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HEAT TRANSFER AT HIGH PRESSURE DROPS AND LOW REYNOLDS NUMBERS

A THESIS

Presented to

the Faculty of the Graduate Division

by

Henry Barden Allison, II

In Partial Fulfillment

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HEAT TRANSFER AT HIGH

PRESSURE DROPS AND LOW REYNOLDS NUMBERS

APPROVED: _____
Thesis Advisor

_____

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TABLE OF CONTENTS

	Page
ACKNOWLEDGEMENTS	ii
LIST OF FIGURES	iv
LIST OF TABLES	v
SYMBOLS AND SUBSCRIPTS	vi
INTRODUCTION	1
DESCRIPTION OF TESTING	3
ANALYTICAL RESULTS	5
CORRELATION OF HEAT TRANSFER DATA	7
CORRELATION OF PRESSURE DROP DATA	10
CONCLUSIONS	12
RECOMMENDATIONS	14
APPENDIX A, CALCULATION OF ADIABATIC TEMPERATURE RISE	29
APPENDIX B, BIBLIOGRAPHY	31

LIST OF FIGURES

Figure		Page
1.	Test Apparatus Schematic Diagram	22
2.	Variation of Viscosity with Temperature	23
3.	Variation of Specific Gravity with Temperature	24
4.	Adiabatic Temperature Difference Versus Pressure Drop	25
5.	Variation of Nusselt Number with Peclet Number x $\frac{D}{L}$	26
6.	Variation of Theoretical Nusselt Number with Peclet Number x $\frac{D}{L}$	27
7.	Variation of Friction Factor with Reynolds Number	28
8.	Variation of Friction Factor with Reynolds Number	29

LIST OF TABLES

Table		Page
I	Data Sheet for SAE 30 with Tube Wall Insulated	15
II	Data Sheet for SAE 70 with Tube Wall Insulated	16
III	Data Sheet for SAE 30 with Tube Wall 208°F	17
IV	Data Sheet for SAE 70 with Tube Wall 208°F	18
V	Data Sheet for SAE 70 with Tube Wall 151°F	19
VI	Gage Calibration	20

SYMBOLS

A	area, ft^2
B	constant
C	specific heat, BTU/lb-^oF
D	diameter, ft
f	friction factor
Gz	Graetz number
h	heat transfer coefficient, BTU/hr-ft^2-^oF
J	proportionality constant
L	length, ft
N,n	constants
Nu	Nusselt number
p	pressure, psi
Pe	Peclet number
q	heat flow rate, BTU/hr
r	radius, ft
Re	Reynolds number
T	temperature, ^oF
ΔTa	average temperature difference for heat transfer, ^oF
v	specific volume, ft^3/lb
V	velocity, ft/sec
w	mass rate of flow, lb/hr
Z	elevation, ft
ρ	density, lb/ft^3

τ shear stress

SUBSCRIPTS

a average

c centerline

L exit condition

o inlet condition

T theoretical

w wall

INTRODUCTION

The problem under investigation is that of determining experimentally heat transfer coefficients for laminar flow at pressure drops so high that viscous dissipation cannot be neglected and to compare the results with theory.

In Graetz's (1)* solution for heat transfer in laminar flow in a circular tube it is assumed that the velocity distribution is fully developed, that the wall temperature is constant, that all fluid properties are independent of temperature and pressure, and that viscous dissipation is negligible. Both Singh (2) and Toor (3) have investigated the above problem including heat generation and viscous dissipation. For investigation of the effect of viscous dissipation at high pressure drops on the heat transfer coefficient, a system was developed to provide constant wall temperature and high pressure drops.

To obtain pressure drops of large magnitude, a test apparatus consisting of a ten foot section of 0.075 in. O.D. x 0.025 in. I.D. 320 stainless steel tubing, a motor driven positive displacement pump, a surge tank, vapor jacket, thermocouples, and a pressure gage was constructed to provide pressures up to 10,000 psig as shown schematically in Figure 1. Texaco SAE 30 automotive oil and Standard SAE 70 bus oil were used as test fluids. Steam and alcohol vapor were used as condensing vapors to maintain the wall temperature constant.

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* Numbers in parenthesis refer to the Bibliography at the end of the thesis.

Adiabatic test runs were accomplished by filling a pipe enclosing the tubing with vermiculite insulation.

DESCRIPTION OF TESTING

The viscosity of each oil used was determined at several temperatures by means of a Saybolt Universal Viscosimeter. Specific gravity of each oil was obtained by direct weight measure. Values of viscosity and specific gravity plotted against temperature are presented in Figures 2 and 3 respectively.

Test runs were made with both the SAE 30 oil and the SAE 70 oil with the tube wall insulated. Pressure drops up to 9000 psi were obtained, and the data from these tests are presented in Tables I and II.

Two heating runs were made using saturated steam as the condensing vapor to maintain a constant wall temperature of 208°F. With steam, the exit oil temperature closely approached the wall temperature for all flow conditions. Because of the marked change in viscosity of the oil with temperature, as shown in Figure 2, the maximum pressures obtainable were 1400 psig for the SAE 30 oil and 3500 psig for the SAE 70 oil. Data for these runs are presented in Table III and IV.

In order to obtain data at high pressures, SAE 70 oil was run using alcohol vapor at 151°F to maintain a constant wall temperature. Since the viscosity of the SAE 70 oil at the average bulk temperature of this test remained high, pressure drops up to 9000 psi were attained. Data for this run are presented in Table V.

During the test runs, oil temperatures were measured at the inlet and outlet of the tube using copper-constantan thermocouples located as shown in Figure 1 and read on a Leeds and Northrop potenti-

ometer to the nearest 0.5°F. Pressure was measured in a drilled block some ten inches upstream of the test section inlet, and the diameter of the tube between the drilled block and test section was large compared to the diameter of the test section. Operation of the apparatus with the test section disconnected confirmed that there was no measurable pressure drop between the drilled block and the test section connection. The calibration of the pressure gage is presented in Table VI.

Although a surge tank was provided in the system to reduce the pressure surges from the piston type pump, some fluctuation of pressure about the measured pressure was obtained. The maximum fluctuation obtained was ± 100 psi at 9000 psig, and the magnitude of the fluctuations reduced with reduction in pressure to a value of ± 5 psi at 3000 psig and below.

The oil flow rate was obtained by measuring the amount of oil collected in a graduated cylinder during a given period of time. Flow rates are presented in milliliters per minute. The smallest division on the graduated cylinder was one-half milliliter, and the amount of fluid was read to the nearest one-fourth milliliter. Timing of the flow rate was accomplished using the sweep second hand of a wrist watch. Time intervals of one minute were used for the high flow rates, and two minute time intervals were used for flow rates less than 25 milliliter per minute.

ANALYTICAL RESULTS

In the analytical solution of forced convection with viscous dissipation by Toor (3), a value of B is defined by the equation

$$B = \frac{T_c - T_w}{T_o - T_w} \tag{1}$$

where T_c is the centerline temperature at a value of tube length equal to infinity and T_o and T_w are fluid inlet and wall temperatures respectively.

A dimensionless constant n is defined by the stress-rate-of-strain relation

$$\frac{\partial v}{\partial r} = - N \tau^{n-1} \tag{2}$$

For n = 2 this equation is the relation for a Newtonian fluid.

Toor showed that at low values of Graetz number

$$\frac{T_L - T_o}{T_w - T_o} = 1 - \frac{n + 3}{n + 4} B \tag{3}$$

It is evident that when B is negative the mean temperature rises above the wall temperature. The point at which the mean temperature reaches the wall temperature depends upon B and the Graetz number. For B = - 1 the mean temperature reaches the wall temperature at a Graetz number of 20 and exceeds it for lower Graetz numbers.

The values of Nusselt number were found to reach asymptotic values at small Graetz numbers and are represented by

$$Nu = \frac{h_a D}{k} \quad \frac{2}{\pi} \quad \frac{(n + 4) - B\ (n + 3)}{(n + 4) + B\ (n + 3)} \quad Gz \tag{4}$$

where h_a is defined by

$$WC\ (T_L - T_o) = h_a\ A\ \frac{(T_w - T_o) + (T_w - T_L)}{2} \qquad (5)$$

These asymptotic relations for $n = 2$ are plotted for Nusselt number versus Peclet number x $\frac{D}{L}$ in Figure 6.

CORRELATION OF HEAT TRANSFER DATA

For the adiabatic test runs the temperature rise versus pressure drop has been plotted in Figure 4 for two runs of SAE 30 oil and one run of SAE 70 oil. Figure 4 shows that there is a significant difference between the values of temperature rise at any given pressure drop for the three test runs. Reviewing the data shown in Table I and II it is noted that the lowest value of temperature rise at the high pressure drops occurs in the run of SAE 70 oil. The flow rates for the SAE 70 oil run are very low compared to those of the SAE 30 oil. The temperature rise is highest for the run of SAE 30 oil at the elevated inlet temperature and this run also has the highest flow rate. The increase in temperature rise with increased volume flow rates indicates the possibility that the heat transfer from the outside tube wall becomes significant at low flow rates.

Two test runs were made with the steam temperature held constant at 208°F. Using SAE 30 oil a maximum pressure of 1400 psig was obtained and using SAE 70 oil a maximum pressure of 3500 psig was obtained. From Figure 4 it is evident that there is only a very small viscous effect on fluid temperature below 3500 psig. The outlet temperatures for these runs approach the wall temperature and were 204.5^{o}F for SAE 30 oil and 205.5^{o}F for SAE 70 oil.

For correlation of these data the specific heat and coefficient of thermal conductivity are assumed constant at 0.45 BTU/lb-^{o}F and 0.078 BTU/hr-ft-^{o}F respectively. These numerical values were selected

after reviewing data in the International Critical Tables (4), Maxwell (5), and McAdams (6).

The heat added to the fluid during these tests was calculated from the flow rate, the specific heat, and the temperature difference using the equation

$$q = WC\Delta T \tag{6}$$

where

$$\Delta T = T_L - T_o \tag{7}$$

The heat transfer coefficient, h_a, is defined by the equation

$$q = h_a A \Delta T_a \tag{8}$$

where

$$\Delta T_a = \frac{(T_w - T_o) + (T_w - T_L)}{2} \tag{9}$$

The Nusselt number was computed using the value of h_a obtained from equation 8. Values of Nusselt number were plotted against the computed value of Peclet number x $\frac{D}{L}$ and are presented in Figure 5. These results for the test runs using steam closely correlate the data presented in Jakob (1) for the Graetz solution based on average temperature difference.

Since there was no apparent viscous effect during the runs using steam, alcohol vapor was used to reduce the constant wall temperature so that the viscosity of the oil was high enough to give the high pressures desired. With alcohol vapor and SAE 70 oil pressures up to 9000 psig were obtained, and at 9000 psig with an inlet oil temperature of 105°F and a wall temperature of 151°F an exit oil temperature of 166°F was reached. Data for this test run is presented in Table V.

Defining the heat transfer coefficient for this test run as in equation 5

$$WC(T_L - T_o) = h_a A \frac{(T_w - T_o) + (T_w - T_L)}{2} \qquad (5)$$

the heat transfer coefficient was calculated and the values of Nusselt number were plotted versus Peclet number x $\frac{D}{L}$ as shown in Figure 5.

From the data presented in Table V, it is apparent that the term $(T_w - T_L)$ becomes negative at all pressures above 3500 psig. The values of Nusselt number above 3500 psig for this run are above the Graetz solution by a significant amount and are, at the 9000 psig point, double the value which would be anticipated for flow at low pressure drops and a corresponding value of Peclet x $\frac{D}{L}$.

Using the relationship of equation 3, the values of B were calculated for this test run. With the calculated values of B and Peclet number x $\frac{D}{L}$ the theoretical values of Nusselt number (Nu_T) were obtained from Figure 6 and tabulated in Table V. At these low values of Peclet number x $\frac{D}{L}$, the values of Nusselt number based on the heat transfer coefficient of equation 5 agree very closely with the values of theoretical Nusselt number obtained from Figure 6. The maximum deviation from the theoretical value is ten per cent.

CORRELATION OF PRESSURE DROP DATA

A friction factor defined by the equation

$$\Delta P = f \frac{L}{D} \frac{\rho V^2}{2} \tag{10}$$

was calculated for each of the test runs. The corresponding Reynolds number was calculated and both friction factor and Reynolds number are included in the data sheets for the test runs presented in Tables I through V. The resulting friction factors were plotted against Reynolds numbers and are presented as Figure 7 and 8.

The analytical solution for friction factor as a function of Reynolds number in fully developed isothermal laminar flow is shown in Figures 7 and 8 and is represented by

$$f = \frac{64}{Re} \tag{11}$$

Values of friction factor for the adiabatic test runs are slightly higher than the values determined from equation 11. Friction factors for the cases of heating with the wall temperature at 208°F and low pressure drops, represented by the solid line in Figure 7, fall well below the values represented by equation 11 but compare favorably with the values of friction factor presented by Colburn (7) for the case of constant wall temperature heat transfer.

For the case of heating with the wall temperature at 151°F it is evident from Figure 7 that the friction factor at low pressures is well below the line representing equation 11 and that as pressure is increased the value of friction factor approaches and crosses the line represented by equation 11.

The fact that the values of the friction factor for the heating case fall below those of equation 11 can be explained by the fact that the friction factor and Reynolds number are based on the average bulk temperature of the fluid and the actual temperature of the fluid at the wall is substantially higher than the average bulk temperature used in the calculations.

As the pressure of the system is increased, as in the test run of SAE 70 oil with the wall temperature at 151°F, the average bulk temperature rises due to viscous effect and the difference between wall temperature and average fluid bulk temperature is reduced as indicated in Table V. This effect may explain the return of the friction factor values to those of equation 11 at high pressures. Fluid viscosity is also a function of pressure, and data presented in the International Critical Tables (4) indicates that an increase in pressure is accompanied by an increase in viscosity for petroleum oils. Although the magnitude of the viscosity change with pressure is not known for the working fluids of this test, this characteristic increase may account for the fact that the friction factor values without heating are above those of equation 11.

CONCLUSIONS

When high pressure laminar flow of viscous fluids at low Reynolds numbers is considered in conjunction with high transfer, care must be taken in defining a heat transfer coefficient. In petroleum oils flowing at pressures above 3500 psig a significant temperature rise caused by viscous effect occurs in the fluid.

For laminar flow of oils through a tube having a constant wall temperature, the test data for high pressure drops substantiates the values of theoretical Nusselt number obtained by Toor at values of Peclet number x $\frac{D}{L}$ between 1.0 and 3.0. At pressure drops below about 3500 psi, the data obtained substantiates that presented in the Graetz solution over the range of Peclet number x $\frac{D}{L}$ from 0.4 to 5.0.

The friction factor values for a system subjected to heating are dependent upon variation in average bulk temperature with respect to wall temperature and on the viscous effect caused by system pressure. Heating of fluids in laminar flow at low pressures results in a friction factor well below the analytical value of 64/Re based on average fluid bulk temperature. As system pressure is increased the viscous effect causes the average fluid bulk temperature to rise. This viscous temperature rise reduces the difference between average fluid bulk temperature and wall temperature and results in the friction factor approaching the value of 64/Re with increasing pressures. With low wall temperatures and high pressures the friction factor may exceed the value of 64/Re.

If increased pressure is assumed to increase viscosity, the values of friction factor without heating could be expected to be above those given by 64/Re if Reynolds number is based on the viscosity at the average fluid bulk temperature and atmospheric pressure.

RECOMMENDATIONS

If continued investigation of the characteristics of fluid flow and heat transfer at high pressure drops and low Reynolds numbers is considered, it is recommended that a more refined test apparatus be used. A new test apparatus should include intermediate pressure and temperature indications along with the inlet and exit conditions monitored during these tests. A new test apparatus should also include a heat exchanger ahead of the test section inlet so that fluid inlet temperature can be held constant or varied as desired for any test run.

The values of adiabatic temperature rise with pressure should be obtained using a more satisfactory method of maintaining an adiabatic wall or reducing the percentage error caused by system losses. A system of larger proportions capable of operating at substantially higher flow ratas would be desirable.

It is also recommended that any pump used in future testing be able to provide high flow rates over the desired pressure range so that different diameter test sections could be accommodated. This type pump would definitely be required if it is desirable to maintain high pressures and vary the parameters of Peclet number x $\frac{D}{L}$ or Reynolds number.

Table I

Data Sheet for SAE 30 with Tube Wall Insulated

Pressure PSI	Inlet Temp. °F	Outlet Temp °F	Flow Rate Ml/Min	f	Re
7750	68	94.5	14.5	43.5	2.03
7000	68	91	13.0	48.5	1.68
6000	69	87	10.5	63.5	1.26
5000	69	82.5	8.5	81.0	0.96
4000	69	79	6.2	117	0.66
3000	69	76.5	4.8	157	0.50
8000	94	128	43	5.1	16.1
7000	91	118	31	8.6	9.65
6000	88	105.5	22	14.5	5.29
5000	87.5	96	15.5	21.5	3.26
4000	84	88	10.5	42.7	1.77
3000	81	83.5	6.2	91.0	0.84

Table II

Data Sheet for SAE 70 with Tube Wall Insulated

Pressure PSI	Inlet Temp. °F	Outlet Temp. °F	Flow Rate Ml/Min	f	Re
9000	78	88	4.50	551	0.150
8500	78	88	4.00	660	0.134
8000	78	87	3.75	710	0.114
7500	78	86	3.37	820	0.100
7000	78	86	3.12	892	0.093
6500	78	85	3.00	892	0.088
6000	78	84	2.75	980	0.078
5500	78	83	2.37	1230	0.067
5000	78	82	2.25	1230	0.063
4500	78	81.5	2.00	1390	0.056
4000	78	81	1.75	1620	0.048
3500	78	80	1.62	1650	0.044
3000	78	80	1.37	1970	0.038
2500	78	79	1.12	2480	0.031
2000	78	78	0.87	3260	0.024

Table III

Data Sheet for SAE 30 with Tube Wall 208°F

Pressure PSI	Inlet Temp °F	Outlet Temp °F	Flow Rate Ml/Min	Nu	Pe $\frac{D}{L}$	f	Re
1400	88	203.5	52	2.17	4.40	0.63	47.5
1200	89	204.5	46	1.96	3.89	0.69	43.0
1000	90	204.5	40	1.70	3.39	0.76	37.5
800	91	204.5	33	1.40	2.80	0.91	31.8
600	91	204.5	25	1.06	2.11	1.19	24.1
400	91	204.5	18	0.77	1.52	1.53	17.4

Table IV

Data Sheet for SAE 70 with Tube Wall 208°F

Pressure PSI	Inlet Temp °F	Outlet Temp °F	Flow Rate Ml/Min	Nu	Pe $\frac{D}{L}$	f	Re
3500	96	204.5	50	2.10	4.25	1.68	18.7
3250	99	205.5	48.5	2.08	4.07	1.66	18.2
3000	99	205.5	47	2.02	4.00	1.64	17.6
2750	103	205.5	45	1.94	3.81	1.64	16.8
2500	105	205.5	40.5	1.73	3.43	1.83	15.2
2250	105	205.5	38	1.63	3.22	1.91	14.3
2000	105	205.5	34.5	1.49	2.95	2.02	13.0
1750	105	205.5	31	1.36	2.62	2.20	11.6
1500	105	205.5	27.5	1.18	2.33	2.38	10.3
1250	104	205.5	23	0.99	1.95	2.85	8.6
1000	104	205.5	21	0.84	1.71	2.85	7.9
750	100	204.0	15	0.63	1.28	4.04	5.6
500	96	203.0	9.5	0.39	0.81	6.68	3.6

Table V

Data Sheet for SAE 70 with Tube Wall 151°F

Pressure PSI	Inlet Temp. °F	Outlet Temp. °F	Flow Rate Ml/Min	Nu	Pe $\frac{D}{L}$	f	Re	B	Nu_T
9000	105	166	31	2.64	2.50	11.7	7.72	-.402	2.60
8500	105	166	30	2.55	2.45	12.4	7.48	-.402	2.55
8000	105	166	29	2.47	2.37	12.6	7.21	-.402	2.48
7500	105	165	28	2.27	2.27	11.9	6.98	-.363	2.10
7000	105	163	26.5	1.96	2.15	12.4	6.33	-.312	1.82
6500	103	158	25	1.45	2.02	12.9	5.24	-.177	1.35
6000	103	157	24	1.32	1.95	13.0	4.98	-.150	1.23
5500	103	156	23	1.22	1.87	13.0	4.70	-.126	1.13
5000	102	154	21	1.05	1.70	14.1	4.15	-.072	0.96
4500	98	154	19	0.92	1.54	15.6	3.54	-.066	0.85
4000	95	153	17	0.79	1.38	17.2	2.93	-.044	0.74
3500	93	152	15.5	0.69	1.25	18.2	2.47	-.012	0.64
3000	90	150	13.5	0.57	1.10	20.5	1.97	+.018	0.54
2500	87	150	12	0.50	0.98	21.7	1.68	+.072	0.98
2000	87	148	10	0.38	0.82	24.8	1.35	+.056	0.41
1500	87	148	7.5	0.29	0.61	32.8	1.02	+.056	0.30
1000	87	148	5.5	0.21	0.45	41.3	0.74	+.056	0.22

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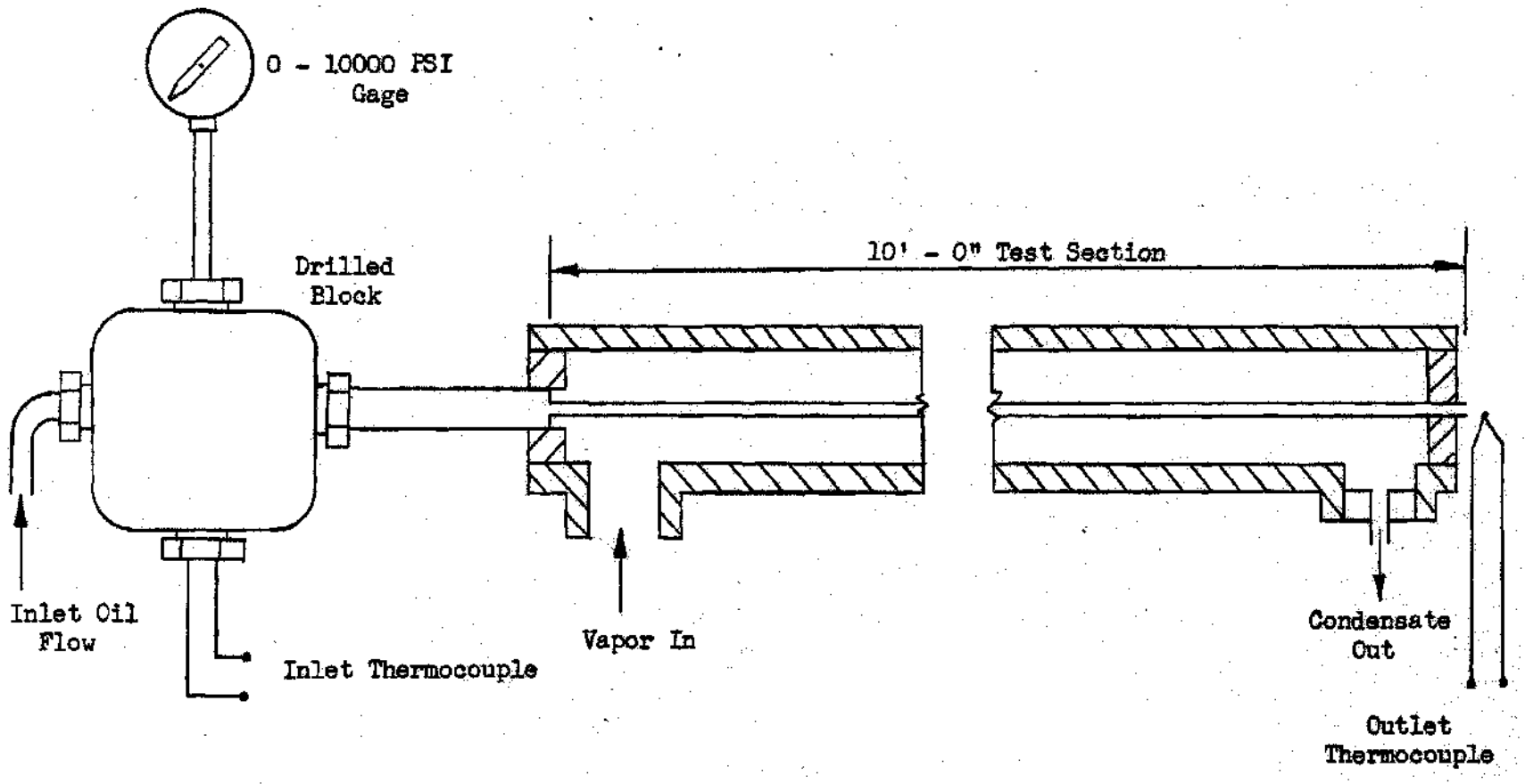
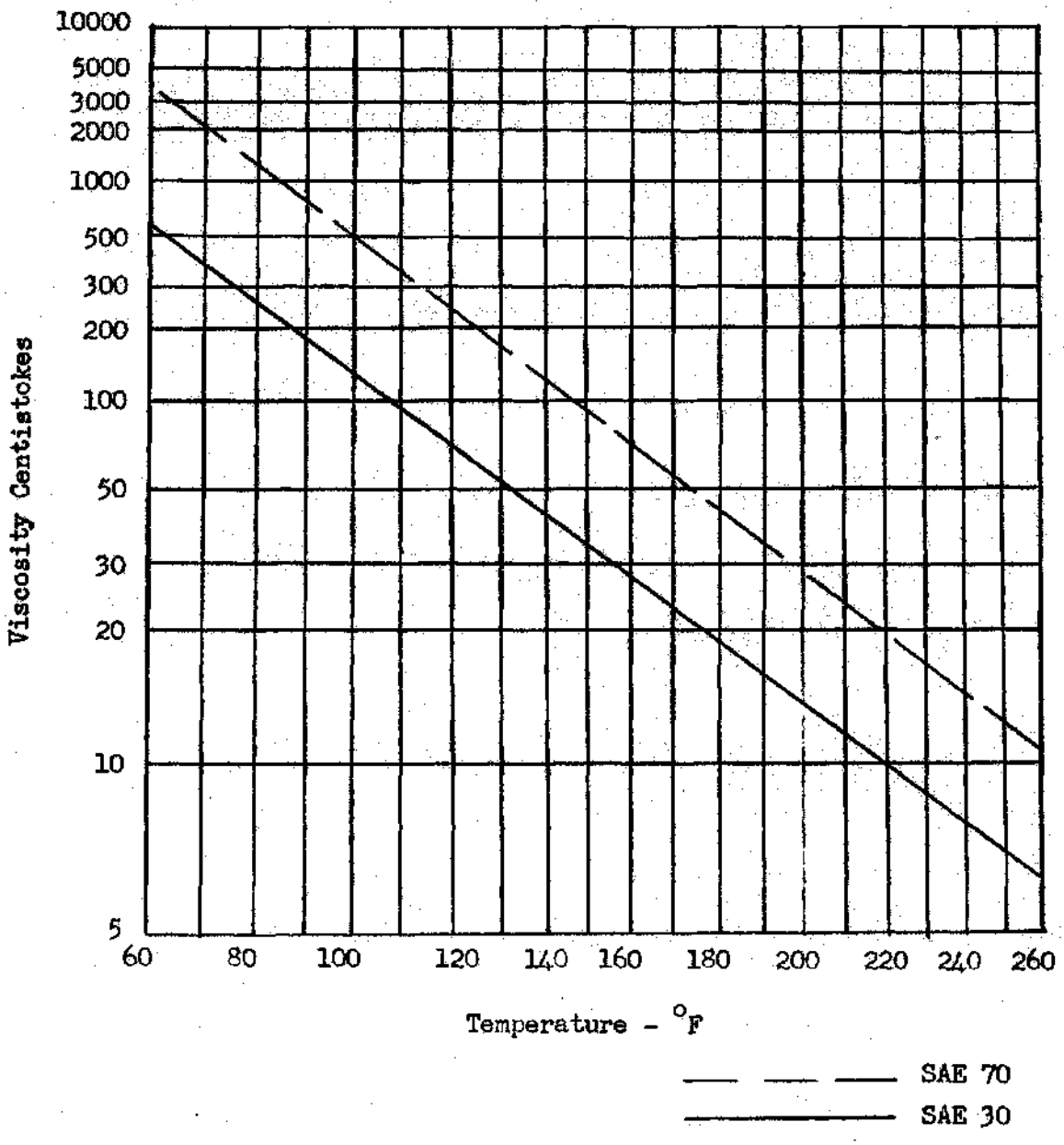


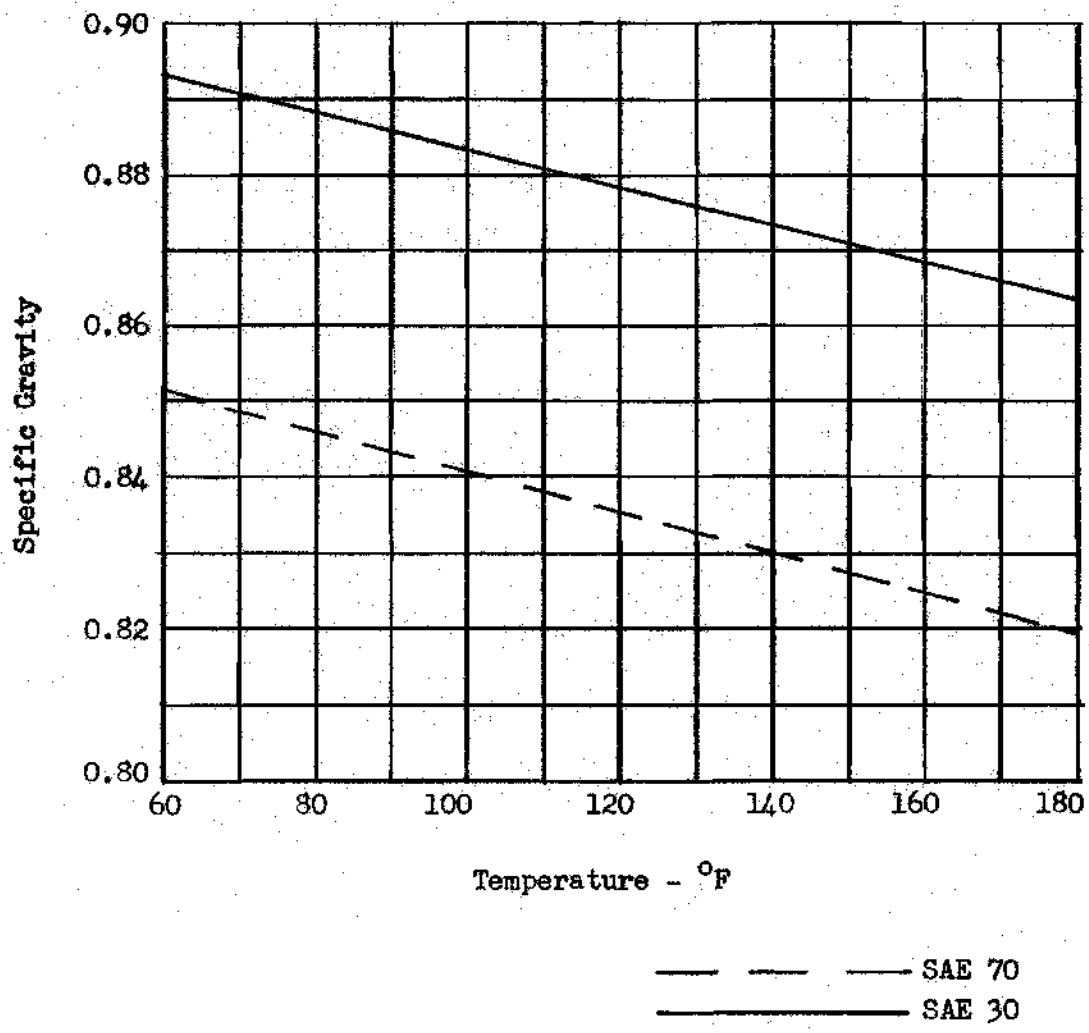
Test Apparatus Schematic Diagram

Figure 1



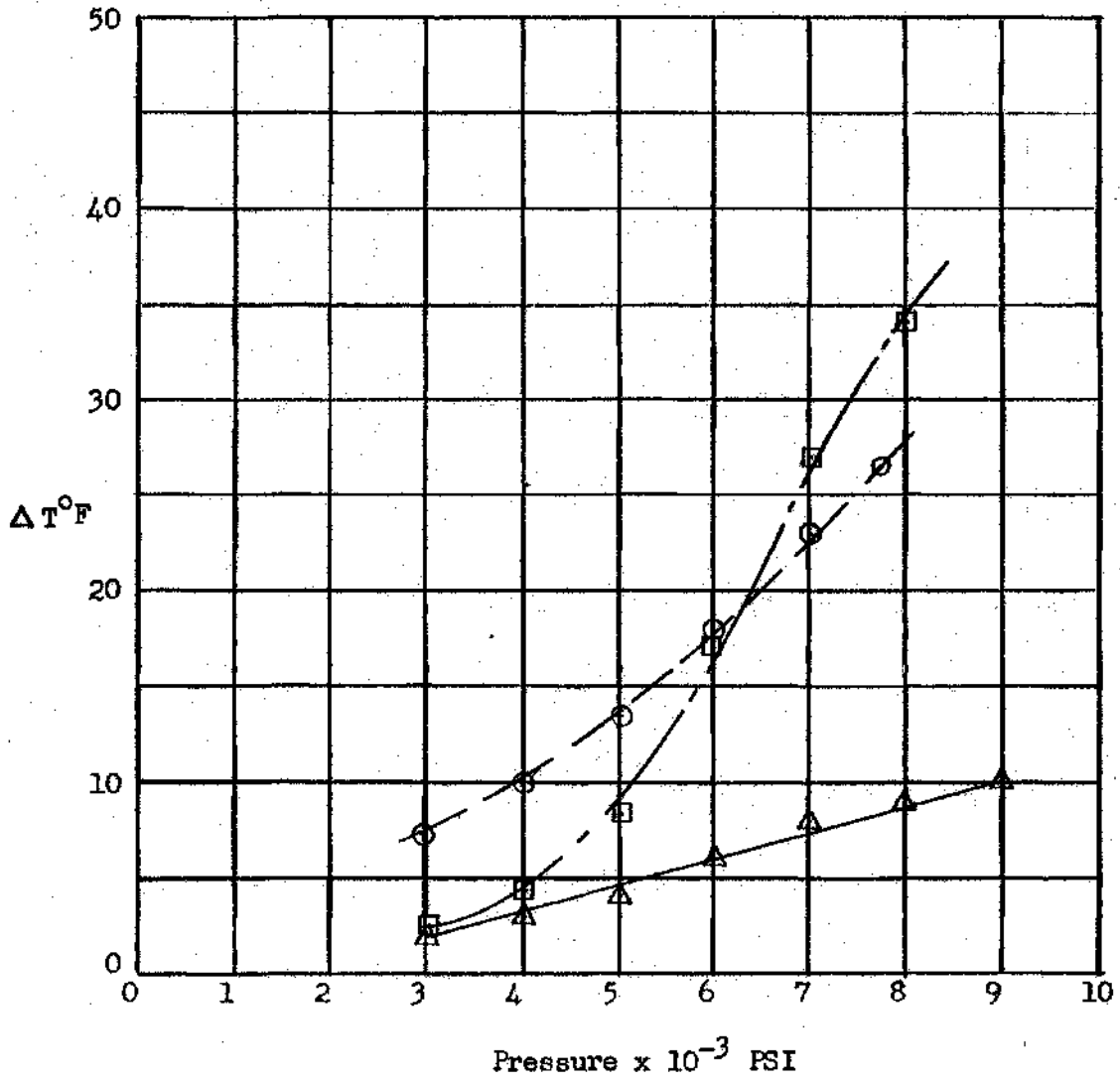
Variation of Viscosity with Temperature

Figure 2



Variation of Specific Gravity with Temperature

Figure 3



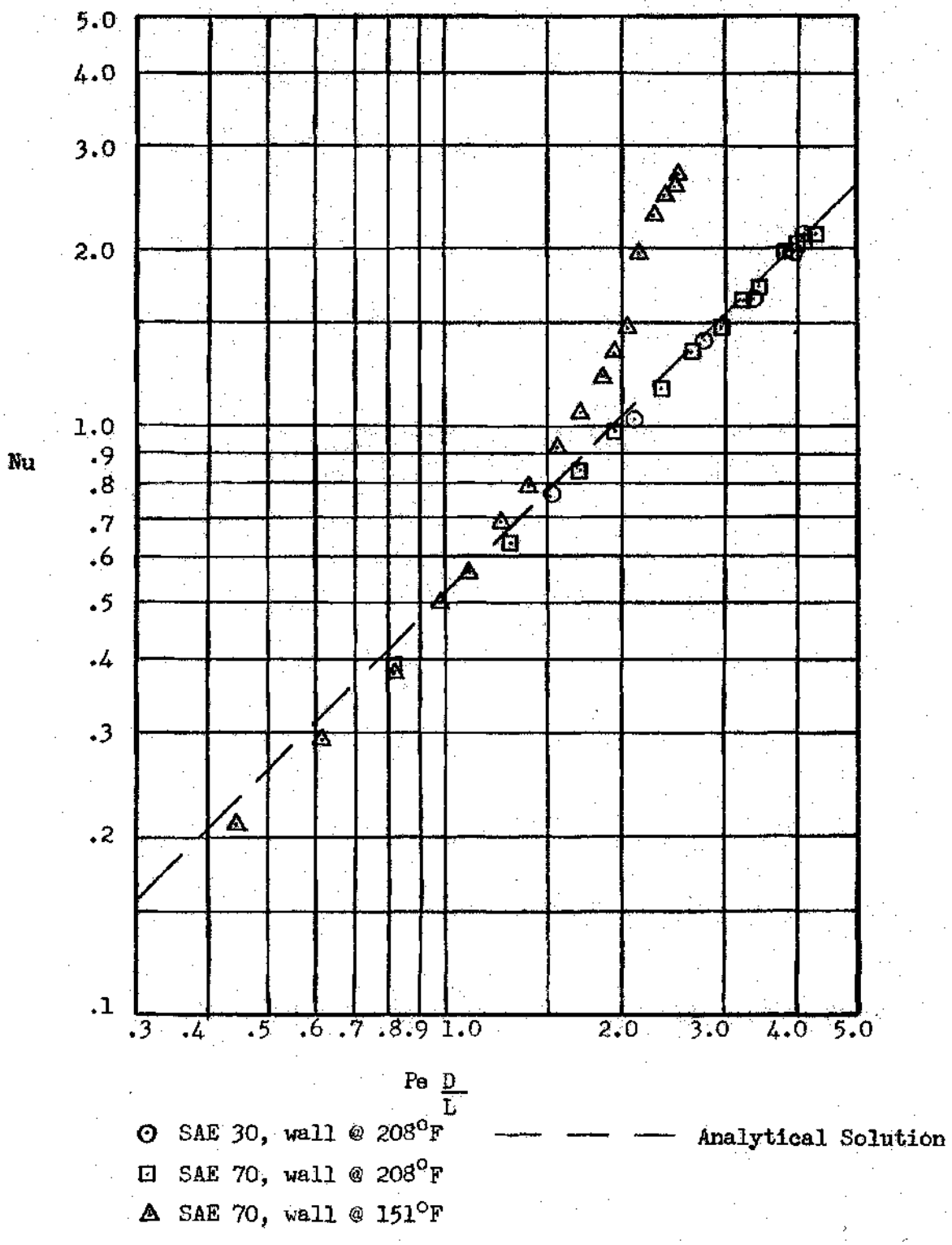
⊙ SAE 30, Inlet Temp 69°F

⊡ SAE 30, Inlet Temp 81-94°F

△ SAE 70, Inlet Temp 78°F

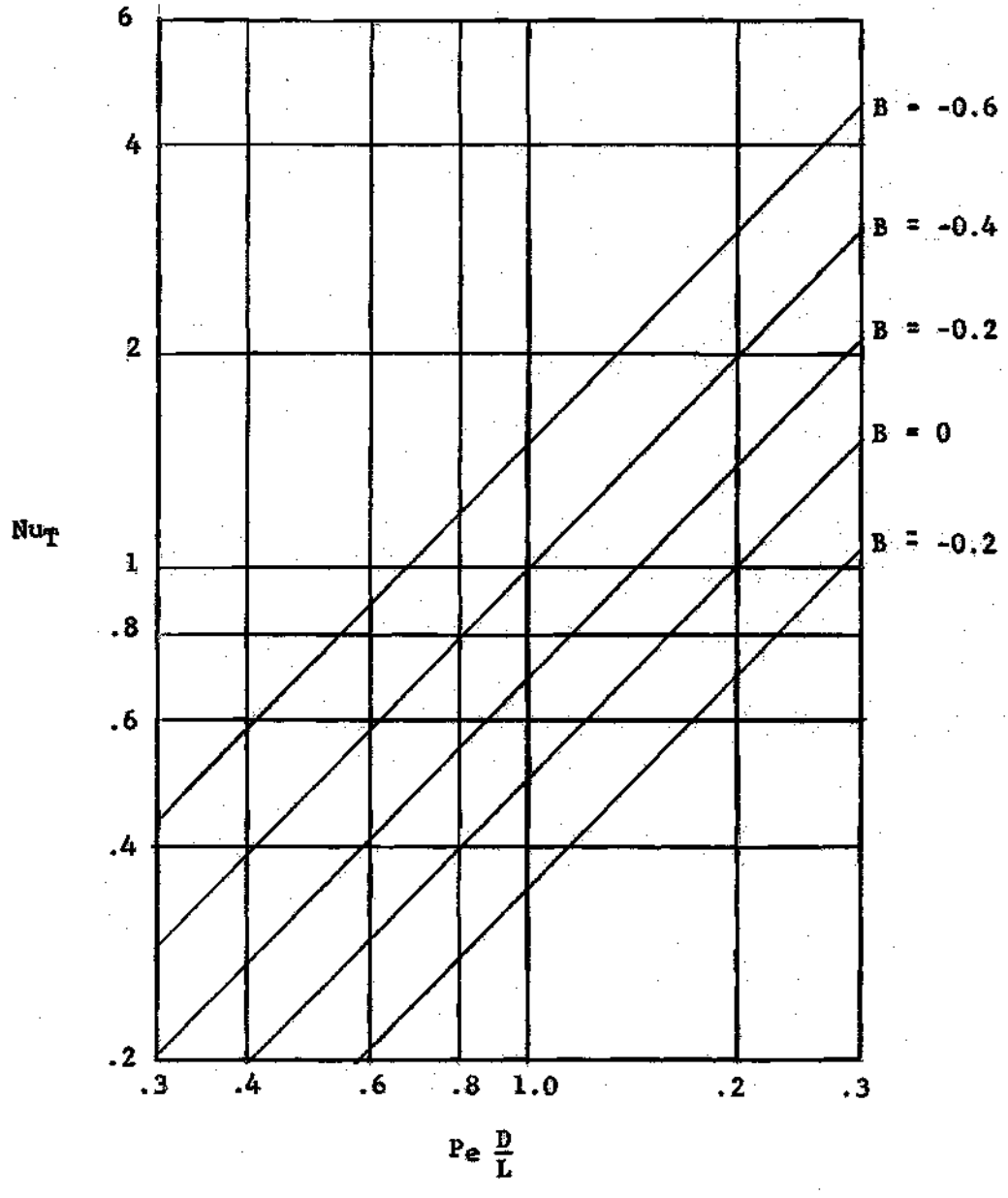
Adiabatic Temperature Difference Versus Pressure

Figure 4



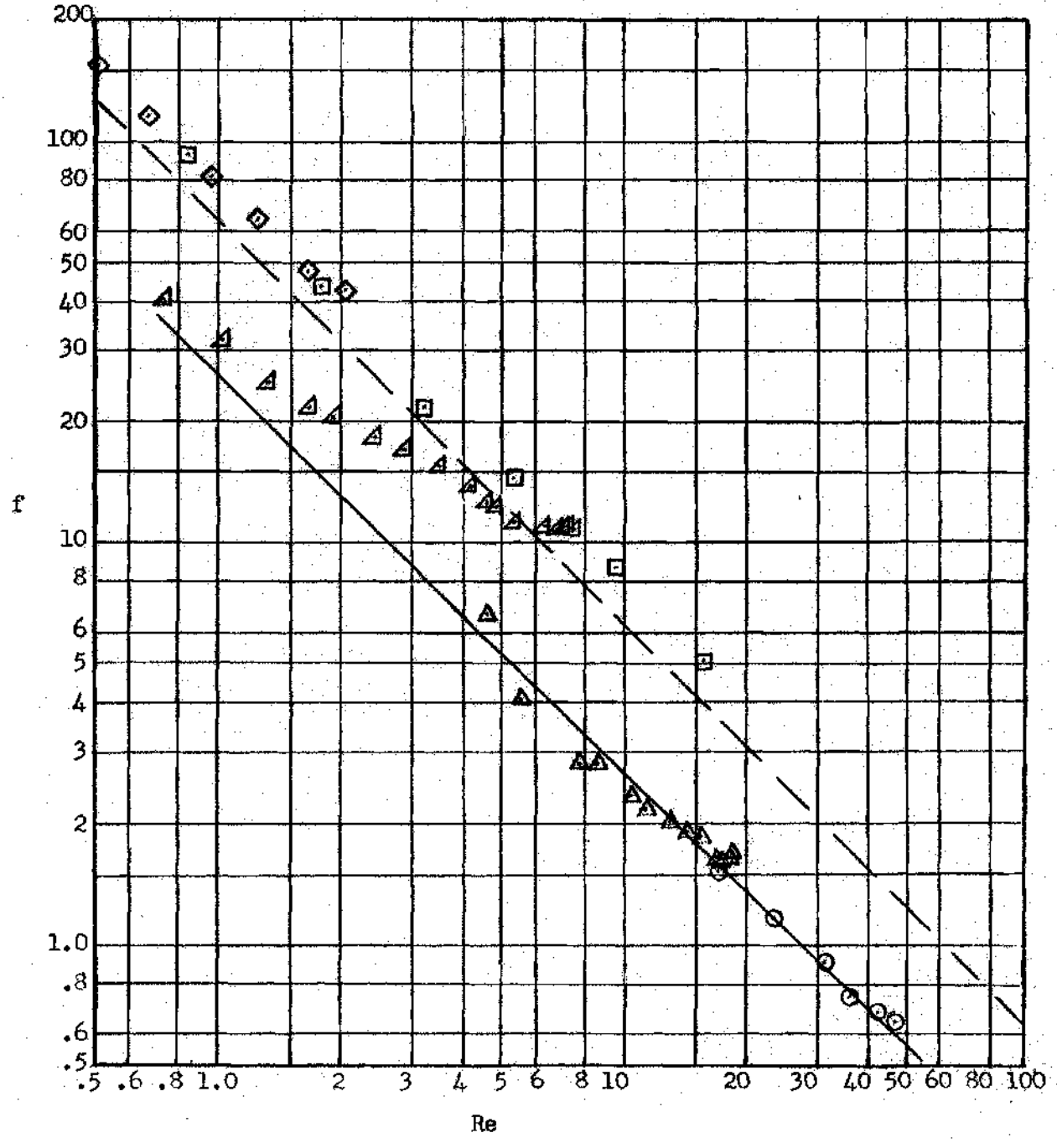
Variation of Nusselt Number with Peclet Number X $\frac{D}{L}$

Figure 5



Variation of Theoretical Nusselt Number With Peclet Number x $\frac{D}{L}$

Figure 6



⊙ SAE 30, wall @ 208°F

⊡ SAE 30, wall insulated

◇ SAE 30, wall insulated

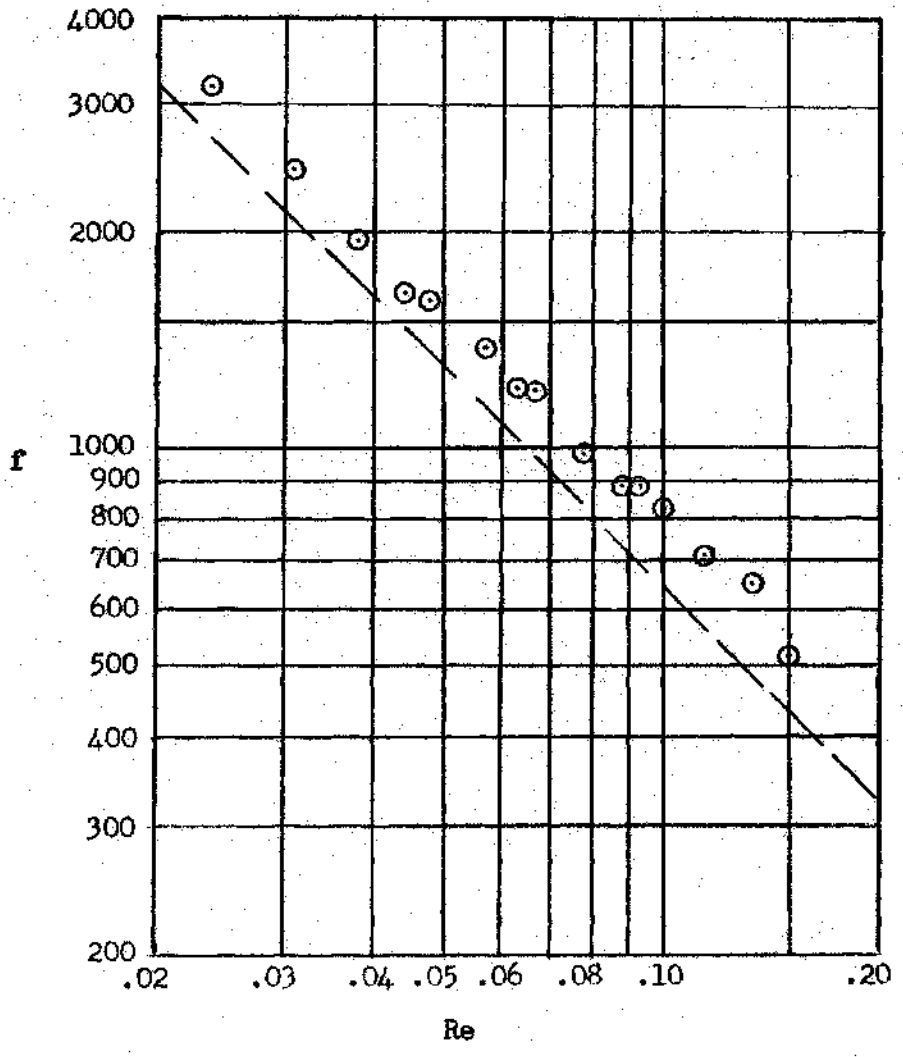
△ SAE 70, wall @ 208°F

◿ SAE 70, wall @ 151°F

— — — 64/Re

Variation of Friction Factor with Reynolds Number

Figure 7



⊙ SAE 70, wall insulated

— — — 64/Re

Variation of Friction Factor with Reynolds Number

Figure 8

APPENDIX A

CALCULATION OF ADIABATIC TEMPERATURE RISE

An analytical investigation of the adiabatic temperature rise is presented in an effort to determine the relative magnitude of temperature rise to be anticipated. The steady flow energy equation without shaft work and with no change in elevation or kinetic energy is given as

$$ {}_1q_2 = H_2 - H_1 \tag{12} $$

For the case considered

$$ V_1 = V_2 \text{ and } Z_1 = Z_2 $$

Therefore,

$$ {}_1q_2 = H_2 - H_1 \tag{13} $$

For the adiabatic case ${}_1q_2 = 0$ and the equation reduces to

$$ H_1 = H_2 \tag{14} $$

The change of states between H_1 and H_2 can be defined by the thermodynamic relation

$$ H_2 - H_1 = \left[\int_{T_1}^{T_2} C\,dt \right]_{P = P_1} + \left[\frac{\int_{P_1}^{P_2} \left(v - T \frac{\partial v}{\partial t}_p \right) dp}{J} \right]_{T = T_2} \tag{15} $$

but $H_2 - H_1 = 0$ and

$$ \left[\int_{T_1}^{T_2} c\,dt \right]_{P = P_1} + \left[\frac{\int_{P_1}^{P_2} \left(v - T \frac{\partial v}{\partial t_p} \right) dp}{J} \right]_{T = T_2} = 0 \tag{16} $$

Assuming:

$$v - T\left(\frac{\partial v}{\partial t}\right)_p = \text{constant}$$

then,

$$C = \text{constant}$$

$$T_2 - T_1 = \frac{\left[v - T\left(\frac{\partial v}{\partial t}\right)_p\right](P_1 - P_2)}{CJ} \qquad (17)$$

At 8000 psi the substitution for SAE 70 reduces to

$$T_2 - T_1 = \frac{(.01913 - .00186)\ (8000)\ (144)}{.45 \times 778}$$

or

$$T_2 - T_1 = 57^{o}F$$

where $v - T\left(\frac{\partial v}{\partial t}\right)_p$ was evaluated at atmospheric pressure using data from Figure 3.

APPENDIX B

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