell Side															
90	192.92	202.28	9.36	33700	316000	4.56	225.75	214.30	11.45	45000	350000	2.76	21.3	198	.00505	0.217						
	195.71	203.02	7.31	44100	323000	5.99	226.08	214.86	11.22	45200	345000	3.11	20.3	208	.00480	0.173						
	197.96	203.61	5.65	57800	326000	7.84	225.99	214.95	11.04	45700	343000	3.22	18.8	223	.00448	0.139		289	92.5	1565	108	16.5
	200.55	204.66	4.11	78800	324000	10.69	227.05	216.05	11.00	45600	341000	3.24	18.1	231	.00433	0.108						
91	192.15	195.75	3.60	50200	180000	6.87	225.00	207.66	17.34	15400	180000	0.59	21.1	108	.00925	0.158	860	127	40.7	502	113	7.25
92	158.27	161.37	3.10	52000	161000	7.05	194.04	176.67	17.37	14350	162000	0.58	24.5	83.0	.01205	0.177	772	95.0	30.8	268	192	4.60
93	158.81	169.22	9.41	32600	307000	4.46	193.62	183.27	10.35	46700	315500	3.58	23.7	163.6	.00611	0.250						
	162.00	168.98	6.98	42500	296000	5.81	193.78	183.45	10.33	47000	316500	3.52	22.7	171.0	.00584	0.202						
	164.10	169.34	5.24	57500	301000	7.89	193.95	183.45	10.50	46000	315000	3.50	21.4	181.5	.00551	0.157		233	75.3	915	180	11.3
	165.70	169.82	4.12	72900	300000	9.98	193.69	183.34	10.35	45600	308000	3.51	20.2	192.8	.00519	0.130						
94	157.46	166.66	9.20	51100	470000	6.94	194.14	184.64	9.50	78000	486000	9.25	24.7	250.	.00400	0.202	771	413	133	1580	178	20.0
95	122.16	126.36	4.20	46000	193000	6.24	157.95	145.96	11.99	26800	199000	1.99	27.2	92.7	.01081	0.226	602	113	36.9	214	421	4.24
96	122.32	129.76	7.44	46100	343000	6.25	157.95	149.81	8.14	70000	355000	8.71	26.3	171	.00585	0.224	608	254	83.0	623	400	9.34
97	122.29	126.07	3.78	37200	140000	5.04	157.10	143.55	13.55	16800	141000	1.08	25.5	69.5	.0144	0.268	508	82.0	26.7	133	442	3.05
98	85.91	89.28	3.37	38000	128000	5.15	130.59	114.66	15.93	13400	128000	1.53	34.4	47.0	.0213	0.318	430	53.5	17.6	40.8	1105	1.51
99	86.40	91.22	4.97	43200	216000	5.86	131.07	119.07	11.81	34900	247000	3.74	35.7	89.2	.00112	0.285	479	113	37.2	118	1016	3.28
100	85.89	92.12	6.23	48900	304000	6.63	130.78	121.57	9.21	55300	306000	8.61	36.7	105	.00953	0.258	530	136	44.6	195	979	4.00

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

1/2" PLAIN TUBES IN 8" SHELL BUNDLE 3

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_1'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
											Water on Shell Side											
153	158.76	168.84	10.08	33600	338000	2.69	193.32	170.64	22.68	14600	331000	0.34	14.88	380	.00263	0.376		708	74.2	10190	2.09	57.3
	161.02	168.57	7.55	44700	338000	3.58	193.48	170.26	23.22	14600	338000	0.34	13.60	421	.00237	0.297						
	163.18	168.68	5.50	62300	342000	4.99	193.57	170.37	23.20	14600	338000	0.34	12.44	461	.00218	0.228						
	164.61	168.61	4.00	86200	345000	6.91	193.14	170.10	23.04	14600	336000	0.34	11.22	513	.00195	0.175						
154	158.34	167.72	9.38	62400	585000	5.00	193.31	171.21	22.10	26700	590000	1.18	16.35	610	.00164	0.229	1390	1100	116.1	18700	2.08	89.0
155	158.97	177.52	18.56	31600	586000	2.54	193.30	182.59	10.71	53600	585000	5.07	17.59	564	.00177	0.387		1777	186.2	37900	2.00	144.5
	164.88	177.89	13.01	45600	594000	3.66	193.23	182.22	11.01	52300	576000	5.06	14.74	670	.00149	0.285						
	167.45	177.85	10.40	57600	600000	4.64	192.96	181.76	11.20	51700	580000	5.06	13.30	750	.00133	0.234						
	170.10	177.35	7.25	87300	633000	7.01	192.85	180.83	12.02	51300	617000	5.05	11.72	901	.00111	0.168						
156	159.24	174.20	14.96	61100	915000	4.88	193.75	180.45	13.30	68500	911000	6.49	16.41	838	.001193	0.231	1380	2245	238	49700	2.01	185
157	160.32	171.19	10.87	60900	661000	4.87	194.78	175.50	19.28	33900	654000	2.25	17.09	650	.00154	0.232	1370	1270	133.6	24350	2.04	103
157a	159.53	170.20	10.67	60900	650000	4.87	193.58	174.52	19.06	34000	649000	2.26	16.86	650	.00154	0.232	1365	1270	134	24150	2.06	103
158	122.79	138.51	15.57	53900	820000	4.24	157.95	145.98	11.97	68200	816000	8.68	19.73	700	.00143	0.302	1060	2130	234	38400	2.70	162
159	122.81	136.85	14.04	36200	509000	2.88	158.20	141.94	16.36	31900	521000	1.92	18.24	477	.00210	0.411		1210	133	17600	2.74	91.7
	127.00	137.86	10.86	46100	501000	3.66	158.23	142.25	16.98	31900	541000	1.92	15.83	556	.00180	0.324						
	129.63	137.93	8.30	60100	499000	4.78	157.82	141.89	15.93	31900	509000	1.92	14.96	571	.00175	0.269						
	132.75	138.63	5.88	85200	501000	6.76	157.80	142.02	15.78	31900	503000	1.92	12.91	657	.00152	0.202						
160	122.85	131.50	8.51	51100	435000	4.07	157.69	135.21	22.48	19100	429000	0.74	16.42	445	.00225	0.317	1010	805	89.0	10290	2.83	60.6
161	122.88	132.06	9.18	33200	305000	2.64	157.77	134.64	23.13	13300	308000	0.37	15.90	326	.00306	0.446		634	70.1	7290	2.85	47.6
	125.20	132.49	7.29	42700	311000	3.40	158.13	134.82	23.31	13300	310000	0.37	14.38	365	.00274	0.361						
	126.48	132.03	5.55	57700	320000	4.59	157.84	134.19	23.65	13300	314000	0.37	13.24	405	.00247	0.284						
	128.79	132.30	3.51	85100	299000	6.78	158.25	134.26	23.99	13300	319000	0.37	11.78	444	.00226	0.207						
162	85.41	94.82	9.41	33700	318000	2.65	121.96	99.14	22.82	13800	315000	0.44	17.03	303	.00330	0.534		650	75.2	5380	4.05	45.1
	87.30	94.55	7.25	46000	333000	3.62	122.09	98.31	23.78	13800	328000	0.44	16.23	344	.00291	0.413						
	90.43	95.85	5.42	58500	317000	4.60	122.43	98.92	23.51	13800	324000	0.39	14.33	377	.00265	0.339						
	92.23	96.13	3.90	85200	332000	6.70	122.72	98.82	23.90	13800	330000	0.39	13.02	430	.00232	0.248						
163	85.41	95.76	10.35	50900	526000	4.01	121.95	102.22	19.73	27300	538000	1.50	19.49	461	.00217	0.383	831	1055	120	10810	4.01	72.2
164	85.82	101.59	15.77	50800	800000	4.00	122.68	111.49	11.19	72400	810000	9.76	21.98	620	.00162	0.376	848	2360	270	30100	3.79	166
165	85.14	103.48	18.13	31700	575000	2.49	121.93	111.31	10.62	52200	562000	5.29	20.86	440	.00228	0.536		1745	199	21400	3.81	122
	90.05	103.48	13.22	43300	573000	3.41	122.16	111.20	10.96	52000	570000	5.29	18.60	520	.00192	0.419						
	95.31	104.86	9.45	60000	566000	4.72	121.96	111.09	10.87	52000	565000	5.29	15.32	625	.00160	0.320						
	99.28	106.20	6.73	84800	570000	6.68	122.34	111.34	11.00	51900	571000	5.29	13.13	735	.00136	0.238						
166	85.39	96.84	11.45	51900	594000	4.08	122.02	104.49	17.53	34700	609000	2.36	20.59	495	.00202	0.378	844	1220	140	14000	3.95	85.0

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

1/2" PLAIN TUBES IN 8" SHELL BUNDLE 3

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_i'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
										Oil on Shell Side												
197	195.67	198.10	2.43	51800	121000	4.20	228.31	212.00	16.31	14200	121000	0.79	22.15	91.8	.0109	0.235	1365	98.5	52.8	316	167	8.10
198	195.39	200.93	5.54	32100	178000	2.58	227.59	215.28	12.31	29400	192000	2.75	22.37	140	.00715	0.342		177	92.5	660	167	14.05
	198.32	202.33	4.01	45500	183000	3.70	228.22	215.46	12.76	29200	197000	2.72	20.85	154	.00650	0.245						
	199.89	202.95	3.06	59500	182000	4.84	228.24	215.64	12.60	29200	195000	2.72	19.56	161	.00619	0.206						
	201.20	203.41	2.21	79100	175000	6.43	227.25	215.10	12.15	29400	189000	2.72	18.14	170	.00589	0.164						
199	195.76	201.99	6.23	45600	285000	3.71	227.55	216.99	10.56	60000	330000	10.12	22.90	227	.00440	0.256	1245	279	147	1325	167	22.4
200	196.07	202.24	6.17	32100	199000	2.61	228.04	216.99	11.05	36100	209000	3.93	22.81	151	.00660	0.338		183	98.5	815	166	15.1
	198.27	202.79	4.52	45700	207000	3.72	228.43	217.53	10.90	36100	207000	3.92	21.99	159	.00627	0.254						
	198.84	202.31	3.47	59700	208000	4.85	228.22	217.18	11.04	36100	209000	3.93	21.57	163	.00611	0.206						
	199.29	201.90	2.61	80100	209000	6.51	227.84	216.75	11.09	36100	210000	3.93	21.14	167	.00597	0.162						
201	159.75	164.82	5.07	36900	187000	2.96	193.42	182.88	10.54	35500	190000	4.32	25.38	126	.00794	0.350		147	76.6	466	272	8.81
	160.83	165.11	4.30	45600	196000	3.66	193.75	183.18	10.57	35500	190000	4.32	25.06	130	.00770	0.294						
	160.48	163.81	3.33	59700	199000	4.79	193.75	182.70	11.05	35500	199000	4.32	25.75	131	.00763	0.239						
	161.82	164.38	2.56	79600	204000	6.39	193.24	182.53	10.71	35500	193000	4.32	24.54	137	.00730	0.190						
202	158.86	164.13	5.27	50300	265000	4.04	192.81	184.19	8.62	59400	259000	10.79	26.64	167	.00599	0.273	1160	196	102	811	272	11.8
203	158.54	161.31	2.77	41200	114000	3.30	193.46	177.53	15.93	13500	108900	1.03	25.46	74.2	.0135	0.325	983	80.2	41.6	173	271	4.81
204	158.58	162.22	3.64	47200	172000	3.78	194.29	181.62	12.67	26800	172000	2.66	27.16	107	.00935	0.287	948	121	63.1	366	271	7.31
205	122.25	125.53	3.28	47500	156000	3.76	158.07	146.64	11.43	26700	149000	3.88	28.19	91.6	.0109	0.339	940	102	52.8	170	550	5.41
206	122.50	125.27	2.77	39700	110000	3.14	158.16	142.90	15.26	14100	104500	1.77	25.84	70.0	.0143	0.391	810	76.7	39.7	87.1	561	4.05
207	122.92	126.50	3.58	51200	183000	4.06	158.38	148.62	9.76	35800	170500	8.52	28.47	105	.00953	0.318	996	118	60.7	238	550	6.21
208	122.38	128.68	6.30	33200	209000	2.63	157.93	150.33	7.60	53300	198000	11.10	28.26	122	.00820	0.449		149	77.2	362	526	8.10
	124.34	129.09	4.75	44100	210000	3.50	158.18	150.57	7.61	53000	197000	10.88	27.34	126	.00794	0.353						
	124.90	128.61	3.71	58500	217000	4.64	157.98	150.22	7.76	53300	202000	11.10	27.94	131	.00764	0.283						
	126.30	129.09	2.79	79800	223000	6.33	158.16	150.46	7.70	53400	201000	11.14	26.36	136	.00735	0.218						
209	85.73	88.89	3.16	51500	163000	4.05	122.32	113.36	8.96	38500	162000	11.73	30.27	90.9	.0110	0.386	823	102	52.4	105	1220	4.19
210	85.17	88.29	3.12	42100	132000	3.31	122.27	111.34	10.93	26600	136000	7.54	29.82	76.0	.0132	0.451	699	85.2	43.9	69.0	1330	3.35
211	85.23	87.77	2.54	39700	101000	3.12	122.54	108.37	14.17	15200	101000	0.61	28.73	59.4	.0168	0.477	639	65.7	33.8	37.8	1400	2.54

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

1/2" PLAIN TUBES IN 8" SHELL BUNDLE 3

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_1'	Shell Side: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
							Glycerine on Shell Side															
139	194.95	199.26	4.30	37200	160000	3.02	227.34	211.05	16.31	13600	151200	0.64	20.9	126.0	.00794	0.303	1050	143.5	35.2	280	110	6.28
140	159.39	202.32	6.93	34700	240500	2.82	227.55	215.49	12.06	31900	263000	2.08	22.1	193.0	.00519	0.318	1000	240	58.6	675	107.5	10.55
141	196.21	203.76	7.54	52000	392000	4.23	227.37	217.74	9.63	63800	419000	6.81	22.0	311	.00322	0.229	1390	404	99.0	1370	106.5	17.80
142	195.62	202.03	6.41	52000	334000	4.23	227.73	216.23	11.50	45800	361000	3.66	22.5	261	.00383	0.229	1385	324	79.2	975	107.0	14.25
143	159.40	164.82	5.40	52000	281000	4.17	193.55	182.01	11.54	38300	280000	3.54	25.2	188.0	.00532	0.267	1195	224	55.2	445	182.5	8.37
144	159.31	166.91	7.60	52000	395000	4.17	193.59	184.68	8.91	70500	409000	9.68	25.6	266	.00377	0.267	1195	343	84.5	865	178.5	12.85
145	160.00	164.17	4.17	31400	131000	2.52	194.31	177.24	17.07	12000	133000	0.71	22.3	100.0	.00100	0.398	798	114.5	28.2	136	190.0	4.21
146	159.22	165.06	5.71	31400	179000	2.52	193.50	180.27	13.23	20700	178000	1.54	23.8	132.5	.00755	0.398	798	159.5	39.4	241	187.0	5.90
147	122.31	126.79	4.22	38600	166000	3.07	158.09	143.71	14.38	18600	167000	2.16	16.4	98.5	.01015	0.394	795	112.5	28.0	77.0	435	3.19
148	122.31	127.24	4.75	47000	223000	3.72	157.71	146.05	11.66	31600	228000	4.19	26.8	143.7	.00696	0.342	930	170.5	42.2	157	423	4.88
149	122.58	128.43	5.69	58800	334000	4.66	157.91	149.32	8.59	62100	332000	11.01	27.8	210	.00476	0.282	1115	260	64.6	327	403	7.51
150	122.45	127.31	4.68	58800	275000	4.66	157.91	147.51	10.40	43000	273000	6.41	27.6	169.0	.00592	0.286	1115	200	50.0	215	413	5.80
151	85.37	90.01	4.64	43700	203000	3.34	122.07	112.41	9.66	36200	223000	10.76	29.3	123.0	.00813	0.450	705	149.5	34.3	64.5	1365	2.61

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

1/2" FINNED TUBES IN 8" SHELL BUNDLE 4

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_i'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
							Water on Shell Side															
167	158.67	177.80	19.13	51900	992000	8.06	192.25	178.25	14.00	70600	989000	4.41	13.77	440	.00227	0.154	680	1400	122	34000	2.04	94.0
168	159.12	176.29	17.17	35900	617000	5.58	193.66	174.94	18.72	32900	616000	1.23	12.59	298	.00336	0.205						
	161.56	175.37	13.81	45200	625000	7.03	193.42	174.00	19.42	32100	624000	1.22	11.35	336	.00298	0.171						
	165.13	175.51	10.38	57900	601000	9.00	194.02	174.36	19.66	31000	610000	1.01	10.00	370	.00270	0.139		780	67.7	14800	2.05	51.9
	168.82	175.62	6.80	85300	580000	13.29	194.00	174.54	19.46	30600	595000	0.98	8.06	445	.00225	0.102						
169	158.86	175.39	16.53	32800	542000	5.10	193.62	173.77	19.85	27700	550000	0.86	12.35	270	.00370	0.223						
	162.75	175.33	12.58	42300	531000	6.59	193.49	173.88	19.61	27700	544000	0.86	10.71	306	.00327	0.181						
	165.51	174.56	9.05	60700	549000	9.45	193.28	173.52	19.76	27700	548000	0.86	9.58	250	.00286	0.136		720	62.5	13300	2.06	47.8
	168.21	174.85	6.64	84200	559000	13.10	193.55	173.91	19.64	27800	546000	0.86	8.24	408	.00245	0.103						
170	159.55	167.58	8.03	50300	403000	7.79	192.96	165.92	27.04	15100	408000	0.27	9.95	251	.00398	0.160	648	435	38.2	7000	2.14	29.1
171	122.86	131.47	8.61	50900	439000	7.81	157.98	130.42	27.56	15600	430000	0.27	11.58	230	.00435	0.187	560	402	36.8	5550	2.90	24.8
172	122.88	139.37	16.49	31500	520000	4.84	158.23	138.96	19.27	26900	519000	0.86	13.78	231	.00434	0.270						
	128.46	139.86	11.40	44400	506000	6.81	158.11	139.42	11.69	26900	503000	0.86	11.27	275	.00365	0.204						
	131.23	139.84	8.61	59300	511000	9.10	158.27	139.59	18.68	26900	503000	0.86	10.13	306	.00327	0.159		615	55.8	10000	2.80	38.2
	133.23	139.64	6.41	82300	529000	12.64	158.23	139.41	18.82	26900	506000	0.86	8.99	352	.00284	0.122						
173	122.77	137.35	14.58	51200	745000	7.85	158.09	137.70	20.39	35700	728000	1.47	14.36	314	.00138	0.184	570	755	68.5	13150	2.80	46.9
174	122.65	141.22	18.43	55000	1010000	8.46	157.39	143.20	14.19	71800	1010000	5.26	15.57	397	.00252	0.172	608	1270	115	26800	2.74	79.3
175	122.72	143.94	21.22	33300	700000	5.11	158.20	145.06	13.14	53100	699000	3.00	15.16	282	.00354	0.256						
	127.54	143.74	16.20	44900	727000	6.89	158.41	144.75	13.66	53100	726000	3.00	13.23	336	.00298	0.199						
	132.04	143.94	11.90	59600	710000	9.15	157.98	144.70	13.28	53100	702000	3.05	11.07	390	.00256	0.157		1030	93.5	20000	2.70	65.0
	134.98	143.94	8.96	80700	720000	12.40	157.93	144.64	13.29	53100	707000	3.05	9.73	444	.00223	0.121						
176	85.21	103.80	18.59	60300	1120000	9.19	122.14	107.35	14.79	77000	1139000	6.18	17.88	386	.00259	0.193	541	1440	136	21100	3.87	82.5
177	85.24	100.89	15.65	43100	675000	6.56	122.18	101.95	20.23	34600	701000	1.33	15.66	269	.00372	0.254	409	828	78.0	9300	4.01	48.0
178	85.51	106.52	21.01	32200	678000	4.90	121.91	109.09	12.82	54000	694000	3.14	16.54	254	.00394	0.318						
	91.06	106.70	15.64	45500	712000	6.94	121.93	108.68	13.25	54000	716000	3.14	14.01	311	.00321	0.235						
	94.82	106.97	12.15	59900	729000	9.14	121.96	108.43	13.53	54000	731000	3.14	12.11	368	.00272	0.188		1110	105	14800	3.86	63.6
	98.47	107.64	9.17	79900	731000	12.20	122.09	109.79	13.30	54000	719000	3.14	10.37	427	.00234	0.147						
179	85.68	95.86	10.18	40100	409000	6.11	122.34	95.11	27.23	15200	414000	0.27	12.91	196	.00511	0.272	394	410	39.1	3900	4.19	23.0
180	85.53	102.04	16.51	31700	524000	4.83	121.98	102.63	19.35	27600	534000	0.84	15.28	212	.00471	0.326						
	89.67	101.73	12.06	45200	545000	6.90	122.16	102.16	20.00	27600	552000	0.84	13.18	255	.00393	0.236						
	93.90	102.76	8.86	60600	537000	9.25	122.40	102.92	19.48	27600	538000	0.84	11.08	297	.00337	0.187		661	62.9	7400	4.01	37.6
	95.77	102.54	6.77	79900	541000	12.20	122.34	102.74	19.60	27600	541000	0.84	10.05	330	.00303	0.150						

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

1/2" FINNED TUBES IN 8" SHELL BUNDLE 4

Run No.	Water on Tube Side: Temperature, °F in	Water on Tube Side: Temperature, °F out	Water on Tube Side: Temperature, °F rise	Water on Tube Side: Pounds per hr.	Water on Tube Side: Heat trans. BTU per hr.	Water on Tube Side: Water Velocity Ft./sec.	Shell Side: Temperature, °F in	Shell Side: Temperature, °F out	Shell Side: Temperature, °F drop	Shell Side: Pounds per hr.	Shell Side: Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_i'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
											Oil on Shell Side											
181	195.50	203.50	8.20	49700	408000	7.81	226.94	212.94	14.00	56000	411000	5.67	19.51	128	.00781	0.140	744	156	67.5	851	176	10.0
182	194.79	203.92	9.00	31700	285000	4.99	227.88	211.86	16.02	35200	295000	2.50	19.24	92.3	.01084	0.201		117	50.7	536	176	7.55
	197.01	203.73	6.72	42800	288000	6.73	227.79	211.86	15.93	35200	294000	2.50	18.07	98.5	.01015	0.157						
	198.88	203.81	4.93	59000	290000	9.28	227.80	211.95	15.85	35200	292000	2.50	17.07	104.4	.00958	0.122						
	199.75	203.74	3.99	76100	303000	12.00	227.79	211.82	15.97	35200	294000	2.50	16.75	109.0	.00918	0.0995						
183	195.51	201.00	5.49	45700	251000	7.19	227.62	209.25	18.37	31300	297000	1.42	18.40	97.1	.01097	0.150	694	105	45.7	461	178	6.80
184	195.65	199.62	3.97	39100	155000	6.15	227.59	206.87	20.72	13800	150000	0.57	17.52	53.3	.01876	0.171	610	58.6	25.4	197	180	3.77
185	159.31	164.32	5.11	45700	234000	7.10	193.89	176.05	17.84	25300	228000	1.72	21.95	64.5	.01550	0.175	599	72.5	31.1	218	283	3.97
186	159.11	162.90	3.79	38600	146000	5.99	193.57	173.30	20.27	13900	142000	0.71	20.76	44.5	.02245	0.201	521	48.8	21.0	114	291	2.64
187	122.83	126.46	3.63	39600	144000	6.08	158.32	139.60	18.72	14800	134000	1.18	22.98	37.1	.02695	0.221	455	40.5	17.2	60.4	575	1.71
188	122.99	127.67	4.68	46000	215000	7.06	158.23	142.50	15.73	26000	199000	2.56	24.16	52.4	.01909	0.205	510	58.5	24.8	108	565	2.49
189	159.00	166.86	7.86	50500	397000	7.84	193.64	179.65	13.99	55800	394000	6.40	22.93	105.2	.00949	0.160	655	127	54.6	480	281	7.00
190	159.03	164.88	5.85	51300	300000	7.96	193.78	177.24	16.54	35800	299000	2.85	22.39	82.1	.01219	0.159	656	94.1	40.4	318	284	5.15
191	123.25	130.12	6.87	51600	355000	7.93	157.64	145.74	11.90	56500	327000	8.38	24.50	85.3	.01171	0.185	566	101	43.0	242	560	4.32
192	121.77	127.13	5.36	50700	272000	7.79	158.50	143.44	15.06	36000	265000	3.93	23.32	64.2	.01558	0.190	550	72.8	31.0	151	560	3.14
193	85.28	90.19	4.91	50700	249000	7.76	121.55	109.17	12.38	43300	250000	8.92	27.23	56.0	.01785	0.228	456	64.1	27.2	71.9	1340	2.06
195	85.69	89.58	3.89	32100	125000	4.91	122.41	105.39	17.02	14800	117500	2.78	25.17	29.4	.03400	0.331	318	32.4	13.7	24.6	1397	1.01
196	85.01	89.62	4.61	32400	149000	4.96	121.28	106.57	14.71	22100	152000	4.46	25.68	35.9	.0279	0.328		42.8	18.1	35.6	1400	1.34
	86.29	89.76	3.47	45800	159000	7.01	121.48	106.93	14.55	22500	153000	4.46	25.46	37.5	.0266	0.247						
	86.79	89.60	2.81	60000	169000	9.20	121.77	106.99	14.78	22500	155000	4.46	25.50	38.9	.0257	0.199						
	87.24	89.62	2.38	73300	174000	11.23	121.86	106.88	14.98	22700	158500	4.46	25.18	39.2	.0255	0.171						

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

1/2" FINNED TUBES IN 8" SHELL BUNDLE 4

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_i'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
	Glycerine on Shell Side																					
125	190.13	202.95	12.82	55500	712000	8.71	225.64	211.39	14.25	75000	725000	5.50	20.5	213.5	.00469	0.111	805	297	59.8	1015	113	10.5
126	189.54	198.37	8.83	55500	490000	8.70	224.71	206.69	18.02	42800	527000	1.94	20.4	151.4	.00660	0.112	770	192	38.7	567	116.5	6.78
127	191.26	196.43	5.04	46200	233000	7.25	225.18	203.50	21.68	16300		0.44	18.4	77.8	.01290	0.152	665	88.7	17.8	188	116	3.12
128	158.79	163.11	4.32	45600	197000	7.08	195.04	173.12	21.92	13800	196000	0.49	21.2	56.6	.01765	0.175	595	62.8	12.8	102	196	1.88
129	158.65	163.56	4.77	71800	342000	11.13	194.58	175.28	19.30	27200	340000	1.33	22.4	94.6	.01057	0.121	858	107	21.7	206	194	3.20
130	158.99 162.95 166.24 168.37	175.01 174.65 174.92 175.15	16.02 11.70 8.55 6.62	32100 45500 62300 82000	515000 532000 534000 543000	4.98 7.07 9.68 12.74	194.79 194.72 194.85 195.01	183.85 183.67 183.83 183.97	10.94 11.05 11.02 11.04	73800 74100 74800 75500	525000 533000 538000 543000	5.89 6.00 6.10 6.19	21.0 19.3 17.8 16.9	152 169 184 196	.00659 .00592 .00544 .00510	0.227 0.170 0.132 0.105		237	48.1	549	177	7.20
131	158.90	165.94	6.90	71900	496000	11.15	194.67	177.84	16.83	46000	[illegible]2000	2.80	22.9	134	.00746	0.390	864	160	32.5	336	188	4.88
132	122.47	130.73	7.81	71800	560000	11.02	158.18	146.32	11.86	74200	547000	7.91	24.9	140	.00716	0.450	733	174	35.6	244	419	4.05
133	121.91	127.72	5.34	71400	381000	10.97	158.11	143.29	14.82	42700	393000	3.43	25.3	97.8	.01022	0.452	720	114	23.4	137	447	2.59
134	122.14	127.83	5.22	34500	180000	5.29	157.86	139.73	18.13	16200	182000	1.08	22.3	51.6	.01938	0.284	405	59.4	12.2	49.5	460	1.34
135	121.86	128.93	6.59	45900	302000	7.05	158.54	143.17	15.37	30600	292000	2.43	24.6	76.6	.01305	0.341	510	91.0	18.7	100	435	2.12
136	85.48	91.08	5.60	60900	341000	9.26	121.98	109.99	11.99	48800	348000	9.21	27.2	77.5	.01290	0.490	526	91.5	19.0	48.3	1437	1.42
137	85.03	90.19	5.16	46400	239000	7.06	121.96	108.48	13.48	29800	238000	5.32	27.1	53.9	.01860	0.414	424	61.8	12.8	27.8	1485	0.945
138	85.82	90.54	4.72	35300	167000	5.36	122.56	107.24	15.32	18300	167000	3.10	26.2	39.1	.0256	0.346	340	44.3	8.97	17.0	1505	0.660

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

5/8" PLAIN TUBES IN 6" SHELL BUNDLE 5

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_i'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
							Water on Shell Side															
8	122.94	136.29	13.35	28600	382000	4.46	176.88	144.66	32.22	11300	365000	0.91	28.1	526	.00190	0.299						
	124.47	135.46	10.99	41000	451000	6.40	177.46	143.55	33.91	11300	383000	0.91	26.9	612	.00163	0.217						
	127.09	135.23	8.14	51500	419000	8.04	177.35	142.92	34.43	11300	394000	0.91	25.1	640	.00156	0.179		1070	143	13050	2.50	102
	130.89	136.59	5.70	75500	430000	11.80	177.08	143.17	33.91	11400	386000	0.91	22.2	715	.00140	0.130						
9	122.45	138.74	16.29	27700	451000	4.32	177.15	149.31	27.84	17000	470000	1.55	30.0	608	.00164	0.296						
	128.79	139.86	11.07	40400	446000	6.30	176.70	149.13	27.57	17200	474000	1.55	25.8	707	.00141	0.215						
	131.61	140.36	8.75	52400	459000	8.18	177.06	148.80	28.26	17200	486000	1.55	23.7	789	.00127	0.173		1465	195	20200	2.45	142
	134.60	140.81	6.21	75300	467000	11.75	176.59	148.39	28.20	17250	485000	1.55	21.5	875	.00114	0.129						
10	121.60	139.86	18.26	28900	527000	4.51	176.97	152.78	24.19	23800	576000	2.79	32.3	675	.00148	0.285						
	126.81	140.22	13.41	41100	551000	6.41	176.65	151.84	24.81	23900	592000	2.80	28.6	792	.00126	0.213						
	132.48	142.48	10.00	53500	535000	8.35	176.76	152.53	24.23	23900	580000	2.78	24.9	885	.00113	0.169		1800	239	28400	2.41	175
	136.26	143.42	7.16	75500	540000	11.80	177.13	152.44	24.69	23900	590000	2.80	22.4	996	.001003	0.127						
11	122.67	142.79	20.12	28900	581000	4.51	178.88	157.19	19.69	31500	620000	5.13	32.3	735	.00136	0.283						
	128.77	143.69	14.92	40600	605000	6.34	177.15	156.83	20.32	31300	635000	5.13	28.9	846	.00118	0.212						
	136.38	147.07	10.69	53500	561000	8.35	176.77	157.80	18.87	31300	591000	5.15	24.0	950	.00105	0.166		2160	286	35000	2.36	211
	141.28	148.91	7.63	75300	579000	11.80	177.10	158.05	19.05	31200	595000	5.19	20.9	1107	.000905	0.124						
12	149.13	159.73	10.60	28700	304000	4.48	177.04	166.75	10.29	31100	320000	5.19	16.4	750	.00133	0.259						
	153.32	160.84	7.52	40400	305000	6.31	177.17	166.87	10.30	31100	320000	5.18	14.1	880	.00113	0.195						
	155.52	161.22	5.70	53100	303000	8.30	177.06	166.48	10.58	31400	332000	5.25	12.7	991	.00101	0.156		2110	278	39300	2.29	205
	158.74	162.64	3.90	75300	294000	11.80	176.90	167.07	9.83	31500	310000	5.28	10.4	1150	.000870	0.117						
13	149.79	159.30	9.51	28700	272000	4.48	177.24	165.11	12.13	23800	289000	3.05	15.5	716	.00139	0.259						
	153.55	160.02	6.97	40600	283000	6.34	177.17	165.29	11.88	23800	283000	3.05	13.2	844	.00119	0.194						
	155.70	160.75	5.05	54700	276000	8.55	176.92	165.34	11.58	23700	275000	3.04	11.7	927	.00108	0.152		1840	243	29600	2.29	179
	158.09	161.69	3.60	75600	272000	11.80	177.62	165.81	11.81	23700	280000	3.04	10.6	1030	.000970	0.117						
14	149.11	157.10	7.99	28600	229000	4.47	177.21	162.48	14.73	16550	243000	1.52	15.4	610	.00164	0.261						
	152.53	158.04	5.51	41000	224000	6.40	177.22	162.55	14.67	16500	242000	1.52	13.1	704	.00142	0.194						
	153.82	158.07	4.25	53400	228000	8.50	177.13	162.27	14.86	16550	245000	1.52	12.1	770	.00130	0.154		1305	172	20400	2.31	126
	155.80	158.83	3.03	75500	229000	11.80	176.92	162.34	14.58	16500	241000	1.52	10.5	883	.00113	0.118						
15	148.48	155.19	6.71	29500	198000	4.61	176.99	159.60	17.39	11950	208000	0.81	14.6	547	.00183	0.256						
	152.22	156.72	4.50	40800	184000	6.38	177.08	160.47	16.61	11950	199000	0.81	12.3	612	.00163	0.195						
	152.76	156.40	3.64	51700	188000	8.07	176.94	159.88	17.06	11950	204000	0.81	11.7	664	.00151	0.161		1030	136	14650	2.33	99.3
	154.22	156.76	2.54	75100	191000	11.70	176.47	159.78	16.69	11950	199000	0.81	10.5	734	.00136	0.120						
16	122.79	145.24	23.45	28600	671000	4.46	177.28	160.90	16.38	39600	649000	8.01	33.3	781	.00128	0.282						
	130.33	146.50	16.17	40400	654000	6.33	177.13	160.27	16.86	39600	668000	8.01	29.2	923	.00108	0.210						
	137.23	149.05	11.82	53400	633000	8.33	176.70	160.54	16.16	39600	640000	8.02	24.2	1040	.000960	0.165		2680	355	48600	2.33	263
	141.89	150.60	8.71	75600	659000	11.80	176.03	160.47	15.56	39600	616000	8.02	20.6	1222	.000818	0.124						
17	148.46	161.26	12.80	27800	356000	4.36	176.94	168.48	8.46	39900	338000	8.46	16.7	820	.00122	0.265						
	154.27	162.23	7.96	40500	322000	6.32	177.28	168.64	8.64	39700	343000	8.43	13.7	951	.00105	0.193						
	157.26	163.24	5.98	53400	320000	8.33	177.53	168.98	8.55	39800	340000	8.43	12.3	1061	.000942	0.154		2520	332	50000	2.25	245
	159.37	163.65	4.28	75700	332000	11.80	177.21	168.64	8.57	39600	339000	8.40	10.9	1220	.000820	0.116						

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

5/8" PLAIN TUBES IN 6" SHELL BUNDLE 5

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_1'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
											Oil on Shell Side											
57	122.295	128.284	5.989	32300	193500	4.97	195.81	180.68	15.13	26000	199000	5.75	63.0	123.8	.00808	0.271						
	124.522	128.808	4.286	45500	195000	7.00	195.81	180.68	15.13	26100	201000	5.75	61.6	127.0	.00788	0.205						
	126.345	129.524	3.179	62200	197000	9.58	195.96	180.86	15.10	26100	200000	5.75	60.4	130.0	.00770	0.158		143	91.8	668	263	13.4
	128.208	130.660	2.452	81200	198000	12.50	196.03	181.02	15.01	26100	199000	5.75	59.0	133.1	.00752	0.127						
58	122.245	127.366	5.121	31700	162000	4.88	196.14	178.61	17.53	18400	163500	3.24	62.5	103.0	.00971	0.275						
	124.815	128.320	3.505	45300	159000	6.98	195.75	178.36	17.39	18400	162000	3.24	60.4	104.8	.00953	0.206						
	126.682	129.256	2.574	62400	161000	9.60	196.03	178.72	17.30	18400	161000	3.24	59.3	107.3	.00930	0.158		115.5	74.3	461	268	10.9
	127.486	129.476	1.990	81200	161000	12.50	195.66	178.38	17.28	18400	161000	3.24	58.4	109.0	.00919	0.128						
59	122.608	126.729	4.121	32000	132000	4.93	195.84	175.62	20.22	13300	136000	1.86	60.8	87.2	.01148	0.273						
	124.617	127.568	2.951	45300	134000	6.90	196.48	176.31	20.17	13300	135500	1.89	60.1	88.5	.01130	0.208						
	126.343	128.466	2.123	62000	131500	9.55	195.93	175.93	20.00	13300	134500	1.89	58.4	90.2	.01109	0.159		96.0	61.6	325	277	9.05
	127.256	128.907	1.651	81200	134000	12.50	196.02	176.06	19.96	13300	134000	1.90	57.8	91.7	.01091	0.128						
60	122.479	129.200	6.721	32000	215000	4.93	195.60	182.00	13.60	32500	224000	8.25	62.9	138.1	.00725	0.272						
	125.065	129.836	4.771	45300	216000	6.90	195.51	181.82	13.69	32500	225500	8.30	61.2	142.5	.00701	0.207						
	127.243	130.736	3.493	62100	217000	9.56	195.30	181.76	13.54	32500	223000	8.30	59.5	146.5	.00683	0.158		162	104	856	259	15.3
	127.535	130.271	2.736	81400	222000	12.55	195.60	181.87	13.73	32500	226000	8.28	59.8	148.2	.00675	0.127						
61	122.106	129.861	7.755	32000	248000	4.93	195.67	183.52	12.15	42000	259000	12.90	63.5	157.5	.00635	0.272						
	124.900	130.412	5.512	45300	250000	6.90	195.51	183.38	12.13	42000	259000	13.01	61.7	163.0	.00613	0.206						
	126.853	130.941	4.088	61200	254000	9.56	195.42	183.27	12.15	42000	259000	13.02	60.5	167.8	.00596	0.159		188	121	1122	256	17.8
	128.620	131.704	3.084	81400	251000	12.55	195.25	183.15	12.10	42000	258000	13.00	59.0	171.3	.00585	0.127						
67	71.566	72.932	1.366	46600	63700	7.14	112.93	104.72	8.21	14600	55500	6.41	36.3	65.0	.0154	0.268						
	73.162	74.210	1.048	61300	64200	9.39	113.31	104.63	8.67	14200	57000	5.97	35.6	67.3	.0149	0.214						
	74.437	75.180	0.743	84200	62500	12.90	113.04	104.68	8.36	14600	56600	6.12	34.2	68.5	.0146	0.165		71.5	45.2	55.9	1603	3.37
	75.614	76.339	0.725	88300	63600	13.51	113.83	105.53	8.30	14300	54900	6.10	33.6	69.9	.0143	0.158						
68	70.484	71.508	1.024	46600	47800	7.14	113.31	101.05	12.26	8500		2.87	35.8	52.6	.01900	0.270						
	72.126	72.909	0.783	61800	48400	9.45	113.45	101.19	12.26	8600		2.96	34.4	55.6	.01798	0.214						
	72.963	73.627	0.664	73900	49000	11.30	113.20	100.90	12.30	8600		2.99	33.4	58.0	.01724	0.185		59.0	37.3	30.1	1673	2.75
	72.813	73.387	0.574	87500	50200	13.40	113.04	100.72	12.32	8800		2.92	33.5	59.0	.01695	0.161						
											Glycerine on Shell Side											
107	194.31	200.17	5.86	45500	266000	7.21	227.34	219.47	7.87	52000	278000	14.86	25.9	416	.00240	0.151	1990	548	159	2180	105	31.8
108	194.85	202.69	7.83	28000	219000	4.44	227.61	219.65	7.96	46000	250000	11.27	24.4	379	.00264	0.221						
	195.49	201.83	6.34	35800	227000	5.68	227.14	219.42	7.72	46000	242000	11.35	24.2	383	.00261	0.182						
	196.56	201.18	4.62	51000	236000	8.10	227.05	218.98	8.07	46000	254000	11.37	24.1	399	.00250	0.136		548	159	1940	105	30.1
	198.09	201.60	3.51	72000	253000	11.41	227.71	219.49	8.22	45800	262000	11.40	23.5	432	.00231	0.104						
109	194.23	197.38	3.15	38800	122000	6.15	277.55	214.39	13.16	14800	133000	1.42	25.2	200	.00500	0.173	1740	230	69.2	601	108	12.55
110	158.61	161.85	3.24	34100	110500	5.33	193.48	180.84	12.64	14900	122500	1.79	26.4	167	.00599	0.219	1355	193	57.5	338	185	8.71
111	158.58	163.74	5.17	38300	198000	5.99	193.44	185.25	8.19	40000	213000	9.40	27.9	292	.00342	0.200	950	374	113.5	954	178	17.5
112	158.65	163.74	5.09	45900	234000	7.17	193.50	186.08	7.42	51700	249000	14.90	28.3	337	.00297	0.173	1730	431	130.5	1257	176	20.1
121	77.20	80.53	3.15	33100	104000	5.10	122.67	111.99	10.68	16400	105000	7.52	38.4	110.8	.00896	0.336	887	129	39.9	47.2	1365	3.23
122	122.52	125.94	3.38	30000	101000	4.64	158.27	147.56	10.71	15200	102000	2.64	28.5	142.2	.00704	0.288	1040	166	50.8	149	411	5.93
123	122.59	127.18	4.52	30000	136000	4.64	157.69	149.66	8.03	26200	131000	6.01	28.7	185.2	.00540	0.288	1045	229	70.1	257	402	8.20
124	123.51	128.68	5.11	35500	181000	5.50	158.67	152.31	6.36	45700	181000	14.30	29.2	246.5	.00405	0.248	1200	318	97.2	475	383	11.5

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

5/8" FINNED TUBES IN 6" SHELL BUNDLE 6

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_i'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
										Water on Shell Side												
24	148.71	156.30	7.59	31700	241000	7.85	177.38	157.06	20.32	12200	248000	0.61	11.60	387	.00258	0.166						
	150.82	156.03	5.21	45500	237000	11.25	177.40	156.88	20.52	12100	248000	0.61	10.33	431	.00232	0.124		996	114	12000	2.37	83.0
	152.02	156.15	4.13	58000	240000	14.34	177.20	156.88	20.32	12100	246000	0.61	9.43	473	.00211	0.102						
	152.96	156.27	3.31	72500	240000	17.92	177.26	156.92	20.34	12100	246000	0.61	8.78	508	.00197	0.0858						
25	149.92	158.59	8.67	31200	270000	7.70	177.39	160.52	16.87	16800	283000	1.11	12.37	410	.00244	0.168						
	153.52	159.37	5.85	45100	264000	11.13	177.71	160.95	16.76	16800	282000	1.11	10.50	477	.00210	0.124		1230	145	17100	2.33	106
	154.92	159.46	4.54	57400	260000	14.17	177.42	160.85	16.57	16800	278000	1.11	9.48	520	.00193	0.102						
	155.83	159.49	3.66	71600	262000	17.70	177.31	160.73	16.58	16800	278000	1.11	8.78	564	.00177	0.0856						
26	149.18	159.58	10.40	31400	326000	7.75	177.01	162.66	14.35	23800	341000	2.30	13.58	451	.00222	0.167						
	152.56	159.96	7.40	44700	331000	11.03	177.30	162.66	14.64	23800	348000	2.31	11.89	524	.00191	0.125		1620	188	24200	2.31	138
	154.63	160.25	5.62	58000	326000	14.31	177.15	162.61	14.54	23900	348000	2.31	10.59	584	.00171	0.101						
	155.92	160.43	4.51	71900	324000	17.73	177.51	162.59	14.92	23800	355000	2.31	9.91	636	.00157	0.0855						
27	148.93	160.50	11.57	32300	374000	7.98	176.95	164.35	12.60	31400	396000	3.92	14.27	495	.00202	0.165						
	153.32	161.42	8.10	45600	370000	11.28	177.75	164.88	12.87	31500	405000	3.94	12.43	572	.00175	0.122		2040	240	32300	2.30	176
	154.83	161.35	6.52	58000	378000	14.31	177.03	164.37	12.66	31400	398000	3.94	11.13	640	.00156	0.101						
	156.49	161.80	5.31	72500	385000	17.90	177.04	164.52	12.52	31500	395000	3.92	10.16	704	.00142	0.0841						
28	149.01	162.76	13.75	32300	445000	7.98	177.44	168.23	9.21	49600	457000	9.70	15.50	534	.00187	0.163						
	154.04	163.67	9.63	45400	437000	11.20	177.60	168.32	9.28	49600	460000	9.66	12.98	633	.00159	0.123		2900	340	51800	2.25	253
	156.78	164.30	7.52	57600	434000	14.23	177.62	168.50	9.12	49600	452500	9.70	11.53	704	.00142	0.0998						
	158.46	164.57	6.11	71700	438000	17.70	177.29	168.23	9.06	49600	450000	9.70	10.30	792	.00126	0.0845						
										Oil on Shell Side												
31	157.86	160.53	2.67	31500	84100	7.77	180.30	170.56	9.74	18500	90000	2.68	16.85	91.4	.01093	0.164	774	106	60.5	310	334	7.15
32	158.33	163.08	4.75	31700	150500	7.82	196.11	179.82	16.29	18300	151000	2.60	26.5	103.2	.00969	0.162						
	160.23	163.54	3.31	45500	150600	11.22	195.54	179.62	15.92	18400	148000	2.49	24.8	110.4	.00906	0.0120		132	75.7	384	269	9.84
	161.11	163.81	2.70	57900	156000	14.30	195.51	179.73	15.78	18300	146000	2.55	24.3	112.3	.00890	0.0990						
	161.67	163.92	2.25	71800	169000	17.72	195.58	179.64	15.94	18300	148000	2.55	23.6	115.2	.00867	0.0834						
33	158.16	163.89	5.73	31900	183000	7.87	195.33	181.13	14.20	26300	189000	4.75	26.4	129.0	.00775	0.161						
	159.85	163.98	4.13	45600	188000	11.26	195.45	181.09	14.36	26500	192000	4.69	25.8	135.5	.00738	0.120		170	97.5	565	263	12.8
	161.59	164.88	3.29	57400	189000	14.19	195.53	181.40	14.13	26200	188000	4.66	24.6	140.2	.00713	0.0997						
	162.03	164.66	2.63	72300	189000	17.85	195.24	181.22	14.02	26100	185500	4.66	24.0	142.0	.00705	0.0831						
34	122.14	128.25	6.11	32100	196000	7.94	196.14	165.42	30.72	13300	205000	1.57	53.8	68.2	.01466	0.187						
	124.16	128.57	4.41	45800	202000	11.31	196.21	165.31	30.90	13300	206500	1.57	52.7	71.0	.01408	0.140		81.5	46.5	243	300	6.30
	126.06	129.56	3.50	58000	203000	14.31	196.02	165.42	30.60	13300	206000	1.57	51.4	73.0	.01370	0.115						
	127.32	130.12	2.80	71900	201000	17.75	195.99	165.49	30.50	13300	204000	1.57	50.4	73.9	.01350	0.0964						
35	122.32	130.50	8.18	32100	262000	7.94	195.91	168.03	27.87	18500	260000	2.90	54.4	88.0	.01136	0.186						
	124.70	130.61	5.91	45500	269000	11.22	196.11	167.92	28.19	18500	262000	2.90	52.9	92.0	.01087	0.141		110	62.7	347	294	8.45
	127.11	131.85	4.74	57900	274000	14.30	195.75	168.10	27.65	18500	257500	2.99	51.2	95.2	.01050	0.114						
	128.50	132.24	3.74	72100	269000	17.80	196.02	168.01	28.01	18500	261000	2.80	50.4	96.5	.01037	0.0954						
36	122.72	132.69	9.97	32100	320000	7.94	196.08	171.00	25.08	26200	330000	4.74	54.8	108.6	.00921	0.184						
	126.73	133.92	7.19	45500	327000	11.22	196.05	170.96	25.09	26200	330000	4.76	51.8	116.4	.00858	0.139		143	81.7	504	286	11.1
	128.98	134.60	5.62	57900	326000	14.30	195.94	170.98	24.96	26200	328000	4.76	50.5	118.7	.00842	0.113						
	130.02	134.52	4.50	72000	324000	17.78	195.72	170.87	24.85	26100	326000	4.96	49.7	120.0	.00833	0.0947						

TABLE IV

SUMMARY OF EXPERIMENTAL DATA AND CALCULATED RESULTS

5/8" FINNED TUBES IN 6" SHELL BUNDLE 6

Run No.	Water on Tube Side: Temperature, °F in	out	rise	Pounds per hr.	Heat trans. BTU per hr.	Water Velocity Ft./sec.	Shell Side: Temperature, °F in	out	drop	Pounds per hr.	Heat trans. BTU per hr.	Pres. drop psi	Mean temp. difference °F	U_o	$\frac{1}{U_o}$	$\frac{N}{V^{0.8}}$	Calc. h_i'	Average Shell Side Values: h_o	Nu	Re	Pr	$(Nu)(Pr)^{-.375}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$
										Oil on Shell Side												
37	122.41	134.13	11.72	32100	376000	7.94	195.93	172.76	23.17	33000	388000	7.20	54.9	127.9	.00782	0.184						
	126.06	134.49	8.43	45500	384000	11.22	196.10	172.72	23.38	33300	392000	7.17	53.6	132.6	.00754	0.138						
	128.28	134.98	6.70	57400	384000	14.19	195.75	172.47	23.28	32900	388000	7.23	50.3	141.0	.00710	0.114		173	98.5	655	283	13.4
	130.06	135.42	5.36	72000	386000	17.78	196.08	172.83	23.25	32900	386000	7.35	50.5	140.0	.00713	0.0944						
38	122.54	135.97	13.43	31400	421000	7.76	196.05	174.61	21.44	39200	425000	9.65	55.4	140.0	.00713	0.183						
	127.86	136.92	9.06	45800	415000	11.31	196.02	174.58	21.44	39300	426000	9.65	52.0	148.5	.00673	0.137						
	130.23	137.37	7.14	58000	415000	14.31	195.81	174.39	21.42	39100	424000	9.63	50.6	151.8	.00659	0.112		194	111	791	277	15.2
	131.88	137.75	5.87	71800	421000	17.72	195.90	174.56	21.34	39500	425000	9.70	49.4	157.0	.00637	0.0936						
72	69.478	72.194	2.716	32800	89000	7.95	113.21	101.03	12.18	15500	87400	5.50	35.8	45.5	.0220	0.248						
	70.722	72.869	2.147	42900	92100	10.38	113.75	101.34	12.41	15600	89600	5.57	35.7	46.7	.0214	0.199						
	71.537	73.116	1.579	58300	92000	14.10	113.84	101.61	12.23	15700	89000	5.39	34.3	48.4	.0206	0.156		53.2	30.0	47.5	1675	2.19
	71.960	73.193	1.233	72800	89900	17.61	113.30	101.03	12.27	16000	90900	5.42	34.2	48.5	.0201	0.129						
73	69.800	72.145	2.345	32000	75100	7.76	113.44	99.68	13.76	10600	67500	3.86	35.3	37.0	.0270	0.253						
	69.620	71.420	1.800	42900	77200	10.38	113.57	99.77	13.80	10500	67100	3.95	35.5	37.3	.0268	0.201						
	70.599	71.946	1.346	58500	78700	14.16	113.68	99.99	13.69	11000	69600	4.00	35.0	38.8	.0258	0.156		41.9	23.6	34.0	1697	1.72
	73.042	74.102	1.060	72700	77100	17.60	114.20	101.01	13.19	11500	70300	4.15	33.4	40.4	.0248	0.128						
										Glycerine on Shell Side												
80	123.08	137.93	14.85	51300	764000		194.61	173.91	20.70	58200	780000	14.40	52.8	268	.00373	0.124	1030	400	108	1222	192	16.9
81	122.43	141.15	18.72	32500	609000		193.86	173.44	20.42	47500	629000	9.49	50.5	224	.00445	0.180						
	126.64	141.13	14.49	42900	620000	10.60	193.69	173.16	20.53	48000	638000	9.84	48.3	239	.00418	0.142						
	131.47	142.27	10.80	57900	625000	14.30	194.14	173.39	20.75	48100	648000	9.84	46.0	254	.00394	0.109		375	101	917	199	12.4
	134.08	142.77	8.69	72600	631000	17.97	193.73	173.26	20.47	48700	646000	10.05	44.4	263	.00380	0.091						
82	122.16	133.74	11.58	51300	595000	12.69	194.79	169.65	25.14	38000	619000	6.61	53.2	210	.00477	0.126	1020	272	73.5	701	204	11.6
83	122.68	132.49	9.81	51300	504000	12.69	193.62	167.13	26.49	30000	514000	4.57	51.9	180	.00556	0.127	1020	229	62.1	532	211	9.76
84	122.13	129.87	7.74	51300	397000	12.69	193.53	164.08	29.45	21900	418000	2.70	51.2	146	.00684	0.127	1010	178	48.3	379	218	7.55
85	121.55	128.07	6.52	51300	334000	12.69	193.41	162.23	31.18	16700	334000	1.90	51.6	119	.00840	0.128	1010	139	37.7	288	224	5.81
86	194.82	198.22	3.40	51300	174000	12.69	227.34	207.86	19.48	13800	182000	1.05	18.8	158	.00632	0.096	1340	187	50.1	484	111	8.90
87	194.68	204.01	9.33	33700	314000	8.34	226.11	216.23	9.88	46900	315000	8.96	21.5	268	.00374	0.133						
	197.38	204.41	7.03	43200	305000	10.69	226.33	216.36	9.97	46100	313000	8.90	19.8	287	.00349	0.108						
	199.99	205.32	5.33	58100	311000	16.85	226.36	216.50	9.86	45500	305000	8.59	18.2	310	.00322	0.075		450	121	1709	108	21.4
	201.92	205.88	3.96	78900	312000	19.50	225.72	215.71	10.01	44500	303000	8.34	16.1	347	.00288	0.066						
88	194.16	201.58	7.42	51900	385000	12.82	225.39	215.79	9.68	60100	396000	14.80	21.8	332	.00302	0.095	1360	490	131	2225	108	23.3
101	86.04	90.18	4.14	42300	175000	10.46	131.04	117.61	13.43	23500	189000	5.73	35.7	93.4	.0107	0.180	715	110	30.3	80	1040	2.65
102	86.09	91.83	5.74	42200	242000	10.41	131.11	119.71	11.40	35500	243000	10.98	35.9	124.0	.00806	0.178	715	155	42.6	127	1005	3.76
103	85.96	89.60	3.64	32900	120000	8.14	130.86	114.51	16.35	13500	132000	3.10	34.2	67.6	.01482	0.221	585	77.5	21.3	42.5	1100	1.70

TABLE VI

WEIGHTED FLOW AREA SHELL SIDE

Example Calculations for 8-in. Exchanger with 3/4-in. Finned Tubes, Bundle No. 2

Longitudinal-Flow Area

Baffle cut 1.94., add 0.02 in. for clearance between baffle and shell. Baffle cut based on shell I.D. = 1.96 in. Area of baffle window (Ref. 3, page 32) = 9.60 sq in. Number of tubes in window $6 + (2 \times 0.5) + 0.75 = 7.75$. Area of tubes = $\pi/4\ (0.735)^2 \times 7.75 = 3.28$ sq in. Net flow area $9.60 - 3.28 = 6.32$ sq in.

Cross-Flow Area

Number of tubes in row nearest to a diameter normal to the direction of flow = 8

Equivalent diameter of tube $D_e = 0.660$ in.

Cross-flow area = $(7.972 - 8 \times 0.660)4 = 10.60$ sq in.

Geometric mean $= \sqrt{10.60 \times 6.32} = 8.18$ sq in.

Flow area end spaces, 7.31 in. long.

Cross-flow area $7.31 \times 7.972 - 8 \times 0.660 = 19.40$ sq in.

Geometric mean $= \sqrt{19.40 \times 6.32} = 11.1$ sq in.

Weighted average of end spaces 14.6 in. long and baffled space, 32 in. long:

$$\frac{11.1 \times 14.6 + 8.18 \times 32}{46.6} = 9.25 \text{ sq in.} \qquad \text{or} \qquad 0.0643 \text{ sq ft, } A_m$$

Flow areas for all exchangers are listed in Table I.

APPENDIX II

Reproduced from Alco Heat Exchanger Price Book, with tubing prices furnished by the Wolverine Tube Division, for copies of the report distributed to students in Course CM 121 at the University of Michigan.

TEMA CLASS "R" UNIT PRICING SEQUENCE

DESIRED:

150 lb. TEMA Standard Exchanger
Size 36" dia. by 16' long
Tube size 3/4" O.D. x 14 gauge 30% Cupro-Nickel
on triangular pitch
Tube sheets naval rolled bronze
Baffles 20 - 3/16" segment type Muntz metal
Passes shell side 1, tube side 4
Nozzles - Shell (1 - 8" dia. radial)
(1 - 8" dia. impingement)
Nozzles - channel 2 - 8" dia. radial

EXAMPLE:

1.	Price of shell from page 135.........	$ 1,957.00
2.	Price of tube side from page 136.....	879.00
3.	Price of nozzles from page 137.......	402.00
4.	Price of tube sheets from page 138...	2,088.00
5.	Price of baffles from page 139.......	1,414.00
6.	Price of tubes from page 144-46......	7,899.54
		$14,639.54

Price F.O.B. Factory

*If supports plates are also desired, select price from pages 140-43.

SHELL SIDE - STEEL
(16' 0" LONG)
1 PASS

		150 # STANDARD			300 # STANDARD		
Nom. Size	Suggested Noz. Size	Shell Thick.	Price	Extra Per Ft.	Shell Thick.	Price	Extra Per Ft.
12"	3"	3/8"	$ 840.	$ 6.40	3/8"	$ 855.	$ 6.40
14"	3"	3/8"	920.	6.75	3/8"	920.	6.75
16"	3"	3/8"	937	7.70	3/8"	947.	7.70
18"	3"	3/8"	962.	8.29	3/8"	973.	8.29
20"	4"	3/8"	1087.	9.56	7/16"	1108	10.58
22"	4"	3/8"	1168.	10.10	7/16"	1185.	11.25
24"	6"	3/8"	1251.	11.60	1/2"	1272.	12.74
27"	6"	3/8"	1414.	13.01	1/2"	1440.	14.34
30"	8"	3/8"	1561.	14.19	9/16"	1623.	17.52
33"	8"	7/16"	1645.	15.00	5/8"	1746	20.32
36"	8"	7/16"	1957.	16.40	5/8"	2104.	22.07
39"	10"	7/16"	2324.	17.53	11/16"	2497.	25.57
42"	10"	1/2"	2684.	23.27	3/4"	2960.	30.40

Above prices include shell, flanges, shell cover, 2 saddles, gaskets and bolts, 3 outside clamp rings, assemble and test, skids, seal strips and miscellaneous. For shells under 16' 0" long, deduct price per foot for shorter shell.

TUBE SIDE
(4 PASS)

		150# STANDARD	300# STANDARD	EXTRA FOR 1 RIB	
Nom. Size Nom. Size	Suggested Noz. Size	Price	Price	Channel	Float,Head
12"	3"	$ 379.	$ 386.	$ 11.54	$ 5.05
14"	3"	396.	404.	11.70	5.23
16"	3"	426.	438.	12.60	5.50
18"	3"	428.	442.	12.23	5.35
20"	4"	465.	482.	12.59	6.04
22"	4"	492.	515.	12.75	6.13
24"	6"	528.	549.	13.63	6.09
27"	6"	610.	627.	14.90	7.31
30"	8"	705.	731.	15.48	8.07
33"	8"	775.	817.	15.60	8.34
36"	8"	879.	929.	18.23	9.35
39"	10"	1149.	1231.	18.70	12.20
42"	10"	1278.	1398.	20.03	12.71

Above prices include channel barrel, channel flanges, channel cover, floating head cover, 1 floating head closure, bolts, gaskets, and grooving.

For 2 Pass deduct one channel and floating head rib.
For 6 Pass add three channel and floating head ribs.

NOZZLES

SHELL SIDE NOZZLES

Nozzle Size	150# Radial Type	150# Impingement Type	300# Radial Type	300# Impingement Type
1	53.	----	53.	----
1-1/2	53.	107.	54.	107.
2	54.	80.	55.	80.
2-1/2	57.	81.	57.	81.
3	60.	89.	60.	90.
4	65.	105.	66.	106.
5	69.	114.	70.	115.
6	81.	118.	82.	119.
8	98.	156.	100.	159.
10	116.	200.	120.	203.
12	143.	222.	148.	228.
14	174.	238.	186.	250.

TUBE SIDE NOZZLES

Nozzle Size	150# Radial Type	150# Tangential Type	300# Radial Type	300# Tangential Type
1	32.	39.	32.	40.
1-1/2	33.	40.	33.	41.
2	34.	41.	34.	42.
2-1/2	36.	44.	36.	44.
3	39.	47.	39.	47.
4	45.	53.	46.	54.
5	48.	57.	50.	58.
6	59.	75.	61.	76.
8	74.	95.	77.	97.
10	93.	132.	97.	136.
12	116.	145.	121.	160.

TUBE SHEETS (PER PAIR)

150# STANDARD

Nom. Size	STEEL 5/8" Tubes	STEEL 3/4" Tubes	STEEL 1" Tubes	N.R. 5/8" Tubes	N.R. 3/4" Tubes	N.R. 1" Tubes
12"	$ 172.	$ 158.	$ 133.	$ 272.	$ 257.	$ 238.
14"	190.	163.	144.	311.	288.	272.
16"	240.	197.	178.	401.	366.	349.
18"	278.	227.	191.	480.	436.	410.
20"	349.	293.	242.	651.	596.	546.
22"	422.	341.	284.	782.	704.	651.
24"	485.	390.	325.	911.	819.	757.
27"	645.	521.	434.	1210.	1085.	999.
30"	789.	634.	514.	1513.	1362.	1240.
33"	960.	775.	627.	1853.	1665.	1523.
36"	1285.	1024.	808.	2332.	2088.	1899.
39"	1500.	1195.	931.	2803.	2623.	2289.
42"	1880.	1500.	1172.	3428.	3093.	2896.

300# STANDARD

Nom. Size	STEEL 5/8" Tubes	STEEL 3/4" Tubes	STEEL 1" Tubes	N.R. 5/8" Tubes	N.R. 3/4" Tubes	N.R. 1" Tubes
12"	$ 178.	$ 164.	$ 139.	$ 315.	$ 301.	$ 276.
14"	208.	180.	160.	369.	342.	324.
16"	261.	216.	197.	488.	446.	426.
18""	305.	252.	217.	595.	540.	505.
20"	379.	322.	269.	762.	704.	651.
22"	464.	383.	324.	917.	833.	774.
24"	529.	433.	366.	1070.	975.	907.
27"	766.	619.	502.	1448.	1298.	1183.
30"	1003.	804.	635.	1937.	1738.	1569.
33"	1208.	976.	768.	2373.	2137.	1930.
36"	1423.	1145.	912.	2903.	2623.	2386.
39"	1692.	1368.	1070.	3493.	3173.	2876.
42"	2419.	1930.	1471.	4547.	4127.	3754.

Above prices include a pair of tube sheets (fixed and floating), drilled, reamed, and grooved.

SEGMENT BAFFLES

	5/8" Tubes			3/4" Tubes			1" Tubes		
STEEL - PER BAFFLE									
Size	1/8"	3/16"	1/4"	1/8"	3/16"	1/4"	1/8"	3/16"	1/4"
12"	$6.63	$8.21	$9.90	$6.34	$7.83	$9.41	$6.00	$7.45	$8.95
14"	7.48	9.46	11.64	7.10	8.98	11.02	6.68	8.45	10.39
16"	8.35	10.34	13.70	7.88	9.76	12.90	7.40	9.15	12.12
18"	9.67	11.11	14.42	8.93	10.42	13.54	8.34	9.73	12.63
20"	11.00	14.10	17.78	10.28	13.17	16.64	9.58	12.27	15.45
22"	12.80	15.99	22.08	11.90	14.88	20.53	11.00	13.75	18.96
24"	13.89	17.79	23.12	12.86	16.49	22.05	11.83	15.18	19.68
27"	19.60	23.55	33.60	17.85	21.65	30.15	16.50	19.80	28.30
30"	25.05	30.00	43.35	22.95	27.50	39.70	20.90	25.05	36.18
33"	30.26	34.95	48.70	27.65	31.80	44.50	25.00	28.75	40.35
36"	33.80	39.90	57.90	30.30	36.30	52.75	27.28	32.65	47.48
39"	42.80	47.40	68.65	38.75	42.90	62.20	34.62	38.40	55.70
42"	47.50	56.40	82.50	42.75	50.80	74.40	38.10	45.20	66.08
MUNTZ METAL - PER BAFFLE									
Size	1/8"	3/16"	1/4"	1/8"	3/16"	1/4"	1/8"	3/16"	1/4"
12"	12.87	15.83	18.57	12.40	15.08	17.69	11.80	15.00	16.86
14"	16.47	18.31	21.68	14.21	17.39	20.62	13.47	16.48	19.58
16"	17.48	21.73	25.88	16.54	20.53	24.59	15.45	19.40	23.14
18"	19.70	24.28	28.80	18.53	22.85	27.15	17.40	21.45	25.57
20"	22.52	28.70	34.79	21.09	27.05	32.67	19.67	25.25	30.14
22"	26.24	33.68	40.84	44.47	31.54	38.24	22.77	29.38	35.59
24"	29.97	38.93	45.89	27.95	35.33	42.84	25.82	32.75	39.69
27"	35.72	47.23	57.29	33.17	43.78	53.19	30.59	40.38	48.99
30"	42.38	57.94	68.81	39.18	53.54	63.56	35.83	49.04	58.51
33"	48.48	65.22	78.76	44.63	60.17	72.46	40.68	54.87	66.21
36"	57.13	77.30	93.91	52.33	70.70	86.11	47.48	64.20	78.26
39"	66.30	90.81	109.70	60.35	82.66	100.15	54.40	74.76	90.70
42"	78.60	107.38	128.00	71.25	97.68	116.55	64.10	87.83	104.80

Above prices include drilling, cutting, and 1' length of tie rod with spacers.

LONG BAFFLES AND SUPPORT PLATES

	Liquid Type Expanding Long Baffles (16'long)				Vapor Type Perforated Long Baffles (16'long)				Floating Head Support Pl. 1/2"	
	Steel		N.R.		Steel		N.R.		Steel	N.R.
Size	Price	Extra Per Ft.	Price	Extra Per Ft.	Price	Extra Per Ft.	Price	Extra Per Ft.	Price	Price
12"	$197	$7.70	$480.	$25.42	$61	$.93	$185	$9.90	$30.	$43.
14"	201	7.80	509	26.77	65	.96	193	10.45	35	51.
16"	213	8.43	554	29.13	65	1.03	207	10.41	38	56
18"	206	8.13	558	29.75	62	1.04	209	11.69	40	59
20"	213	8.30	610	31.67	66	1.14	223	12.72	45	68
22"	218	8.47	639	33.46	66	1.21	232	13.27	49	75
24"	220	8.37	660	34.51	66	1.25	233	13.55	50	83
27"	249	9.52	730	41.73	73	1.37	261	15.06	58	100
30"	255	9.68	813	43.07	74	1.49	281	16.32	63	108
33"	265	9.59	870	45.79	75	1.53	286	16.85	70	122
36"	284	10.21	909	47.61	83	1.67	313	18.44	83	144
39"	288	10.45	975	51.32	86	1.78	332	19.58	98	172
42"	299	10.60	1080	56.02	87	1.87	348	20.82	114	195

STEEL 3/8" SUPPORT PLATES
(Price of One)

	Basis of 1 or 2 Plates			Basis of 3 or more Plates		
Size	5/8" Tubes	3/4" Tubes	1" Tubes	5/8" Tubes	3/4" Tubes	1" Tubes
12"	12.50	11.30	9.56	12.00	10.90	9.35
14"	14.25	12.69	10.65	13.62	12.23	10.40
16"	16.70	15.00	12.40	15.80	14.40	12.05
18"	20.00	17.25	14.32	18.85	16.45	13.85
20"	23.70	20.40	16.50	22.20	19.31	15.85
22"	27.50	23.30	18.00	25.70	21.90	17.30
24"	31.80	26.70	20.40	29.60	25.10	19.50
27"	40.25	33.50	25.30	37.50	1.50	24.10
30"	47.80	40.40	28.60	44.20	37.70	27.30
33"	57.80	47.50	34.30	53.50	44.00	32.50
36"	67.25	55.20	40.00	60.50	51.30	37.80
39"	78.00	63.80	45.40	71.50	59.20	42.80
42"	90.00	73.20	52.00	82.60	67.75	49.00

STEEL 1/2" SUPPORT PLATES
(Price of One)

	Basis of 1 or 2 Plates			Basis of 3 or more Plates		
Size	5/8" Tubes	3/4" Tubes	1" Tubes	5/8" Tubes	3/4" Tubes	1" Tubes
12"	14.10	12.60	10.40	12.70	11.50	9.83
14"	16.20	14.22	11.65	14.50	12.96	11.00
16"	17.55	17.10	13.75	16.30	15.38	12.81
18"	21.10	18.30	16.15	19.40	17.05	14.85
20"	25.10	21.70	18.75	23.01	20.00	17.10
22"	29.60	24.80	20.00	26.60	22.82	18.20
24"	34.00	28.60	21.90	30.70	26.10	22.40
27"	42.90	36.00	27.30	38.80	32.90	25.50
30"	52.20	43.40	32.30	47.00	39.40	30.00
33"	62.20	51.20	37.30	55.50	46.30	34.50
36"	72.40	59.40	43.40	64.50	53.60	40.10
39"	83.80	68.90	49.60	74.75	61.90	45.60
42"	96.50	78.80	56.60	85.75	70.75	52.00

STEEL 5/8" SUPPORT PLATES
(Price of One)

Basis of 1 or more Plates

Size	5/8" Tubes	3/4" Tubes	1" Tubes
12"	18.30	16.38	13.58
14"	20.80	18.38	15.08
16"	25.60	22.00	17.75
18"	31.30	26.50	20.80
20"	37.50	31.80	24.10
22"	44.00	36.70	27.50
24"	51.50	42.40	31.40
27"	65.80	53.90	39.10
30"	80.25	65.30	47.00
33"	96.00	77.80	54.60
36"	101.18	90.50	63.80
39"	130.50	105.20	73.00
42"	150.50	121.00	83.25

TUBING CHARACTERISTICS											
O.D OF TUBING	WALL THICKNESS		OUTSIDE AREA SQ. FT. PER FT.	CROSS SECTION AREA OF BORE SQ. IN.	I.D INCHES	WEIGHT PER FOOT (AVERAGE) LBS.					
	B.W.G GUAGE	INCHES				ADMIRALTY & 70-30 BRASS	COPPER & CUPRO-NICKEL 30%	ALUMINUM BRASS	ALUMINUM BRONZE 5%	RED BRASS 85/15	MUNTZ METAL
5/8"	18	.049	.164	.218	.527	.338	.354	.330	.323	.346	.332
	16	.065		.192	.495	.435	.457	.425	.417	.447	.428
	14	.083		.165	.459	.538	.564	.526	.515	.552	.529
3/4"	18	.049	.196	.334	.652	.411	.431	.401	.393	.421	.404
	16	.065		.302	.520	.532	.558	.520	.510	.546	.524
	14	.083		.268	.584	.662	.694	.647	.634	.629	.651
	12	.109		.222	.532	.836	.876	.817	.800	.857	.822
1"	18	.049	.262	.639	.902	.557	.584	.545	.534	.572	.548
	16	.065		.594	.870	.727	.762	.710	.696	.746	.715
	14	.083		.546	.834	.910	.955	.890	.872	.934	.895
	12	.109		.480	.782	1.16	1.22	1.14	1.11	1.19	1.14
	10	.134		.421	.732	1.39	1.46	1.36	1.33	1.42	1.37

CONDENSER TUBES - HEAT EXCHANGER TUBES
BASIC SCHEDULE - CENTS PER POUND
LENGTHS 1 FT TO 30 FT, INCL.

OUTSIDE DIAMETER IN INCHES	THICKNESS OF WALL B.W.G. GUAGE & IN INCHES-NET PER POUND								
	10 UNDER .148 TO .134 INC.	11 UNDER .134 TO .120 INC.	12 UNDER .120 TO .109 INC.	13 UNDER .109 TO .095 INC.	14 UNDER .095 TO .083 INC.	15 UNDER .083 TO .072 INC.	16 UNDER .072 TO .065 INC.	17 UNDER .065 TO .058 INC.	18 UNDER .058 TO .049 INC.
INC. 5/8 TO 3/4	$.5200	$.5200	$.5200	$.5200	$.5200	$.5200	$.5262	$.5262	$.5312
INC. 3/4 TO 7/8	.5150	.5087	.5087	.5087	.5087	.5087	.5087	.5150	.5200
INC. 7/8 TO 1"	.5087	.5087	.5087	.5087	.5087	.5087	.5087	.5150	.5200
INC. 1 TO 1 1/4	.5062	.5062	.5062	.5062	.5062	.5062	.5062	.5137	.5200

QUANTITY SCHEDULE

The Base Schedule Prices are subject to the following deductions and additions on account of the quantity of a single ORDER for shipment to one destination at one time.

Term "order" means the amount contained in one order for one alloy only, in varying sizes and lengths for shipment at one time to one destination.

30,000 lbs. and over	Less	$ .0255	per pound
15,000 lbs. to 30,000 lbs.	Less	.02	per pound
10,000 lbs. to 15,000 lbs.	Less	.01	per pound
5,000 lbs. to 10,000 lbs.	Less	.005	per pound
2,000 lbs. to 5,000 lbs.	Above prices		
1,000 lbs. to 2,000 lbs.	Add	.01	per pound
500 lbs. to 1,000 lbs.	Add	.025	per pound
300 lbs. to 500 lbs.	Add	.05	per pound
100 lbs. to 300 lbs.	Add	.08	per pound
Under 100 lbs.	Consult Mill		

ALLOY SCHEDULE

The base prices in this schedule are subject to additional or deductions for the alloy required. All tubes of the following alloys within the range of sizes covered by the BASE SCHEDULE will be priced accordingly, regardless of their ultimate use.

Admiralty 70/29/1	Base Schedule Prices
Inhibited Admiralty	Base Schedule Prices
Arsenical Copper	Base Plus $ .0156 Net
Cupro-Nickel - 30%	Base Plus .1902 Net
Aluminum Brass 76/22/2	Base Plus .0706 Net
Aluminum Bronze 95/5	Base Plus .1586 Net
70/30 Brass	Base Schedule Prices
85/15 Red Brass	Base Schedule Prices
Muntz Metal	Base Less $ .0100 Net

WOLVERINE TRUFIN PRICES

COPPER — 19 Fins Per Inch

Nominal Size

Plain End Diameter	Root Diameter		Wall Thickness	Base Price Per Ft
1/2"	3/8"	x	.032"	$ .2125
5/8"	1/2"	x	.035"	.2550
3/4"	5/8"	x	.042"	.3225
3/4"	5/8"	x	.065"	.3900
3/4"	3/4"	x	.049"	.3900

ALLOYS — 19 Fins Per Inch

Nominal Size

	Plain End Diameter	Root Diameter		Wall Thickness	Base Price Per Ft
85-15 Red Brass	3/4"	5/8"	x	.049"	$ .3700
70-30 Cupro-Nickel	3/4"	5/8"	x	.049"	.5775
70-30 Cupro-Nickel	3/4"	5/8"	x	.065"	.6550
Admiralty	3/4"	5/8"	x	.065"	.4125

QUANTITY SCHEDULE

The quantity of each item for delivery at one time determines the price.

		Per Foot
250,000 ft and over	Base Less	$ .04
Inc. 175,000 ft to 250,000	Base Less	.03
Inc. 100,000 ft to 100,000	Base Less	.025
Inc. 50,000 ft to 100,000	Base Less	.02
Inc. 10,000 ft to 50,000	Base Less	.01
Inc. 5,000 ft to 10,000	Base Less	.005
Inc. 2,000 ft to 5,000	BASE PRICE	
Inc. 1,000 ft to 2,000	Base Plus	.005
Inc. 500 ft to 1,000	Base Plus	.01
Inc. 300 ft to 500	Base Plus	.02
Inc. 100 ft to 300	Base Plus	.04
Less than 100 ft	Base Plus	.01

PRICES AND WEIGHT OF MIN. WALL STEEL TUBES					
Sq. Ft. Per Lin. Ft.	Dia.	Gage	Weight Lin. Ft.	Price	
				Lin.Ft.	Sq.Ft.
.1636	5/8	18	.340	.1562	.96
	5/8	16*	.435	.639	1.01
	5/8	14	.538	.1712	1.05
.1963	3/4	18	.415	.1629	.83
	3/4	16	.534	.1703	.87
	3/4	14*	.665	.1788	.91
.2617	1	18	.565	.1833	.70
	1	16	.732	.1899	.725
	1	14	.918	.1986	.76
	1	13	1.037	.2063	.79
	1	12*	1.168	.2150	.82

Above prices for steel tubes are <u>Base Prices</u>. See Extras below. On items marked* on 16' 0" long tubes only, use <u>Base Price</u> regardless of quantity.

<u>Extras</u>

40,000 lbs or Feet or over = Base	
30,000 lbs to 39,999 lbs or Ft	5%
20,000 lbs to 29,999 lbs or Ft	10%
10,000 lbs to 19,999 lbs or Ft	20%
5,000 lbs to 9,999 lbs or Ft	30%
2,000 lbs to 4,999 lbs or Ft	45%
Under 2000 lbs of Ft	65%

Example:

16,480 Ft of 3/4" x 14 gauge, Cupro-Nickel 30%, plain tubing
Weight = 16,480 x 0.694 = 11.437 lbs

Base price	$0.5087 per lb
Alloy Extra	0.1920 per lb
	$0.7007 per lb
Quantity Discount	0.0100 per lb
Net	$0.6907 per lb

Net price = 0.6907 x 11,437 = $7899.54

ENGINEERING DATA

5/8" O.D.TUBE ON 13/16" TRIANGULAR PITCH
(Single Pass Shell)

Unit Size	2 Pass Channel				4 Pass Channel					6 Pass Channe		
		Surface				Surface				Surface		
	No. Tubes	8'	12'	16'	No.Tubes	8'	12'	16'	No.Tubes	8'	12'	1
12"	138	181	271	382	130	171	256	342	124	163	244	3
14"	172	226	339	452	156	205	307	410	150	197	295	3
16"	242	318	477	636	216	284	426	568	210	275	412	5
18"	316	415	622	830	288	378	567	756	284	373	559	7
20"	404	530	795	1060	372	488	732	976	362	475	712	9
22"	490	644	966	1288	460	604	906	1208	454	595	892	11
24"	586	768	1152	1536	558	732	1098	1464	542	711	1066	1
27"	770	1010	1515	2020	732	960	1440	1920	716	939	1408	18
30"	970	1272	1908	2544	926	1215	1822	2430	912	1195	1792	2
33"	1188	1560	2340	3120	1144	1500	2250	3000	1118	1467	2200	2
36"	1378	1810	2715	3620	1338	1755	2632	3510	1318	1730	2595	3
39"	1644	2160	3240	4320	1594	2090	3135	4180	1576	2070	3105	41
42"	1922	2520	3780	5040	1862	2440	3660	4880	1844	2420	3630	48

3/4" O.D. TUBE ON 15/16" TRIANGULAR PITCH
(Single Pass Shell)

Unit Size	2 Pass Channel				4 Pass Channel				6 Pass Channel			
		Surface				Surface				Surface		
	No.Tubes	8'	12'	16'	No.Tubes	8'	12'	16'	No.Tubes	8'	12'	16'
12"	106	167	250	334	98	154	231	308	96	151	226	302
14"	126	198	297	396	114	179	268	358	108	170	255	340
16"	172	271	406	542	156	245	367	490	150	236	354	472
18"	232	365	547	730	210	330	495	660	204	321	481	642
20"	298	468	702	936	278	437	655	874	270	425	637	850
22"	358	562	843	1124	338	532	798	1064	328	516	774	1032
24"	432	680	1020	1360	410	645	967	1290	398	626	939	1252
27"	566	890	1335	1780	538	845	1267	1690	526	827	1240	1654
30"	720	1132	1698	2264	682	1072	1608	2142	670	1052	1578	2104
33"	882	1385	2077	2760	844	1330	1995	2660	824	1295	1942	2590
36"	1030	1620	2430	3240	986	1550	2325	3100	976	1535	2252	3070
39"	1224	1925	2887	3850	1184	1862	2793	3724	1162	1830	2745	3660
42"	1426	2242	3363	4484	1378	2165	3247	4330	1358	2140	3210	4280

1" O.D. TUBES ON 1-1/4" TRIANGULAR PITCH
(Single Pass Shell)

Unit Size	2 Pass Channel				4 Pass Channel				6 Pass Channe			
		Surface				Surface				Surface		
	No.Tubes	8'	12'	16'	No.Tubes	8'	12'	16'	No.Tubes	8'	12'	16
12"	52	109	163	218	52	109	163	218	50	105	157	210
14"	72	150	225	300	60	126	189	252	54	113	169	226
16"	98	205	307	410	86	180	270	360	82	172	258	344
18"	126	264	396	528	116	243	364	486	110	231	346	462
20"	164	343	514	686	152	318	477	636	146	306	459	612
22"	195	407	610	814	186	390	585	780	176	319	478	638
24"	242	506	759	1012	228	478	717	956	220	461	691	922
27"	315	660	990	1320	298	624	936	1248	290	607	910	1214
30"	400	836	1254	1772	378	792	1188	1584	366	967	1150	1534
33"	486	1020	1530	2040	464	970	1455	1940	462	968	1452	1936
36"	570	1191	1786	2382	546	1145	1717	2290	542	1135	1702	2270
39"	680	1422	2133	2844	652	1365	2047	2730	648	1360	2040	2720
42"	792	1655	2482	3310	764	1600	2400	3200	752	1575	2362	3150

5/8" O.D. ON 13/16" SQUARE PITCH
(Single Pass Shell)

Unit Size	2 Pass Channel				4 Pass Channel				6 Pass Channel			
	No.Tubes	Surface			No.Tubes	Surface			No.Tubes	Surface		
		8'	12'	16'		8'	12'	16'		8'	12'	16'
12"	124	163	244	326	116	152	228	304	116	152	228	304
14"	158	207	307	414	140	184	276	368	132	173	259	346
16"	208	273	409	546	196	257	385	514	188	246	369	492
18"	274	359	538	718	262	344	516	688	252	330	495	660
20"	352	462	693	924	332	436	654	872	320	420	630	840
22"	424	555	832	1110	406	532	798	1064	396	520	780	1040
24"	518	680	1020	1380	488	640	960	1280	484	635	1007	1270
27"	664	872	1308	1744	644	845	1267	1690	628	824	1236	1648
30"	844	1106	1659	2212	816	1070	1605	2140	804	1055	1582	2110
33"	1032	1355	2032	2710	1004	1320	1980	2640	980	1285	1927	2570
36"	1204	1581	2371	3162	1176	1545	2317	3090	1140	1500	2250	3000
39"	1430	1875	2812	3750	1398	1835	2752	3670	1372	1800	2700	3600
42"	1664	2182	3273	4364	1620	2130	3195	4260	1600	2100	3150	4200

3/4" O.D. ON 1" SQUARE PITCH
(Single Pass Shell)

Unit Size	2 Pass Channel				4 Pass Channel				6 Pass Channel			
		Surface				Surface				Surface		
	No.Tubes	8'	12'	16'	No.Tubes	8'	12'	16'	No.Tubes	8'	12'	16
12"	80	125	188	251	76	119	179	238	76	119	179	[illegible]
14"	100	157	235	314	92	144	216	289	84	131	197	[illegible]
16"	132	207	310	414	124	194	292	389	116	187	273	[illegible]
18"	178	279	419	558	168	263	395	527	160	251	376	[illegible]
20"	224	351	527	703	216	339	508	678	212	327	499	[illegible]
22"	276	433	650	866	270	423	635	847	260	407	612	[illegible]
24"	336	527	791	1055	324	508	763	1017	312	489	734	[illegible]
27"	436	684	1026	1369	420	609	989	1318	408	640	960	[illegible]
30"	554	869	1304	1739	532	835	1252	1670	528	829	1243	[illegible]
33"	676	1061	1592	2122	656	1029	1544	2059	640	1004	1507	[illegible]
36"	792	1243	1865	2487	762	1196	1794	2392	752	1180	1771	[illegible]
39"	934	1466	2199	2932	918	1441	2161	2882	896	1406	2110	[illegible]
42"	1100	1727	2590	3454	1064	1671	2505	3341	1048	1645	2468	[illegible]

1" O.D. ON 1-1/4" SQUARE PITCH
(Single Pass Shell)

Unit Size	2 Pass Channel				4 Pass Channel				6 Pass Channel			
		Surface				Surface				Surface		
	No.Tubes	8'	12'	16'	No.Tubes	8'	12'	16'	No. Tubes	8'	12'	16'
12"	52	108	163	217	48	100	150	201				
14"	60	125	188	251	60	125	188	251	56	117	174	234
16"	82	171	257	343	78	163	245	326	72	150	226	301
18"	112	234	351	469	104	217	326	435	100	209	314	418
20"	144	301	452	603	136	284	427	569	128	218	402	536
22"	170	356	533	712	166	347	521	695	158	331	496	662
24"	212	444	665	888	204	427	640	854	196	410	675	821
27"	274	574	860	1148	266	557	835	1114	260	543	815	1087
30"	344	721	1080	1441	336	703	1055	1407	324	678	1017	1357
33"	428	896	1344	1793	414	862	1300	1734	402	842	1262	1684
36"	494	1034	1551	2069	484	1012	1519	2025	476	996	1494	1992
39"	596	1247	1871	2495	576	1205	1808	2411	564	1181	1771	2361
42"	688	1441	2160	2882	664	1391	2085	2782				

APPROXIMATE SHIPPING WEIGHT

Std. Each Size	AVERAGE WEIGHT (Less Tubes)			
	150# Series		300# Series	
	Wt. 8'0"	Added Wt./Foot	Wt. 8'0"	Added Wt./Foot
12"	1690	70	1835	70
14"	1876	77	2040	77
16"	2456	97	2658	97
18"	2888	109	3232	109
20"	3348	126	4035	140
22"	3846	137	4621	152
24"	4768	166	5641	182
27"	5884	188	6998	206
30"	6810	210	8458	251
33"	8116	237	10278	306
36"	9461	257	12028	331
39"	11030	280	14336	382
42"	12409	328	16360	410

Weights Are Based On

1)-Type 2-34-42H Exchangers.
2)-3/16"Steel Baffles on 12" centers.
3)-Normal Size Nozzles.
4)-Domestic Type Skids.

Add weight per foot for Exchangers over 8' 0" Long.
Add weight of Tubes from Sheet No. 9.

EXAMPLE: - 30" - 192 Exchanger, 150 series
1840 sq. ft. of Tubes 3/4" O.D. x 16 Ga. Steel

Shell weight of 8' 0" = 6,810
Add 8' 0" @ 210# = 1,680#
Tubes 1840 sq.ft. x 5.094 x .534 = 5,000
13,490 Exchanger Shipping W