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Cooling of a hydrocarbon oil.

Edward Groth
University of Louisville

UNIVERSITY OF LOUISVILLE

COOLING OF A HYDROCARBON OIL

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Edward Groth, Jr.

1937

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LIST OF SYMBOLS

Symbol	Quantity	Units
A	Heat transfer surface	sq. ft.
C_p	Specific heat	B.T.U./(hr.)(°F.)
D	Diameter of pipe	ft.
G	Mass velocity	lbs./(hr.)(sq. ft.)
h	Heat transfer coefficient	B.T.U./(hr.)(sq. ft.)(°F.)
k	Thermal conductivity	B.T.U./(hr.)(sq.ft.)(°F./ft.)
N	Length of test section	ft.
Q	Quantity of heat transferred	B.T.U./hr.
t	Average pipe temperature	°F.
t_1	Test liquid entrance temperature	°F.
t_2	Test liquid exit temperature	°F.
t_3	Refrigerant entrance temperature	°F.
t_4	Refrigerant exit temperature	°F.
V	Velocity of test liquid in pipe	ft./hr.
W	Weight rate of flow	lbs./hr.
z	Viscosity	centipoises
Δt	Temperature difference	°F.
ρ	Density of fluid	lbs./cu. ft.
μ	Viscosity of fluid	lbs./(ft.)(hr.)

Symbols are from the system recommended by the Council of the American Institute of Chemical Engineers.

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I. INTRODUCTION

The purpose of this investigation was to study the transmission of heat in a liquid-liquid heat exchanger in regions of viscous and critical flow and at temperatures in the region of 32° F. The experimental work was carried on under conditions approaching industrial practice, so that the results would be useful in the design of practical coolers, crystallisers, and refrigerating units.

The fluid used in this investigation was "Eocene", a high grade of kerosene. The heat exchanger was designed by Browne and Finger, (1) from the plans of J. F. D. Smith (10), and built by Browne (2) and Donahew (3).

Previous investigators (5-10) have correlated similar heat transfer data by use of a form of the Nusselt equation:

$$\frac{hD}{k} = a\left(\frac{DG}{\mu}\right)^{n}\left(\frac{C_p\mu}{k}\right)^{m} \qquad (1)$$

where a, n, and m are constants and the other terms are defined in the List of Symbols, page 3. This equation involves three dimensionless groups, i. e., the Nusselt number, (hD/k), the Reynolds number, (DG/μ), and the Prandtl number, $(C_p\mu/k)$, and illustrates the use of dimensional analysis for the correlation of heat transfer data. This method is used as a convenient way of relating and correlating data where many variables are involved. Variables encountered in heat transfer of this type are the viscosity, specific heat, density, and thermal conductivity of the fluid, the diameter and length of the pipe, and the velocity with which the fluid flows through the pipe.

Of the three dimensionless groups, the Reynolds number, (DG/μ), is one of the most frequently occurring in fluid dynamics and heat transfer, and indicates the degree of turbulence of flow. With a single fluid flowing in a pipe of constant diameter, the density and viscosity are substantially constant, and the controlling factor is seen to be the velocity of flow. For values of the Reynolds number below 2100, the flow is considered viscous or streamline, while for values above 4000, the flow is called turbulent. The intermediate region is called the critical region, and is a transient zone in which the flow may have the characteristics of either of the two types.

The Prandtl number, $(C_p\mu/k)$, is obtained from the physical properties of the fluid itself— specific heat, viscosity, and thermal conductivity. These properties are nearly constant over small ranges of temperatures, so the magnitude of the Prandtl number will be found to change but little during investigation of a single system. Its magnitude will change chiefly as the complexity of the system is increased. That is, the value of the Prandtl number will increase in passing from a gas with a simple molecule to a gas with a complex molecule, then to water and aqueous solutions, and then to oils and organic liquids. As previously stated, the Prandtl number remains almost constant in the investigation of a single fluid, and thus could be omitted entirely, or replaced by a constant in the Nusselt equation, without any appreciable error. Its inclusion is required in order that a general relation may be shown, and in order that a comparison with data on other systems may be obtained.

The Nusselt number, (hD/k), contains the film coefficient, the pipe diameter, and the thermal conductivity of the fluid. In a single experiment, where the pipe diameter and the conductivity remain constant, the magnitude of the Nusselt number is dependent entirely on the film coefficient. Since the coefficient is defined as $h = k/L$, where k is the thermal conductivity and L is the thickness of the film, the magnitude of the coefficient is dependent entirely on the thickness of the film. This thickness is influenced by several factors, including the velocity of flow, the density of the fluid, the viscosity of the fluid, and the smoothness of the inner surface of the metal wall. The turbulence resulting from a high velocity reduces the film thickness, while a high density also tends to bring about the same result. The film thickness is also influenced by the fluid viscosity, especially at the temperature of the film. The thickness of the film bears a direct relation to the viscosity— a high viscosity produces a thick film, while a thin film accompanies a low viscosity. Lastly, the thickness is influenced by the smoothness of the metal pipe. A roughness of the surface has been shown to increase the effective film thickness.

To the three groups in the equation, a fourth— N/D, the ratio of tube length to diameter— has been added by several investigators. McAdams and Frost (4), after analysis of various data on heating water in tubes in which the N/D ratio varied from 34 to 100, found an apparent effect of tube length, and proposed the equation:

$$\frac{hD}{k} = 0.0272 \left(1 + \frac{50\,D}{N}\right)\left(\frac{DG}{\mu_f}\right)^{0.8} \qquad (2)$$

The omission of the Prandtl number was compensated for to a certain extent by the use of the viscosity, μ_f, at the mean film temperature.

However, Lawrence and Sherwood (5) found the length to have a negligible effect, although their investigation covered a range of lengths equivalent to 59 to 224 diameters. The data on water obtained by Lawrence and Sherwood were found to be reasonably well correlated by the equation:

$$\frac{hD}{k} = 0.056 \left(\frac{DG}{\mu}\right)^{0.7} \left(\frac{C_p\mu}{k}\right)^{0.5} \qquad (3)$$

Morris and Whitman, (6), after a study covering the heating and cooling of water and several oils, proposed a value of 0.37 as the exponent of the Prandtl number, and presented their correlation graphically by plotting $(hD/k)/(C_p\mu/k)^{0.37}$ against DG/μ. Rice (7) studied the data of several previous investigators on fluids ranging from air and simple gases through water and oils, both in heating and cooling. He first proposed the equation:

$$\frac{hD}{k_f} = \frac{1}{63} \left(\frac{DG}{\mu_f}\right)^{0.83} \left(\frac{C_f\mu_f}{k_f}\right)^{0.5} \qquad (4)$$

where the physical properties of the fluid are all taken at the arithmetic mean film temperature. He later modified the original equation to:

$$\frac{hD}{k_f} = \frac{1}{60} \left(\frac{DG}{\mu_f}\right)^{0.83} \left(\frac{C_f\mu_f}{k_f}\right)^{0.33} \qquad (5)$$

After extended study of data on water and air, and the Morris and Whitman data on oils, Dittus and Boelter (8) in 1930 proposed two equations to be used in the correlation of data on heat transfer:

$$\text{for heating,} \quad \frac{hD}{k} = 0.024 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{C_p\mu}{k}\right)^{0.4} \qquad (6)$$

$$\text{for cooling,} \quad \frac{hD}{k} = 0.026 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{C_p\mu}{k}\right)^{0.3} \qquad (7)$$

Two years later, Sherwood and Petrie (9) presented data on heat transfer to water, acetone, benzene, kerosene, and n-butyl alcohol in both viscous and turbulent flow. In the turbulent region, their data were found to be in excellent agreement with the empirical Dittus and Boelter equation, but the agreement of points in the viscous and critical regions was not very close. They observed, however, that the slope of the kerosene line in and slightly above the critical region was "definitely greater" than 0.8. Since most of their work was done in the turbulent region, the data on viscous flow were not extensive enough to warrent a general conclusion. A direct comparison with the equation was made by plotting $(hD/k)/(C_p\mu/k)^{0.4}$ against DG/μ. In 1935, Smith (10) presented similar data on hydrocarbon oils, and also endorsed the Dittus and Boelter equation as suitable for correlating heat transfer data in turbulent flow. He also plotted $(hD/k)/(C_p\mu/k)^n$ against DG/μ, using $n = 0.4$ for heating and $n = 0.3$ for cooling runs.

In correlation of viscous flow data, the Graetz number, (WC_p/kN), is often used in place of the Reynolds number. After studying and recalculating the data of a number of previous investigators, Drew, Hogan, and McAdams (11) concluded that in viscous flow, where the heat transferred by convection was negligible compared with that transferred by conduction, a good correlation could be obtained by plotting WC_p/kN against $(t_2-t_1)/(t-t_1)$. While the use of the Graetz number and the (temperature-rise)-(temperature-difference) ratio may be satisfactory in single cases, there is no indication that such a correlation will be applicable over as wide ranges as that satisfied by the Nusselt equation.

Sherwood, Kiley, and Mangsen, (12), in working with oil flowing in horizontal pipes, attempted a correlation by means of the Graetz relation. They found, however, that the method was inadequate for the correlation of their data. They found that a plot of $(t_2-t_1)/(t-t_1)$ against DG/μ indicated a sudden and large increase in the amount of heat transferred as the flow passed from the viscous to the critical and then to the turbulent region. Smith (10) used the Graetz relation for the correlation of his data in the viscous region, and found a reasonably close agreement. The correlation in the viscous region by this method was not so satisfactory, however, as that in the turbulent region, where the Nusselt equation was used.

In the present investigation, heat transmission at low velocities and at temperatures below room temperature will be studied. Purified kerosene will be used as a test liquid on the inside of the pipe, and also as a refrigerant on the outside. The data will be compared with the results of previous investigators obtained at higher temperatures.

II. APPARATUS AND MATERIALS

A complete and detailed description of the apparatus is given by Browne (2) and Donahew (3). Figure I, page 15, is a view of the complete heat exchanger, including the potentiometer and galvanometer used in taking the thermocouple readings. Figure II, page 17, shows the heat exchanger with the parts labeled to correspond with the details mentioned in the following discussion. Figure III, page 19, shows the ammonia compressor which was used for cooling the refrigerant.

A. <u>Heat Exchanger</u>.

The test section is a standard one-inch brass pipe, contained in a cast iron shell. The total length of the brass pipe is nine feet, six inches— a test length of six feet preceded by a calming section of twenty-three inches and followed by an exit section of nineteen inches. The cast iron shell is made up of two inlet and outlet sections for the refrigerant, C 1 and C 2; six split sleeves, D 1, D 2, D 3, D 4, F 1, and F 2; and two thirty-inch sleeves, E 1 and E 2. The split sleeves are installed at the thermocouple junctions so that any changes or repairs may be facilitated.

Calming sections in the refrigerant inlet and outlet sections are provided to prevent turbulence near the thermocouple junctions at the pipe ends, and to make the flow of the refrigerant streamline after passage through the mixing chambers.

Inner steel sleeves prevent the transfer of heat between the refrigerant and the test liquid in the refrigerant inlet and outlet sections. These sleeves extend through the inlet and outlet sections to the centers of the split cast iron sleeves, D 1, D 2, D 3, and D 4, thus forming an annular space around each end of the brass pipe. The

Figure I.

Heat Exchanger and Accessories

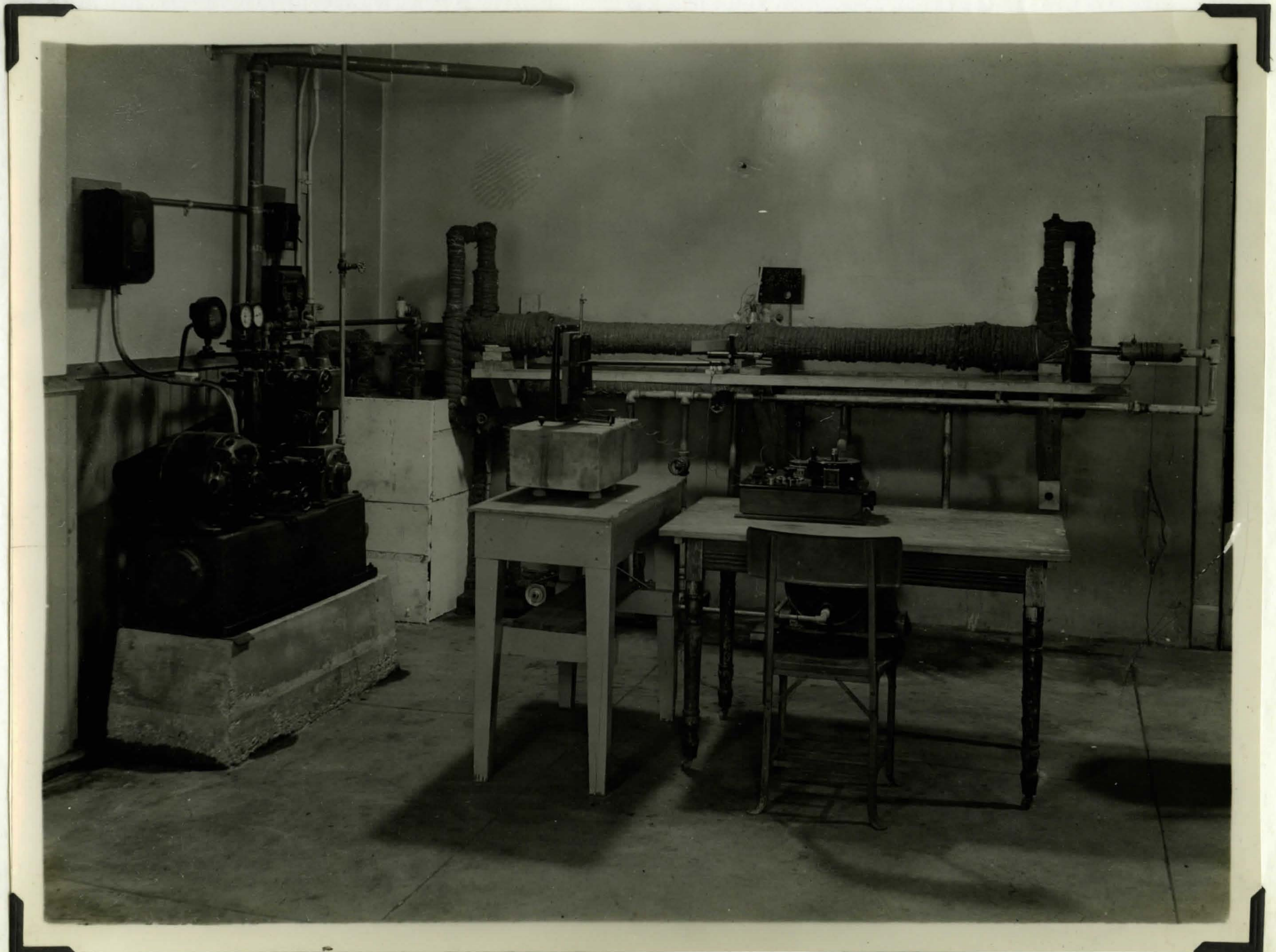


Figure II.

Heat Exchanger

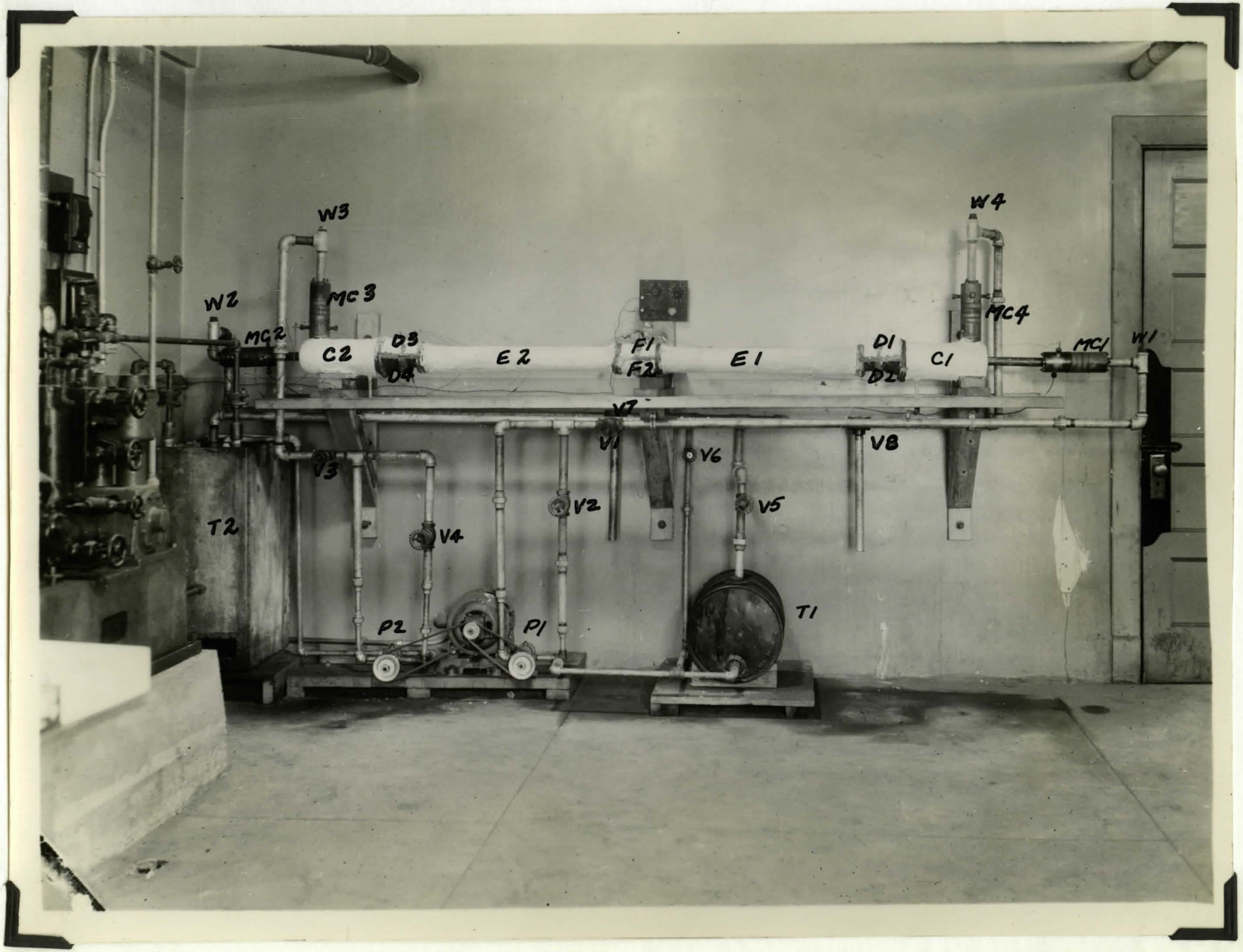
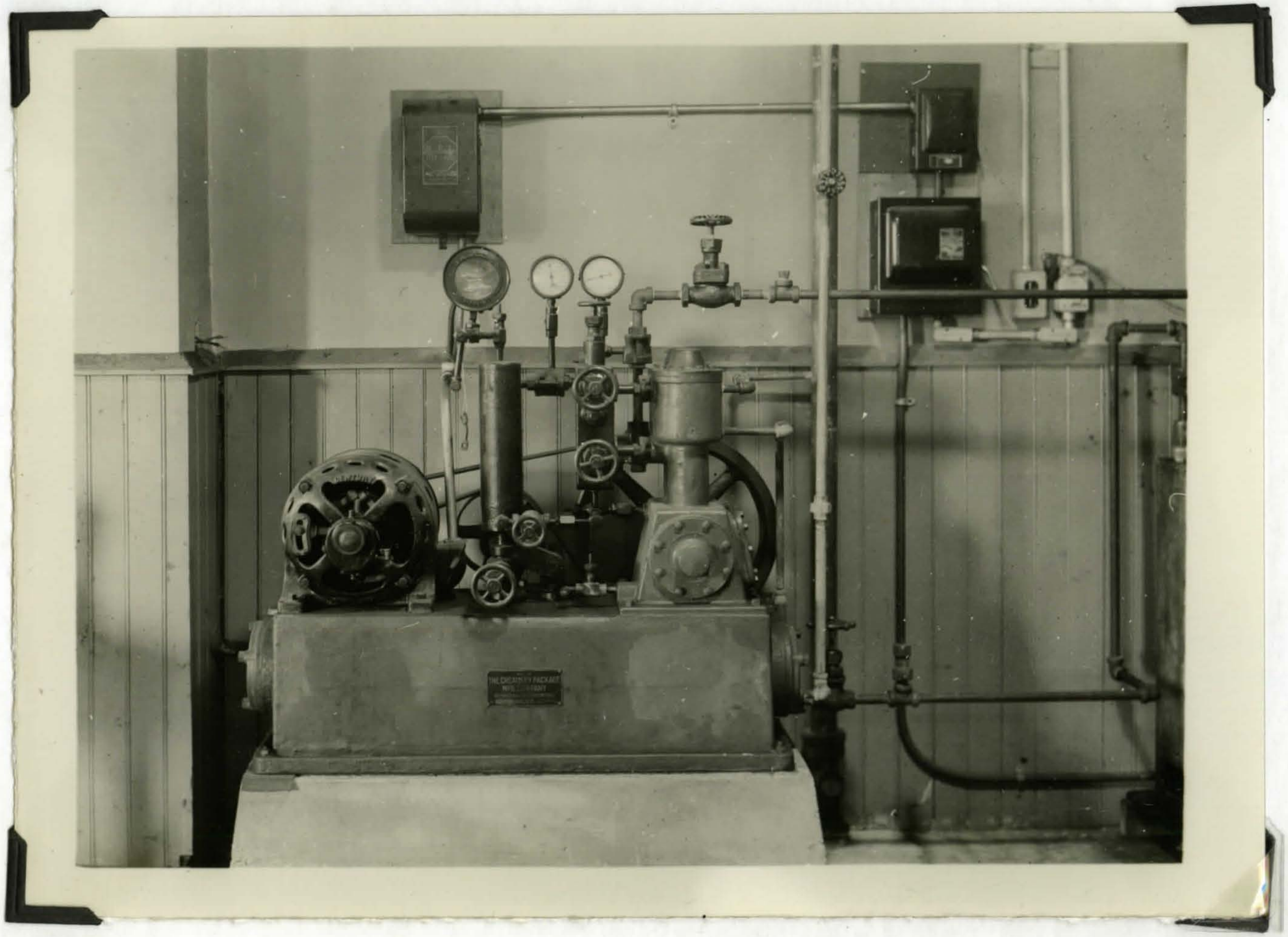

W3
W2
MC3
MC2
C2
D3
D4
E2
F1
F2
E1
D1
D2
C1
W4
MC4
MC1
W1
V7
V1
V8
V3
V6
V2
V5
V4
T2
T1
P2
P1

Figure III.

Ammonia Compresser



annular spaces are packed with rock wool, so that heat transfer is reduced, and more nearly accurate measurement of temperatures is made possible.

In order that heat transfer by conduction between the brass pipe and the cast iron shell may be minimized, asbestos strips are placed between the surfaces making contact. Such conduction transfer is of little importance, since areas of contact are small and relatively distant from the test section.

The mixing chambers are made up of slotted steel and copper collars bolted together. Mixing chambers in the test liquid line, MC 1 and MC 2, and in the refrigerant inlet and outlet lines, MC 3 and MC 4, bring about a complete mixing and uniform fluid temperatures, so that the temperature of the entire fluid body may be measured by a thermocouple placed in the center of the stream of liquid.

The calming section is a cross made of sheet copper, placed inside the brass pipe between the test section and the mixing chamber at the entrance end. The calming cross tend to eliminate any turbulence in the pipe that may result from the passage of the fluid through the mixing chamber.

Sixteen thermocouples are installed in the heat exchanger—one is placed in each of the four mixing chambers, and twelve, in groups of four, are fastened to the brass pipe. The thermocouples are made of No. 28 B & S gage iron and constantan wire, welded together at one end and carried to a cold junction on a shelf behind the exchanger. From the cold junction, copper leads are run to two manually controlled dials with sixteen terminals, so that any thermocouple of the group may be connected to the potentiometer by a proper adjustment of the dials.

The thermocouples are installed in the mixing chambers through one-eighth inch copper tubing, centered in one-half inch steel bolts. The junctions are located in the center of the liquid stream, and are protected by the copper tube, which extends to within one-sixteenth inch of the junction.

The twelve thermocouples attached to the pipe also pass through one-half inch bolts in the split sleeves, four at each end and four in the center of the test section. Each group of four is equally spaced around the circumference of the pipe. Each junction is soldered to the pipe at one end of a one-sixteenth inch by one inch slot cut in the pipe. The slots are filled with litharge and glycerine cement, and the surface is dressed down even with the surface of the pipe.

B. Accessories.

Two reservoirs are provided for the refrigerant and the test liquid. The test liquid reservoir is a horizontal steel drum with a capacity of ten gallons. A by-pass line around the reservoir affords a means of controlling the temperature of the liquid passing through the exchanger. The refrigerant reservoir is a vertical rectangular tank, holding about thirty gallons. The expansion coils of the ammonia compressor are contained in this tank.

Two pumps, P 1, in the test liquid line, and P 2, in the refrigerant line, are provided for the circulation of the two liquids. They are bronze gear pumps, having a capacity of six gallons per minute at a speed of 1000 R. P. M. The pumps are driven by a motor, operated at constant speed. The rate of flow of refrigerant and test liquid is regulated by means of valves V 1, V 2, V 3, and V 4 in the lines.

The weighing tank is simply an open container, used to collect a portion of the test liquid over a measured period of time, for the determination of the rate of flow. The test liquid is taken from the return line through the three-way valve, V 7. A similar three-way valve, V 8, is provided for the determination of the rate of flow of the refrigerant. A stop watch was used to measure accurately the period of time over which the fluid was collected, which ranged from thirty to sixty seconds.

For the attainment of low temperatures, a three-quarter ton ammonia compressor was installed, with the expansion coils placed in the refrigerant reservoir. The compressor was manually operated, and the refrigeration was controlled entirely by adjusting a needle-point expansion valve in the liquid ammonia line.

Four thermometer wells, W 1, W 2, W 3, and W 4, were added to the exchanger to provide a means for readily determining the conditions of operation, and to furnish a check on the thermocouple readings. The wells are made up of quarter-inch copper tubing, with a plug brazed in one end. The tubing passes through a hole drilled in a standard one-inch cast iron plug.

The refrigerant reservoir was insulated with sheet cork and Celotex, sealed into place with hot asphalt. Irregularities in the surface of the heat exchanger were filled with asbestos fiber pulped with water and plaster of paris, and allowed to dry. The exchanger and all the refrigerant lines were then covered with a layer of hair felt.

The thermocouple readings were taken with a Leeds and Northrup Type K potentiometer and a wall galvanometer. The galvanometer was mounted on a heavy concrete base, which in turn was supported by four rubber cushions, in order to eliminate disturbances due to vibration.

C. Operation.

In the operation of the heat exchanger, the test liquid is drawn from the tank, T 1, by the pump, P 1. The rate of flow is controlled by manipulation of the valves, V 1 and V 2. The test liquid flows next through the mixing chamber, MC 1, and its temperature is measured by thermocouple # 1. After flowing through the calming section it enters the test section, where its temperature is lowered. Its temperature is again measured by thermocouple # 2, after passing through the mixing chamber, MC 2. The test liquid is then returned to the reservoir, where it re-enters the cycle.

In the same way, the refrigerant is taken from the tank, T 2, by the pump, P 2, has its velocity regulated by manipulation of the valves, V 3 and V 4, and is pumped through the mixing chamber, MC 3, where its temperature is measured by thermocouple # 3. After passing through the annular space between brass pipe and the cast iron shell, the refrigerant is mixed in mixing chamber MC 4, has its temperature measured by thermocouple # 4, and is then returned to the tank, T 2, to be cooled again.

D. Materials.

The physical properties of the Nocene used in the calculation of the results are listed in Table I, page 25. The density was determined experimentally by use of a Westphal balance. Since the Reynolds

Table I

PHYSICAL PROPERTIES OF EOCENE

Density (60° F.)	50.1 lbs./cu. ft.
Specific heat	0.504 B.T.U./(lb.)(°F.)
Thermal conductivity	0.0875 B.T.U./(hr.)(sq. ft.)(°F./ft.)

(Distillation ranges and a temperature-viscosity curve are included in the appendix.)

number was calculated directly from the weight rate of flow, no assumptions of a constant density were made. The specific heat was determined by measuring the increase in temperature of a known weight of Eocene, brought about by the introduction of a measured quantity of electricity. The heating was accomplished in a small thermos bottle, using a nichrome heating element; an electric stirrer insured complete mixing during the heating. Viscosities were determined at several temperatures using a Hoepler viscosimeter, and a smooth curve was drawn to show the variation of the viscosity with a change in temperature. The value of the thermal conductivity was taken as representative of several found in the literature.

III. EXPERIMENTAL WORK

Prior to obtaining data, the refrigerant was circulated in the unit until the apparatus was thoroughly cooled. This generally required about six to eight hours. The cooling was followed by circulation of the test fluid until equilibrium conditions had been attained. Conditions of steady state heat transfer were indicated by constant test liquid and refrigerant entrance temperatures, and usually were established in from fifteen minutes to two hours, depending on the rate of flow of the test liquid.

When equilibrium had been reached, the cold junctions of the thermocouples were adjusted to exactly 32° F. in an ice bath. The potentiometer circuit was balanced against a standard cell, and the thermocouple temperatures were determined and recorded. The four thermometers in the thermometer wells were read and their temperatures likewise were recorded. After all the temperature readings had been taken, a portion of the test liquid was collected, usually over a period of thirty seconds, and weighed. When this sample had been returned to the test liquid reservoir a new rate of flow was obtained by regulation of the valves, and operation was continued until equilibrium conditions were again attained. By carefully adjusting the valves, it was possible to obtain eight to ten sets of data between the minimum and maximum rates of flow.

The experimental data and the calculated results are contained in Table II, pages 28-33. The test liquid inlet and outlet temperatures are recorded directly as determined by the thermocouples. The average pipe temperature is the temperature corresponding to the average millivolt reading of the twelve thermocouples on the pipe. Other investigators

Table II

HEAT TRANSFER DATA

Run No.	Test Liquid Inlet Temp.	Test Liquid Outlet Temp.	Average Pipe Temp.	Δt_1	Δt_2	Log Mean Δt	Test Liquid Δt
Units	°F.	°F.	°F.	°F.	°F.	°F.	°F.
1	56.438	53.046	39.638	16.800	13.408	15.056	3.392
2	57.997	55.019	39.546	18.451	15.473	16.936	2.978
3	54.223	51.211	37.365	16.858	13.846	15.318	3.012
4	59.300	54.154	38.865	20.435	15.289	17.756	5.146
5	64.067	58.827	34.870	29.197	23.957	26.518	5.240
6	54.292	50.138	29.923	24.369	20.215	22.250	4.154
7	61.167	56.612	39.382	21.785	17.230	19.438	4.555
8	55.400	52.285	34.008	21.392	18.277	19.814	3.115
9	60.567	57.997	47.380	13.187	10.617	11.868	2.570
10	59.200	56.992	47.768	11.432	9.224	10.299	2.208
11	50.312	48.407	34.712	15.600	13.695	14.642	1.905
12	55.607	53.011	28.573	27.034	24.438	25.740	2.596
13	60.867	49.377	19.452	41.415	29.925	35.395	11.490
14	57.650	54.638	25.899	31.751	28.379	30.250	3.012
15	55.365	52.458	23.038	32.327	29.420	30.882	2.907
16	52.215	50.588	36.877	15.338	13.711	14.524	1.627
17	56.231	54.223	35.512	20.719	18.711	19.717	2.008
18	56.889	54.500	38.135	18.754	16.365	17.550	2.389
19	59.867	56.231	38.114	21.753	18.117	19.899	3.636
20	58.031	55.607	29.343	28.688	26.264	27.486	2.424
21	56.231	54.362	32.321	23.910	22.041	22.986	1.869
22	55.019	53.600	35.817	19.202	17.783	18.502	1.419
23	54.050	53.081	38.642	15.408	14.439	14.933	0.969
24	53.774	52.665	38.145	15.629	14.520	15.084	1.109
25	56.715	54.535	37.569	19.146	16.966	18.052	2.180
26	54.673	52.631	37.288	17.385	15.343	16.359	2.042
27	58.827	56.715	36.870	21.957	19.845	20.905	2.112
28	54.085	52.250	34.781	19.304	17.469	18.390	1.835
29	54.050	52.215	35.577	18.473	16.638	17.557	1.835
30	54.812	52.873	34.473	20.339	18.400	19.373	1.939

Table II

HEAT TRANSFER DATA

Run No.	Test Liquid Inlet Temp.	Test Liquid Outlet Temp.	Average Pipe Temp.	Δt_1	Δt_2	Log Mean Δt	Test Liquid Δt
Units	°F.	°F.	°F.	°F.	°F.	°F.	°F.
31	54.742	52.907	33.666	21.076	19.241	20.165	1.835
32	55.123	53.184	32.459	22.664	20.725	21.702	1.939
33	50.830	50.035	36.855	13.975	13.180	13.587	0.795
34	62.833	61.400	45.801	17.032	15.599	16.321	1.433
35	61.067	57.788	46.811	14.256	10.977	12.568	3.279
36	62.267	58.862	46.167	16.100	12.695	14.345	3.405
37	62.733	59.100	47.085	15.648	12.015	13.766	3.633
38	64.000	59.600	47.339	16.661	12.261	14.363	4.400
39	65.400	61.533	46.352	19.048	15.181	17.067	3.867
40	66.600	61.833	45.686	20.914	16.147	18.447	4.767
41	65.933	60.200	32.757	33.176	27.443	30.249	5.733
42	62.300	57.546	32.181	30.119	25.365	27.702	4.754
43	60.767	56.785	32.373	28.394	24.412	26.380	3.982
44	59.567	55.607	33.643	25.924	21.964	23.910	3.960
45	57.892	54.396	34.574	23.318	19.822	21.544	3.496
46	57.096	53.566	35.077	22.019	18.489	20.233	3.530
47	56.127	52.977	35.691	20.436	17.286	18.836	3.150
48	55.780	52.250	34.505	21.275	17.745	19.476	3.530
49	55.780	51.869	34.644	21.136	17.225	19.133	3.911
50	54.950	50.796	35.323	19.627	15.473	17.485	4.154
51	54.673	50.415	36.116	18.557	14.299	16.352	4.258
52	54.569	50.277	36.902	17.667	13.375	15.437	4.292
53	66.700	61.100	35.747	30.953	25.353	28.088	5.600
54	65.167	60.933	37.045	28.122	23.888	25.974	4.234
55	63.833	60.200	38.599	25.234	21.601	23.394	3.633
56	63.033	60.000	41.519	21.514	18.481	19.976	3.033
57	62.133	59.500	42.930	19.203	16.570	17.872	2.633
58	61.333	59.167	44.522	16.811	14.645	15.719	2.166
59	60.767	58.862	47.175	13.592	11.687	12.628	1.905
60	61.533	59.833	49.749	11.784	10.084	10.923	1.700

Run No.	Weight Test Liquid per Min.	Weight Test Liquid per Hr.	Average Test Liquid Temp.	s	μ	Q
Units	Lbs. per min.	Lbs. per hr.	^{o}F.	Centi-poises	$\frac{\text{Lbs.}}{(\text{hr.})(\text{ft.})}$	B.T.U. per hr.
1	19.2500	1155.00	54.742	2.27	5.49	1971
2	29.8750	1792.50	56.508	2.22	5.37	2695
3	17.2500	1035.00	52.717	2.33	5.64	1570
4	9.1250	547.50	56.727	2.22	5.37	1419
5	25.2500	1515.00	61.447	2.10	5.08	3995
6	14.0000	840.00	52.215	2.35	5.69	1760
7	32.3750	1942.50	58.890	2.16	5.23	4470
8	12.0000	720.00	53.843	2.30	5.57	1130
9	32.3750	1942.50	59.282	2.15	5.20	2520
10	15.6250	937.50	58.096	2.18	5.27	1042
11	32.0000	1920.00	49.360	2.45	5.93	1842
12	17.1875	1031.25	54.309	2.28	5.52	1350
13	8.2500	495.00	55.122	2.26	5.47	2870
14	17.0000	1020.00	56.144	2.23	5.40	1545
15	21.0000	1260.00	53.912	2.29	5.54	1848
16	31.0000	1860.00	51.402	2.37	5.73	1525
17	16.7500	1005.00	55.227	2.26	5.47	1020
18	19.2500	1155.00	55.695	2.24	5.42	1390
19	16.6250	997.50	58.049	2.18	5.27	1825
20	11.6250	697.50	56.819	2.21	5.35	851
21	14.5625	873.75	55.297	2.26	5.47	822
22	17.8125	1068.75	54.310	2.28	5.52	763
23	24.6250	1477.50	53.566	2.30	5.57	720
24	31.6250	1897.50	53.220	2.31	5.59	1060
25	33.3750	2002.50	55.625	2.25	5.45	2200
26	26.6250	1597.50	53.652	2.30	5.57	1640
27	24.5625	1473.75	57.771	2.19	5.30	1568
28	21.6250	1297.50	53.168	2.32	5.61	1200
29	18.0000	1080.00	53.133	2.32	5.61	998
30	17.1875	1031.25	53.843	2.30	5.57	1010

Run No.	Weight Test Liquid per Min.	Weight Test Liquid per Hr.	Average Test Liquid Temp.	z	μ	Q
Units	Lbs. per min.	Lbs. per hr.	^{o}F.	Centi-poises	$\frac{\text{Lbs.}}{(\text{hr.})(\text{ft.})}$	B.T.U. per hr.
31	15.5000	930.00	53.825	2.30	5.57	859
32	14.0000	840.00	54.154	2.29	5.54	821
33	33.0000	1980.00	50.433	2.41	5.83	793
34	33.0000	1980.00	62.117	2.09	5.06	1430
35	31.8750	1912.50	59.428	2.15	5.20	3160
36	24.8750	1492.50	60.565	2.12	5.13	2560
37	24.0000	1440.00	60.917	2.12	5.13	2640
38	18.3750	1102.50	61.800	2.10	5.08	2445
39	14.5000	870.00	63.467	2.06	4.98	1695
40	12.3750	742.50	64.217	2.04	4.94	1784
41	8.8750	532.50	63.067	2.07	5.01	1539
42	11.0000	660.00	59.923	2.14	5.18	1581
43	12.8750	772.50	58.776	2.17	5.25	1550
44	15.5000	930.00	57.587	2.19	5.30	1854
45	17.0000	1020.00	56.144	2.23	5.40	1797
46	18.3750	1102.50	55.331	2.26	5.47	1960
47	20.5000	1230.00	54.552	2.28	5.52	1950
48	23.1250	1387.50	54.015	2.29	5.54	2470
49	26.1250	1567.50	53.825	2.30	5.57	3090
50	27.2500	1635.00	52.873	2.32	5.61	3420
51	31.8750	1912.50	52.544	2.33	5.63	4110
52	32.0000	1920.00	52.423	2.34	5.66	4150
53	8.0000	480.00	63.900	2.05	4.96	1352
54	9.5000	570.00	63.050	2.07	5.01	1214
55	13.2500	795.00	62.017	2.09	5.06	1453
56	15.7500	945.00	61.517	2.10	5.08	1441
57	19.1250	1147.50	60.817	2.12	5.13	1520
58	22.2500	1335.00	60.250	2.13	5.15	1458
59	29.2500	1755.00	59.815	2.14	5.18	1682
60	32.1250	1927.50	60.683	2.12	5.13	1650

Run No.	h	$\frac{hD}{k}$	$\frac{C_p \mu}{k}$	$\left(\frac{C_p \mu}{k}\right)^{0.3}$	$\frac{\frac{hD}{k}}{\left(\frac{C_p \mu}{k}\right)^{0.3}}$	$\frac{DG}{\mu}$
Units	B.T.U./(hr.)(sq. ft.)(°F.)	None	None	None	None	None
1	79.4	79.2	31.6	2.82	28.1	3060
2	96.5	96.3	30.9	2.80	34.4	4850
3	62.0	61.9	32.5	2.84	21.8	2670
4	48.5	48.4	30.9	2.80	17.3	1480
5	91.5	91.3	29.3	2.76	33.0	4340
6	48.0	47.9	32.7	2.84	16.9	2150
7	139.2	139.0	30.1	2.78	50.0	5410
8	34.6	34.5	32.0	2.83	12.2	1880
9	128.8	128.6	29.9	2.77	46.4	5430
10	61.4	61.3	30.3	2.78	22.1	2590
11	76.4	76.2	34.1	2.88	26.4	4710
12	31.8	31.7	31.8	2.83	11.2	2720
13	49.2	49.1	31.5	2.82	17.4	1320
14	31.0	30.9	31.1	2.80	11.0	2740
15	36.3	36.2	31.9	2.83	12.8	3510
16	63.6	63.5	33.0	2.86	22.3	4720
17	31.3	31.2	31.5	2.82	11.1	2670
18	48.0	47.9	31.2	2.81	17.0	3100
19	55.6	55.5	30.3	2.78	20.0	2750
20	18.7	18.7	30.8	2.79	6.7	1900
21	21.6	21.5	31.5	2.82	7.6	2320
22	25.0	24.9	31.8	2.83	8.8	2810
23	29.2	29.1	32.0	2.83	10.3	3860
24	42.6	42.5	32.1	2.83	15.0	4940
25	73.9	73.7	31.4	2.82	26.2	5330
26	60.9	60.8	32.0	2.83	21.5	4170
27	45.5	45.4	30.5	2.79	16.3	4040
28	39.6	39.5	32.3	2.84	13.9	3360
29	34.4	34.3	32.3	2.84	12.1	2800
30	31.6	31.5	32.0	2.83	11.1	2690

Run No.	h	$\frac{hD}{k}$	$\frac{C_p\mu}{k}$	$\left(\frac{C_p\mu}{k}\right)^{0.3}$	$\frac{hD/k}{\left(\frac{C_p\mu}{k}\right)^{0.3}}$	$\frac{DG}{\mu}$
Units	B.T.U./(hr.)(sq. ft.)(°F.)	None	None	None	None	None
31	25.8	25.7	32.0	2.83	9.1	2430
32	22.9	22.8	31.9	2.83	8.1	2210
33	35.4	35.3	33.5	2.87	12.3	4950
34	53.1	53.0	29.2	2.75	19.2	5690
35	152.4	152.1	29.9	2.77	54.9	5360
36	108.0	107.8	29.6	2.76	39.0	4230
37	116.3	116.1	29.6	2.76	42.1	4080
38	103.2	103.0	29.3	2.75	37.5	3160
39	60.1	60.0	28.7	2.74	21.9	2540
40	58.6	58.5	28.5	2.73	21.4	2190
41	30.8	30.7	28.8	2.74	11.2	1550
42	34.6	34.5	29.8	2.77	12.5	1850
43	35.7	35.6	30.2	2.78	12.8	2140
44	47.1	47.0	30.5	2.79	16.8	2550
45	50.5	50.4	31.1	2.81	17.9	2750
46	58.7	58.6	31.5	2.82	20.8	2930
47	62.8	62.7	31.8	2.83	22.2	3240
48	77.0	76.8	31.9	2.83	27.2	3640
49	98.0	97.8	32.0	2.83	34.5	4090
50	118.6	118.3	32.3	2.84	41.6	4230
51	152.4	152.1	32.4	2.84	53.5	4950
52	163.0	162.7	32.6	2.84	57.2	4930
53	29.2	29.1	28.6	2.74	10.6	1410
54	28.3	28.2	28.8	2.74	10.3	1650
55	37.7	37.6	29.2	2.75	13.7	2290
56	43.8	43.7	29.3	2.75	15.9	2710
57	51.6	51.5	29.6	2.76	18.7	3250
58	56.2	56.1	29.7	2.76	20.3	3760
59	80.8	80.6	29.8	2.77	29.1	4930
60	91.5	91.3	29.6	2.76	33.1	5460

have used the temperatures at the ends of the pipe in the calculation of the coefficients, but since, in the present investigation, the temperature rise of the refrigerating fluid outside the pipe was very small, and due largely to heat losses in the apparatus, this average temperature was assumed to represent constant conditions along the length of the pipe, and was further assumed to be suitable for the calculation of average coefficients. Δt_1 is the difference between the test liquid inlet temperature and the average pipe temperature, while Δt_2 is the difference between the test liquid outlet temperature and the average pipe temperature. The logarithmic mean temperature difference is the logarithmic mean of Δt_1 and Δt_2. The test liquid Δt represents the decrease in temperature brought about by passage through the test section. The weight of the test liquid per minute was either weighed directly or else obtained by a simple multiplication, when the period over which the sample was collected was less than a minute. The weight of test liquid per hour was obtained by multiplying the weight per minute by 60. The average test liquid temperature is the average of the inlet and outlet temperatures, and was calculated in order that an average viscosity could be obtained. The viscosity in centipoises was taken from the curve at the average test liquid temperature, while the absolute viscosity was obtained by conversion to pounds per foot-hour.

The quantity of heat transferred per hour was calculated as the product of the temperature change, the specific heat, and the weight per hour. The film coefficient, h, was calculated by use of Newton's law— $Q/\theta = h\ A\ \Delta t$; in which Q/θ is the quantity of heat transferred per hour, h is the film coefficient, A is the heat transfer area, and Δt

is the temperature differential. In this case, the area was taken as the inside area of exactly six feet of standard one-inch pipe, and the logarithmic mean temperature differential was used as Δt.

The values of the Nusselt and Prandtl numbers were calculated using the film coefficients and the physical properties of the fluid. The Reynolds number, as previously stated, was calculated directly from the weight rate of flow as $4W/\pi D\mu$, where W and μ have the units pounds per hour and pounds per foot-hour respectively.

A sample calculation, using the data of Run No. ~~1~~ 34 is as follows:

<u>Δt_1</u>:

$$\Delta t_1 = t_1 - t$$

(Terms are defined in the list of symbols.)

$$= 62.833 - 45.801$$

$$\Delta t_1 = 17.032^\circ \text{ F.}$$

<u>Δt_2</u>:

$$\Delta t_2 = t_2 - t$$

$$= 61.400 - 45.801$$

$$\Delta t_2 = 15.599^\circ \text{ F.}$$

<u>Log Mean Δt</u>:

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

$$= \frac{17.032 - 15.599}{2.303 \log \frac{17.032}{15.599}}$$

$$\Delta t_m = 16.321^\circ \text{ F.}$$

<u>Test Liquid Δt</u>:

$$\Delta t = t_1 - t_2$$

$$= 62.833 - 61.400$$

$$\Delta t = 1.433^\circ \text{ F.}$$

Weight Rate of Flow: $W = w \times 60$ Where w is the weight of test liquid per minute

$= 33.00 \times 60$

$W = 1980.00$ lbs./hr.

Average Test Liquid Temperature:

$$t_a = \frac{t_1 + t_2}{2}$$

$$= \frac{62.833 + 61.400}{2}$$

$$t_a = 62.117^\circ \text{ F.}$$

Absolute Viscosity: $\mu = z \times 2.42$

$= 2.09 \times 2.42$

$\mu = 5.06$ lbs./ft.-hr.

Heat Transferred: $Q = W \times \Delta t \times C_p$

$= 1980.0 \times 1.433 \times 0.504$

$Q = 1430$ B.T.U./hr.

Film Coefficient: $Q/\theta = h \times A \times \Delta t$

$$h = \frac{Q}{A \times \Delta t}$$

$$= \frac{1430}{1.65 \times 16.321}$$

$h = 53.1$ B.T.U./(hr.)(sq. ft.)(°F.

Nusselt Number:

$$Nu = \frac{h \times D}{k}$$

$$= \frac{53.1 \times 0.0874}{0.0875}$$

$$Nu = 53.0$$

Prandtl Number:

$$Pr = \frac{C_p \times \mu}{k}$$

$$= \frac{0.504 \times 5.06}{0.0875}$$

$$Pr = 29.2$$

<u>(Prandtl Number)$^{0.3}$</u>:

$$Pr^{0.3} = \text{antilog } (0.3 \times \log Pr)$$
$$= \text{antilog } (0.3 \times \log 29.2)$$
$$Pr^{0.3} = 2.75$$

<u>Ratio of Nusselt to Prandtl Number</u>:

$$\frac{\frac{hD}{k}}{\left(\frac{C_p\mu}{k}\right)^{0.3}} = \frac{53.0}{2.75}$$
$$= 19.2$$

<u>Reynolds Number</u>:

$$Re = \frac{4 \times W}{\pi \times D \times \mu}$$
$$= \frac{4 \times 1980.00}{3.1416 \times 0.0874 \times 5.06}$$
$$Re = 5690$$

The experimental data is shown graphically by Curve I, page 38, where $(hD/k)/(C_p\mu/k)^{0.3}$ is plotted against DG/μ. The straight lines represent the Dittus and Boelter equation and the empirical equation derived from the present data. The supplementary points are taken from the kerosene data of Sherwood and Petrie.

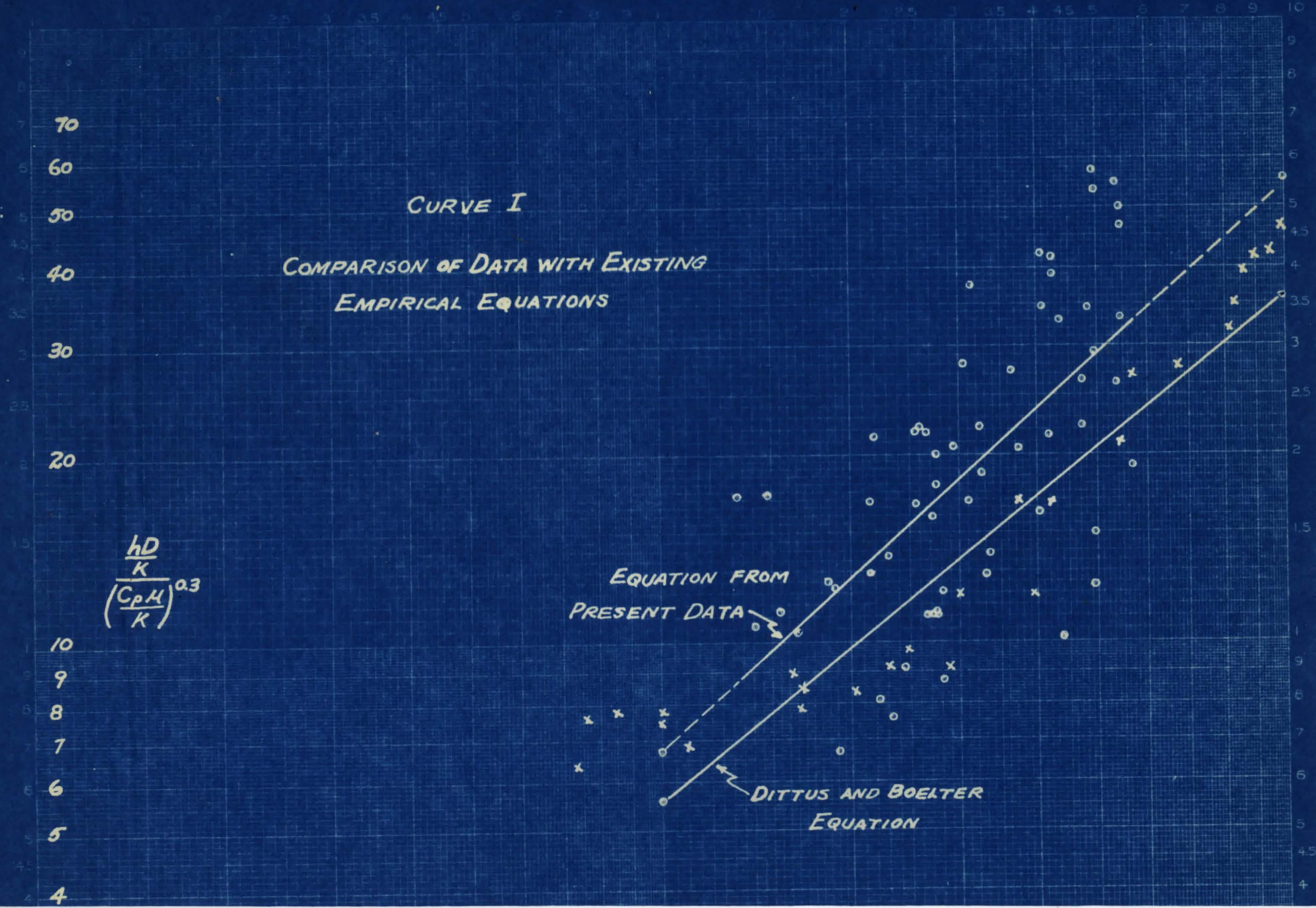
The method of least squares was used to obtain the equation of the straight line best correlating the data, and the equation obtained was:

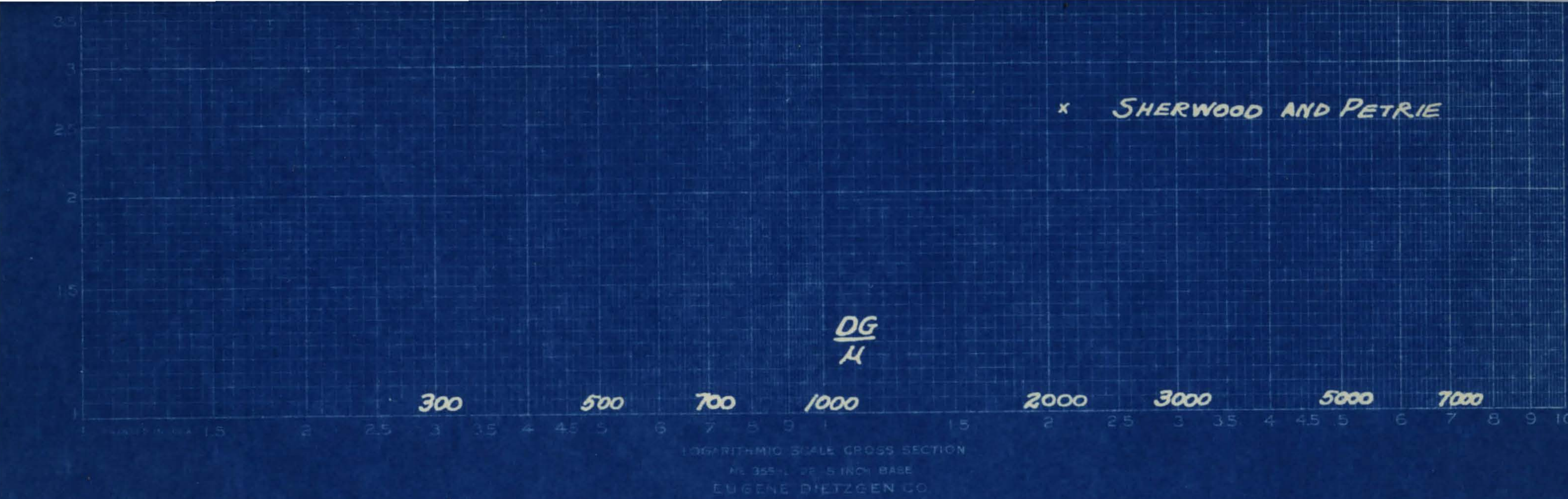
$$\frac{hD}{k} = 0.0121 \left(\frac{DG}{\mu}\right)^{0.915} \left(\frac{C_p\mu}{k}\right)^{0.3}$$

as compared with the Dittus and Boelter equation:

$$\frac{hD}{k} = 0.026 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{C_p\mu}{k}\right)^{0.3}$$

The coefficient of correlation, which is a measure of the tendency of two variables to vary together, was found to be 0.652, as compared with a value of 1.000 for perfect correlation.


CURVE I
COMPARISON OF DATA WITH EXISTING EMPIRICAL EQUATIONS
$\dfrac{\frac{hD}{K}}{\left(\frac{C_p\mu}{K}\right)^{0.3}}$
70
60
50
40
30
20
10
9
8
7
6
5
4
EQUATION FROM PRESENT DATA
DITTUS AND BOELTER EQUATION


× SHERWOOD AND PETRIE
DG/μ
300
500
700
1000
2000
3000
5000
7000
LOGARITHMIC SCALE CROSS SECTION
EUGENE DIETZGEN CO.

IV. CONCLUSIONS

Heat transfer coefficients were determined for Reynolds numbers ranging from about 1500 to 5500. The groups, $(hD/k)/(C_p \mu/k)^{0.3}$ were plotted against the Reynolds numbers, and the method of least squares was used to obtain the equation of the best straight line through the points. It was found that the data could be correlated by use of the equation:

$$\frac{hD}{k} = 0.0121 \left(\frac{DG}{\mu}\right)^{0.915} \left(\frac{C_p \mu}{k}\right)^{0.3}$$

which compares very favorably with the Dittus and Boelter equation:

$$\frac{hD}{k} = 0.026 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{C_p \mu}{k}\right)^{0.3}$$

As predicted by Sherwood and Petrie, the slope of the empirical curve is greater than 0.8, by about 14 %.

The value of the coefficient of correlation, 0.652, indicates that the Nusselt equation may be satisfactorily used for the correlation of heat transfer data in the viscous and critical regions, as well as in regions of turbulent flow. It was found that the present data could not be satisfactorily correlated by means of the Graetz relation.

The most accurate experimental data is obtained only when equilibrium conditions are established before the readings are taken. The installation of an orfice meter in the test liquid return line would result in greater accuracy in the determination of the rate of flow. A check on the quantity of heat transferred could be obtained if the rate of flow and the temperature change of the refrigerant were determined.

The author is indebted to Wilson R. Barnes, who aided with the experimental work in this investigation.

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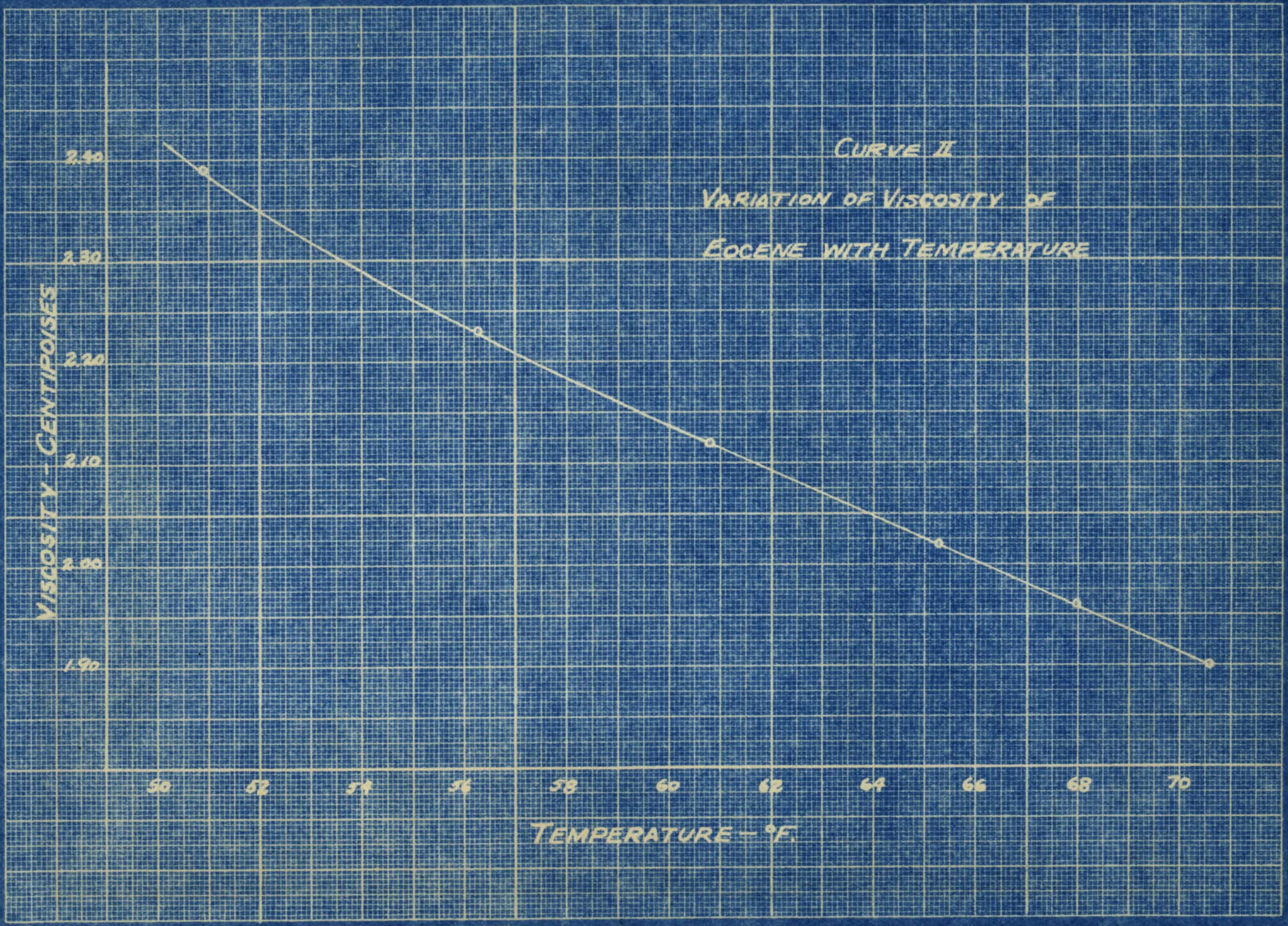
APPENDIX

Table III

DISTILLATION RANGES OF EOCENE

Percent Distilled	Temperature, °F. New Sample	Used Sample
0	320	332
10	367	368
20	376	384
30	386	407
40	398	424
50	424	439
60	448	455
70	466	471
80	484	490
90	509	510



CURVE II
VARIATION OF VISCOSITY OF
EOCENE WITH TEMPERATURE
VISCOSITY - CENTIPOISES
2.40
2.30
2.20
2.10
2.00
1.90
50
52
54
56
58
60
62
64
66
68
70
TEMPERATURE - °F.