THE RELATION OF HEAT TRANSFER TO POWER CONSUMPTION OF LIQUIDS IN TURBULENT FLOW

BY

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ABSTRACT

The relationship between the heat transfer ability and the power input requirement of liquid coolants in a turbulent heat transfer condition was investigated. A correlation in terms of the liquid physical properties and the required volumetric flow was proposed and the use of this correlation in selection of liquids as coolant was discussed. The optimum condition for a high heat transfer/power input ratio was also discussed.

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INTRODUCTION

When heat transfer is applied in a system where space, weight or power supply is limited, the mode of heat transfer and the selection of coolants are critical. In view of the numerous coolants available, a quick concise method of screening coolants by comparing their heat transfer properties would be highly desirable.

The selection of coolants, of course, depends on many other properties aside from their heat transfer properties. Often, these properties, such as toxicity, flammability, and many others have to be considered first before their merits as coolants are evaluated.

The purpose of the present investigation is to find a simple correlation between the heat transfer characteristics of liquids and the power requirements to effect this heat transfer. As much as possible, the correlation should be expressed in terms of the fundamental physical properties of the liquid that relate to heat transfer. These are thermal conductivity, viscosity, specific heat, coefficient of thermal expansion and density. If the correlation prove to be valid, the selection of coolants can be facilitated because many of these physical properties are available.

THEORY

The heat transfer ability of a coolant depends on many factors, such as the heat exchangers configuration, the mode of heat transfer, and naturally, the properties of the fluid itself. In this investigation, heat transfer from a liquid in turbulent flow in a horizontal pipe was studied. Within the confinement, the coolant properties would then be the major factor in determining the heat transfer ability.

To compare the heat transfer ability of various coolants, the heat transfer coefficient is compared with the power input required per unit heat transfer area. The relation is derived below. The viscosity correction factor, $\left(\frac{\mu_w}{\mu}\right)^{0.14}$, was ignored because the variation in viscosity was small for the liquid studied.

Let Total Heat Transferred $= Q$

and Total Pressure Drop $= -\Delta p$

and Power Requirement $= (-\Delta p)V$

Now

$$f = \frac{(-\Delta p)g_c D}{2\,\bar{v}^2 L\rho}. \qquad (1)$$

Rearrange (1)

$$-\Delta p = \frac{2\,\bar{v}^2 L\rho}{g_c D}\cdot f \qquad (2)$$

Using the empirical fluid friction equation

$$f = 0.046\left(\frac{D\bar{v}\rho}{\mu}\right)^{-0.2} \qquad (3)$$

and substituting (3) into (2)

We have $$-\Delta p = \frac{2 \times 0.046\bar{v}^2 L\rho}{g_c D} \left(\frac{D\bar{v}\rho}{\mu}\right)^{-0.2} \qquad (4)$$

Now, in a fixed system, L, g_c, D are constants, hence,

$$-\Delta p = K_1 \ \bar{v}^2 \rho \left(\frac{D\bar{v}\rho}{\mu}\right)^{-0.2} \qquad (5)$$

with

$$K_1 = \frac{2 \times 0.046L}{g_c D} \qquad (6)$$

Since $$Q = h \ A_T \Delta T \qquad (7)$$

and

$$\frac{hD}{k} = 0.023 \left(\frac{D\bar{v}\rho}{\mu}\right)^{0.8} \left(\frac{C_p \mu}{k}\right)^{1/3} \qquad (8)$$

then

$$Q = 0.023 \left(\frac{D\bar{v}\rho}{\mu}\right)^{0.8} \left(\frac{C_p \mu}{k}\right)^{1/3} \frac{k}{D} A_T \Delta T \qquad (9)$$

If the Total Heat Transferred is divided by Total Power Requirement

$$\frac{Q}{(-\Delta p)\bar{V}}$$

and

$$\frac{Q}{(-\Delta p)\bar{V}} = \frac{0.023\left(\frac{Dv\rho}{\mu}\right)^{0.8}\left(\frac{C_p\mu}{k}\right)^{1/3}\frac{k}{D}A_T\Delta T}{K_1\ \bar{v}^2\left(\frac{D\bar{v}\rho}{\mu}\right)^{-0.2}\bar{V}} \quad (10)$$

Rearrange (10) gives

$$\frac{Q}{(-\Delta p)\bar{V}} = K_2\left(\frac{Dv\rho}{\mu}\right)\left(\frac{C_p\mu}{k}\right)^{1/3}\left(\frac{k}{\bar{v}^2}\right)\frac{A_T\Delta T}{\bar{V}} \quad (11)$$

and

$$\frac{Q}{(-\Delta p)\bar{V}} = K_2\Delta T\ A_T\ D\left(\frac{k}{\mu}\right)\left(\frac{C_p\mu}{k}\right)^{1/3}\frac{1}{\bar{V}} \quad (12)$$

Rearranging (12) gives

$$\frac{\frac{Q}{(\Delta T\ A_T)}}{(-\Delta p)\bar{V}} = K_2\ D\left(\frac{k}{\mu\bar{v}}\right)\left(\frac{C_p\mu}{k}\right)^{1/3}\frac{1}{\bar{V}} \quad (13)$$

and from (13)

$$\frac{h}{(-\Delta p)\bar{V}} = K_2 D \left(\frac{k}{\mu \bar{v}}\right) \left(\frac{C_p \mu}{k}\right)^{1/3} \frac{1}{\bar{V}} \tag{14}$$

If the power input term is converted to unit power input, P_A, by multiplying (14) by A_T,

$$\frac{h}{P_A} = K_2\, A_T\, D \left(\frac{k}{\mu \bar{v}}\right) \left(\frac{C_p \mu}{k}\right)^{1/3} \frac{1}{\bar{V}} \tag{15}$$

since

$$\bar{v} = \frac{\bar{V}}{A},$$

then

$$\frac{h}{P_A} = \frac{K_2 A_T D A \left(\frac{k}{\mu}\right) \left(\frac{C_p \mu}{k}\right)^{1/3}}{\bar{V}} \tag{16}$$

Because A_T, D and A are constants,

let

$$K_3 = K_2 \cdot A_T \cdot D \cdot A \text{ and} \tag{17}$$

From (16),

$$\frac{h}{P_A} = K_3 \frac{\left(\frac{k}{\mu}\right)\left(\frac{C_p \mu}{k}\right)^{1/3}}{\bar{V}^2} \tag{18}$$

where

$$K_3 = K_2 A_T DA$$

$$= \frac{0.023\ A_T\ DA}{K_1 D}$$

$$= \frac{g_c\ A_T\ AD}{4L}$$

$$= \frac{g_c \pi DL \frac{\pi D^2}{4} D}{4L}$$

$$= \frac{1}{16} g_c\ \pi^2\ D^4$$

Equation (18), therefore, relates the heat transfer coefficient to the unit power consumption for turbulent flow in a tube. This equation can therefore be used to screen coolants and determine which will give the highest heat transfer to power consumption ratio

DESCRIPTION OF APPARATUS

The apparatus is shown in Figure 1 and Figure 2. The test section consists of a 36 inch length of schedule 80, 1/4 inch, 304 stainless steel pipe. The heat transfer takes place in this section and this section is connected to the whole system by means of flanges and bushings.

Heat is provided by means of two Briskeat Samox Fiber insulated heating tapes wrapped around the pipe. Each of these tapes measures 1 inch by 48 inches and has a rating of 576 watts at 115 volts. The electrical power supply to the tape is controlled by two Powerstats with a variable output range of 0 to 140 volts. The power supply is measured by means of two Weston voltmeters and two Weston ammeters. The whole test section is insulated with one layer of asbestos cloth and a one and a half inch thick layer of Air Cell pipe insulation.

The temperature of the pipe wall is monitored by five thermocouples spot welded onto the pipe wall. Two other thermocouples, inserted into the flowing channel, measure the inlet and exit fluid temperature of the test section. All thermocouples are made of 30 gauge, glass fiber insulted, Iron-Constantan thermocouple wires. The temperature readings are measured by a Leeds and Northrop No. 8662 portable precision Potentiometer through a 15 point selector switch.

The pressure measurement is taken by means of an 80 inch manometer. The pressure taps are made through the testing bushings and are connected to the manometer, by copper tubings. Fluids used in the manometer are Meriam Fluid No. 3 for the water runs and Mercury for the "Freon 113" and carbon tetrachloride runs.

An Eastern centrifugal pump, Model D-11, is used to circulate the fluids and the flow rate is monitored by a Fisher and Porter Model 10A3565A flowrator rated for 3.55 GPM for water. Other accessories include a shell and tube heat exchanger to cool the fluid from the heating section and a one gallon size reservoir to store the fluid.

Figure 1: Apparatus - Test Section

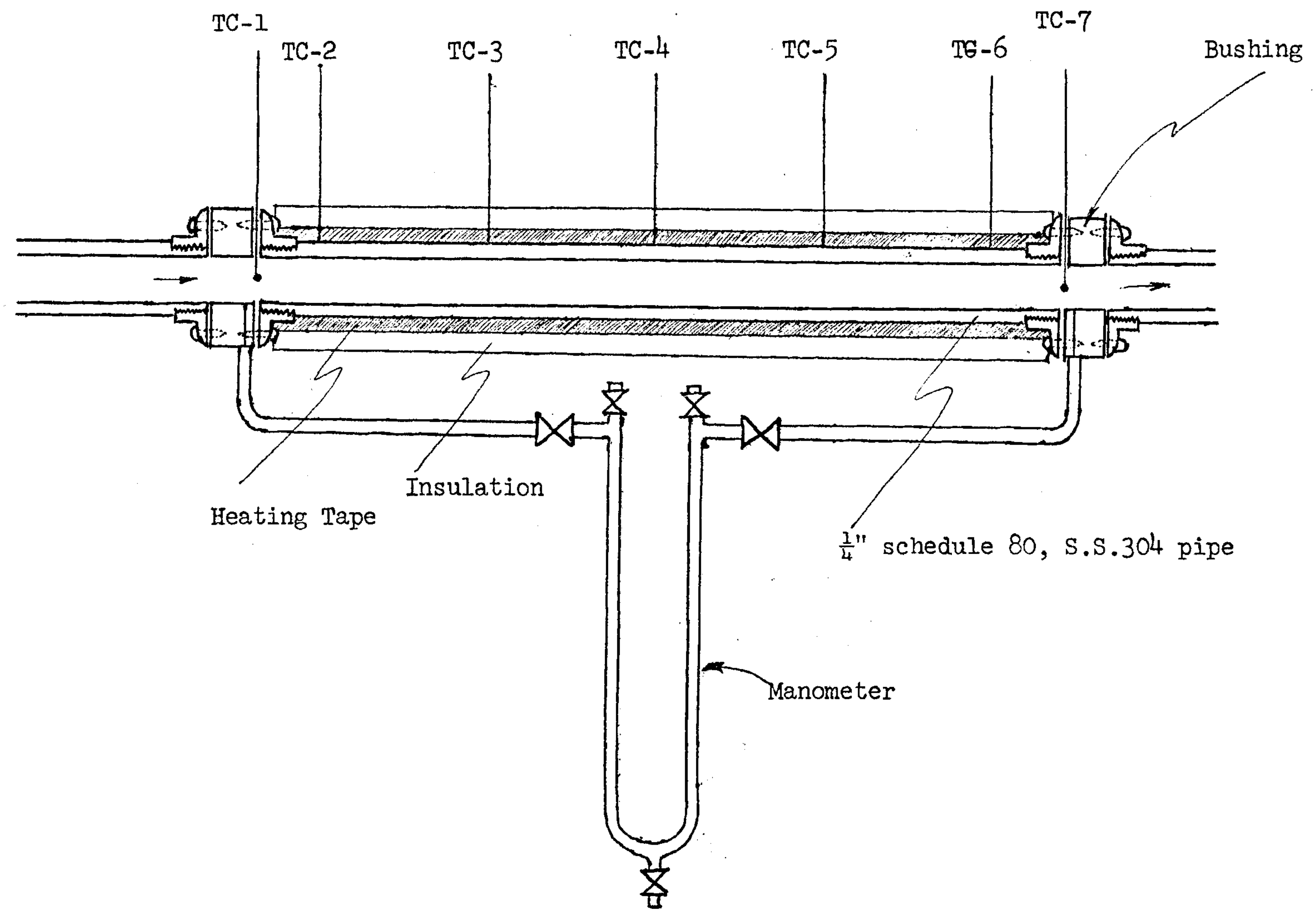
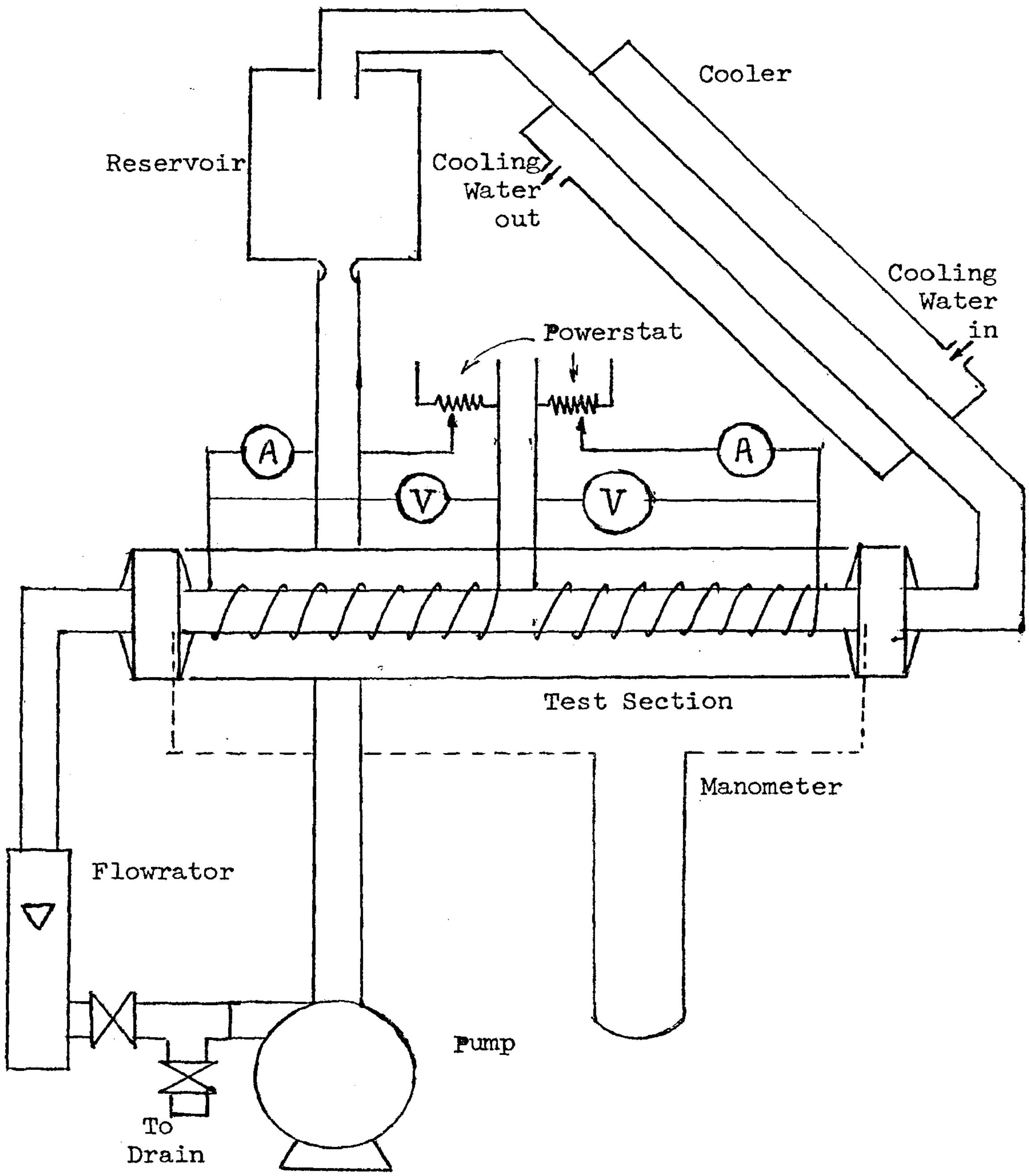


Figure 2: Apparatus - Complete Flow Sheet



EXPERIMENTAL PROCEDURE

The experimental equipment was assembled as described in the previous section and is shown in Figure 1 and Figure 2. The equipment was first set-up for the water runs. For this purpose the manometer was filled to a suitable height with Meriam Fluid No. 3 which has a specific gravity of 2.95. The rest of the manometer and the copper tubing joining the manometer and the pressure taps were filled with water and the whole pressure measuring system was checked to ensure that there was no air in the system. Once this was done, the circulation of the coolant was started and the Rotameter was calibrated (Figure 9).

Prior to the setting up of the testing section each thermocouple was calibrated by comparing with a mercury thermometer in a water bath. Once the test section was assembled, the temperature measuring system was checked for readings with the whole system at room temperature. The potentiometer was also checked. The voltmeters and ammeters had been standardized but the zero point was corrected prior to each run.

To start a run, the following procedure was followed:

1. Check the whole system visually.
2. Fill the reservoir with fluid to be tested.
3. If everything appears in order, start the pump with the flow rate set at a minimum by means of a needle valve.
4. If the system appears correct, increase the flow gradually to the desired rate.
5. Check the flowrator and the manometer at this point.
6. Turn on the cooling water and set the desired flow rate by means of a needle valve and a flowrator.
7. Turn on the electrical supply to the heating tapes at a low voltage supply.
8. Adjust the potentiometer, then check the thermocouples. A defective thermocouple can be detected by drifting potentiometer readings or a reading that stays the same even when the test section warms up.
9. Increase the electrical supply to the voltage desired.
10. Take pressure, temperature, flow rate and power supply readings at convenient time intervals.

11. A steady state generally is reached within 2--3 hours.
12. Frequent checking should be made on the fluid flow rate and the cooling water flow rate to ensure a steady state condition for each run.

EXPERIMENTAL RESULTS AND DISCUSSION

Altogether twenty experimental runs were made, of which seven runs were made with water as the coolant, eight runs with "Freon 113" as the coolant and five runs with carbon tetrachloride. An attempt was also made to use ethylene glycol as the coolant but the pumping required to make the heat transfer run in the turbulent flow region could not be achieved.

As a first step, the experimental data were checked against published results. First of all, the Fanning friction factor, f, was calculated from the relationships:

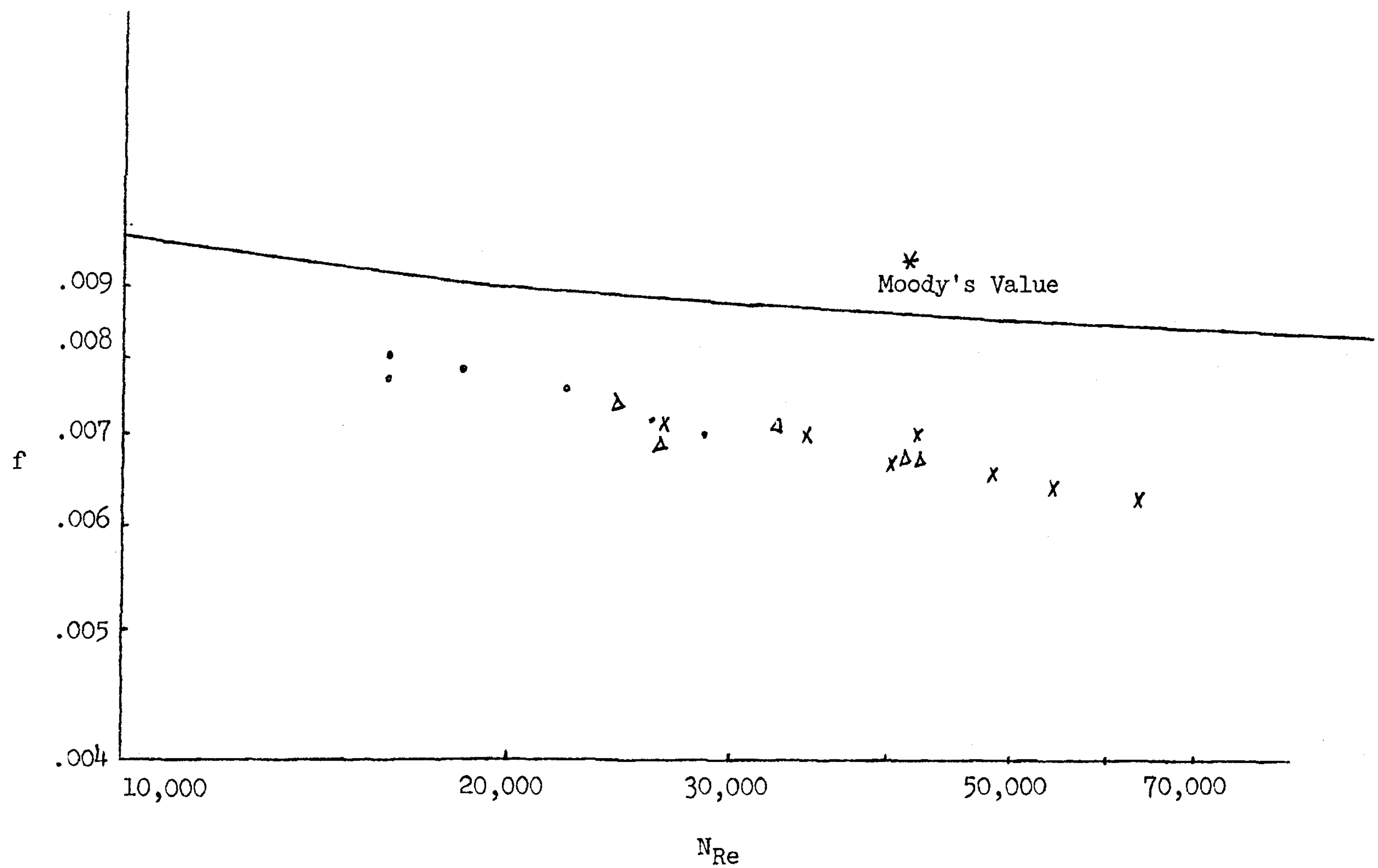
$$f = \frac{(-\Delta P) g_c D}{2 \bar{v}^2 \rho L} \qquad (1)$$

and

$$(-\Delta P) = R\,(\rho_A - \rho_B) \qquad (20)$$

where, "R", the measured manometer reading, and $\bar{v}$, coolant linear velocity, can be obtained from the run data. The Fanning friction factor was then plotted against the Reynolds number and compared to Moody's (10) data, (as in Figure 3). The results were also tabulated (Table 1) and the average deviation calculated.

Figure 3: Fanning Friction Factor, f, vs. N_{Re}



* Moody used a friction factor which is four times larger than fanning friction factor, its value has been adjusted here.

Table 1

COMPARISON OF f FROM EXPERIMENTAL DATA AND f CALCULATED FROM N_{Re} VALUES

COOLANT	RUN#	N_{Re}	f/2 (From N_{Re})	f/2 Experimental	% DEVIATION
	1	16115	.0092	.0079	16.5
	2	16211	.0090	.0080	12.5
WATER	3	18505	.0090	.0079	13.0
	4	22263	.0088	.0075	14.2
	5	28400	.0085	.0070	18.5
	6	26100	.0085	.0071	16.1
	8	26617	.0088	.0070	19.6
	9	34355	.0085	.0069	18.6
FREON 113	10	40482	.0084	.0067	20.7
	11	42276	.0084	.0070	16.7
	12	48369	.0083	.0065	21.8
	13	53983	.0083	.0064	23.7
	14	54230	.0083	.0064	23.7
	15	63036	.0082	.0062	24.7
	16	24744	.0087	.0073	15.9
CARBON TETRA.	17	26254	.0087	.0068	21.4
	18	32409	.0085	.0071	17.3
	19	42656	.0084	.0067	20.7
	20	42028	.0084	.0067	20.2

AVERAGE DEVIATION, WATER 15.1
FREON 113 21.2
$C\,Cl_4$ 19.1
AVERAGE DEVIATION, OVERALL 18.7

Secondly, the " j " factor for heat transfer, j_H, was calculated. The " j " factor is related to the Reynolds number by the following relation:

$$j_H = (N_{St}) \, (N_{Pr})^{2/3} \left(\frac{\mu_w}{\mu}\right)^{0.14} = 0.023 \, (N_{Re})^{-0.2} \tag{21}$$

that is:

$$j_H = \left(\frac{h}{GC_p}\right)_b \left(\frac{C_p\mu}{K}\right)^{2/3}_{t\,=\,0.5} \left(\frac{\mu_w}{\mu}\right)^{0.14} = 0.023 \left(\frac{DG}{\mu}\right)^{-0.2}_{t\,=\,0.5} \tag{22}$$

To obtain the j , the value of h must be calculated. The "h" was estimated through the relation.

$$Q = h \cdot A \cdot \Delta T$$

"Q" was calculated from the rise in temperature of the coolant by the relation.

$$Q = w \, C_p \Delta t_b \tag{23}$$

To estimate Δt_b, first the average of the outside pipe wall temperature was found based on the readings of thermocouples two through six. The inside pipe wall temperature was then evaluated, after calculating the temperature drop through the pipe wall. From the inside pipe wall temperature and the fluid temperature, ΔT was estimated. The arithmetic mean instead of the log mean temperature difference was used in this case because for most of the runs the coolant temperature rise was not large and the logarithmic mean can lead to large calculational errors. The heat transfer coefficient h, was then calculated and the value of j_H was derived from the relation.

$$j_H = \left(\frac{h}{GC_p}\right)_b \left(\frac{C_p \mu}{k}\right)^{2/3} \left(\frac{\mu_w}{\mu}\right)^{0.14} \tag{22}$$

To compare with the published results, j_H was correlated against N_{Re} (Figure 4), and the graph was compared with that in Foust (1). The values were also tabulated (Table 2) and the deviation calculated.

As a third check, the heat input in each run was calculated from the voltmeter and ammeter readings recorded. This heat input value was then compared with the value Q, the heat absorbed by the coolant, and then the heat loss was calculated and is tabulated in Table 3.

Figure 4: j_H vs. N_{Re}

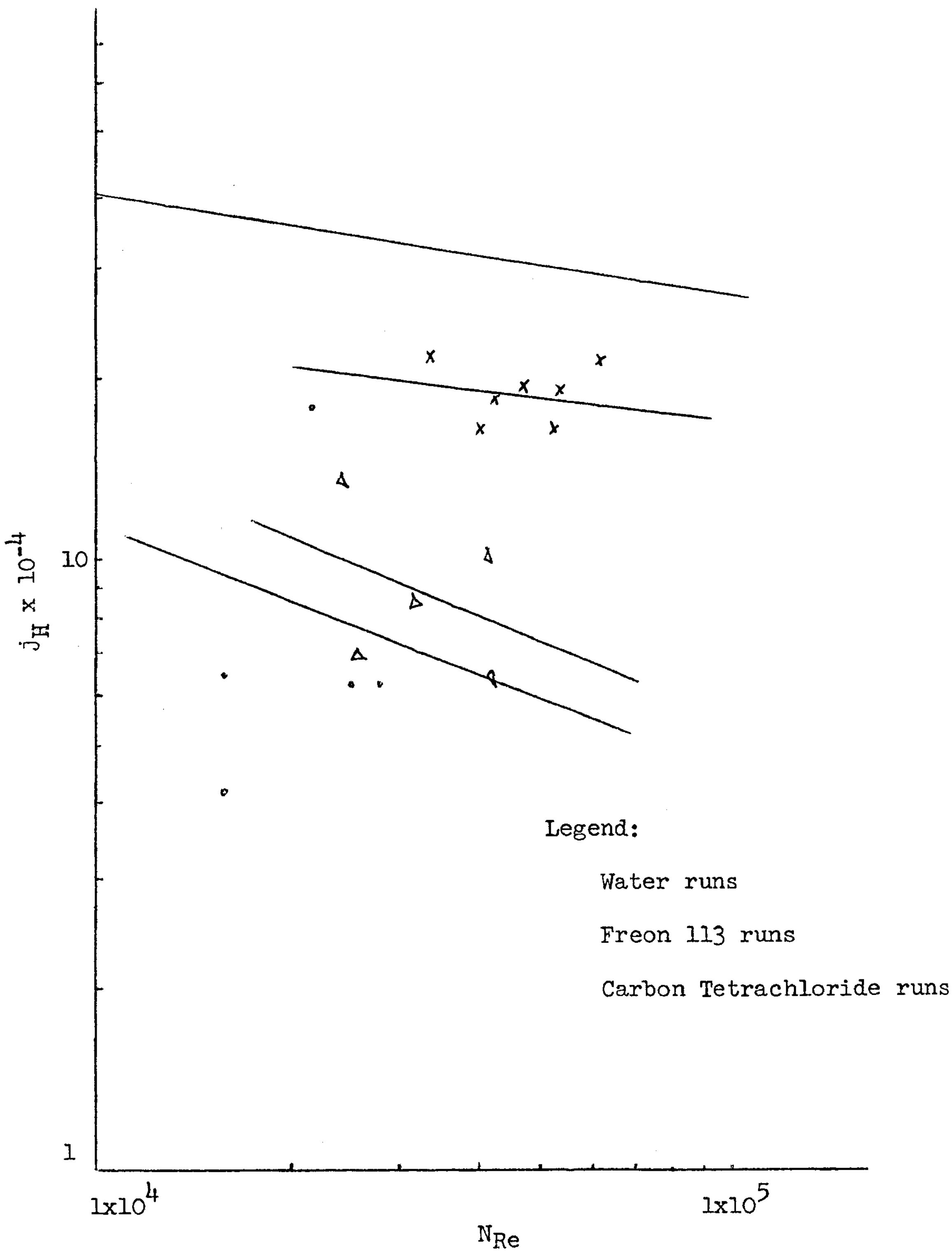


Table 2

COMPARISON OF 'j_H' EXPERIMENTAL WITH 'j_H' CALCULATED

COOLANT	RUN #	N_{Re}	'j_H' Experimental	'j_H' Calculated	% DEVIATION
WATER	1	16115	.00064	.00331	80.8
	2	16211	.000408	.00331	87.9
	3	18505	.001138	.00322	64.7
	4	22263	.001391	.00310	44.9
	5	28400	.00061	.00297	79.3
	6	26100	.000618	.00310	79.6
FREON 113	7	26617	.00942	.00300	
	9	34355	.00213	.00285	25.4
	10	40482	.00162	.00276	41.4
	11	42276	.00182	.00273	33.4
	12	48369	.00191	.00266	28.4
	13	53983	.00164	.00260	36.9
	14	54230	.00189	.00260	27.3
	15	63036	.00209	.00252	16.7
CARBON TETRA-CHLO-RIDE	16	24744	.00133	.00171	56.2
	17	26254	.00069	.00301	77.1
	18	32409	.00085	.00288	70.3
	19	42656	.00063	.00273	77.1
	20	42028	.00100	.00273	63.3

AVERAGE DEVIATION: WATER	76.2
FREON 113	29.9
$C\,Cl_4$	68.8
AVERAGE DEVIATION, OVERALL	53.2

Table 3

COMPARISON OF HEAT OUTPUT BY HEATING TAPES AND HEAT TRANSFERRED TO COOLANT

COOLANT	RUN #	HEAT OUTPUT	HEAT TRANS.	% HEAT LOSS
WATER	1	384.9	176	54.3
	2	1085	353	67.3
	3	2130	1397	34.5
	4	3439	2896	15.6
	5	3439	1788	48.1
	6	3439	1733	49.7
FREON	8	696	740	-
	9	696	570	18.1
	10	1126	900	20.1
	11	1196	1290	24.0
	12	1696	1345	20.7
	13	1696	1250	26.4
	14	1696	1390	18.0
	15	2334	1965	15.8
CARBON TETRA.	16	812	615	24.2
	17	812	385	52.6
	18	1324	734	44.5
	19	1324	625	52.6
	20	1851	1177	36.4

AVERAGE HEAT LOSS%: WATER	45.3
FREON 113	20.5
$C\ Cl_4$	42.1
OVERALL	34.7

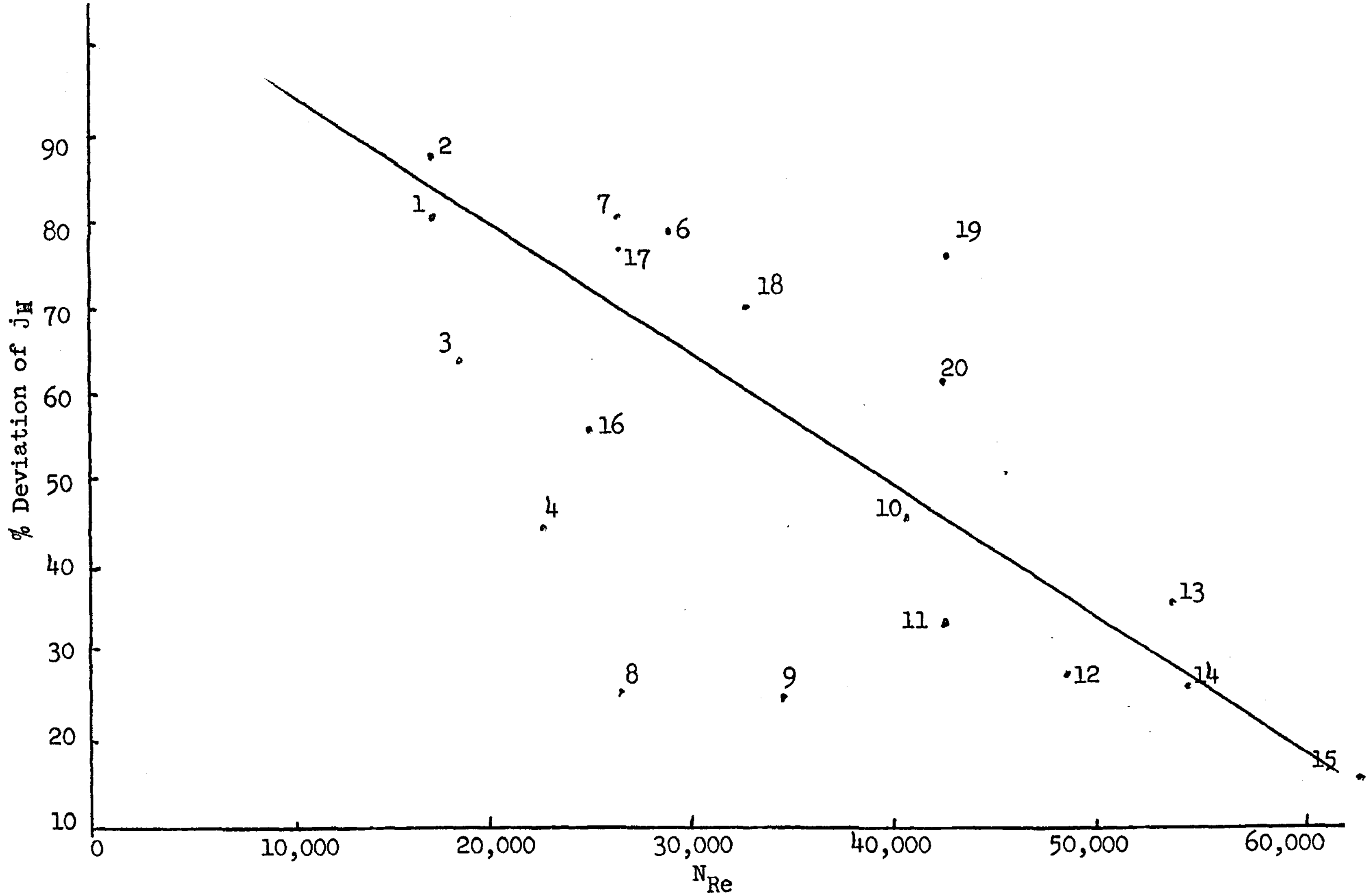
For the three liquids, it can be seen from Table 1 that the pressure measuring system was operating fairly consistently. The experimental friction factors measured were approximately nineteen per cent less than those of the values calculated from their corresponding Reynolds number. Those values were calculated from the published plot of "f" vs "N_{Re}" (Foust (1)). The roughness factor was taken to be .00015 (Moody (10)). The water runs gave the best result with fifteen percent average deviation.

The "j" factor correlation was much more different to interpret. The indication was that the system worked well with the "Freon 113" runs. An average deviation of approximately thirty percent is within the reasonable range. However, the deviations of the water and carbon tetrachloride runs were much too large. The most probable source of error was from the temperature measuring devices. First of all, the bulk temperature of the coolants was measured with two thermocouples placed into the coolant flowing channel at each end of the test section. The thermocouples were placed in such a position that the junctions were near the center of the flowing channel. Consequently, the temperatures measured were really the temperature of the turbulent region. The temperature thus measured at the exit end of the test section may be lower than the true bulk temperature. Theoretically, as the Reynolds number increases, turbulent flow is developed more fully and the deviation in temperature should be less, and the deviation in the "j" factor would in turn be smaller. A plot of percent deviation of the j_H factor against the Reynolds number bears out this statement (Figure 5).

Another source of error can be traced to the measuring of the pipe wall temperature. The measurements were effected by spot welding five thermocouples, evenly spaced, from one end of the test section to the other end. The wall temperature was taken to be the average of the five temperatures recorded. The test section was heated by two heating tapes wound around the pipe. It is conceivable that there exists a discrepancy between the average of the measured temperatures and the true average temperature of the pipe wall. This, coupled with the corrosion of the thermocouple wire, especially in the water runs could result in a "bad junction" and faulty readings.

A comparison of heat input as measured from the voltmeter and ammeter readings and as calculated from the temperature rise of the coolant points strongly to the fluid temperature as the source of error. A

Figure 5: % Deviation of Experimental j_H from theoretical j_H vs. N_{Re}



50 percent loss of heat through insulation seems to be high.

The magnitude of the deviation could also be affected by the magnitude of the bulk temperature rise of the coolant through the tube section. For example, if the bulk temperature rose by $0.5^{o}F$ and assuming the error of bulk temperature rise measured is $0.1^{o}F$, the deviation is twenty percent. Meanwhile, for a bulk temperature rise of $4^{o}F$, with the same measuring error, the deviation is only two and a half percent.

Since data were available, the experimental results were treated with some of the well known heat transfer and momentum transfer analogies. First, the Reynolds Analogy was tested. The results are shown as in Figure 6 by plotting 2fL/D vs. $\ln \frac{t_w - t_{b1}}{t_w - t_{b2}}$ and also in Table 4 where the Stanton number and the f/2 values were tabulated and compared. The correlations used were given by Knudsen (3) and are:

$$\frac{2fL}{D} = \ln \frac{t_w - t_{b1}}{t_w - t_{b2}} \qquad (24)$$

and

$$\frac{h}{\bar{v} C_p \rho} = N_{St} = \frac{f}{2} \qquad (25)$$

Figure 6: Comparison of Experimental Data with Reynolds Analogy

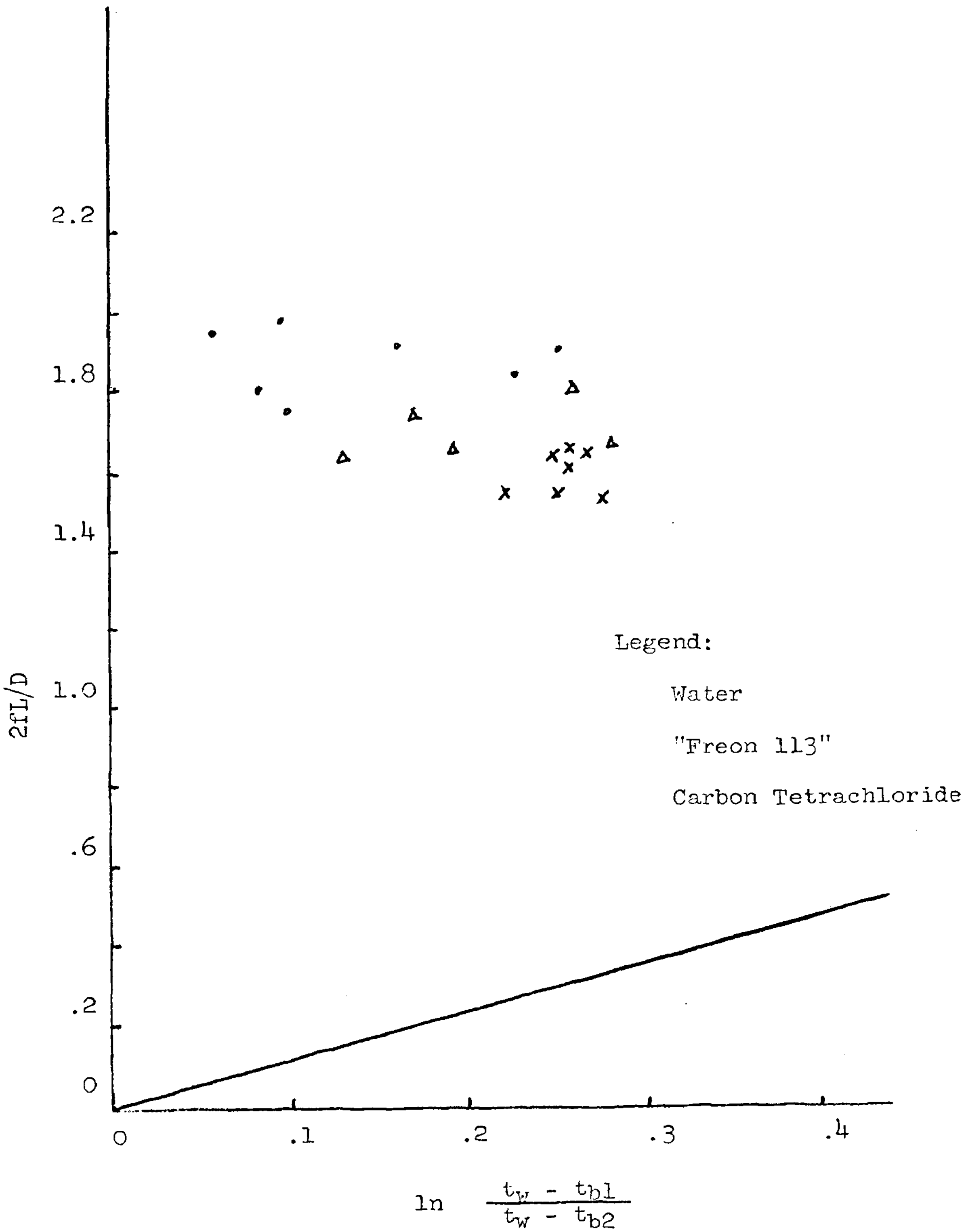


Table 4

COMPARISON OF EXPERIMENTAL DATA WITH REYNOLDS ANALOGY

COOLANT	RUN #	N_{St}	f/2	% DEVIATION
WATER	1	.000178	.00384	95.4
	2	.000125	.00400	96.9
	3	.00038	.00388	90.2
	4	.00066	.00376	82.4
	5	.00021	.00348	94.0
	6	.0002	.00355	94.4
FREON	8	.00272	.00352	22.7
	9	.000632	.00347	81.8
	10	.000477	.00333	85.7
	11	.000568	.00335	83.0
	12	.000572	.00326	82.6
	13	.000485	.00318	84.8
	14	.000560	.00318	82.4
	15	.000635	.00310	79.5
CARBON TETRA.	16	.000546	.00366	85.1
	17	.000288	.00342	91.5
	18	.000363	.00354	89.7
	19	.000265	.00333	92.0
	20	.000419	.00335	87.5

AVERAGE DEVIATION: WATER 92.2
FREON 113 75.3
$C\ Cl_4$ 89.2
OVERALL 84.7

The average deviation of the N_{St} to the f/2 value was very large, however, large deviations from the Reynolds Analogy are not uncommon as shown by Knudsen (3).

The Prandtl Analogy is represented by the following correlation (Knudsen (3)).

$$\ln \frac{t_w - t_{b1}}{t_w = t_{b2}} = \frac{2fL}{D} \cdot \frac{1}{1 + 5 \sqrt{f/2}\,(N_{Pr}-1)} \tag{26}$$

and

$$\frac{h}{C_p \rho \bar{v}} = N_{St} = \frac{f/2}{1 + 5\sqrt{f/2}\,(N_{Pr}-1)} \tag{27}$$

To apply the analogy to the experimental data, the value of $\ln \dfrac{t_w - t_{b1}}{t_w - t_{b2}}$ was plotted against the right hand side of the equation (26) in Figure 7. A tabulated comparison of the N_{St} and $\dfrac{f/2}{1 + 5\sqrt{f/2}\,(N_{Pr}-1)}$ is also shown in Table 5 along with the calculated deviation. As with Reynold's analogy large deviations are shown.

The third analogy used was Colburn's Analogy. The results were presented both in graphical and tabulated form. The correlation was given by Knudsen (3) as:

Figure 7: Prandtl Analogy - Experimental Data and Published Value

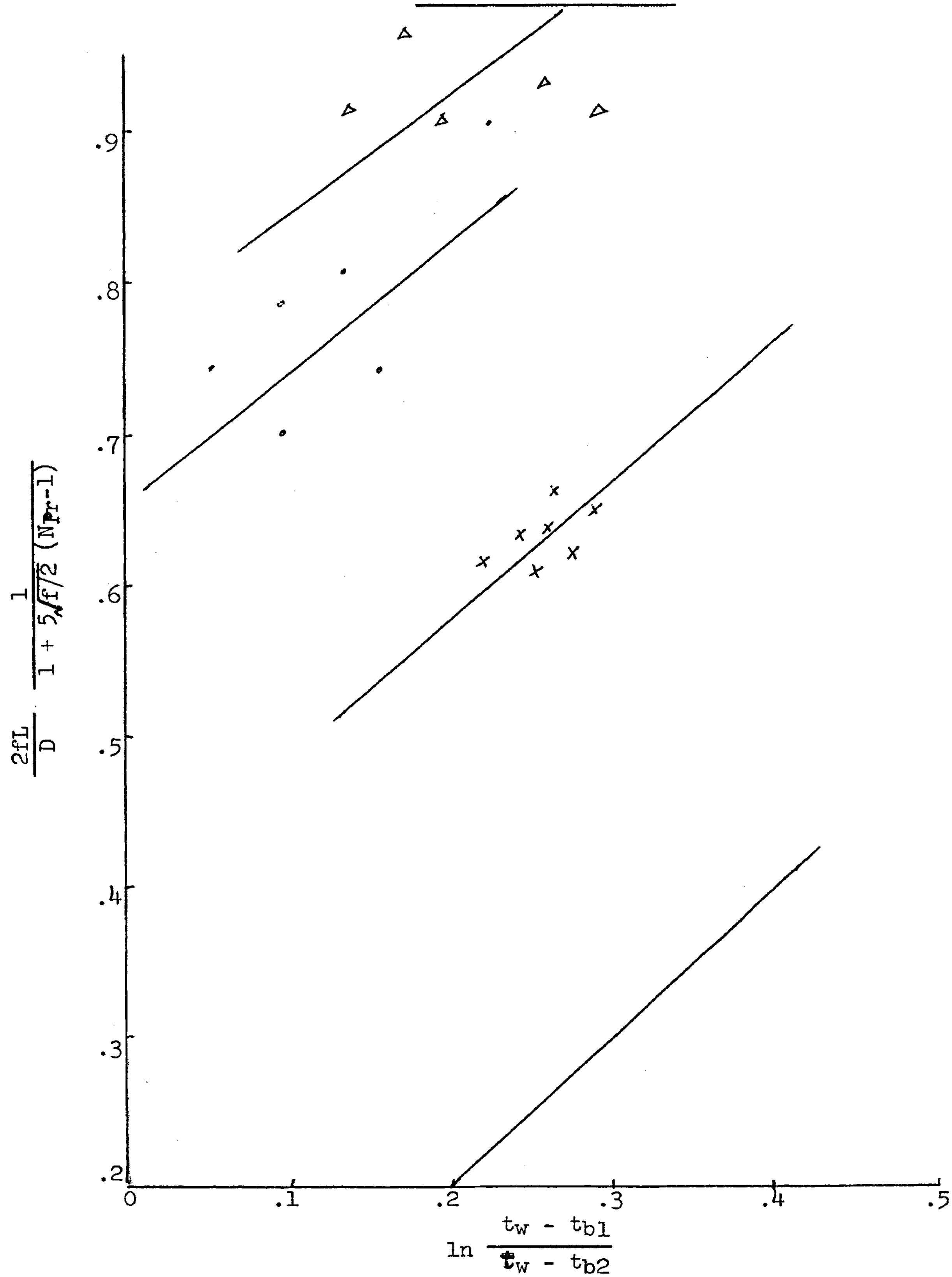


Table 5

COMPARISON OF EXPERIMENTAL DATA WITH PRANDTL ANALOGY

Coolant	RUN #	N_{St}	$\frac{f/2}{1+5\sqrt{\frac{f}{2}}(N_{Pr}-1)}$	% DEVIATION
	1	.000178	.00136	86.9
	2	.000125	.00150	91.7
	3	.000380	.00164	76.8
WATER	4	.000660	.00185	64.3
	5	.00021	.00155	86.5
	6	.00020	.00147	86.4
	8	.00271	.00134	102.2
	9	.000646	.00136	52.5
	10	.000477	.00130	63.2
	11	.000568	.00135	57.9
FREON	12	.000571	.00130	56.1
	13	.000451	.00126	64.2
	14	.000560	.00126	55.6
	15	.000636	.00127	49.9
	16	.000546	.00191	71.4
	17	.000288	.00188	84.7
CARBON TETRA	18	.000363	.00197	81.6
	19	.000265	.00187	85.8
	20	.000419	.00185	77.4

AVERAGE DEVIATION: WATER	82.1
FREON 113	62.7
$C\,Cl_4$	80.1
OVERALL	73.4

$$j_H = \left(\frac{h}{C_pG}\right)(N_{Pr})^{2/3} = .023(N_{Re})^{-0.2} \tag{28}$$

The results are shown in Figure 4 and Table 6. The deviations were also calculated and presented.

From the three analogies, it is apparent that although the deviations were large, a conclusion can still be made. The Colburn's Analogy is the most applicable one and gives the best result. It is followed by the Prandtl Analogy and then the Reynolds Analogy, as is expected. Colburn's Analogy is an empirical one and has a wide range of application. The Reynold's Analogy postulates that the mechanism of heat transfer and momentum transfer are the same and neglects the fluid flow profile. The Prandtl Analogy is an extension of the Reynolds analogy. It considers the lamina sub-layer in the turbulent region flow and thus gives slightly better results.

The heat transfer-power imput correlation developed in this study was applied to the experimental data. The results are shown in Figure 8 and Table 7. Figure 8 shows that a higher ratio of heat transfer to power consumption is obtained by higher values of thermal conductivity and heat capacity and lower values of viscosity. The correlation also shows that in the

Table 6

COMPARISON OF EXPERIMENTAL DATA WITH COLBURN'S ANALOGY

COOLANT	RUN #	J_H	f/2	% DEVIATION
WATER	1	.00064	.00384	83.3
	2	.000408	.004	89.8
	3	.001138	.00388	70.7
	4	.001709	.00376	54.5
	5	.000610	.00348	82.5
	6	.000618	.00355	82.6
FREON	8	.00942	.003516	167.9
	9	.002128	.003472	38.7
	10	.001620	.003332	51.4
	11	.001820	.003348	45.6
	12	.001905	.003264	41.6
	13	.001641	.003180	48.4
	14	.001890	.003180	40.6
	15	.002089	.003100	32.6
CARBON TETRA.	16	.001332	.00366	63.6
	17	.000692	.00342	79.8
	18	.000853	.00354	75.9
	19	.000629	.00333	81.1
	20	.001002	.00335	70.1

AVERAGE DEVIATION: WATER	75.6
FREON 113	58.5
$C\ Cl_4$	74.1
OVERALL	68.5

Figure 8: Heat Transfer Rate and Power Requirement Rate Correlation

Experimental and Theoretical Values

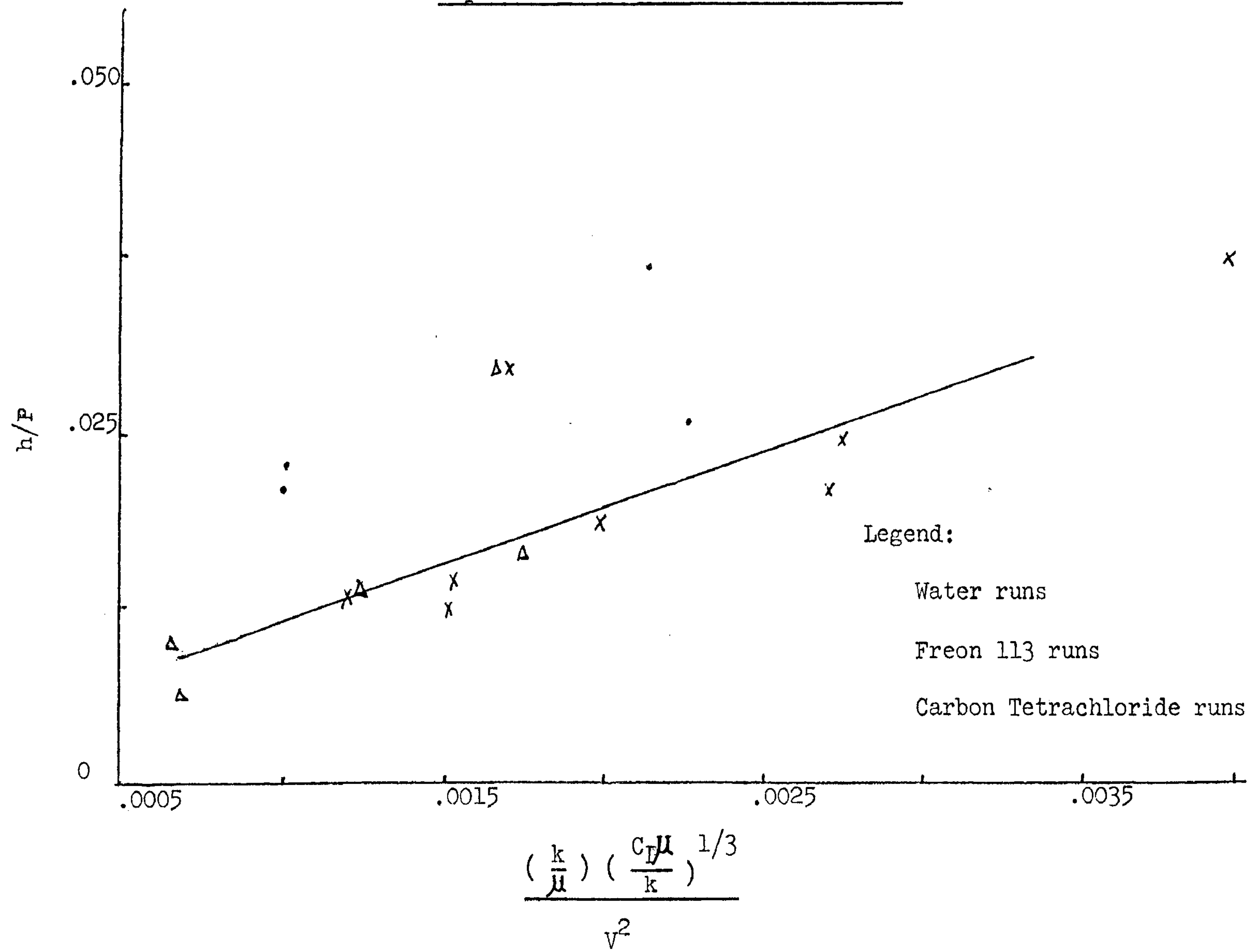


Table 7

DEVIATION OF EXPERIMENTAL VALUES IN CORRELATION GIVEN BY EQUATION 18

COOLANT	RUN #	h/P_A	$\dfrac{K_3\left(\frac{k}{\mu}\right)\left(\frac{C_p\mu}{k}\right)^{1/3}}{\psi^2}$	% DEVIATION
WATER	1	.036	.224	83.9
	2	.026	.237	89.0
	3	.080	.262	69.5
	4	.144	.300	52.0
	5	.022	.118	81.4
	6	.021	.111	81.1
FREON	8	.248	.640	61.3
	9	.0384	.412	90.3
	10	.021	.283	92.6
	11	.024	.287	91.6
	12	.018	.210	91.4
	13	.012	.159	92.4
	14	.014	.160	91.3
	15	.013	.127	89.8
CARBON TETRA.	16	.026	.178	85.4
	17	.016	.185	91.4
	18	.014	.132	89.4
	19	.0059	.0733	91.9
	20	.0095	.0724	86.9

AVERAGE DEVIATION: WATER 76.2
FREON 113 87.5
$C\,Cl_4$ 89.0

OVERALL 84.3

turbulent region higher velocities are detrimental to the ratio.

CONCLUSION

1. The unusually large deviations make a conclusion difficult. However, the study indicates that in general the physical properties of the coolants can be used as a screening tool.

2. The study indicates that the physical properties of the coolants alone are insufficient for comparing the merits of the coolants. The volumetric flow, $\overline{V}$, is also needed. A lower fluid velocity would increase the heat transfer coefficient/power input ratio in the turbulent region.

3. Lower viscosity, higher density, high heat capacity and high heat conductivity are advantageous in the selection of a coolant.

NOMENCLATURE

A = Cross Section Area of Testing Section, ft^2.

A_L = Log mean heat transfer area, ft^2.

A_T = Heat transfer area based on the inside wall of the pipe, ft^2.

c_p = Specific heat at constant pressure, Btu/lb/^oF.

D = Test Section inside diameter, ft.

f = Fanning friction factor.

g_c = Newton's Law Conversion Factor.

G = Mass velocity in lb/hr/ft^2.

h = Heat transfer coefficient, Btu/hr/$^oF/ft^2$.

j_H = Colburn heat-transfer factor.

k = Thermal conductivity of fluid, Btu/hr ft^2(^oF/ft)

k_w = Thermal conductivity of pipe wall, Btu/hr ft^2 (^oF/ft)

L = Length of testing section, ft.

N_{Pr} = Prandtl number.

N_{Re} = Reynolds number.

N_{St} = Stanton number.

p = Pressure, lb/ft^2.

P = Work done due to skin friction--$(-\Delta P)\overline{V}$, ft lb_f/hr.

P_A = P/unit heat transfer area, ft lb_f/hr ft^2.

Q = Total heat transferred, Btu/hr.

r_i = Inside radius of pipe, ft.

r_o = Outside radius of pipe, ft.

R = Manometer height, cm.

t_b,avg. = Average bulk fluid temperature, ^oF.

t_{b1} = Bulk temperature at outlet of test section, ^oF.

t_{b2} = Average temperature of inside pipe wall, ^oF.

t_w = Average temperature of inside pipe wall, ^oF.

t_{wo} = Average temperature of outside pipe wall, ^oF.

$\bar{v}$ = Linear velocity, ft/hr.

$\bar{V}$ = Volumetric flow, ft^3/hr.

w = Mass velocity, lb/hr.

Greek Letters

ρ = Average density of fluid, lb/ft^3.

ρ_A = Density of manometer fluid, lb/ft^3.

ρ_B = Density of lighter fluid in manometer, lb/ft^3.

μ = Viscosity of fluid, centipoise.

μ_w = Viscosity of fluid at wall temperature, centipoise.

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APPENDIX

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APPENDIX A

Flowrator Calibration

Figure 9: Flowrator Calibration, Water and "Freon 113"

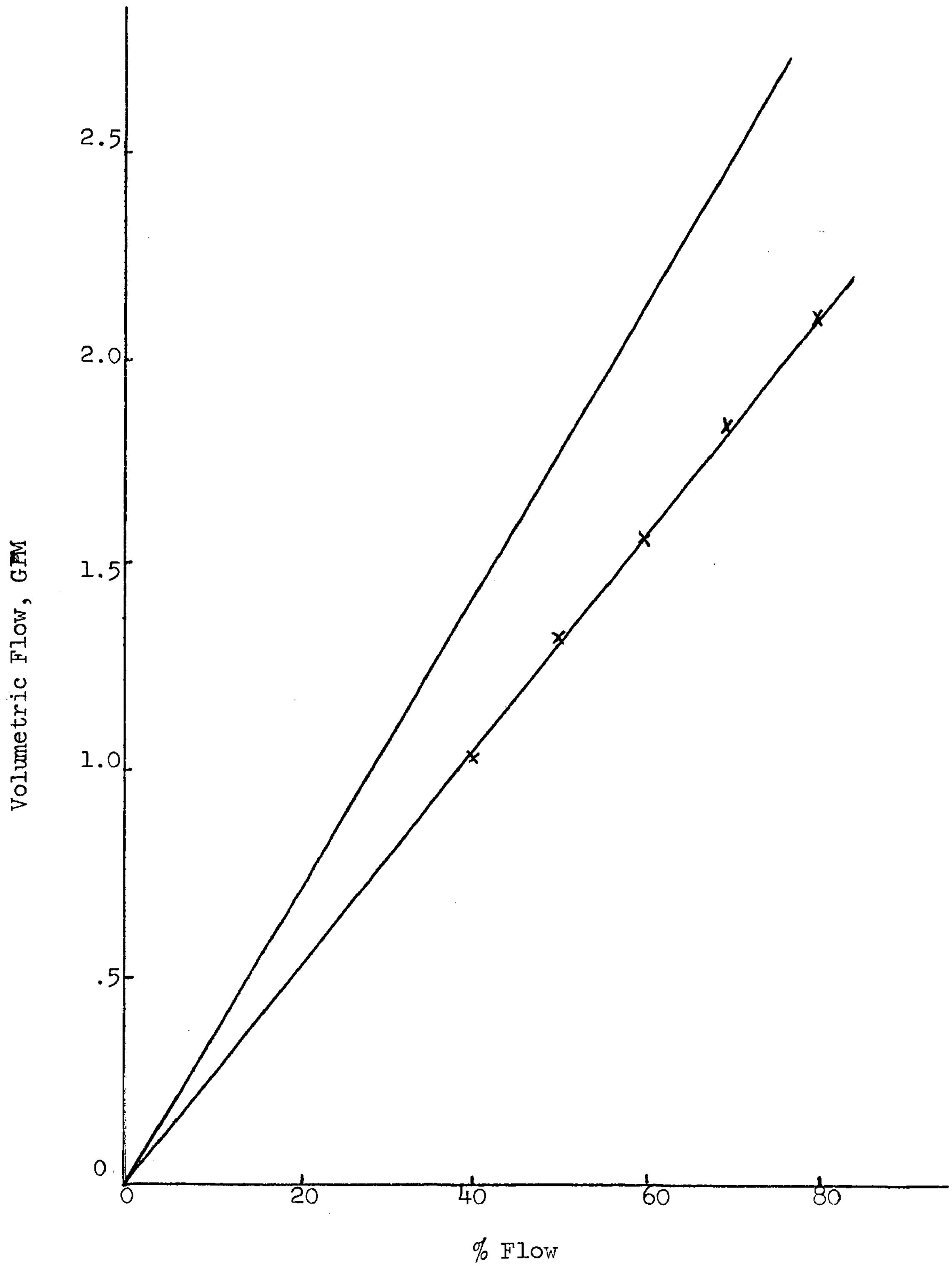
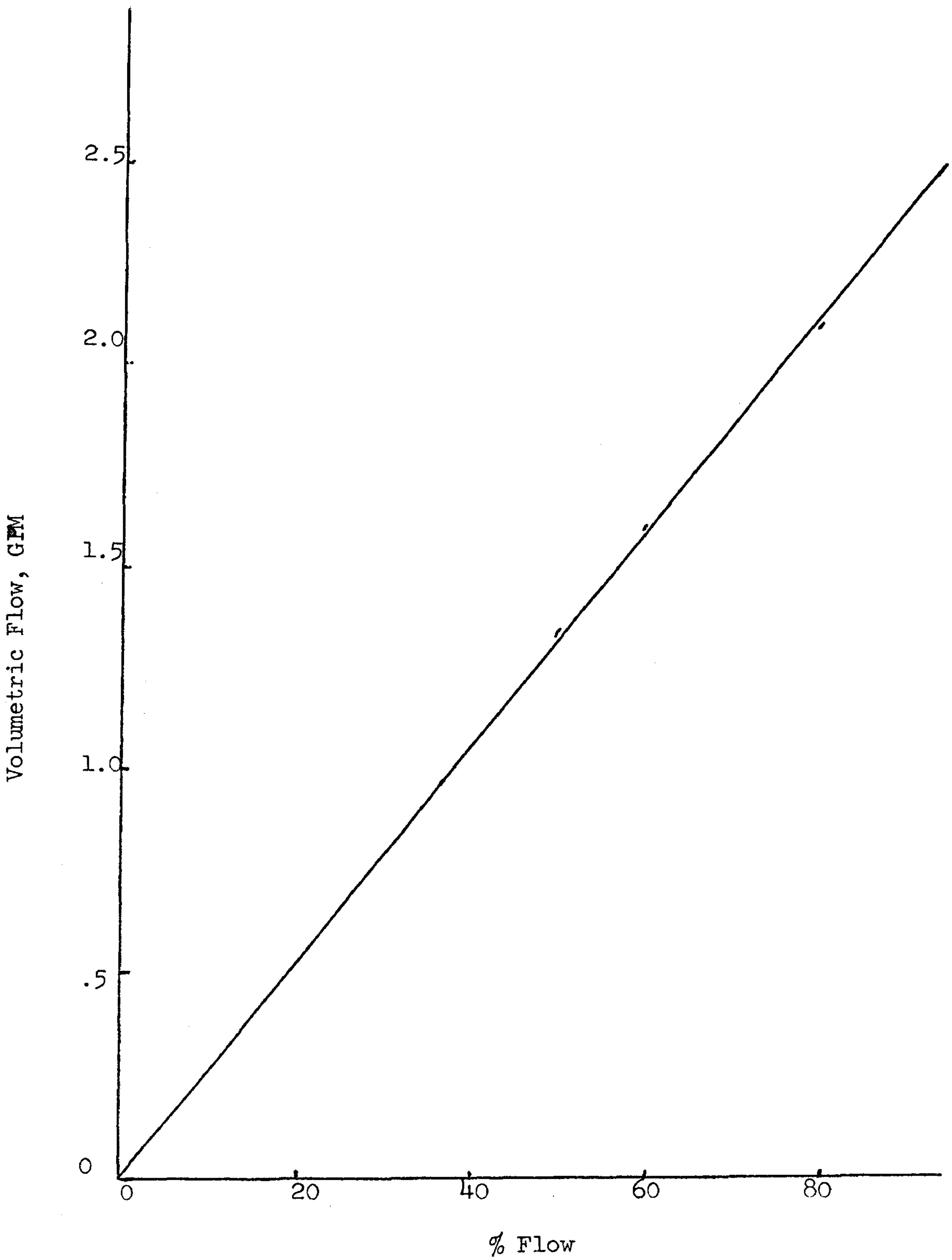


Figure 10: Flowrator Calibration, Carbon Tetrachloride



APPENDIX B

Physical Properties

Figure 11: Viscosity of Liquid Water
(From McCabe & Smith, Reference (9))

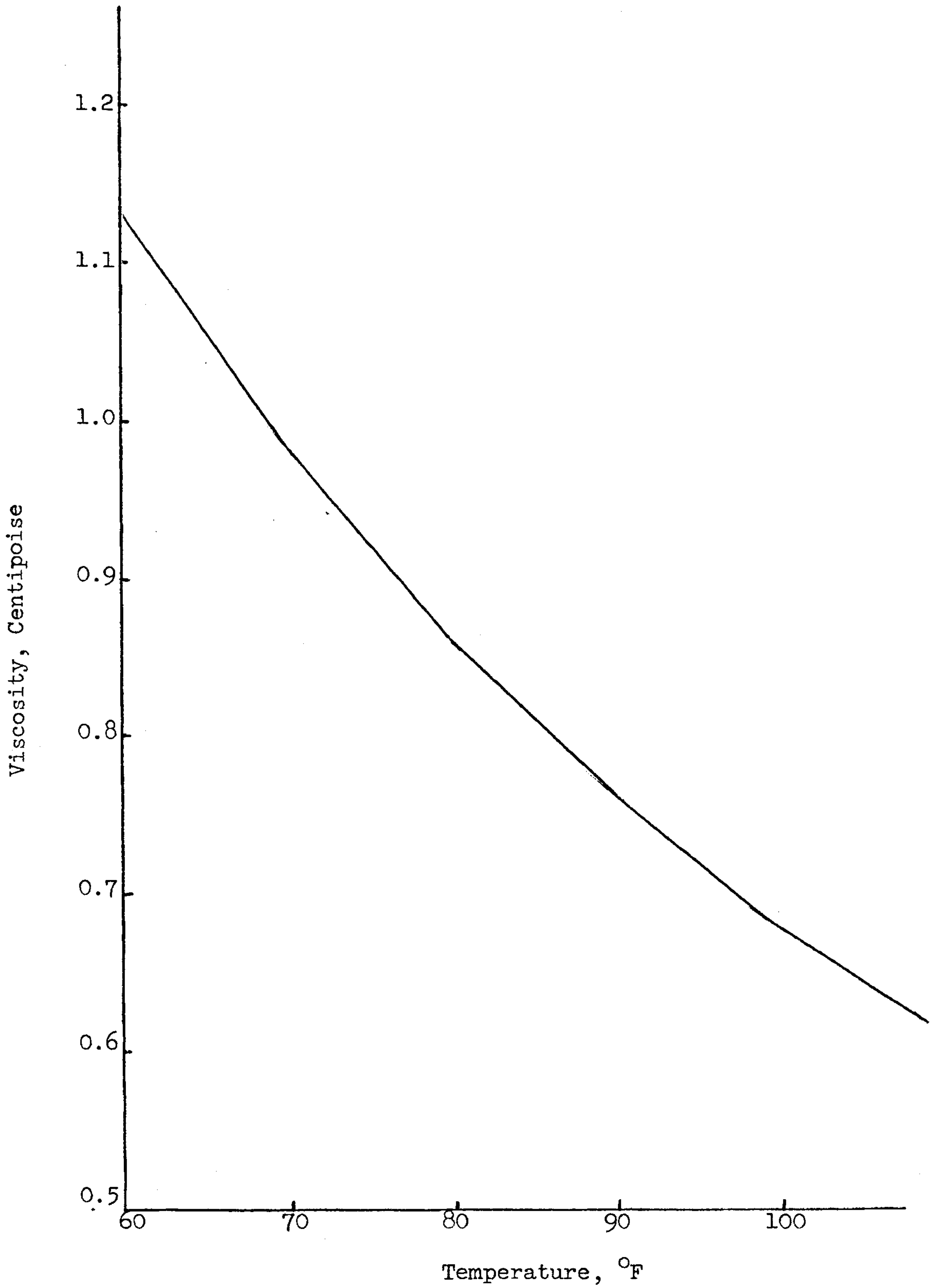


Figure 12: Thermal Conductivity of Liquid Water
(From McCabe & Smith, Reference (9))

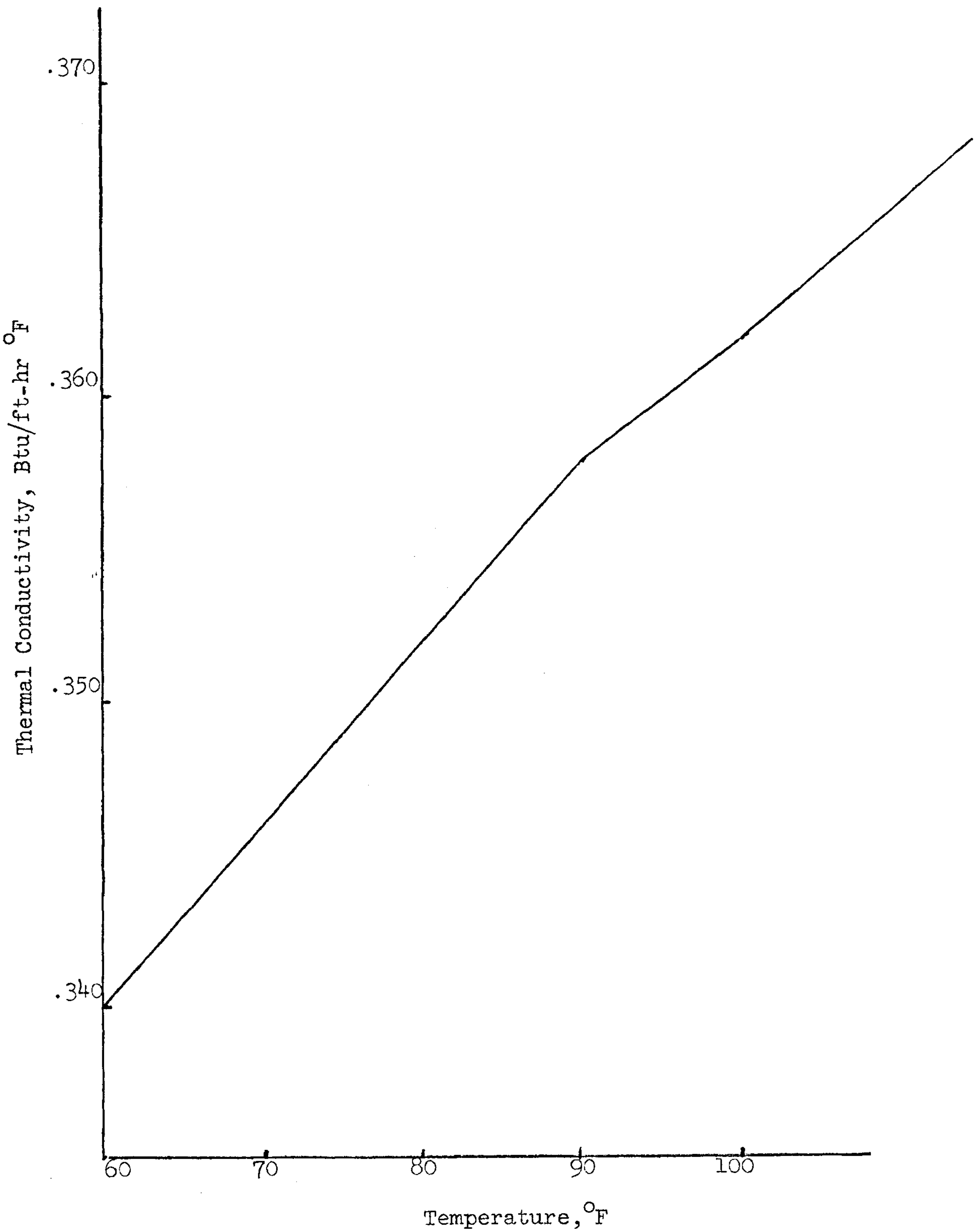


Figure 13: Liquid Density of "Freon 113"
(Estimated, based on du Pont Technical Bulletin (8))

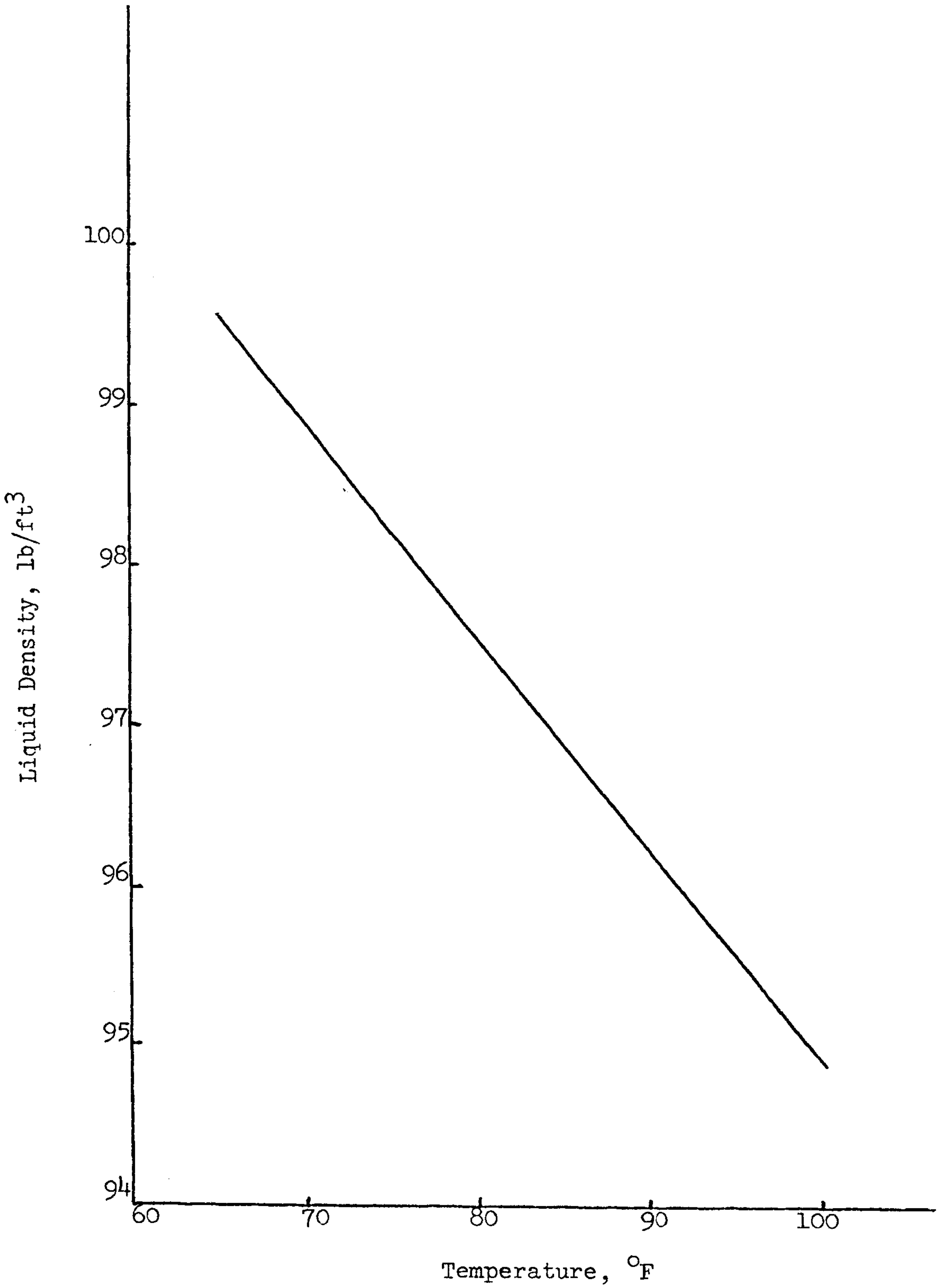


Figure 14: Liquid Heat Capacity of "Freon 113"
Estimated, based on: (1) McCabe & Smith (9)
(2) du Pont Technical Bulletin (8)

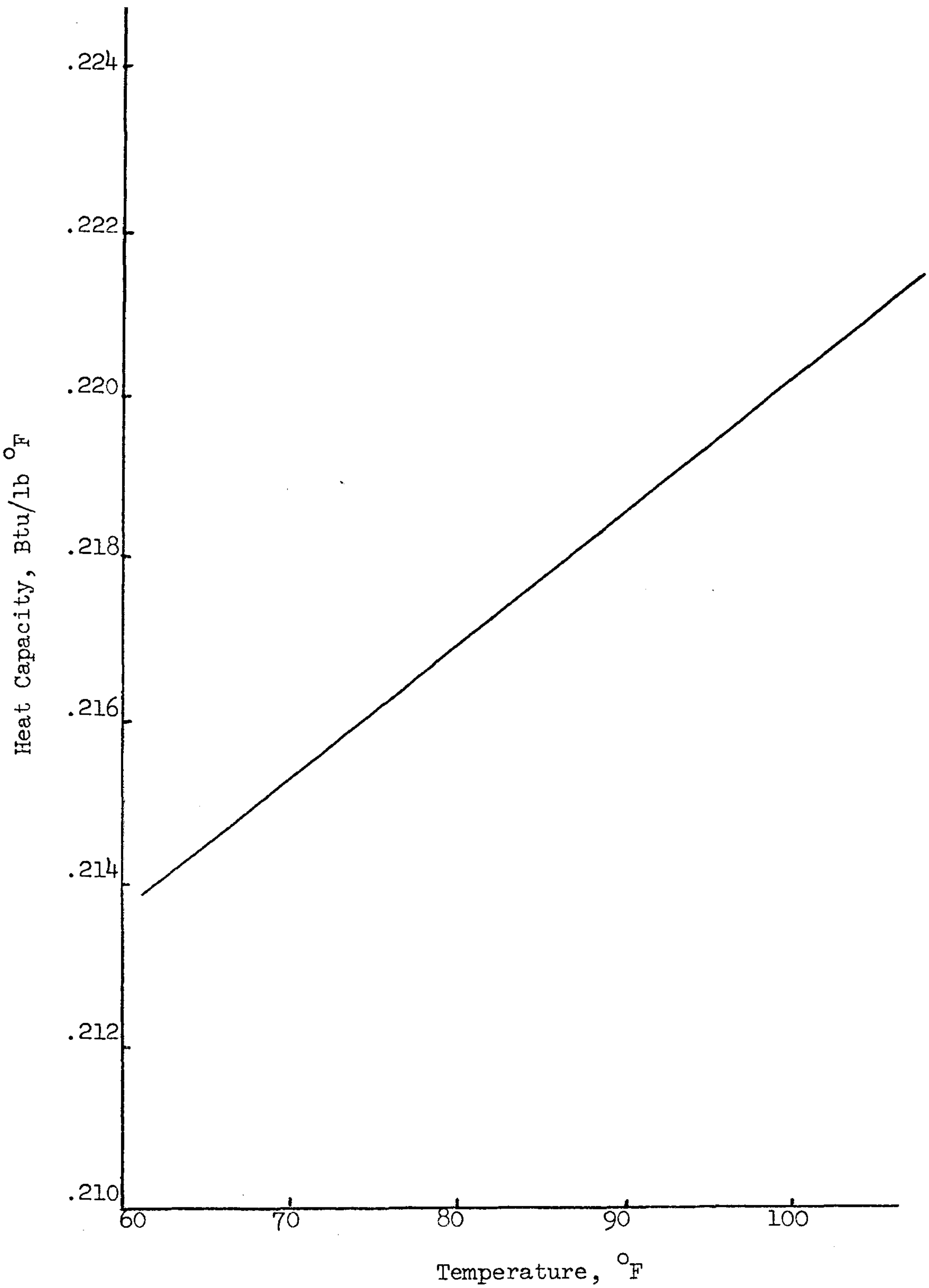


Figure 15: Liquid Thermal Conductivity of "Freon 113"
(du Pont "Freon" Technical Bulletin, Reference (8))

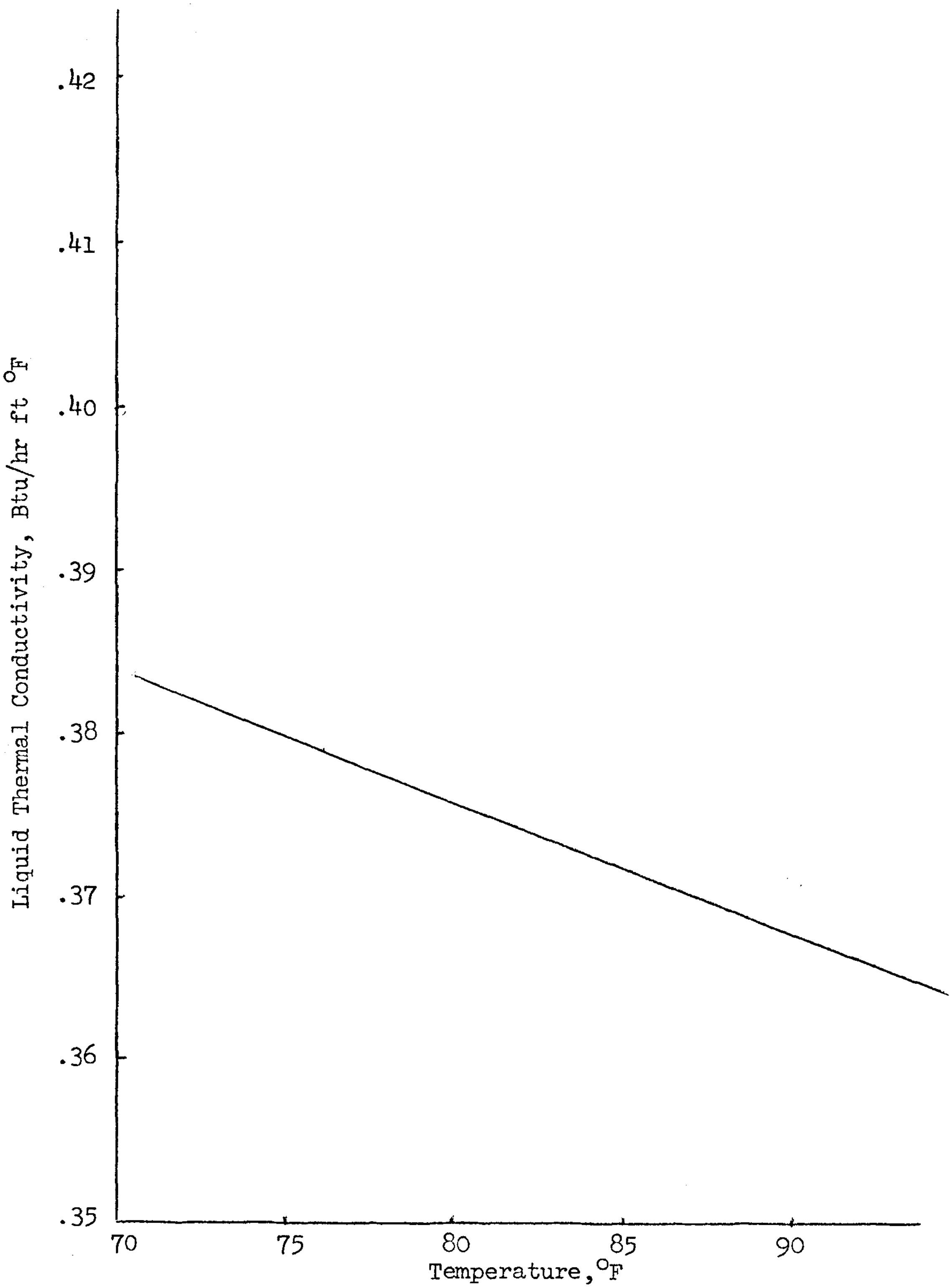


Figure 16: Liquid Viscosity of "Freon 113"
(du Pont "Freon" Technical Bulletin, Reference (8))

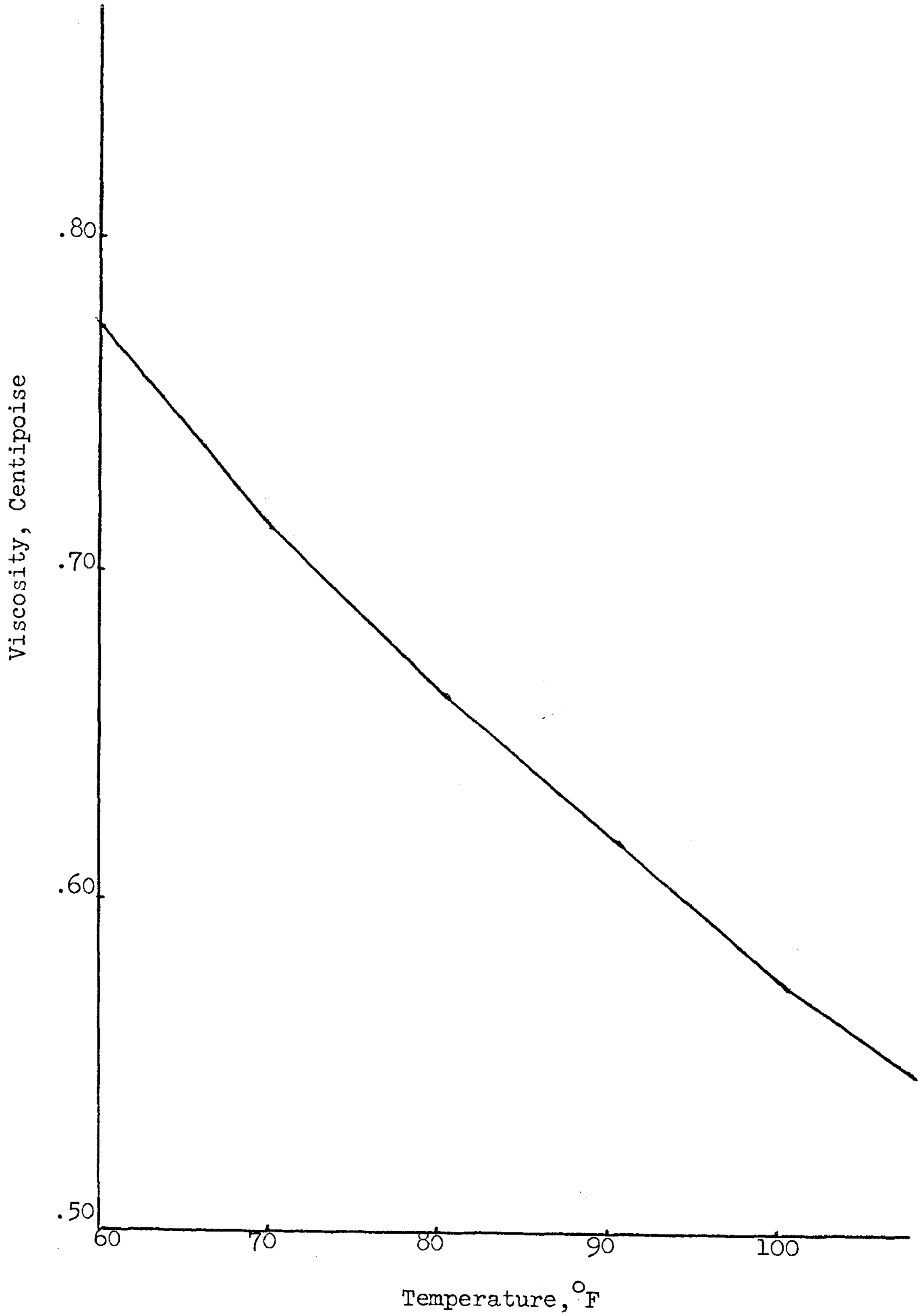


Figure 17: Density of Carbon Tetrachloride
(From International Critical Table)

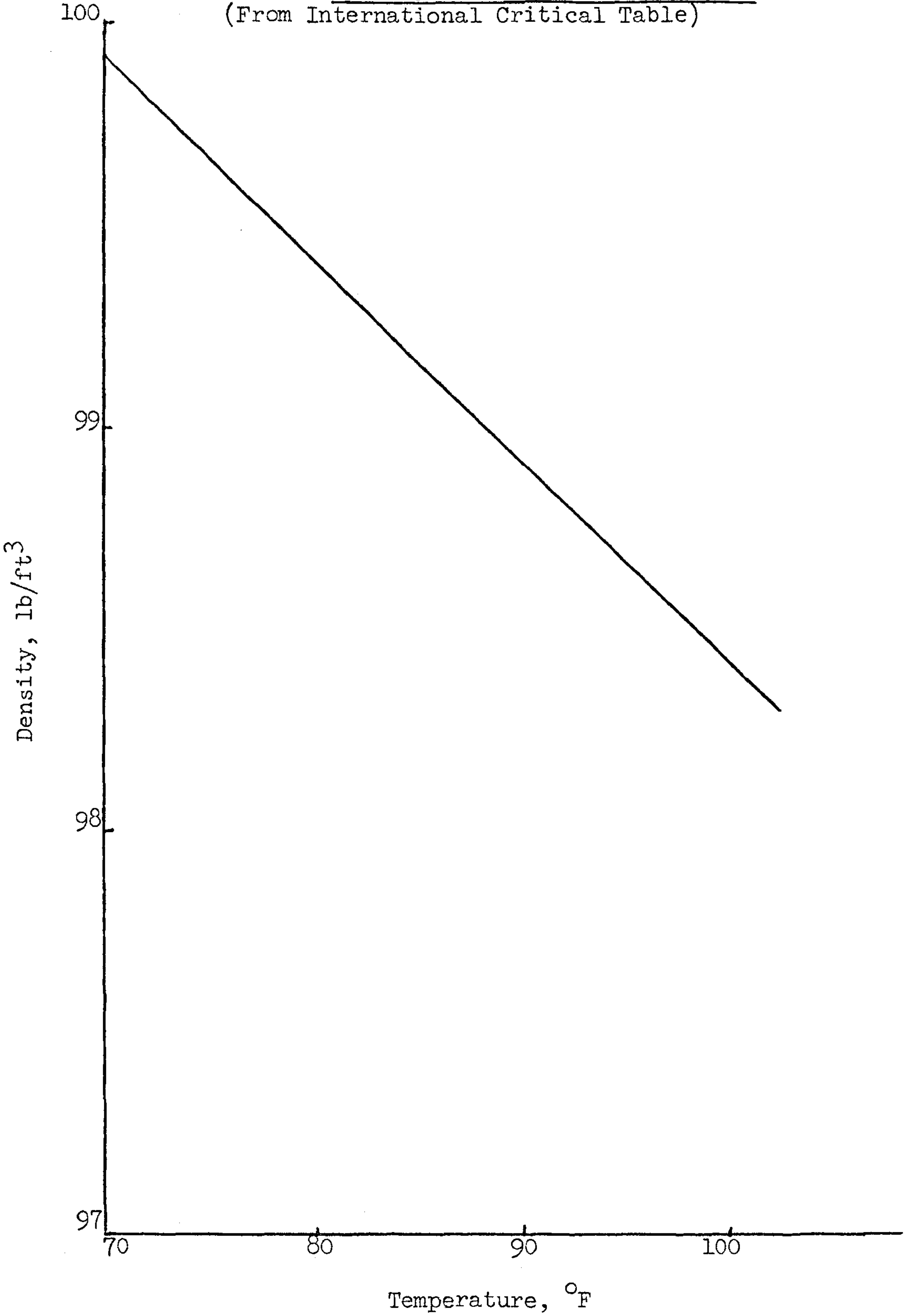
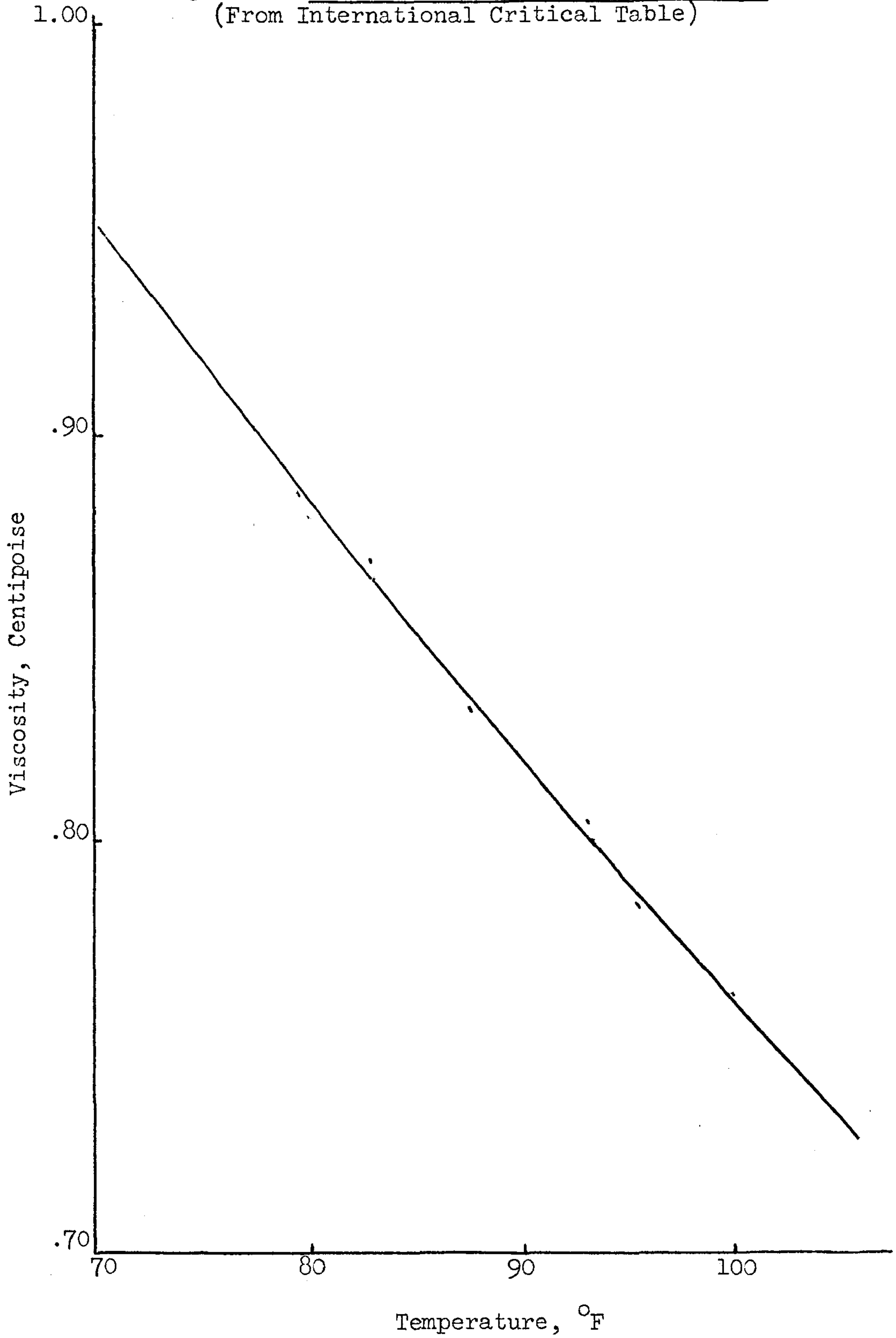


Figure 18: Viscosity of Carbon Tetrachloride
(From International Critical Table)



APPENDIX C

Sample Calculation

To show the steps of calculation, run #3 is picked as the demonstration set.

I. Original Data and other important data:

a.) Flow Rate:-

1. Measured flow rate: 40% or 1.416 GPM.

2. Volumetric flow rate, $\overline{V}$, ft^3/hr:

$$\overline{V} = 1.416 \times 60 \times \frac{1}{7.48} = 11.36 \ ft^3/hr.$$

3. Linear flow rate, $\overline{v}$, ft/sec:

$$\overline{v} = \overline{V}/A - 11.36/(.0005 \times 60 \times 60)$$

$$= 6.31 \ ft/sec.$$

4. Mass flow rate, w, lb/hr:

$$w = \overline{V} \cdot \rho = 11.36 \times 62.15 = 707 \ lb/hr.$$

5. Mass velocity, G, lb/ft^2hr.

$$G = w/A = 707/.0005 = 1,414,000 \ lb/ft^2hr.$$

b.) Temperature:- (Average value of sets 4 & 5 are used)

1. t_{b1}:- Inlet temperature of fluid = 85.83°F

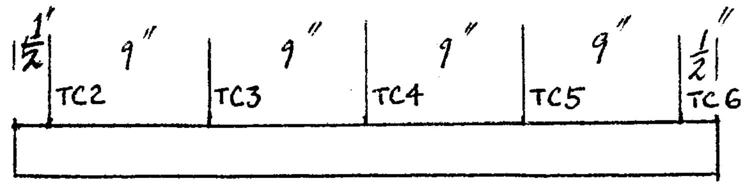
2. t_{b2}:- Outlet temperature of fluid= 87.81°F

3. $t_{b\ avg}$:- $(t_{b1} + t_{b2})/2$ = 86.82°F

4. t_b:- $(t_{b2} - t_{b1})$ = 1.98°F

5. t_{wo}:- Average outside pipe wall temperature

The thermocouples are positioned on the test section as in figure below:



However, thermocouple #3 was found not in function, as a result, the remaining thermocouples were assigned the following weighing factors:

thermocouples	weighing factor
# 2	2
# 4	3
# 5	2
# 6	1

therefore:

$$t_{wo} = \frac{104.3x2+105.5x3+100.6x2+102.2}{2+3+2+1}{}^{o}F$$

$$= 103.6^{o}F$$

C.) Pressure Drop:-

$$-\Delta p = R(\rho_A - \rho_B) = \frac{36.8}{30.48}(184.2 - 62.3)$$

$$= 147.23 \text{ lb/ft}^2$$

II. Reynolds number and Fanning friction factor:

a.) Reynolds number:-

$$N_{Re} = \frac{DG}{\mu} = \frac{.302}{12}x\ 1,414,000x\frac{1}{0.795x2.42}$$

$$= 18,500.$$

b.) Fanning Friction Factor:-

1. theoretical: from published chart

ϵ, roughness factor for steel pipe

$= .00015$

$$\frac{\epsilon}{D} = \frac{.00015}{\frac{.302}{12}} = .006$$

$$f = f'/4 = .036/4 = .0090.$$

2. calculated: from pressure drop data

$$f = \frac{(-\Delta p) g_c D}{2 \bar{v}^2 L} \qquad (1)$$

$$= \frac{147.23 \times 32.17 \times .302}{2 \times 6.31^2 \times 62.15 \times 37 \times}$$

$$= .0079.$$

III. Total Heat Input, Total Heat Transferred and Heat Transfer Coefficient:

a.) Total Heat Input:- This is calculated from the Voltmeter and ammeter readings.

Btu/hr = 3.4129 watts = 3.4129 x volts x amps.

$Q_T = 3.4129 \times 2(70 \times 4.46) = 2130$ Btu/hr.

b.) Total Heat Transferred, Q:-

$Q = w \times C_p \times \ t_b = 707 \times .998 \times 1.98 =$ 1397 Btu/hr.

c.) Heat lost: 2130 - 1397 = 733

733/2130 = 34.5%

d.) Heat Transfer Coefficient:-

$$Q = h\, A_T\, \Delta t$$

$$h = Q/A_T\Delta t = Q/A_T\left(t_w - \frac{t_{b1} + t_{b2}}{2}\right)$$

$$= 1397/\left[.244\left(\frac{t_w - t_{b1} + t_w - t_{b2}}{2}\right)\right]$$

t_w has to be estimated from t_{wo}

$$t_w \doteq t_{wo} - \frac{r_o - r_i}{k_w \cdot A_T} \cdot Q$$

$$= 103.6 - \frac{r_o - r_i}{9.4\,\frac{2\,L(r_o - r_i)}{\ln(r_o/r_i)}} \times 1397$$

$$= 103.6 - \frac{\ln(r_o/r_i)}{9.4 \times 2\,L} \times 1397$$

$$= 103.6 - \frac{\ln(.270/.151)}{9.4x2x3.142x\,\frac{37}{12}} \times 1397$$

$$= 103.6 - 4.4 = 99.2$$

$$h = 1397/\left[.244\left(\frac{99.2 - 85.85 + 99.2 - 87.81}{2}\right)\right]$$

$$= 1397/3.001 - 466\ \text{Btu/hr.}^{o}\text{F ft}^2$$

IV. Colburn Heat-Transfer Factor, "j_H".

a.) "j_H" calculated:

$$j_H = 0.023\ (N_{Re})^{-0.2} = 0.023\ \left(\frac{DG}{\mu}\right)^{-0.2}$$

$$= 0.023\ (18505)^{-0.2}$$

$$= 0.00322$$

b.) "j_H" experimental:

$$j_H = (N_{St})\ (N_{Pr})^{2/3} \left(\frac{\mu_w}{\mu}\right)^{0.14}$$

$$= \left(\frac{h}{\bar{v}C_p\rho}\right)\left(\frac{C_p\mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14}$$

$$= \left(\frac{466}{6.31x3600x.998x62.15}\right)\left(\frac{.998x.706x2.42}{.356}\right)^{2/3}\left(\frac{.795}{.706}\right)^{0.14}$$

$$= .00325 \times 5.39^{2/3} \times 1.017^{0.14}$$

$$= .001138.$$

V. Reynolds Analogy comparison: N_{Re}, f, $\frac{2fL}{D}$, $\ln \frac{t_w - t_{b1}}{t_w - t_{b2}}$

a.) N_{Re}:- calculated in II.(a.)

b.) f :- calculated in II.(b.)

c.) $\frac{2fL}{D} = \frac{2 \times .0079 \times 37/12}{.302/12} - 1.917$

d.) $\ln \frac{t_w - t_{b1}}{t_w - t_{b2}} = \ln \frac{99.2 - 85.83}{99.2 - 87.81} = \ln 1.174 = .1584$

VI. Prandtl Analogy: N_{St}, $\frac{f/2}{1+5\sqrt{\frac{f}{2}}\,(N_{Pr}-1)}$, $\ln \frac{t_w - t_{b1}}{t_w - t_{b2}}$,

$\frac{2fL}{D}$ $\frac{1}{1+5\sqrt{\frac{f}{2}}\,(N_{Pr}-1)}$.

a.) N_{St}:- $N_{St} = \dfrac{h}{\bar{V}\, C_p \rho} = \dfrac{466}{6.31 x 3600 x .998 x 62.15}$

$= .000325$

b.) $\dfrac{f/2}{1+5\sqrt{\frac{f}{2}}\,(N_{Pr}-1)} = \dfrac{.0039}{1+5\ \sqrt{.0039}\ (5.39-1)} = \dfrac{.0039}{.4219}$

c.) $\ln \dfrac{t_w - t_{b1}}{t_w - t_{b2}}$:- see V. (d.).

d.) $\dfrac{2fL}{D}\ \dfrac{1}{1+5\sqrt{\frac{f}{2}}\ (N_{Pr}-1)} = \dfrac{2x.0079x37}{.302}\ \dfrac{1}{1+5\ \sqrt{.0039}\ (5.39-1)}$

$= 1.917 \times .4219 = .809$

VII. Colburn's Analogy comparison: j_H, $f/2$, N_{Re}

a.) j_H:- calculated in IV.(b.)

b.) $f/2$:- calculated in II. (b.)

c.) N_{Re}:- calculated in II. (a.)

VIII. Correlation comparison: $\dfrac{h}{P_A}$, $\dfrac{\left(\frac{k}{\mu}\right)9\left(\frac{C_p\mu}{k}\right)^{1/3}}{\bar{V}^2}$.

a.) $\dfrac{h}{P_A} = \dfrac{h}{P/\pi DL} = \dfrac{h \times \pi DL}{(-\Delta p)\ \bar{V}}$

$= \dfrac{466x3.142x.302x37}{147.2x11.36x144} = .07$ Btu/ft $=$ $lb_f\ {}^oF$

b.) $$\frac{\left(\frac{k}{\mu}\right)\left(\frac{C_p\mu}{k}\right)^{1/3}}{\bar{V}^2} = \frac{\left(\frac{.356}{0.795 \times 2.42}\right)(N_{Pr})^{1/3}}{11.36^2}$$

$$= .00252 \frac{Btu}{{}^oF\text{-}lb_f}\left(\frac{hr}{ft^3}\right)^2$$

c.) $$K_3 = \frac{g_c A_T AD}{4L} =$$

$$\frac{32.17 ft/sec^2 \times .2376 ft^2 \times .0005 ft^2 \times .302 in}{4 \times 37\ in}$$

$$= \frac{32.17 \times .2376 \times .0005 \times .302}{4 \times 37} \frac{lb\text{-}ft}{lb_f\text{-}sec^2} . ft^4$$

$$= \frac{32.17 \times .2376 \times .0005 \times .302 \times 3600^2}{4 \times 37} \frac{lb\text{-}ft^5}{lb_f\text{-}hr^2}$$

$$= 100.8\ lb\text{-}ft^5/lb_f\text{-}hr^2$$

APPENDIX D EXPERIMENTAL DATA

		Manometer Readings			Thermocouple Readings								Heat Input	
Run #	Set of Value	Left Arm	Right Arm	R	TC1	TC7	TC2	TC4	TC5	TC6	TC1 -TC7	% Flow	Tape I	Tape II
1	1	62.55	24.30	38.25	70.45	69.75	73.3	74.2	73.41	73.0	0.7		30v.	30v.
1	2	62.50	24.35	38.15	70.30	69.80	73.55	74.35	73.63	73.2	0.5	40		
1	3	62.50	24.35	38.15	70.00	69.75	73.3	74.23	73.42	73.0	0.25		1.85a	1.91a
1	4	62.55	24.20	38.35	70.05	69.75	73.3	74.2	73.41	73.0	0.30			
2	1	62.30	24.45	37.85	75.55	75.25	84.15	85.83	84.85	84.05	0.30			
2	2	62.25	24.50	37.75	76.30	75.67	85.06	85.75	85.75	84.80	0.63		50v	50v
2	3	62.25	24.50	37.75	76.50	76.00	85.20	86.00	86.00	85.10	0.50	40		
2	4	62.25	24.50	37.75	76.40	76.05	85.30	86.15	86.15	85.27	0.35			
2	5	62.25	24.50	37.75	76.55	76.07	85.40	86.25	85.30	85.30	0.48			
3	1	61.95	24.8	37.15	85.0	81.60	101.3	101.55	96.80	99.53	3.40			
3	2	61.85	24.9	36.95	87.73	85.20	104.25	104.6	98.20	101.00	2.63		70v	70v
3	3	61.85	24.9	36.95	87.07	85.60	104.7	105.57	98.90	101.60	2.47	40		
3	4	61.80	24.95	36.85	87.88	85.95	104.4	105.61	100.00	102.10	1.93		4.46a	4.46a
3	5	61.75	25.0	36.75	87.73	85.70	104.2	105.40	101.20	102.20	2.03			
4	1	61.30	25.45	35.85	102.1	98.85	129.3	126.35	121.5	122.7	3.25			
4	2	61.05	25.70	35.35	104.5	100.30	131.95	128.65	123.55	125.1	4.15		90v	90v
4	3	60.95	25.80	35.25	105.15	101.00	132.95	129.33	124.00	125.7	4.15	40		
4	4	61.05	25.70	35.35	104.7	100.80	134.60	129.15	123.83	125.4	3.90		5.6a	5.6a
4	5	61.10	25.65	35.45	104.8	100.95	134.30	129.00	123.93	125.65	3.85			
5	1	76.00	1.80	74.20	88.90	86.75	116.7	112.83	108.10	110.45	2.15			
5	2	76.00	1.80	74.20	89.50	87.20	117.10	113.20	108.40	110.00	2.30		90v	90v
5	3	76.00	1.80	74.20	89.95	88.27	117.45	114.00	109.35	112.30	1.68	60		
5	4	76.00	1.80	74.20	89.90	88.10	117.3	114.27	109.35	112.00	1.80		5.6a	5.6a

Run #	Set of Value	Manometer Readings Left Arm	Manometer Readings Right Arm	R	Thermocouple Readings TC1	TC7	TC2	TC4	TC5	TC6	TC1 -TC7	% Flow	Tape I	Tape II
6	1	76.25	1.60	74.65	84.9	82.3	112.23	109.50	104.40	106.50	2.60			
6	2	76.65	1.20	75.45	83.5	81.65	111.00	107.75	103.41	105.80	1.85		90v.	90v.
6	3	76.75	1.10	75.65	83.07	81.40	110.50	107.55	103.10	105.50	1.67	60		
6	4	76.80	1.05	75.75	83.07	81.63	110.60	107.60	103.30	105.70	1.44		5.6a	5.6a
6	5	76.80	1.05	75.75	83.23	81.60	110.63	107.68	103.33	105.75	1.63			
8	1	40.60	35.90	4.70	83.40	79.60	88.10	89.30	88.90	79.60	3.80			
8	2	40.60	35.90	4.70	83.40	79.30	87.85	89.30	89.00	79.30	4.10	40	40v	40v
8	3	40.60	35.90	4.70	83.65	79.50	88.25	89.65	89.50	79.50	4.15			
8	4	40.60	35.90	4.70	84.10	79.60	88.55	90.50	89.70	79.60	4.50		2.5a	2.5a
9	1	41.65	34.55	7.10	86.60	84.05	95.50	97.20	95.75	94.00	2.55		40v	40v
9	2	41.65	34.55	7.10	86.65	84.10	95.50	97.10	95.60	93.95	2.55	50		
9	3	41.65	34.55	7.10	86.65	84.00	95.50	96.90	95.60	93.80	2.65		2.5a	2.6a
10	1	42.55	32.55	10.00	84.70	81.65	98.25	101.10	99.55	97.00	3.05		50v	50v
10	2	42.55	32.55	10.00	85.00	81.55	98.00	100.90	99.20	96.65	3.45			
10	3	42.60	32.50	10.10	84.60	81.25	97.55	100.60	98.90	96.50	3.35	60	3.1a	3.5a
10	4	42.55	32.55	10.00	84.75	81.40	97.90	100.90	99.30	96.90	3.35			
11	1	42.60	32.55	10.05	91.05	85.90	110.0	114.0	110.5	107.6	5.15			
11	2	42.60	32.55	10.05	91.00	86.10	110.0	114.0	110.4	107.65	4.90		60v	60v
11	3	42.60	32.55	10.05	90.90	86.10	109.7	114.0	110.4	107.60	4.80	60		
11	4	42.60	32.55	10.05	90.70	85.95	109.65	113.7	110.15	107.35	4.75			
12	1	44.30	30.85	13.45	88.50	84.00	105.90	109.40	106.55	104.20	4.50		60v	60v
12	2	44.25	30.90	13.45	88.30	84.05	105.90	109.40	106.55	104.25	4.25	70		
12	3	44.25	30.90	13.45	88.40	84.10	105.85	109.30	106.50	104.30	4.35		4.08a	4.20a
13	1	46.05	29.05	17.00	85.20	81.55	102.0	105.10	102.65	100.9	3.65		60v	60v
13	2	46.05	29.05	17.00	84.90	81.40	101.7	104.9	102.40	100.8	3.50	80		
13	3	46.05	29.05	17.00	84.60	81.20	101.45	104.6	102.20	100.65	3.40		4.08a	4.20a
13	4	46.05	29.05	17.00	84.85	81.30	101.50	104.75	102.35	100.8	3.55			

Run #	Set of Value	Manometer Readings Left Arm	Manometer Readings Right Arm	R	TC1	Thermocouple Readings TC7	TC2	TC4	TC5	TC6	TC1 -TC7	Flow	Tape I	Tape II
14	1	46.0	29.15	16.85	85.75	81.70	103.0	105.4	102.5	101.2	9.05		60v	60v
14	2	46.0	29.15	16.85	85.60	81.70	103.0	105.4	102.6	101.4	3.90	80		
14	3	46.0	29.15	16.85	85.75	81.90	103.0	105.5	102.7	101.3	3.85		4.08a	4.20a
15	1	48.0	27.10	20.90	91.00	86.25	110.85	114.55	111.20	109.50	4.75		60v	70v
15	2	48.0	27.10	20.90	90.40	85.60	110.50	114.10	110.60	108.95	4.80	90		
15	3	48.0	27.10	20.90	91.00	85.25	110.40	113.90	110.40	108.90	4.85		4.15a	4.78a
15	4	48.0	27.10	20.90	90.20	85.30	110.40	113.90	110.30	108.90	4.90			
16	1	42.25	34.45	7.80	80.65	77.80	91.55	94.20	92.25	90.10	2.85		40v	46v
16	2	42.25	34.45	7.80	80.70	77.80	91.50	94.15	92.10	90.00	2.90	50		
16	3	42.25	34.45	7.80	80.75	77.80	91.40	94.20	92.30	90.00	2.95		2.5a	3.0a
17	1	42.25	34.95	7.30	91.40	87.85	102.20	104.20	102.10	100.00	3.65		40v	46v
17	2	42.25	34.95	7.30	90.60	87.40	101.70	103.40	101.30	99.10	3.20	50		
17	3	42.25	34.95	7.30	89.70	85.95	100.65	102.10	100.00	97.80	3.75		2.5a	2.5a
17	4	42.25	34.95	7.30	88.80	85.30	99.90	101.75	99.70	97.30	3.50			
18	1	43.72	32.88	10.84	93.50	90.50	108.95	113.70	110.50	108.00	3.00		50v	60v
18	2	43.72	32.88	10.84	92.95	89.80	108.40	113.15	110.00	107.55	3.15	60		
18	3	43.72	32.88	10.84	93.10	90.20	108.70	113.70	110.60	108.00	2.90		3.2a	3.8a
18	4	43.72	32.88	10.84	93.20	90.30	109.10	114.30	110.90	108.30	2.90			
19	1	47.40	29.20	18.20	90.50	88.10	104.50	107.65	105.40	103.70	2.40		50v	60v
19	2	47.40	29.20	18.20	89.90	87.80	104.20	107.70	105.60	104.10	2.10	80		
19	3	47.40	29.20	18.20	90.90	89.00	104.40	108.30	106.20	104.55	1.90		3.2a	3.8a
19	4	47.40	29.20	18.20	90.70	88.90	104.30	108.35	106.10	104.65	1.80			
20	1	47.40	29.20	18.20	89.90	86.65	105.50	112.05	109.30	107.35	3.25		60v	70v
20	2	47.40	29.20	18.20	89.60	86.10	105.70	112.40	109.35	107.40	3.50			
20	3	47.40	29.20	18.20	89.30	85.90	105.65	112.40	109.30	107.30	3.40	80		
20	4	47.40	29.20	18.20	89.50	85.95	105.75	113.15	109.25	107.35	3.55		3.85a	4.45a