A STUDY ON THE PERFORMANCE OF A 1-2 PARALLEL COUNTERFLOW OIL HEAT EXCHANGER

by

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B.S. in M.E., Robert College, 1967

Submitted to the Faculty of Graduate School

of Engineering in partial fulfillment of

the requirements for the degree of

Master of Science

in

Mechanical Engineering

Robert College

1969

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ACKNOWLEDGMENTS

I would like to express my gratitude to Assistant Professor A. Tezel for his guidance and patience as an advisor, to Mechanical Engineering Department for supplying the necessary material and funds, and to Ahmet Özköseoğlu ve Ort. Koll. Şti. for giving the heat exchanger used in the tests. Thanks are also due to T. Gaspar of power laboratory for his help in setting up the experimental assembly.

Bülent Özsoy

NOTATIONS

Q	=	heat flow
U	=	overall coefficient of heat transfer, Btu/hrft$^{2\,o}$F
Δt	=	temperature difference
Δt_2, Δt_1	=	hot and cold terminal temperature difference
t_s	=	temperature of condensing steam, ^{o}F
t_I, t_{II}	=	temperatures in first and second passes
U_h	=	overall coefficient of heat transfer based on the hot side area, Btu/hrft$^{2\,o}$F
A_h	=	hot side area, ft^2
Δt_o	=	temperature difference between the hot and cold fluids, ^{o}F
k_f	=	thermal conductivity of condensate at t_f
λ	=	enthalpy change, latent heat of condensation at the saturation temperature
, ρ_f	=	density, density of condensate film
g	=	gravitational constant, ft/sec^2
D	=	diameter of tubes
μ_f	=	viscosity of condensate film
Δt_f	=	temperature difference between the condensing vapor saturation temperature and the surface temperature
N	=	number of tubes

I. INTRODUCTION

A. Background of the Problem

Heat exchange between fluids in shell and tube type heat transfer equipment is one of the most important processes used in engineering. Petroleum refineries, chemical plants, condensing units, and fuel-oil burning systems are a few examples of places of its application in industry. Thermal design of these equipment requires an iterative procedure. First, an overall coefficient of heat transfer (U) is assumed and the heat exchanging area needed to fulfill the heat load with this U is calculated. Then the mechanical configuration thus obtained is checked with appropriate equations to see whether it shall give the U assumed at the beginning. The procedure is repeated until an unchanging value of U is obtained. Therefore, it is clear that a knowledge about the overall as well as individual coefficients of heat transfer for exchangers used in various process conditions is essential for a reliable thermal design.

Furthermore, after the exchangers are designed and manufactured their performance has to be tested and analyzed both theoretically and experimentally. Conditions prevailing at the time of design may have changed or an exchanger may not be functioning properly. Manufacturers of heat transfer equipmet have to perform sample tests on their products in order to see whether they function properly or not under their design conditions. Much useful information pertaining to mechanical and thermal design of heat exchangers can thus be obtained from such a detailed analysis.

B. Review of Literature

One of the first attempts to formulate a theoretical solution of the temperature profiles for laminar-flow heat transfer in tubes whose walls are maintained at a constant temperature is due to Graetz in 1885. Twenty-five years later in 1910 Nusselt rediscovered the mathematical solutions of Graetz and used them to calculate convective heat transfer coefficients (h) for the flow conditions mentioned above. Graetz's mathematical solution and also some related papers have been reviewed in detail by Jacob[1, p. 451]* and Drew.[2] It is interesting to note here that, in the solution of the aforementioned problem the following assumptions have been made:

(a) the fluid properties are constant;

(b) the laminar parabolic velocity profile is assumed to be established throughout the section in consideration;

(c) at $x = 0$ the temperature of the tube wall changes from T to T_w and is uniform at this value for $x > 0$. T is the fluid temperature as it enters the heating or cooling section.

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*Paranthetical references placed superior to the line of text refer to the bibliography.

Sellars, Tribus, and Klein[3] extended the classical Graetz solution by considering boundary conditions other than constant wall temperature, namely, constant-wall-heat-flux and linear wall temperature. In order to take into account the non-parabolic velocity distribution in the entrance regions of tubes Kay[4] used the laminar velocity profile developed by Langhaar[5] and obtained numerical solutions of energy equations under the following boundary conditions:

(a) constant wall temperature;

(b) constant heat input;

(c) constant temperature difference.

Hausen[6, p. 371] utilizing the Graetz solution suggested the semi-empirical relation for constant wall temperature and parabolic velocity distribution of fluids in laminar flow in tubes given below:

$$Nu_m = 3.66 + \frac{0.0668\left(\frac{D}{x}Pe\right)}{1 + 0.04\left(\frac{D}{x}Pe\right)^{2/3}} \qquad (1)$$

where $Nu_m = \frac{h_m D}{k}$ and $Pe = \frac{\varrho C_p V D}{k}$

are the mean Nusselt number, Nu_m, and Peclet number, Pe, respectively. Mean Nusselt number used here is defined with respect to mean convective heat transfer coefficient, h_m. D is the inside diameter of tube, ϱ density, C_p specific heat, V velocity, k thermal conductivity and h_m mean convective heat transfer coefficient defined over the length of pipe $0 < x < L$. This equation represents a partial modification of previous theoretical solutions in order to take into account the deviations from the assumptions listed above.

Many writers (e.g. Chapman,[7, p. 307] Knudsen and Katz,[5, p. 376] Jacob[1, p. 544]) point out that the theoretical results for temperature distribution for laminar flow in tubes do not represent very well the observed convective heat transfer coefficients. It is also pointed out by these same writers that the departure is primarily due to the assumptions made in deriving the theoretical formulas. First, fluid properties were assumed to be independent of temperature and constant. This assumption is not valid for viscous fluids whose properties (e.g., viscosity, density, specific heat) exhibit a strong dependence on temperature (see Appendix C). Secondly, the thermal entrance length for the temperature profile of the fluid to develop in a pipe is a significant part of the total pipe length for fluids with high Prandtl numbers (i.e., viscous fluids). Therefore, the assumption of a very long pipe as used by Nusselt to formulate the classical Graetz solution is not actually realized in most of the heat transfer equipment used in industry. Thirdly, from an analysis of the experimental results for the convective heat transfer coefficients of Holden and White for laminar flow in heated tubes, Jacob[1, p. 544] shows that no cross-sectional velocity distribution whatsoever could have yielded these results if the flow is assumed to be parallel to the axis of the tube. He concludes therefore that the parabolic velocity profile is not valid and probably radial flow compovapor on the hot side by assuming that the vapor side resistance, R_h, does not remain constant during the runs but varies as a function of velocity according to the following equation:

$$R_h = R_{h_o} + 1/V^{0.8} \quad (3)$$

where, R_{ho} is a reference vapor side resistance at infinite water velocity, V. The justification for this modification was strictly empirical. They claim that the film coefficients they measured by this method are more exact because they have avoided the use of thermocouples, which introduces errors in temperature measurement by distorting the heat flux at the point where temperature is being measured.

Wilson plot is a very helpful method in determining the film coefficient of any heat transfer apparatus if the general behavior of the film coefficient of the fluid passing through its tubes is known. It has been employed by various investigators covering a wide range of fields. To mention a few Beatty and Katz[12] have used it in the measurement of film coefficients of vapors condensing on finned tubes, Gardner[13] has used it to measure the bond resistances of finned tubes.

No application of Wilson plot to laminar flow was found in the literature. However, McAdams[14, o.346] suggested that it can be used for laminar flow if the variation of overall coefficient of heat transfer was taken to be proportional to the one third power of fluid velocity.

II. STATEMENT OF THE PROBLEM

The first objective of this work is to obtain experimentally the overall as well as individual coefficients of convective heat transfer of a 1-2 parallel counterflow heat exchanger that is used as an oil heater in fuel-oil systems in industry. The coefficients obtained are tabulated and also presented in graphical form so that they can be used in the design of simular apparatus. The convective heat transfer coefficients are compared with those calculated from the Sieder-Tate correlation

The second objective is to investigate the possibility of the use of Wilson's graphical method to obtain the convective film coefficients of heat transfer for viscous oils in laminar flow.

Lastly the variation of the centerline fluid temperature along the length of the 1-2 parallel counterflow heat exchanger is studied. A theoretical solution for the variation of the bulk fluid temperature along the exchanger length is obtained by making certain simplifying assumptions. These two variations are compared.

II. GENERAL

A. Heat Exchangers

Any apparatus whose function is to transfer heat from one medium to another can be named as a heat exchanger. They can be named according to their function, the fluids they use, and their flow arrangement. A general classification is:

1. surface heat exchangers (recuperators);
2. regenerators;
3. cooling towers.

In the first and most customary type to which belongs the oil heat exchanger that is studied in this work, two fluids of different temperatures flow in spaces separated by a wall and they exchange heat by convection and conduction through the wall.

In the last two types the fluids which exchange heat occupy the same space, a channel with or without solid barriers. In the regenerator type the fluids are carried alternately through the same channel; in the cooling tower they are carried simultaneously.

The surface heat exchangers may further be classified according to their flow arrangements. Fluids may be in parallel, counter, or cross flow.

In the heat exchanger that is analyzed in this work the shell side fluid flows in one shell pass and the tube fluid in two passes. One of these two passes is the same direction as the shall fluid and the other is in the opposite direction, and thus the exchanger is named as 1-2 parallel counterflow. A schematic drawing of the exchanger is shown on the next page.

B. Mean Temperature Difference in Heat Exchanger Design

The thermal design of a heat exchanger means the determination of the parameter of the exchanger and its fluids (e.g., surface area, flow rates, pressure drops, etc.), so that it shall perform as desired. One of the problems encountered in design is the temperature difference to be used in heat flow equation. This difficulty is solved by introducing the "log mean temperature difference" (LMTD) derived for counterflow as (15, p. 88):

$$\Delta t = \frac{\Delta t_2 - \Delta t_1}{\ln \Delta t_2 / \Delta t_1} = \text{LMTD} \tag{4}$$

where, Δt_2 and Δt_1 are the hot and cold terminal temperature differences respectively. Although LMTD is strictly correct for counterflow, for other flow arrangements correction factors (F_T) are used in the following form:

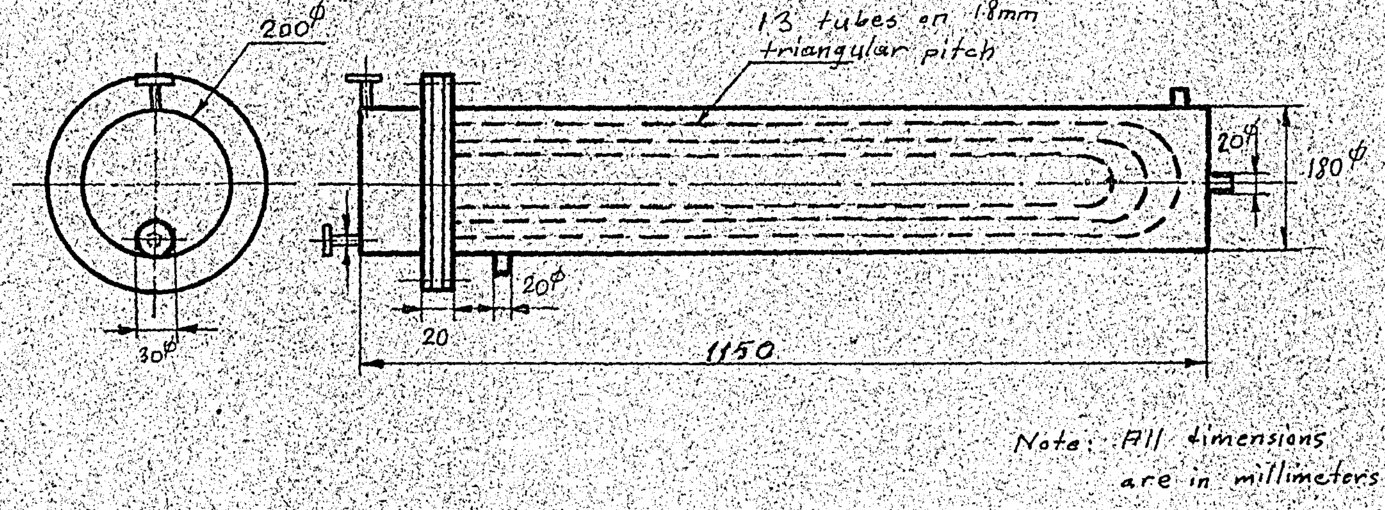


Fig. 1. 1-2 Parallel-Counterflow Heat Exchanger

$$\Delta t_c = F_T \Delta t_m \quad (5)$$

Kern(15, p. 828) tabulates these correction factors to be used for different flow arrangements. It should be mentioned however that for an exchanger which has an isothermally condensing fluid such as steam in its shell side F_T is equal to unity.(15, p. 92)

In the derivations(15, p. 89) of the above mentioned LMTD and F_T, the overall coefficient of heat transfer is assumed to remain constant along the exchanger length. As stated by the various investigators mentioned in section I. - B. this assumption does not hold true for viscous fluids whose properties show a strong dependence on temperature. As the fluid travels along the length of the exchanger it is heated and its viscosity decreases considerably (see Appendix C). According to Jacob(1, p. 460) this has the effect of flattening the velocity profile with respect to isothermal flow as shown in Figure 2 on the next page.

Overall coefficient can be expressed as a function of individual coefficients of heat transfer on the oil and steam sides as follows:

$$1/U = 1/h_o - \Delta x/k - 1/h_i \quad (6)$$

It is seen from equation (6) that U is a function of the oil side film coefficient h_i, and by examining equation (2) we can say that h_i in turn is a function of Reynolds and Prandtl numbers, and dynamic viscosity. Since Reynolds and Prandtl numbers are composed of terms which are temperature dependent, the temperature at which these dimensionless numbers are calculated causes them to assume different values. Therefore, U is to be expected to vary along the length of the exchanger where the temperature of the fluid increases from say t_1 to t_2.

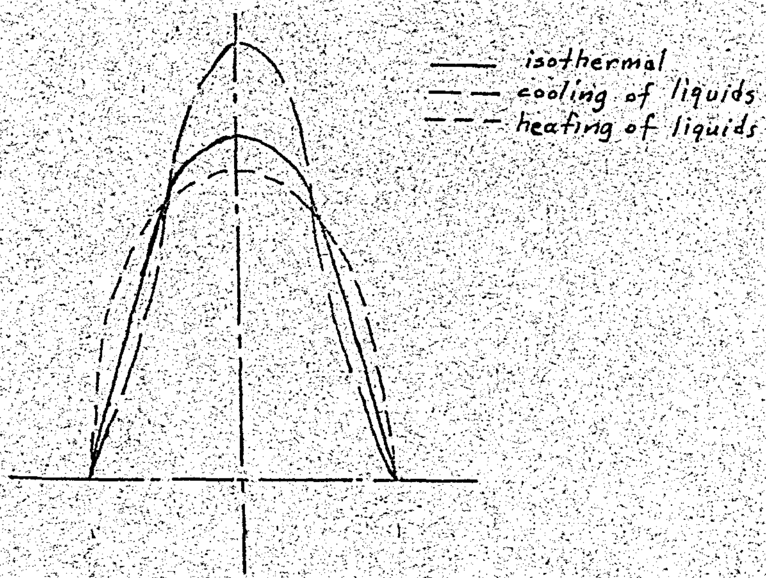


Fig. 2. Temperature profiles

From another point of view it can be argued that as the viscosity of the fluid near the tube wall is decreased while a viscous fluid is being heated in a tube, the velocity near the tube wall increases with respect to the entrance velocity and the velocity profile takes the form shown in Fig. 2. Thus, radial components of velocity may have formed increasing the heat transfer coefficients as temperature increases.

The usual procedure is to evaluate the fluid properties at the arithmetic average of inlet and outlet temperatures. Error in the order of ten per cent[9, p. 94] is introduced in the heat transfer coefficients calculated with this procedure. As pointed out earlier Colbourn assumes a linear variation of U with temperature and derives the following mean temperature difference to be used in design:

$$t_m = \frac{U_1 \Delta t_1 - U_2 \Delta t_2}{\ln U_1 \Delta t_1 / U_2 \Delta t_2} \qquad (7)$$

where U_1 and U_2 are the overall coefficients at the inlet and outlet of the exchanger respectively.

C. Theoretical Variation of Temperature along the Exchanger Length

By assuming the overall coefficient to remain constant along the exchanger length the temperature profiles are derived as (see Appendix B):

$$t_I = t_s - (t_s - t_i) \exp\left(-\frac{Ua''}{wc}x\right) \qquad (8)$$

$$t_{II} = t_s - (t_s - t_e) \exp\left(\frac{Ua''}{wc}x\right) \qquad (9)$$

where t_I and t_{II} are the bulk temperatures in the first and second passes, a'' is the heat exchanging area per linear feet, t_s, t_i and t_e are the steam, inlet, and exit temperatures respectively.

These temperature profiles are plotted in Figs. 12, 13 and 14,

D. Graphical Method of Wilson

From equation (6) we have:

$$\frac{1}{U} = \frac{1}{h_o} + \frac{x}{k} + \frac{1}{h_i}$$

If the individual terms in this expression are analyzed (see Appendix B) the result is:

$$\frac{1}{U} = C_1(\Delta t)^{1/4} + \frac{\Delta x}{k} + C_2 \frac{1}{w^{1/3}} \qquad (10)$$

where Δt is the change in condensate temperature of steam and C_1 and C_2 are constants. If we assume the changes in Δt due to the changes in oil velocity as negligible, the first two terms on the right hand side of equation (10) remain constant. The equation then takes the form:

$$\frac{1}{U} = A + \frac{C_2}{w^{1/3}} \qquad (11)$$

In a series of experiments we can vary the mass rate of flow G and measure the corresponding overall coefficients U. Then we can plot the above equation on graphical paper. As seen from equation (11) there is a linear variation between $1/U$ and $1/w^{1/3}$. The intercept of this straight line gives A and the slope is C_2. Additionally:

$$A - \Delta x/k = h_o \qquad (12)$$

$$C_2/w^{1/3} = h_i \qquad (13)$$

where h_o and h_i are the steam and oil side coefficients of heat transfer respectively.

IV. EXPERIMENTS

A. Experimental Set-up

A schematic drawing of the experimental set-up is shown in Fig. 3 on the next page and Fig. 4 is a photograph of it. The test exchanger is a 1 - 2 parallel counterflow shell and tube type and was obtained from Özköseoğlu Heat Industries Corporation which is a firm dealing with the design and application of fuel-oil burning systems. This exchanger is used in some of their applications to heat fuel-oil ASTM No. 6 in order to improve its pulverization characteristics which is rather crucial in oil-burners.

The oil pump used was a constant displacement type having a capacity of 1.5 m^3/hr (400 gal/hr) and pressure of 70 psig. The 50 m^3 (13000 gal) oil reservoir of Robert College heating system's boilers were used as the source and sink of oil. During the experimentation the thermostatic steam valve which had a temperature range of 20°C - 100°C was used to regulate the outlet temperature of oil. The steam line was insulated heavily with glass wool to prevent steam from condensing inside the pipe. A by-pass line was included from the return line of oil to the pump discharge in order to regulate the amount of oil flowing through the exchanger.

Inlet and outlet temperatures of both fluids were measured with

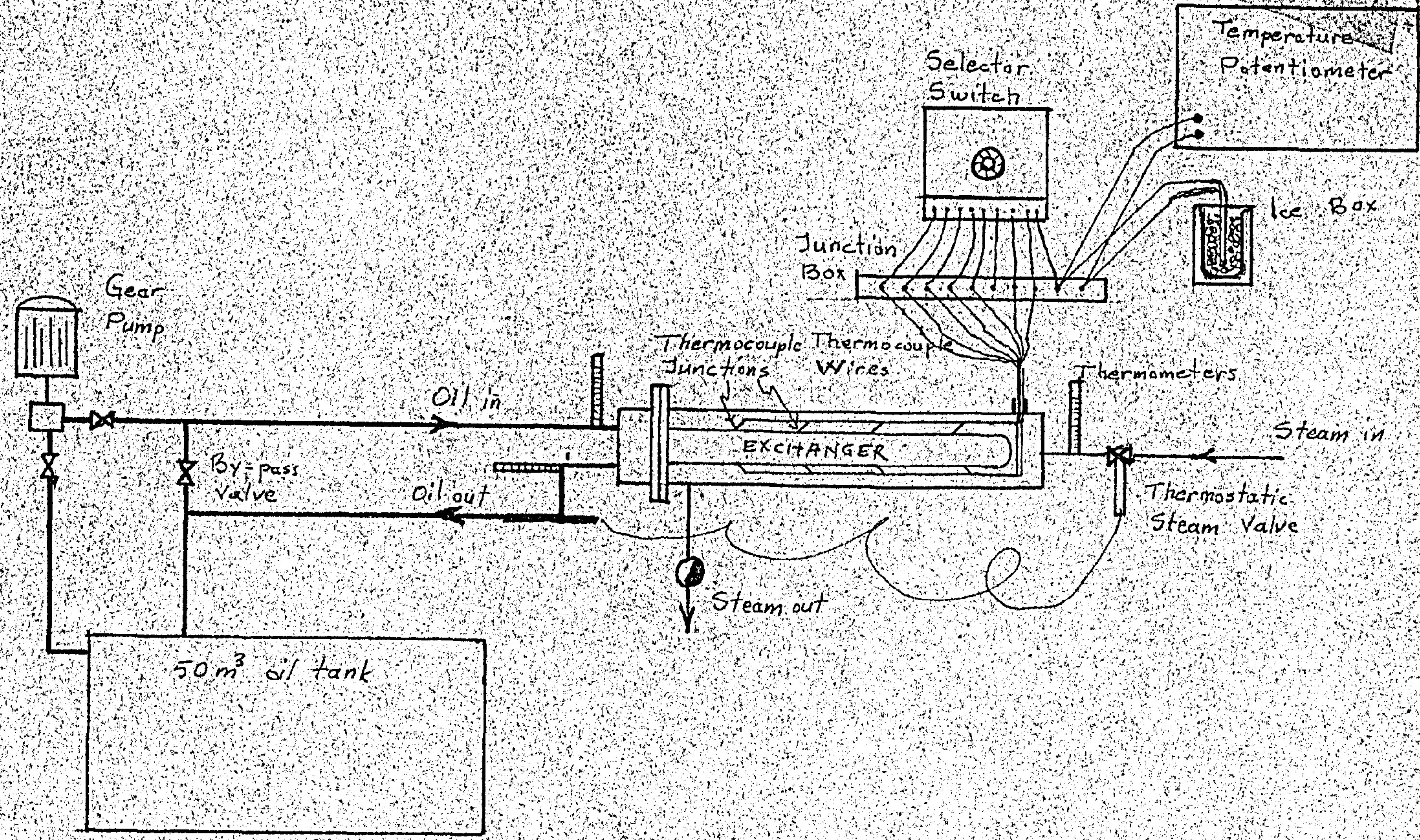


Fig. 3. Experimental set-up

normal mercury in-glass tube thermometers. Thermocouples were used to measure the tube wall and inside fluid temperatures. The mercury in-glass tube thermometers were calibrated by means of the thermocouple circuit.

Steam was obtained from the boilers of Robert College at a maximum pressure of 4 atu (60 psig). Steam trap used was of a vertical bucket type. The condensate was collected in a condensate tank and discharged periodically. Flow rate was measured by collecting the oil in a container during a measured time interval. These measurements were repeated two or three times and the avarage of them was used in the calculations.

1. Tube-wall Thermocouples

In order to measure the tube-wall temperatures thermocouple junctions embedded into the tube wall were used. Thermocouple wires were No. 24 B and W gage iron constantan. Eight thermocouple junctions spaced every twenty centimeter along the length of a tube were used. The junctions were beaded by a 20 V direct current electric battery. Grooves which were two mm wide and one mm deep were cut at the outer surface of the exchanger tube, and thermocouple junctions were placed into them. Grooves were oriented perpendicular to the axis of the tube so that thermocouple wires coming out of the metal would not act as fins conducting away heat since they pass through isothermal regions.

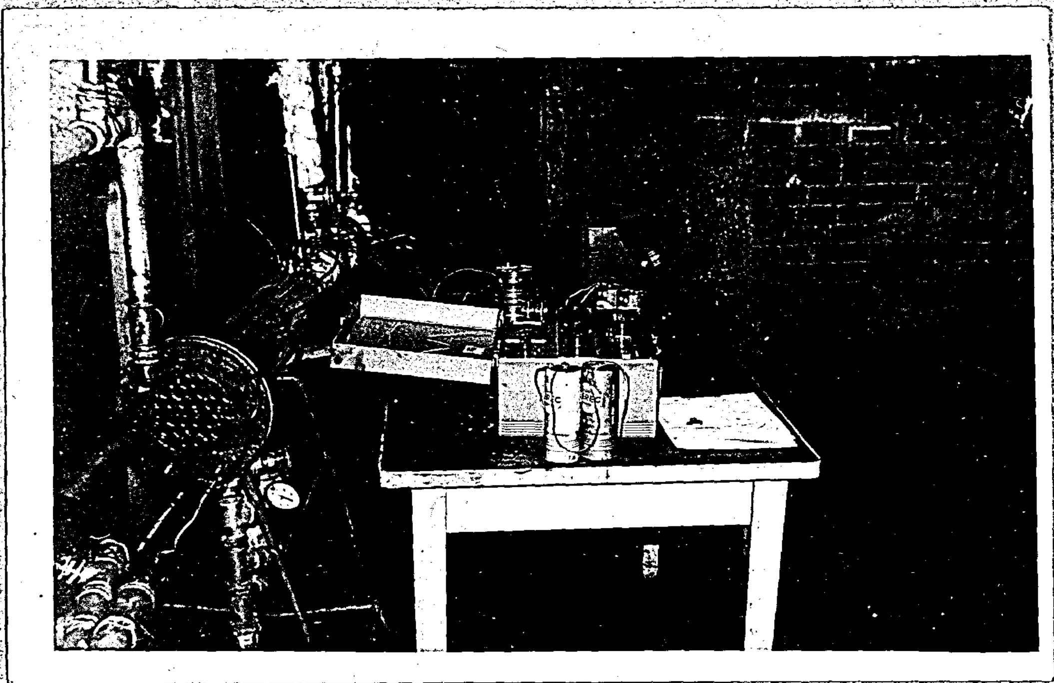


Fig. 4. View of experimental setup

Thermocouple junctions were cemented into the grooves by soldering them with lead-tin solder. The rest of the termocouple wires in the grooves were cemented to the tube wall by "Cement 404". The wires were then insulated with wax coated cotton protector in order to prevent steam condensation on them. The leads were brought out of the exchanger through a hole at one end of the exchanger.

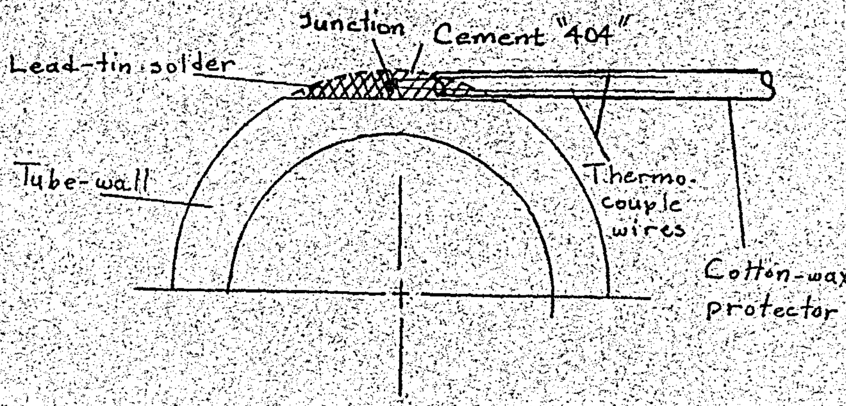


Fig. 5. Tube-wall thermocouple junctions

2. Oil-side Temperature Probes

To measure the centerline temperature of the oil flowing inside the tubes, probes, as shown in Figure 6 were inserted into the tube. The material of the probes was steel wires 1.5 millimeters in diameter. They were placed in a position to oppose the direction of flow so that no flow disturbances before the point where temperature measurements are taken is introduced.

A temperature potentiometer of Leeds and Northrup company was used. It had a range of 1600 millivolts. Its smallest divison was 10 microvolts. The selector switch was also of Leeds and Northrup and it

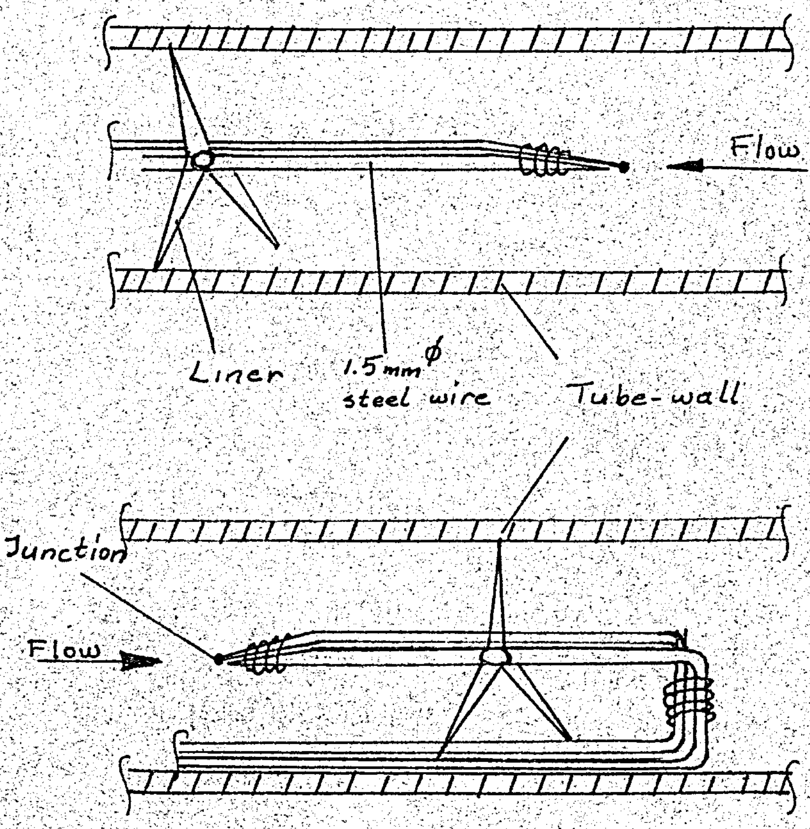


Fig. 6. Oil side temperature probes

had ten leads for transferring the circuit from one junction to another. An ice point and a junction box were the other components of the thermocouple circuit. The junction box was insulated with glass wool in order to keep the junctions at a uniform temperature. The wiring diagram of the thermocouple system is shown on the the next page in Fig. 7.

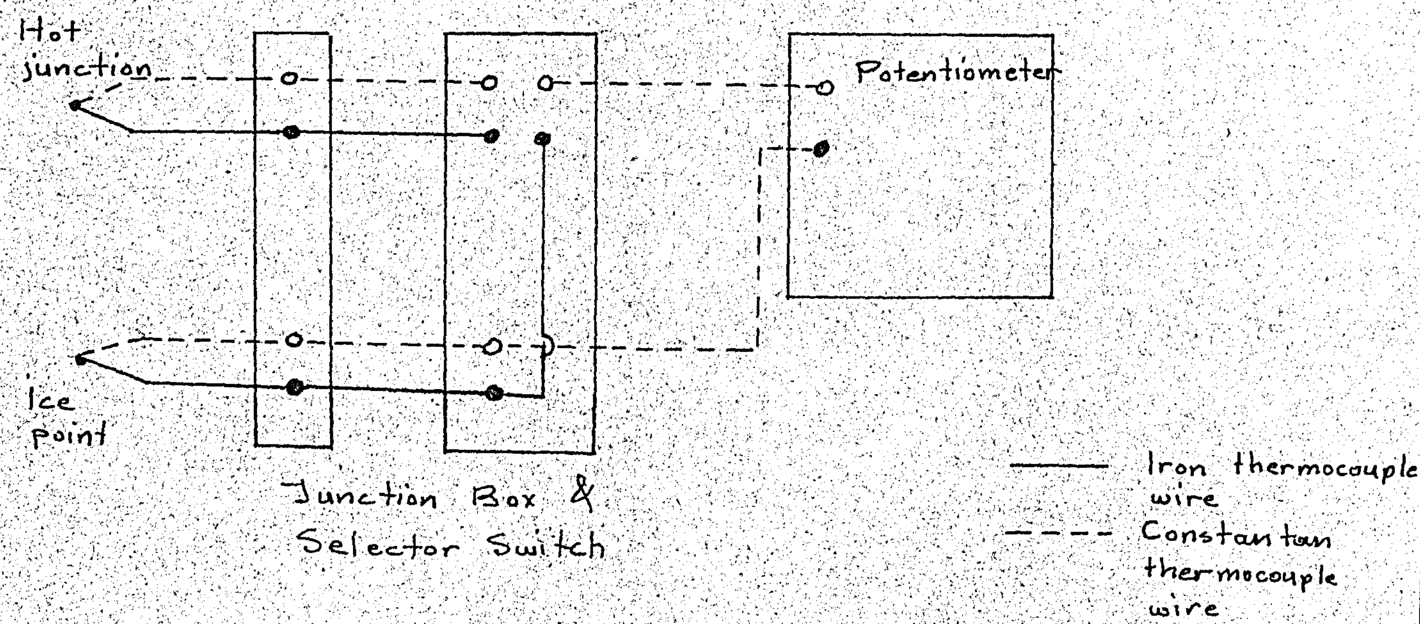


Fig. 7. Thermocouple circuit wiring diagram

B. Procedure and Methods Used

The experiment for the measurement of heat transfer coefficients and oil side centerline temperatures is started by circulating the oil through the exchanger while saturated steam is condensing on the outside of tubes. One-half to one hour is allowed to elapse so that the oil and steam inlet and outlet temperatures reach a stable value. Then tube-wall temperatures at eight junctions and oil-side centerline temperatures at two locations which are measured by means of the thermocouple system are recorded. After a short interval of time (usually of three minutes) these readings are checked again to see if there is any variation in the emf of individual junctions. If the emf readings of any one junction increases or decreases by an amount greater than 10 microvolts,(which corresponds to a $\sim$ 0.2 °F change in temperature) the thermocouple measurements are repeated again in intervals of three minutes until a steady value is obtained. If a steady state value cannot be obtained, sources for this unsteadiness is sought and the experiment is started again after correcting it. The tube wall temperature used in the calculations of heat transfer coefficients is the arithmetic average of the last eight junction readings taken. After the thermocouple measurements are taken the inlet and outlet temperatures of oil and steam are noted and the oil flow rate is measured in the manner described in section IV-A.

At the same time the oil centerline temperaturesare recorded and the probes are moved to a new position and the same procedure is repeated again for the same flow conditions. For new set of readings the oil mass flow rate is varied by adjusting the oil by-pass value from the return line to the pump discharge. The other variables in the runs were the inlet temperatures of steam and oil. The oil inlet temperature varied from 91 to 96 °F and the steam inlet temperature was varied in the interval 219 °F to 263 °F.

C. Results

The results of the experiments are presented in table and graph form. Table 9 gives the data taken during the experiments and the overall (U) and film (h) coefficients of heat transfer calculated from these values. The range of Reynolds number in different runs made with the same oil (ASTM No. 6 Bunker C) is from 58 to 478. So the flow is in the laminar region for all of the experiments. Table 10 lists the oil side centerline temperatures obtained from the readings of the thermocouple probes spaced every 20 centimeter except the return bend in the exchanger tube. The variation of the centerline temperatures on the oil side are plotted as a function of the exchanger tube length in Figures 12, 13, and 14, each figure corresponding to a different Reynolds number. The variation of the bulk temperature of oil along

TABLE 9

Experimental Data and Results

Run No.	t_1	t_2	t_s	w	$(t_w)_{ave}$	t_i	t_m	Q	h_{exp}	Re	Pr	U
-	^oF	^oF	^oF	lbs/hr	^oF	^oF	^oF	Btu/hr	$\frac{Btu}{hr ft^2 {}^oF}$	-	-	$\frac{Btu}{ft^2 hr {}^oF}$
2	93.2	140.0	219.3	75.0	213.4	94.8	100.1	1595	19.8	156	105	18.6
3	91.4	151.0	218.3	43.8	213.3	88.9	95.4	1193	15.7	91	105	14.7
4	91.4	154.3	219.0	41.0	213.4	88.9	96.5	1188	15.5	96.5	105	14.5
5	93.2	117.5	253.5	242.0	246.4	140.0	147.0	2650	22.1	475	116	21.2
7	95.0	149.0	251.6	81.3	246.7	123.0	127.0	2000	18.9	186	105	18.5
8	96.8	163.2	255.2	51.3	252.6	120.0	121.1	1570	15.3	96.5	95	15.1
9	96.8	178.0	252.5	32.5	239.3	96.5	110.0	1240	15.0	88.2	87	13.3
10	93.2	116.6	239.0	238.0	230.2	127.1	134.5	2290	21.0	467	116	20.0
11	93.2	131.0	237.0	115.5	239.8	127.5	125.0	1980	18.1	271	106	18.6
12	95.0	166.0	242.8	42.5	238.8	104.2	109.0	1410	15.8	107	95	15.2
13	95.0	172.5	248.0	34.7	242.8	104.2	109.2	1260	14.1	90.6	87	13.6
14	95.0	178.0	250.0	29.0	245.6	113.2	107.0	1123	9.9	78.6	87	12.3
15	95.0	181.4	251.6	21.2	247.7	104.0	107.3	858	9.6	58.5	87	9.4
16	93.2	118.5	263.4	237.0	255.6	153.0	160.2	2718	20.7	478	111	19.9
17	93.2	147.0	263.0	84.3	259.8	137.5	141.9	1959	16.7	198	98.5	16.0
18	95.0	172.5	263.4	45.3	258.0	123.5	124.5	1625	15.4	123	87	15.3
19	96.8	185.0	263.4	32.8	259.1	112.5	116.5	1340	13.9	96.4	87	13.6
20	96.8	183.0	253.4	27.8	250.1	104.1	108.0	1098	12.3	78.5	87	11.8
21	93.2	121.0	265.0	240.0	259.4	152.5	160.0	3000	22.9	471	116	22.1
22	95.0	145.5	265.0	112.0	260.2	138.5	143.4	1613	22.0	247	105	21.3

(Continued)

TABLE 9 (Continued)

Run No.	t_1	t_2	t_s	w	t_w	t_i	t_m	Q	h_{exp}	Re	Pr	U
--	^oF	^oF	^oF	lbs/hr	^oF	^oF	^oF	$\frac{Btu}{hr}$	$\frac{Btu}{hrft^2 {}^oF}$	—	-	$\frac{Btu}{hrft^2 {}^oF}$
23	96.8	178.0	263.4	45.2	259.0	117.0	122.0	1718	17.2	122	87	16.4
24	97.6	182.0	263.4	34.0	260.5	97.0	103.0	1293	15.6	96	87	13.7
25	96.0	125.5	262.0	236.0	254.1	142.5	148.2	3200	26.2	476	113	25.4
26	95.0	170.6	263.4	46.5	260.2	123.5	126.2	1635	15.4	117	92	15.1
27	95.0	185.0	263.0	34.3	256.0	109.5	115.0	1350	14.4	96.6	87	13.8
28	95.0	187.0	257.0	28.5	253.8	106.0	109.5	1225	13.5	80.5	87	13.1

t_1 = oil inlet temperature
t_2 = oil outlet temperature
t_s = steam condensation temperature
w = weight flow rate of oil
t_w = tube wall temperature averaged from eight thermocouples
t_i = log mean temperature difference between the tube wall and oil
t_m = log mean temperature difference between steam and oil
Q = heat transferred from steam to oil ($Q = wC_p(t_2 - t_1)$)
h_{exp} = experimental film coefficient of heat transfer on the oil side
Re = Reynolds number
Pr = Prandtl number
U = overall coefficient of heat transfer

TABLE 10

Oil Temperatures

Run No.	t_1	t_2	t_s	w	t_I x = 75	t_{II} x = 15	Re
16	93.2	120.5	263.4	237	102	116.6	478
18	95.0	172.5	263.4	45.3	117.5	171.0	123
19	96.8	185.0	263.4	32.8	120.5	174.5	96.4

Run No.	t_1	t_2	t_s	w	t_I x = 15	t_{II} x = 75	Re
21	93.2	121.0	265.0	240	95.0	109.9	471
23	96.8	178.0	263.4	45.2	98.3	165.0	122
24	97.6	182.0	263.4	34	98.8	142.1	96

Run. No.	t_1	t_2	t_s	w	t_I x = 55	t_{II} x = 55	Re
25	95.0	125.5	262.0	236	96.5	112.0	476
26	95.0	176.6	263.4	46.5	106.9	145.0	117
27	95.0	185.0	263.0	34.3	109.0	151.8	96.6

Run. No.	t_1	t_2	t_s	w	t_I x = 35	t_{II} x = 35	Re
29	93.0	120.3	264.0	240	95.1	113	483
31	95.0	175.0	263.0	47.7	102.2	154	129
32	95.0	185.1	265.0	32.5	103.1	161	97

TABLE 11

Run No.	Re	U	h_{exp}	h_{theo}	t_s
2	156	18.6	19.8	16.6	219.3
3	91	14.7	15.7	14.3	218.3
4	96.5	14.5	15.5	14.4	219.0
6	475	21.2	22.1	26.0	253.5
7	186	18.5	18.9	18.1	251.6
8	96.5	15.1	15.3	14.1	255.2
9	88.2	13.3	15.0	13.2	252.5
10	467	20.0	21.0	25.4	239.0
11	271	18.6	18.1	20.2	237.0
12	107	15.2	15.8	14.3	242.8
13	90.6	13.6	14.1	13.3	248.0
14	78.6	12.3	9.9	12.7	250.0
15	58.5	9.4	9.6	11.4	251.6
16	478	19.9	20.7	25.0	263.4
17	198	16.0	16.7	17.4	263.0
18	123	15.3	15.4	14.2	263.4
19	96.4	13.6	13.9	13.0	263.4
20	78.5	11.8	12.3	12.4	253.4
21	471	22.1	22.9	24.4	265.0
22	247	21.3	22.0	19.8	265.0
23	122	16.4	17.2	15.0	263.4
24	96	13.7	15.6	13.4	263.4
25	476	26.4	25.4	25.9	262.0
26	117	15.1	15.4	15.2	263.4
27	96.6	13.8	14.4	13.9	263.0
28	80.5	13.1	13.5	13.1	257.0

h_{th} = film coefficient of heat transfer calculated from the Sider-Tate correlation.

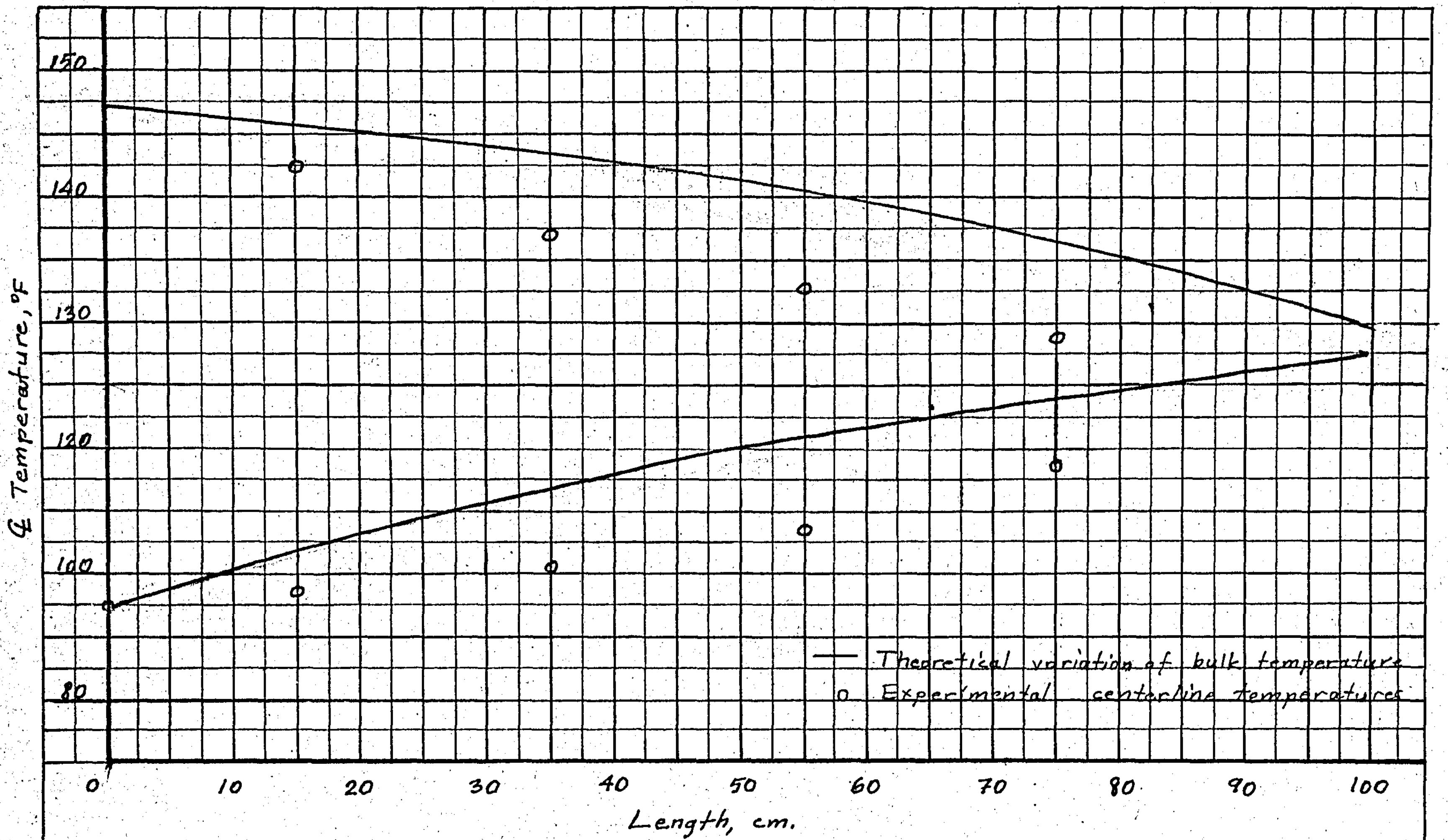


Fig. 12. Oil-side centerline temperature v.s. exchanger length for Re = 123

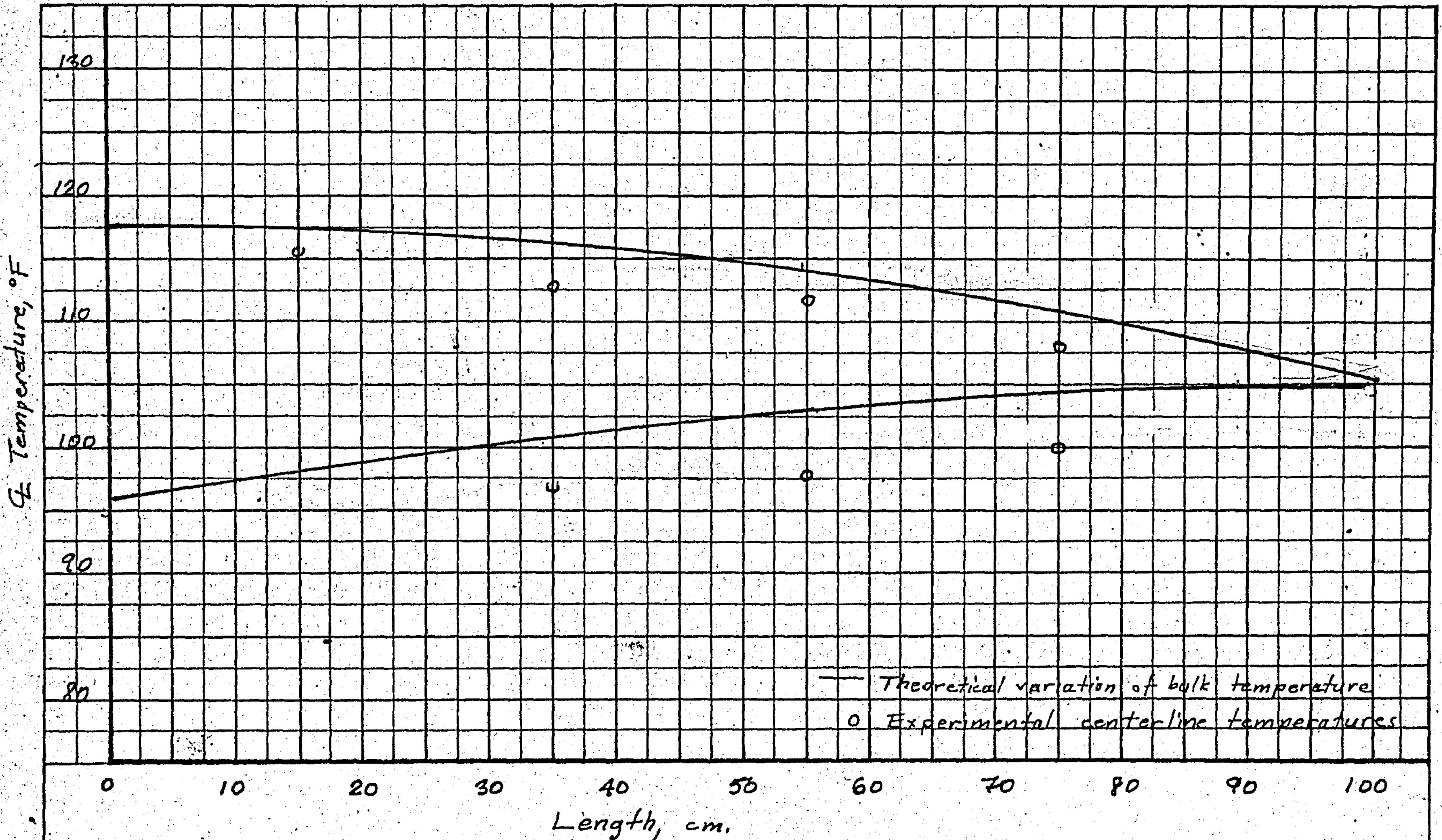


Fig. 13. Oil centerline temperature v.s. exchanger length for Re = 477

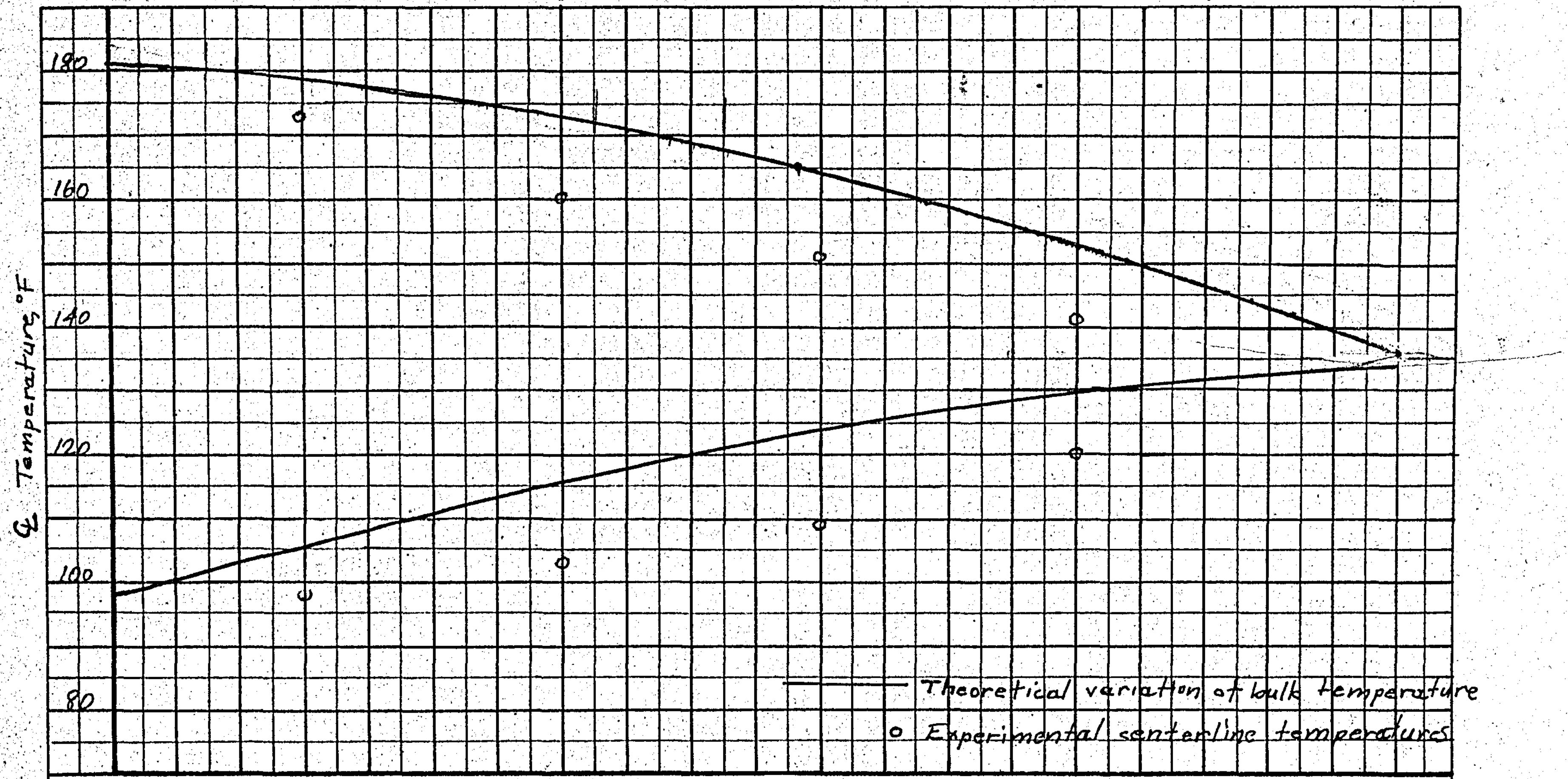


Fig. 14. Oil centerline temperature v.s. exchanger length for Re = 96.5

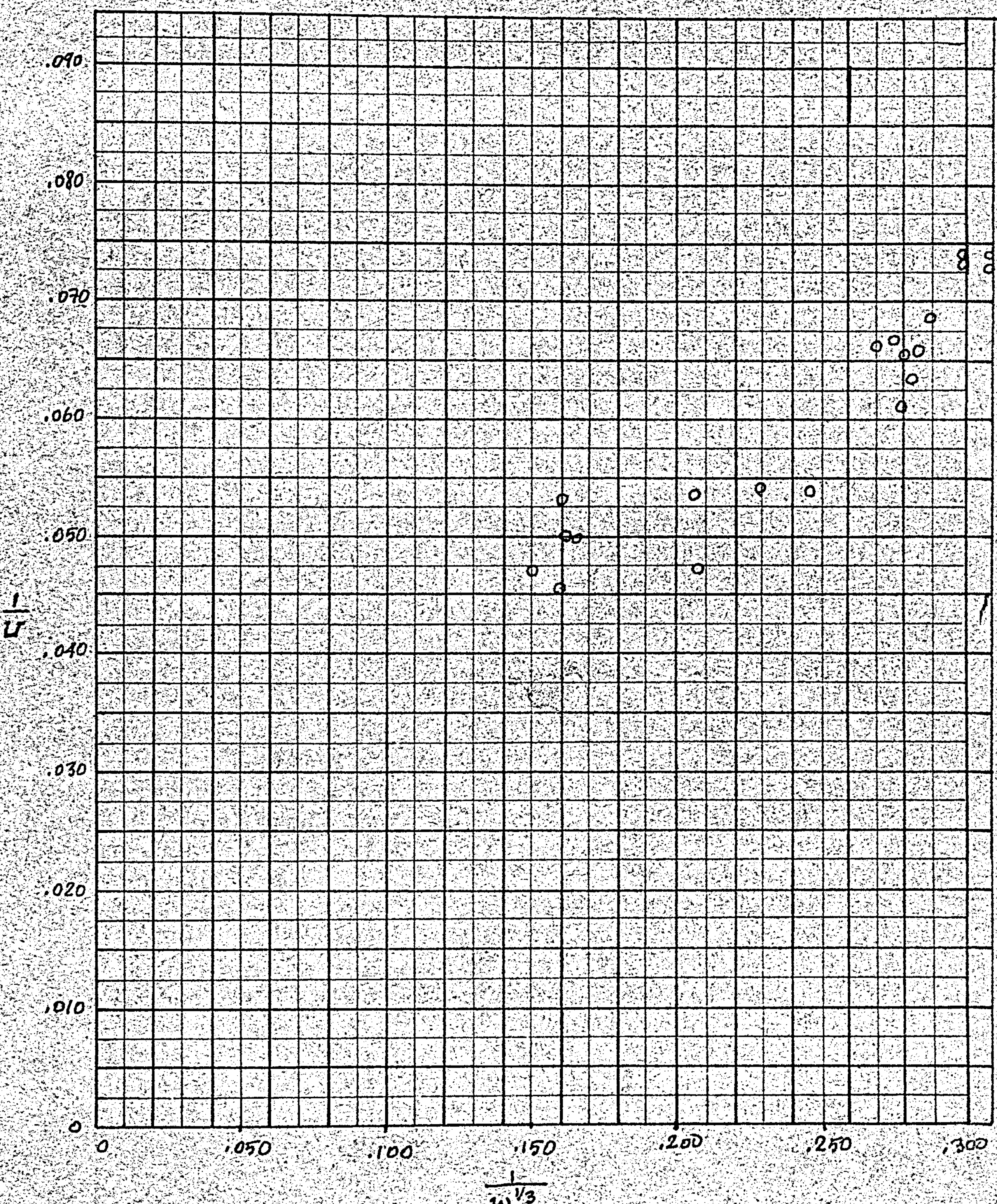
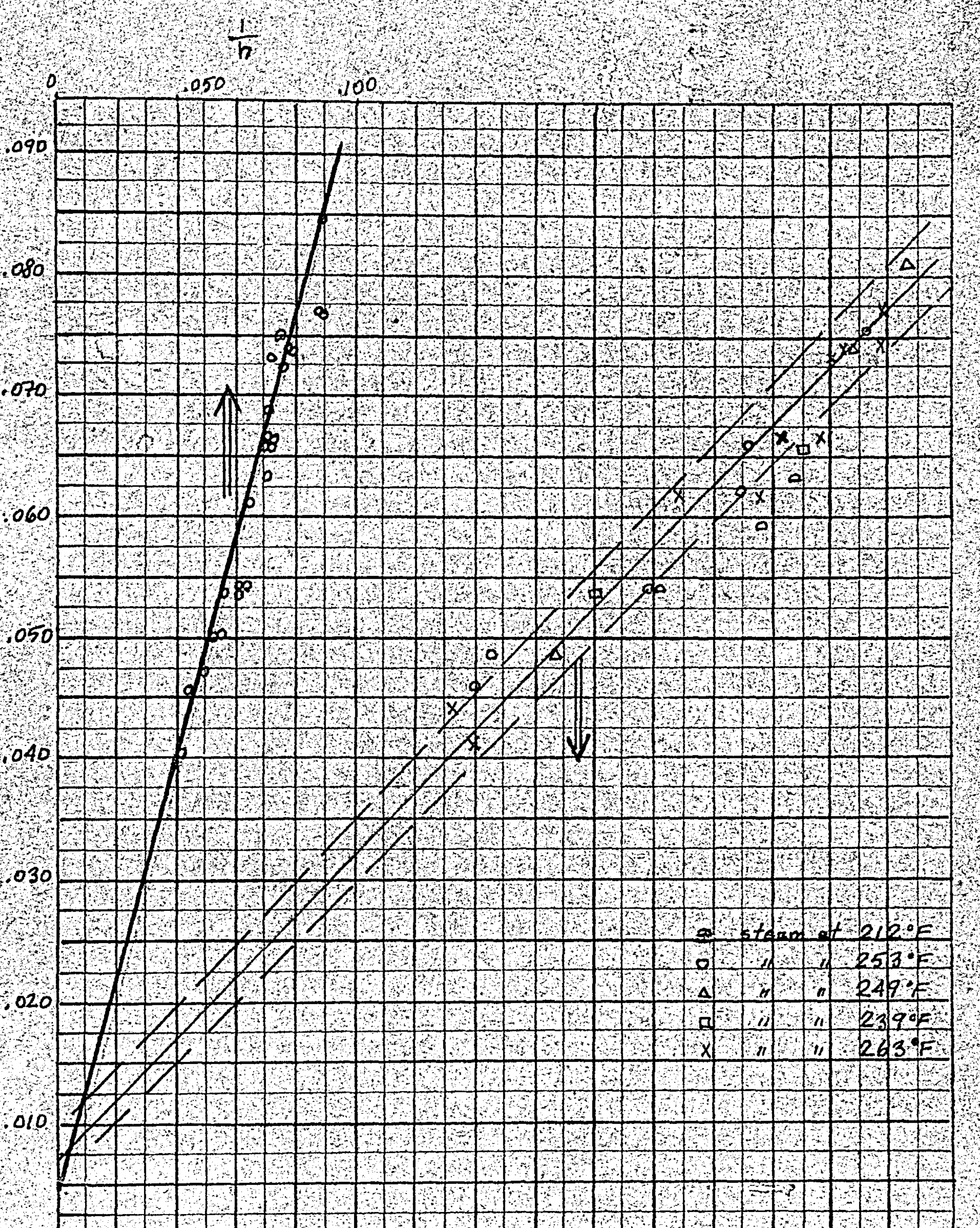


Fig. 15. Wilson Plot



$$\frac{1}{w^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}}$$

Fig. 16. Wilson Plot

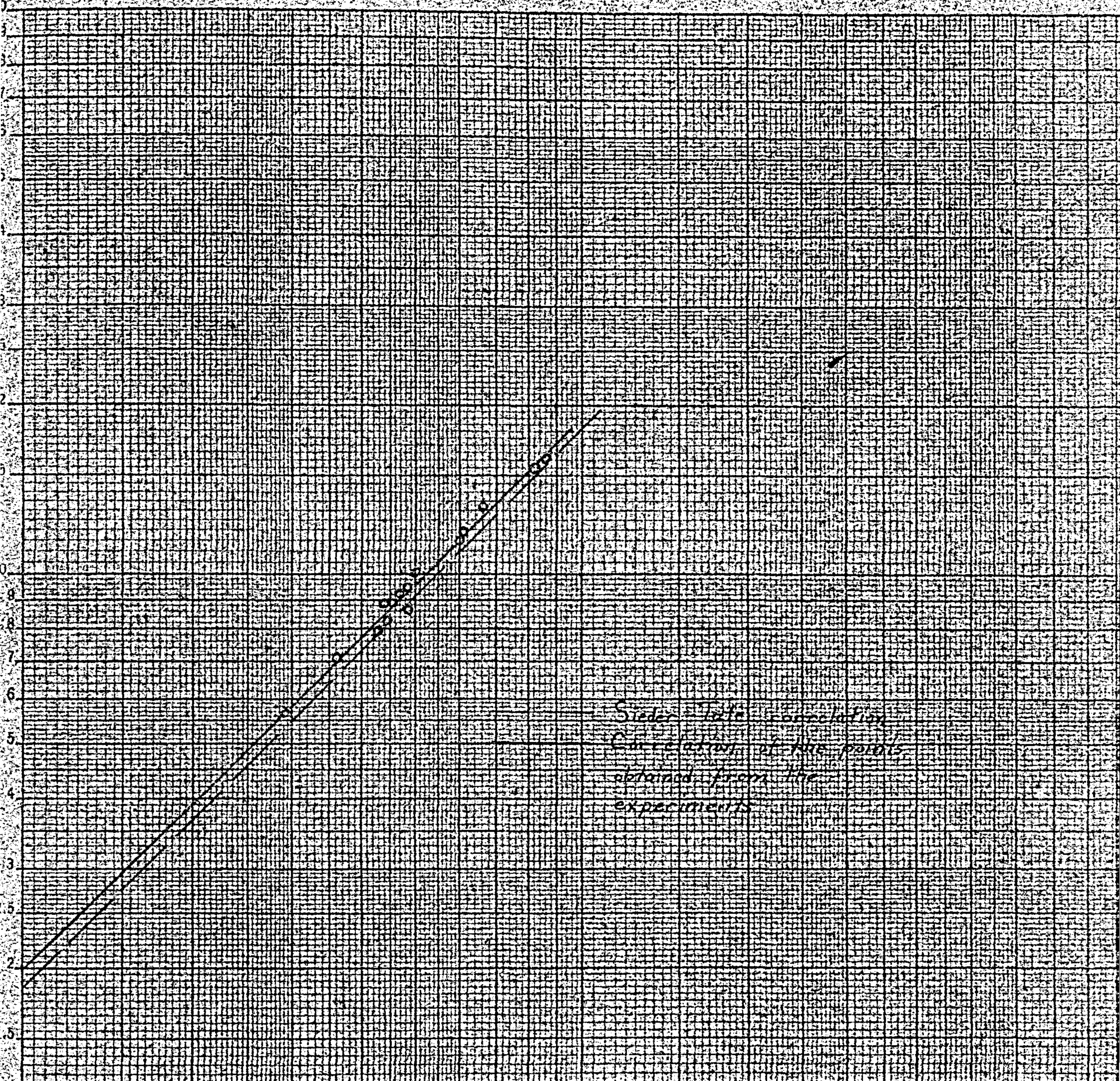


Fig. 17. Correlation of film coefficients of heat transfer

the exchanger length as obtained from equations (8) and (9) are also plotted in these same graphs. The film coefficients of heat transfer calculated by using the Sieder-Tate correlation (see equation number (2)) are tabulated in Table 11. In the same table is also included the values of h obtained from experiments. These two values of h are presented in graph form by plotting Nusselt number v.s. $(Re . Pr . \frac{D}{L})^{1/3} \times (\mu / \mu_w)^{0.14}$ in Figure 17.

The Wilson plots of the experiments are shown in Figures 15 and 16. There are three groups of points on this graph. 1/U is plotted first with $1/w^{1/3}$, then with $1/w^{1/3}(\mu / \mu_w)^{0.14}$ and lastly with 1/h, where the h's are the experimental values of film coefficients of heat transfer.

D. Accuracy and Errors

No attempt is made to analyze rigorously for all possible sources of errors, but rather to estimate the magnitude of the more important ones and use these for arriving at an overall estimate of the accuracy of the results. For the Wilson plot, however, a detailed analysis is attempted as will be shown at the end of this section.

For the first case, the errors considered are as follows:

1. Instrument inaccuracies;
2. Errors introduced by idealizations;
3. Errors inherent in the design and performance of the system.

In the first group is included the temperature and flow rate measurement inaccuracies. The inlet and outlet thermometers for oil and steam had a minimum graduation of 1°F. Their minimum reading was 95°F. Therefore the maximum error in temperature measurement could be 1/95 or 1 per cent. The maximum accuracy of the thermocouple system was ± 0.2°F since differences of 10 microvolts could be read from the potentiometer and 50 microvolts correspond to 1°F. Because the minimum reading of the thermocouple system was 120°F the maximum error from its readings is 0.2/120 or .00167 which is negligible. Flow rate measurements were done by taking an average of the two or three values taken for the same flow conditions. An analysis of the differences in measurement of flow rates for the same run indicates an error on the average of 4.5 per cent.

Some of the idealizations introduced into the calculations are:

1. Steam is assumed to condense uniformly all through the shell of the exchanger;
2. the exchanger is assumed to be clean, i.e., not fouled.
3. Dimensional measurements are uniform.

Errors inherent in the design and performance of the system include the turbulence experienced by the oil in entering and leaving the exchanger, and the fluctuations in steam pressure and steam trap condensate level.

On the basis of the above considerations and following the suggestions of Kays,[16] an estimate of the possible sources of error is:

Flow measurement	4.5 per cent
Temperature measurement	1.0 per cent
Errors inherent in design	1.0 per cent
Errors due to idealizations	0.5 per cent
Other sources of error	0.5 per cent
Total	7.5 per cent

An error of 7.5 per cent is estimated.

An error analysis for the Wilson plot should take into account the errors in the values of the individual terms in the graph shown in Fig. 15. There are three different absissa's on the same graph, namely, $1/w^{1/3}$, $1/w^{1/3}(\mu/\mu_w)^{0.14}$ and $1/h$. The ordinate of the graph $1/U$ in all of these cases.

The error in the overall coefficient U is analyzed by utilizing the definition of U in the following form:

$$(U + \epsilon U) = \frac{(w + \epsilon_1 w)(C_p + \epsilon_2 C_p)(\Delta t + \epsilon_3 \Delta t)}{\ln(\Delta t_1 + \epsilon_3 \Delta t_1)/\Delta t_2 + \epsilon_3 \Delta t_2)} \quad (14)$$

where ϵ's are the errors for each quantity. From the previous analysis of this section, $\epsilon_1 = 0.045$ and $\epsilon_3 = 0.01$. It is seen from Figure 20 in Appendix C that an error of 1^oF in temperature reading causes an error of 0.5 per cent in the readings of specific heat. Therefore ϵ_2 is equal to 0.005. Substitution of these values in the above equation gives a value of 5.6 per cent for the error coefficient of U.

A similar analysis for $1/w^{1/3}$, $1/w^{1/3}(\mu/\mu_w)^{0.14}$ and $1/h$ by utilizing their definitions gives, for $1/w^{1/3}$:

$$\text{Error} = \frac{1}{w^{1/3}} - \frac{1}{(w+\epsilon_1 w)^{1/3}} = \frac{1}{w^{1/3}}\left(1 - \frac{1}{(1+\epsilon_1)^{1/3}}\right) \qquad (15)$$

Since $\epsilon_1 = 0.045$

Error $= 0.017 \times 1/w^{1/3}$

For $1/w^{1/3}(\mu/\mu_w)^{0.14}$:

$$\text{Error} = \frac{1}{w^{1/3}(\mu/\mu_w)^{0.14}} - \frac{(\mu_w + \mu_w\epsilon_4)^{0.14}}{(w+\epsilon_1 w)^{1/3}(\mu + \mu\epsilon_4')^{0.14}} \qquad (16)$$

From Figure 18 in Appendix C it can be seen that a 1°F error in temperature reading causes an error of 0.5/10 or 5 percent error at high temperatures (150 - 250°F) and 1.0/30 or 3.3 per cent error at low temperatures. Since μ_w is always between 200 and 250°F $\epsilon_4 = -0.05$ and $\epsilon_4' = -0.033$. Substituting these values into equation (16) gives:

$$\text{Error} = \frac{1}{w^{1/3}(\mu/\mu_w)^{0.14}}\left(1 - \frac{(1.05)^{0.14}}{(0.955)^{1/3}(0.962)^{0.14}}\right) \qquad (17)$$

$$\text{Error} = \frac{1}{w^{1/3}(\mu/\mu_w)^{0.14}} \times (0.036)$$

These values of errors are used in the Wilson plot in figures 15 and 16 to mark the limits of experimental accuracy.

V. DISCUSSION AND CONCLUSIONS

The overall coefficients obtained from the experiments ranged from 9.4 to 25.4 Btu/hrft^2 ^{o}F in the Reynolds number range of 58 to 478. The oil side film coefficients of heat transfer obtained from the Sieder-Tate correlation have on the average a 7 per cent deviation from the ones obtained from the experiments, the latter being higher. Therefore a re-evaluation of the constant 1.86 in Sieder-Tate equation was done by plotting Nusselt number against $(Re \times Pr \times \frac{D}{L})^{1/3} (\mu / \mu_w)^{0.14}$ on logarithmic paper in Figure 17. The new value of the constant obtained from the graph is 2.0. Sieder-Tate correlation is also plotted in the same figure for comparison purposes.

The Wilson plot was first obtained by plotting 1/U versus $1/w^{1/3}$ in figure 15. As can be seen from the figure, the experimental points lie on a curve rather than a straight line. This behavior leads one to think that the coefficient 1/B which represents the slope of the equation

$$\frac{1}{U} = A + \frac{1}{Bw^{1/3}} \qquad (18)$$

is not constant but varies for different runs. This variation may be due to the fact that the average temperature in each experiment was not the same and the terms which make up "B" are strongly temperature dependent for the viscous oil used in the experiments (see equation (2) and Appendix D). Since viscosity is the term in "B" which changes the most with temperature, its variation in different runs may be taken into account by plotting 1/U against $1/w^{1/3}(\mu/\mu_w)^{0.14}$ as suggested by Sieder-Tate equation. This modified form of the Wilson plot is shown in Figure 16. It then seems possible to draw a straight line through these new points but the points still have a scattered form. The limits of experimental accuracy as obtained in section IV-D are marked on both sides of the lines in Figures 15 and 16. But still some points are seen to lie outside the limits of experimental accuracy. A plot of 1/U versus 1/h in Figure 16 shows that these points lie on a straight line. This means that the constant "A" in equation (18) remains substantially constant during the tests. Therefore, the only source of error that causes the plot of 1/U against $1/w^{1/3}(\mu/\mu_w)^{0.14}$ not to give a straight line is most probably the variation in the value of the coefficient "B" for different runs. This fact leads the author to conclude that Wilson plot cannot be employed to determine film coefficients of heat transfer of viscous fluids in laminar flow because the properties of these fluids, such as viscosity, specific heat, etc., show a strong dependence on temperature.

In analyzing the centerline temperature profiles obtained from experiments, the fact that the exchanger remains in the thermal entrance region should be kept in mind. The thermal starting length is calculated from the equation ( ):

$$\frac{L_e}{D} = \frac{(Re)(Pr)}{20} \tag{19}$$

where, D is the tube diameter and L_e the thermal entrance length.

Pr = 100 for all the runs and D = 0.5 in.

Then from equation (19) for Re = 100, L_e = 21 ft and for Re = 500, L_e = 105 ft.

Therefore, the oil side centerline temperature should not vary much along the exchanger. From Figures 12, 13, and 14 it can be seen that the centerline temperature increases very slowly at first (i.e., a few degrees Fahrenheit for every 20 cm). This increase may be attributed to the effects of turbulence and flow disturbances in the entrance of the exchanger. After the first pass, however, this temperature increases almost linearly, possibly because of the partial mixing at the end of the first pass. The bulk temperatures at each 20 centimeter cross section may be obtained if a temperature profile for the cross section is assumed.

For example, if a parabolic profile is assumed the shape of the parabola for each section may be determined by using the three known points, namely, wall temperatures and centerline temperature. The bulk temperature can then be determined by the usual integration and averaging procedure. The bulk temperatures thus obtained may be used to calculate the average overall coefficients of heat transfer for each 20 centimeter length along the exchanger. Then, a plot of U versus x shall give the variation of U along the exchanger length.

APPENDIX

APPENDIX A

1. Oil Side Heat Transfer Coefficients

Calculation of oil side heat transfer coefficients are done as follows:

We can write for the whole length of the exchanger tube:

$$Q = wc(t_2 - t_1) = h_i A_i \Delta t_i$$

where,

$$\Delta t_i = \frac{\Delta t_2 - \Delta t_1}{\ln \Delta t_2 / \Delta t_1} = \text{LMTD}$$

$$\Delta t_1 = t_w - t_1$$

$$\Delta t_2 = t_w - t_2$$

Therefore,

$$h_i = \frac{wc(t_2 - t_1)}{A_i \Delta t_i} \qquad \text{(A 1)}$$

2. Overall Coefficient of Heat Transfer

Calculation of the average overall coefficient U is done as follows:

$$Q = UA \Delta t_m = wc(t_2 - t_1)$$

where,

$$\Delta t_m = \frac{\Delta t_2 - \Delta t_1}{\ln \Delta t_2 / \Delta t_1} = \text{LMTD}$$

$$\Delta t_2 = T - t_2$$

$$\Delta t_1 = T - t_1$$

therefore,

$$U = \frac{wc(t_2 - t_1)}{A \Delta t_m} \qquad \text{(A2)}$$

APPENDIX B

Theoretical Temperature Profile Along the Length of the Exchanger

Consider a 1 - 2 parallel counterflow heat exchanger in which steam condenses isothermally on the outside and fuel-oil flows through the tubes as shown in Fig. B 1.

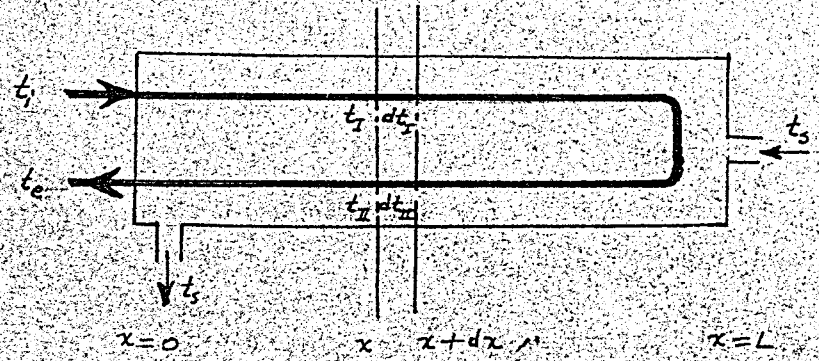


Fig. B 1. 1-2 Parallel-counterflow heat exchanger

If w is the weight flow rate of oil, c its specific heat, t_I and t_{II} the temperatures of the oil in the first and second passes respectively, we can write from a heat balance over dx*,

$$wc\,dt_I = Ua''dx(t_s - t_I) \qquad (B\,1)$$

$$wc\,dt_{II} = -Ua''dx(t_s - t_{II})$$

where a" is the heat exchanging area per unit length.

Furthermore, in order to be able to integrate these equations let us make the following assumptions:

(a) The overall coefficient U is constant throughout the exchanger;

(b) The specific heat and mass flow rate of oil is constant;

(c) There is no heat conduction in the axial direction of tubes;

(d) At any cross section of the exchanger each of the fluids may be characterized by a single temperature.

(e) Heat losses are negligible.

Then, integrating equation (B 1) gives:

$$\frac{dt_I}{t_I - t_s} = -\frac{Ua''}{wc}dx$$

$$\ln(t_I - t_s) = -\frac{Ua''}{wc}x + C_1$$

at $x = 0$, $t_I = t_i$ which gives:

$$C_1 = \ln(t_i - t_s)$$

Substituting back:

$$\ln\left(\frac{t_I - t_s}{t_i - t_s}\right) = -\frac{Ua''}{wc}x \quad \text{or}$$

$$t_I = t_s - (t_s - t_i) \exp(-\frac{Ua''}{wc}x) \qquad (B\ 3)$$

Integration of equation (B 2) gives:

$$\frac{dt_{II}}{t_{II} - t_s} = \frac{Ua''}{wc}dx$$

$$\ln(t_{II} - t_s) = \frac{Ua''}{wc}x + C_2$$

at $x = 0$, $t_{II} = t_e$ which gives:

$$C_2 = \ln(t_e - t_s)$$

Therefore,

$$\ln\left(\frac{t_{II} - t_s}{t_e - t_s}\right) = \frac{Ua''}{wc}x \quad \text{or}$$

$$t_{II} = t_s - (t_s - t_e) \exp(\frac{Ua''}{wc}x) \qquad (B\ 4)$$

Equations (B 3) and (B 4) are the expressions for the temperatures t_I and t_{II} of first and second passes respectively, as a function of the length of the exchanger.

APPENDIX C

Wilson Plot

Consider a series of runs made with a tubular heat exchange equipment in which the tube side velocity is varied. We have from equation (6):

$$\frac{1}{U_h} = \frac{1}{h_h} + \frac{\Delta x}{k \times \frac{A_w}{A_h}} + \frac{1}{h_c \times \frac{A_w}{A_h}}$$

Since a vapor, namely, steam is the heating medium we can write for the hot side coefficient h_h from Nusselts theoretical equation for condensing vapors(16, p.342)

$$(h_h)_{mean} = 0.725 \left(\frac{k_f^3 \lambda \rho_f g}{ND \mu_f \Delta t} \right)^{1/4} \qquad (C\ 1)$$

In a series of runs made by varying the oil velocity all the terms in equation (C 1) except t remains substantially constant. Therefore, we can write:

$$h_h = C_3 \left(\frac{1}{\Delta t} \right)^{1/4} \qquad (C\ 2)$$

where C_3 is a constant.

On the cold side of the exchanger a fluid is flowing in laminar flow. Sieder and Tate(8) have correlated experimental data for heating and cooling of different fluids flowing through tubes by the following formula:

$$\frac{hD}{k} = 1.86\left(\frac{4}{\pi}\,\frac{wc}{kL}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad (C\ 3)$$

In a series of runs made on the same exchanger every term in the above expression remains constant except w and $(\mu_a/\mu_w)^{0.14}$. So we can write:

$$h = C_2(w)\,(\mu_a/\mu_w)^{0.14} \qquad (C\ 4)$$

Substituting expressions (C 2) and (C 4) back into the overall coefficient of heat transfer gives:

$$\frac{1}{U} = C_1(\Delta t)^{1/3} + \frac{\Delta x}{k\,\frac{A_w}{A_h}} + C_2\,\frac{1}{w^{1/3}(\mu_a/\mu_w)^{0.14}}$$

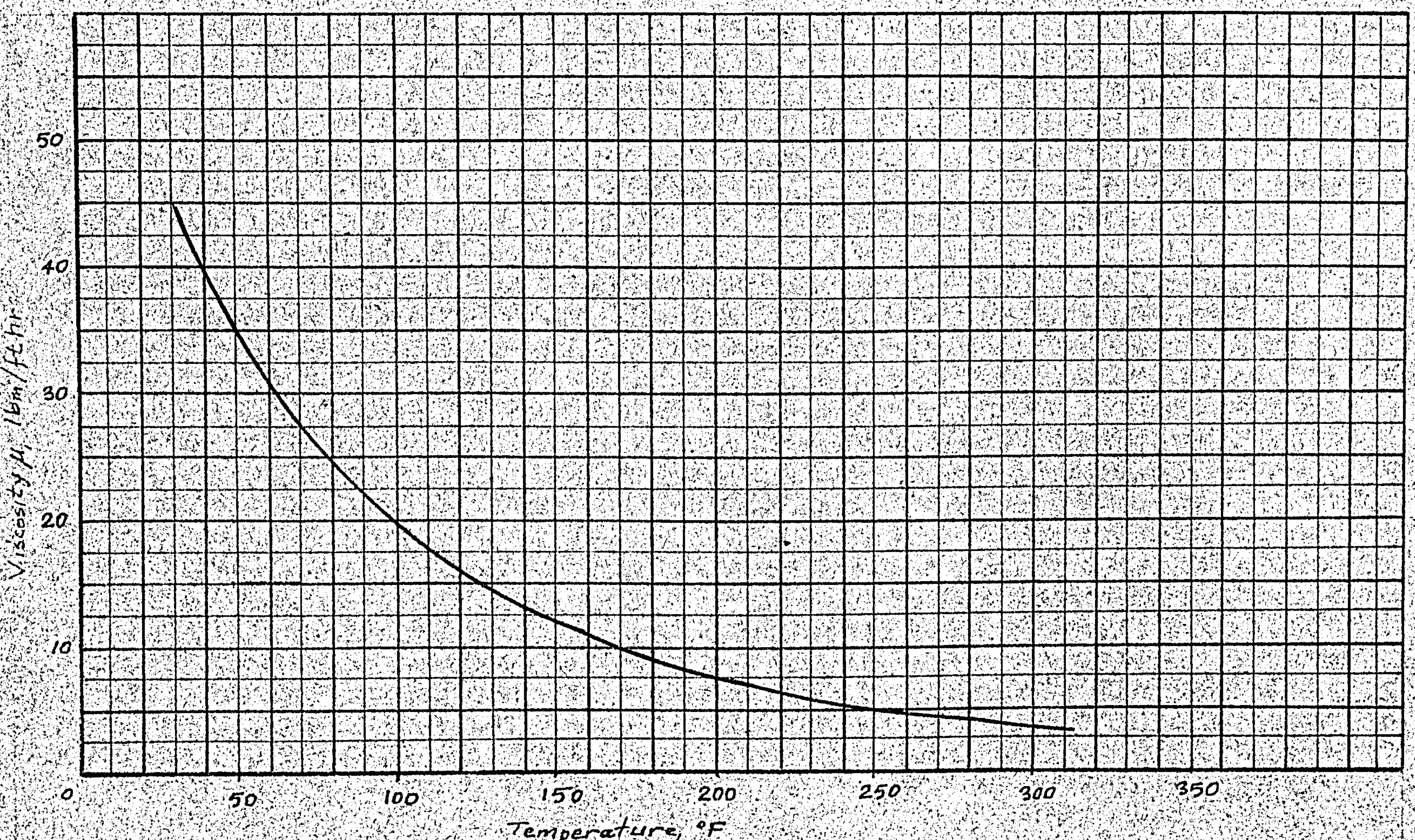


Fig. 18. Temperature v.s. dynamic viscosity μ, of ASTM No. 6 Bunker C, crude oil (from reference 15, pp. 873)

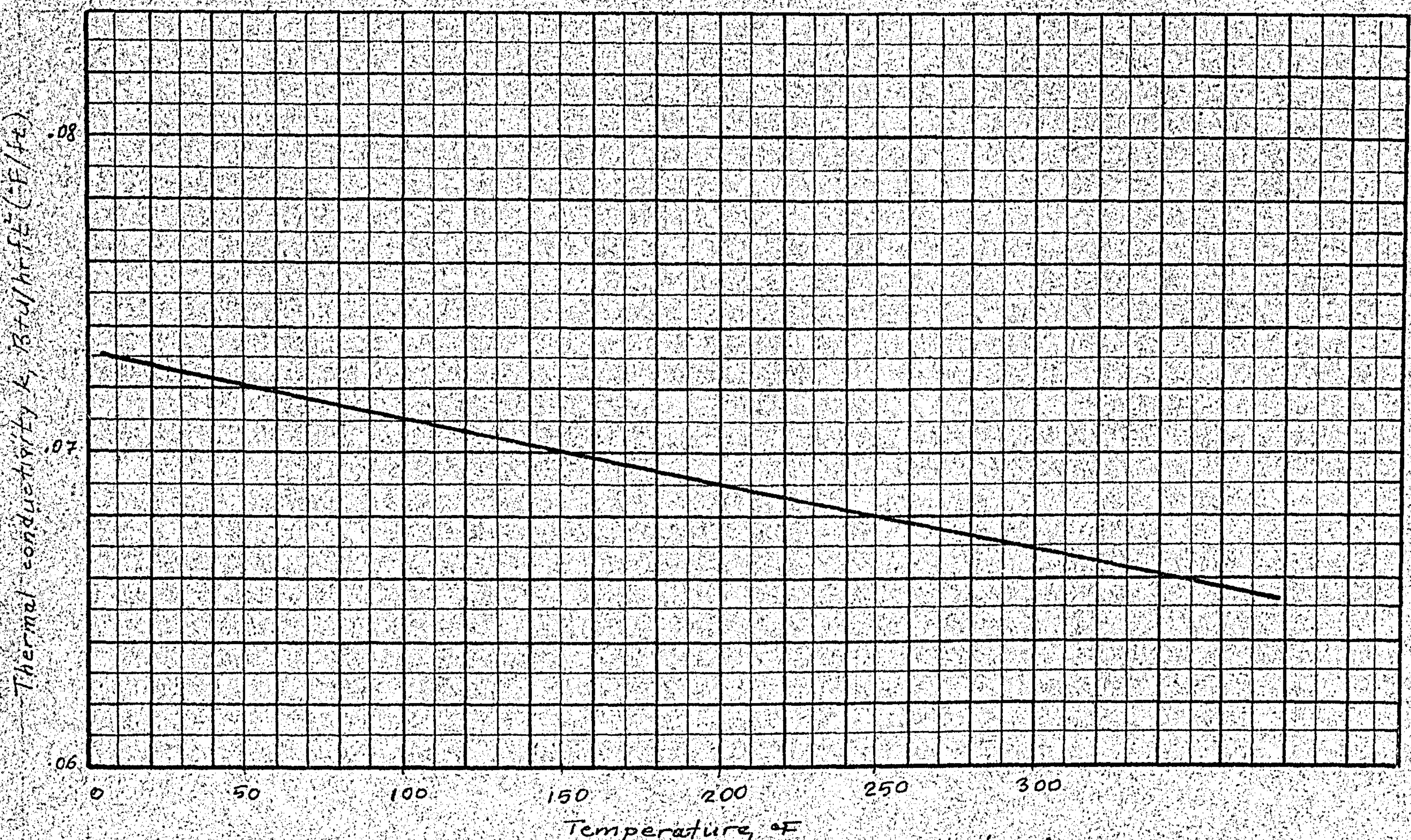


Fig. 19. Temperature v.s. thermal conductivity of ASTM No. 6 Bunker C, crude oil (from reference 15, pp. 876)

Fig. 20. Temperature v.s. specific heat (C_p) of ASTM No. 6 Bunker C, crude oil (from reference 15, pp. 871)

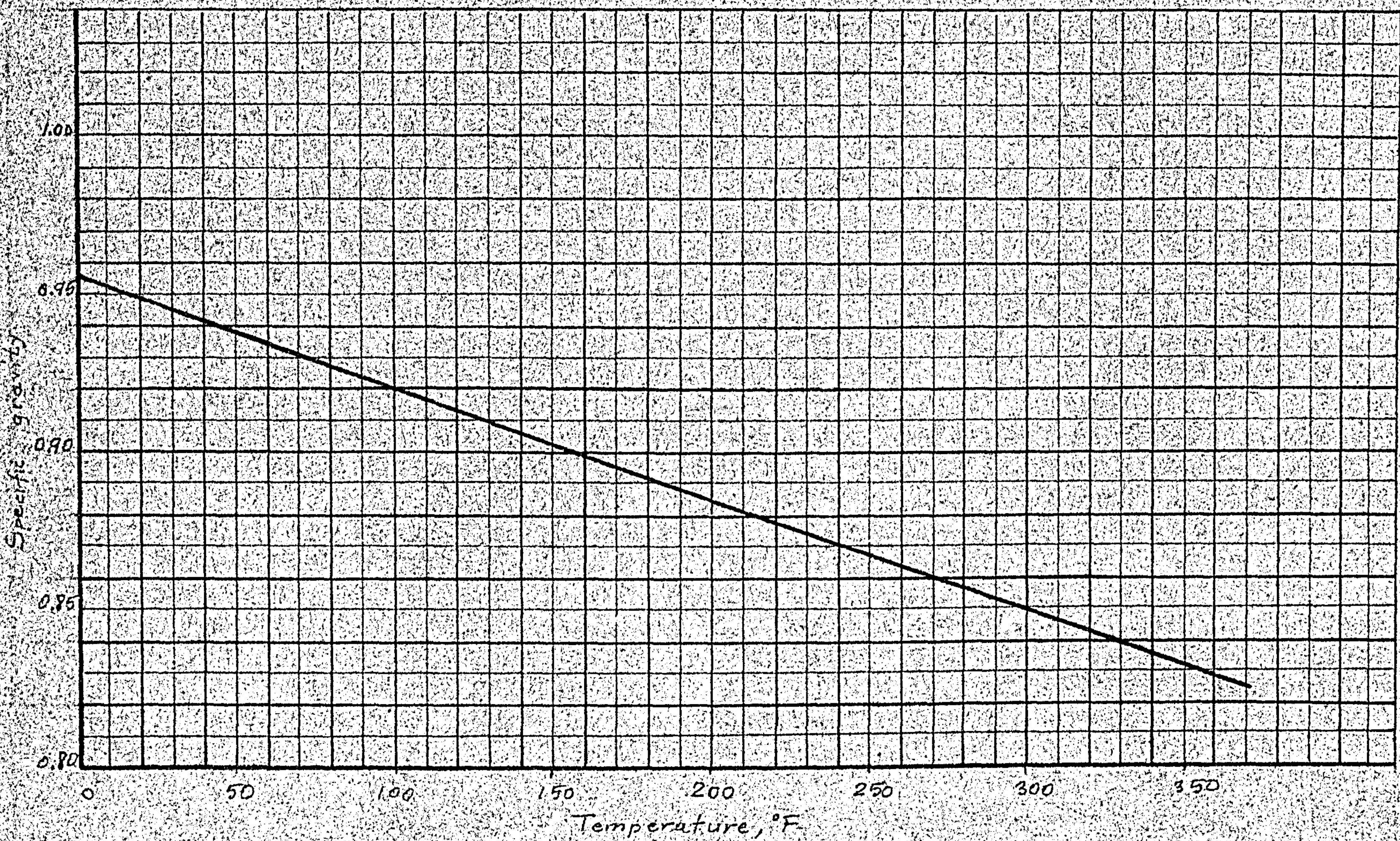


Fig. 21. Temperature v.s. specific gravity of ASTM No. 6 Bunker C crude oil (from reference 15, pp 870)

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