EXPERIMENTAL EVALUATION OF HEAT TRANSFER AND PRESSURE DROP PERFORMANCE OF INTERNALLY FINNED TUBES

by

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ABSTRACT

Heat transfer and pressure drop measurements were made on three test tubes, a quintuplex tube, and two single finned tubes. The single finned tubes tested were of the high spiral type. The spiral finned tubes have been arbitrarily classified in the literature as low spiral with fin height to inside diameter ratio $r < 0.065$, and as high spiral with the ratio $r \geq 0.065$. Lubricating oil was used as the test fluid. Three sets of test runs were performed on each of three test sections, fixing the values of the oil inlet temperature and water flow rate in each set, while varying the oil flow rate to cover as much as possible a wide range of operating conditions. Effects of oil inlet temperature and cooling water flow rate on the performance of test tubes were also investigated.

At almost the same operating conditions it was found that the quintuplex finned tube provides a sizable decrease in heat transfer area, but at the expense of high pumping power requirement. It is, therefore, suggested that quintuplex tube should be preferred in industrial applications in which the size of the heat exchanger is an important factor. But if the pumping power is the criterion for selection, the single-finned tube is superior.

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NOMENCLATURE

A_{xs} = cross-sectional flow area of the tube, ft^2

C_{p_o} = specific heat of oil at constant pressure, $Btu/lb_m F$

C_{p_w} = specific heat of water at constant pressure, $Btu/lb_m F$

D_e = equivalent diameter of tube, ft

D_h = hydraulic diameter of tube, ft

D_i = Inside diameter of tube, ft

D_o = outside diameter of tube, ft

e = heat balance error

f_e = friction factor based on D_e

f_i = friction factor based on D_i

g_c = gravitational constant, $lb_m\ ft/\ lb\ Sec^2$

L = tube length, ft

$\dot{m}_o$ = mass flow rate of oil, lb_m/hr

$\dot{m}_w$ = mass flow rate of water, lb_m/hr

ΔP = pressure drop, psi

P = pumping power, lbft/min

Q = rate of heat transfer, Btu/hr

Q_o = rate of heat lost by oil, Btu/hr

Q_w = rate of heat gained by water, Btu/hr

N_{Re} = Reynolds number

N_{Re} = Reynolds number based on D_e

N_{Ri} = Reynolds number based on D_i

$T_{o,i}$ = oil inlet temperature, F

$T_{o,o}$ = oil outlet temperature, F

$T_{w,i}$ = water inlet temperature, F

$T_{w,o}$ = water outlet temperature, F

ΔT_m = log-mean temperature difference, F

U = over-all heat transfer coefficient, Btu/hr-ft^2-F

v_o = velocity of oil, ft/sec

W = weight of length L of the tube, lb

x = entrance length of tube material, ft

γ = specific weight of tube material, lb /ft^3

ρ_o = density of oil, lb$_m$/ft^3

ν_o = kinematic viscosity, ft^2/sec

1. INTRODUCTION

Common tubular heat exchangers have a long history of industrial applications. Their function is to transfer the heat from one fluid to another. Boilers, condensers, automobile radiators and water heaters are some of the engineering examples in which such transfer occurs.

In general, one tries to maximize the heat transfer rate while keeping down the operating costs and the capital investment. Simultaneous achievement of all these goals is unfortunately impossible, and different applications favor different choices. Thus, for example, where space and weight are at a premium, we have to be prepared to spend more on the equipment in order to make it more compact. Compactness can be achieved in different ways, but the immediate goal is always the same: diminution of convective surface resistance on either side of the tube. A good review of such enhancement techniques can be found in Bergles (1). For our purposes it is sufficient to point out that in an effort to diminish over-all heat transfer resistance, one should pay particular attention to the largest component of that resistance. Thus, if extended surfaces are used, one should apply them on the side on which greater resistance is found. Hence, the need for internally finned tubes for such applications as compact oil coolers which are the object of the present study.

In the earlier studies in that field, Brouillette et al. (2) and Hilding and Coogan (3), fins were produced by cutting notches, or by inserting long metallic strips in the inner surface of smooth tubes. But due to new manufacturing techniques it is easier now to produce integral inner-fin tubing with different fin numbers, height and twist. These integral inner tubes are manufactured by a special process called forged fin process. An American subsidiary of Noranda Metal Industries, Inc. is producing such tubes with different configurations. Single finned, duplex, triplex, quadruplex and quintuplex finned tubes are the examples of multiple finned tubes where two or more single internally finned tubes are fitted one inside the other. We were fortunate to have obtained some of these tubes from Noranda free of charge, thus enabling us to perform the experiments which are the object of this study.

This report deals with the design and construction of a heat-exchange test section, for the study of heat transfer and pressure drop performance of multiple integral inner finned tubes, namely, quintuplex and two single finned tubes, for laminar, transition and turbulent oil flow. Previous study done by Soliman and Feingold (4) could not evaluate the behaviour of quintuplex finned tube at higher oil mass flow rate because the test section was too long for the available pumping power.

It was, therefore, suggested that I should design a shorter test section with which it would be possible to determine the nature of quintuplex tube for laminar, transition, and turbulent oil flows. It was found by Soliman and Feingold that the quintuplex tube could provide a sizable decrease in heat transfer area when compared with other tubes. This led to the concept of "equivalent length" meaning the length of quintuplex tube which would perform the same heat transfer task as a given length of a single-finned tube. Soliman and Feingold arrived at that equivalent length through an analytical transformation of results obtained for a longer quintuplex tube. It was now left to me to prove their calculations by actual experiments.

Likewise, the water inlet temperature in the experiments of Soliman and Feingold was not subject to regulation and, therefore, it was difficult to obtain strictly comparable results, or indeed to repeat the same run on another day when the water supply came at a different temperature. To remedy this, I designed an apparatus permitting the recycled water to be entering the test section at the temperature regulated within ± 1 degree Fahrenheit.

The cooling lenght used by Soliman and Feingold for the single-finned tube was 8-ft and they had calculated that the equivalent lenght of the quintuplex tube should be approximately 2 ft.

My test section design employed 5-ft length of quintuplex tube with the measuring stations separated by a 2-ft distance which is herein referred to as the cooling section. The remaining three feet (1.5 on each side) are the entrance and exit sections similar to the practice adopted by Soliman and Feingold.

My shorter tube having permitted a wide range of flow rates (specifically, adding higher rates), produced interesting results. First of all, it confirmed previous workers' findings regarding the enhancement of heat transfer in quintuplex as compared with single-finned tube. Secondly, when the present shorter quintuplex tube is compared with the previous longer single finned tube, the heat transfer enhancement is still observed at low mass flow rates of oil. But then, at higher rates the advantage begins to disappear. Possible reason for the just described behaviour could be that, in the case of a single-finned tube, transition from one flow regime to another is taking place at higher mass flow rate of oil. The comparison between the present study and the previous work is shown in Figs. 42, 43 and 44. From these figures it is observed that due to change of flow regime in the single-finned tube, a jump in the value of "Q" occurs at higher mass flow rate of oil, while in the quintuplex tube no such behaviour can be observed.

Therefore, the superiority of a very short quintuplex tube over longer single-finned tube is no longer in evidence at higher mass flow rates of oil. This was not predicted by Soliman and Feingold and provides a significant new piece of information for the developers of future heat transfer equipment who will have to choose between a wide variety of internally finned tubes. Also, it is observed that this decrease of size is associated with high pressure drop and is, therefore, affecting the operational and capital costs.

In conclusion, this study proves experimentally the validity of the concept of equivalent length adopted by Soliman and Feingold, but at the same time shows that its applicability is limited to relatively low flow rates.

2. EXPERIMENTAL SET-UP AND PROCEDURE

The schematic diagram of the apparatus is shown in Fig. 1 and its photographic view in Fig. 45. Three tubes were tested, the quintuplex and two single-finned tubes, and experimental data were compiled at almost the same operating conditions. Schematic diagram of the cross-sections of the tubes is shown in Fig. 2 and its dimensions are given in Table 1.

The working fluid used was SAE 10W-30 Motor oil (Shell Rotella T), and its physical properties are give in Table 2, which were measured in the laboratory by Noranda Research Centre of Point Claire, Que., and compared closely with the manufacturing data.

The cross-sectional flow area and the equivalent diameter of the tube is given by

$$A_{xs} = \frac{\pi}{4} D_o^2 - \frac{W}{\gamma L} \qquad (1)$$

$$D_e = \sqrt{\frac{4A_{xs}}{\pi}} \qquad (2)$$

2.1 Description of the Apparatus

Figures 1 and 45 represent respectively the schematic and photographic views of the apparatus.

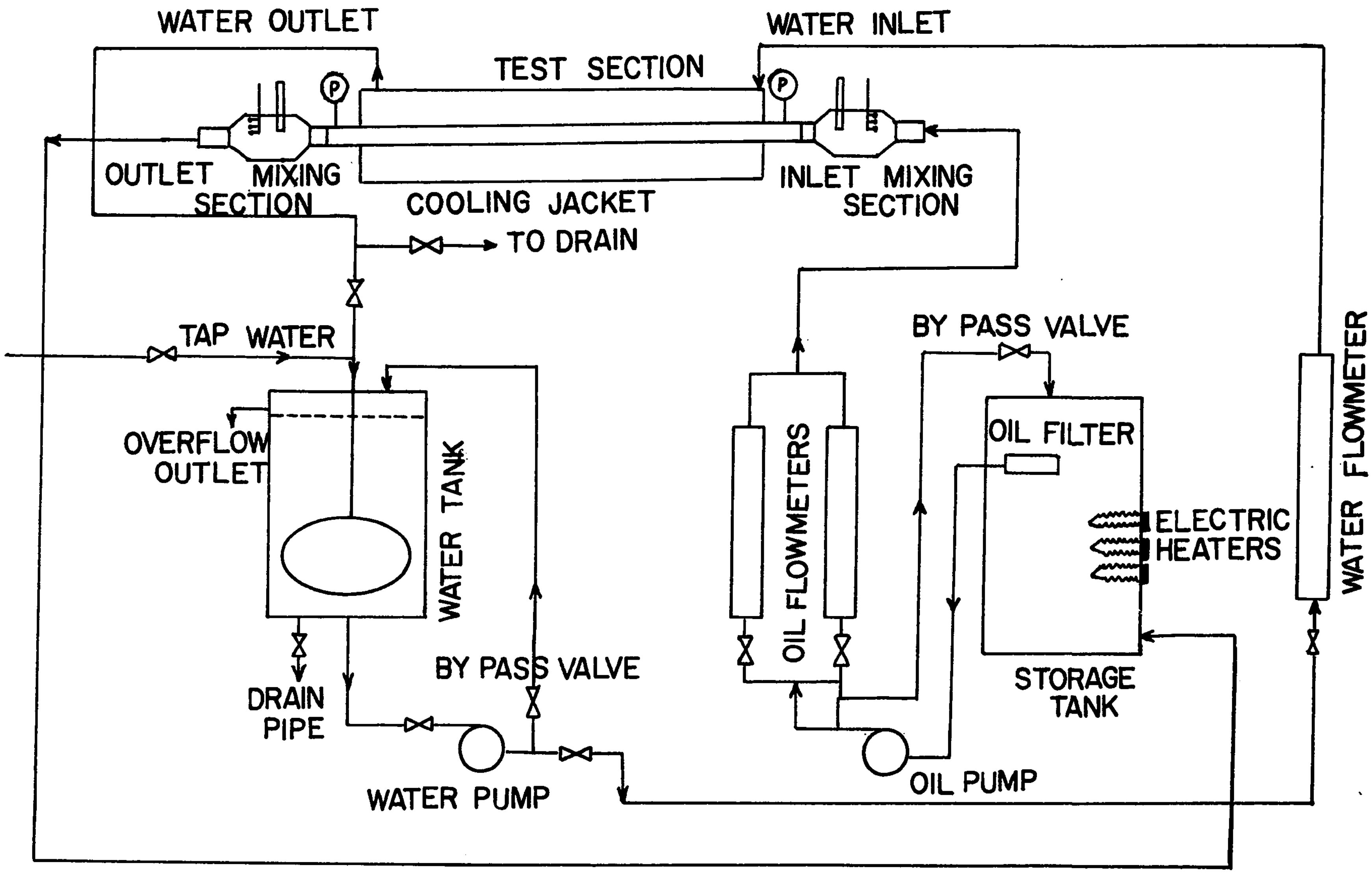
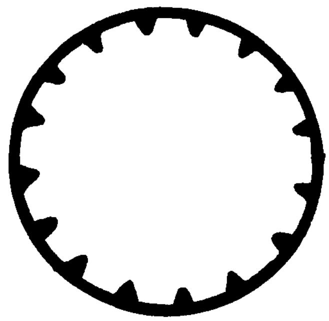
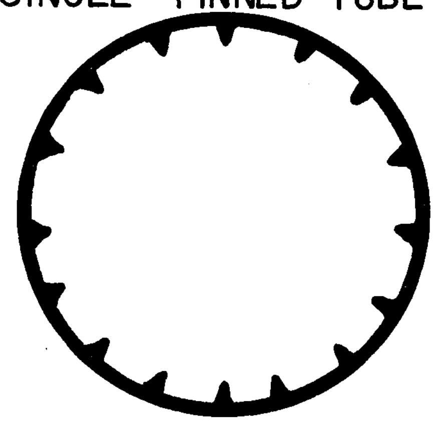


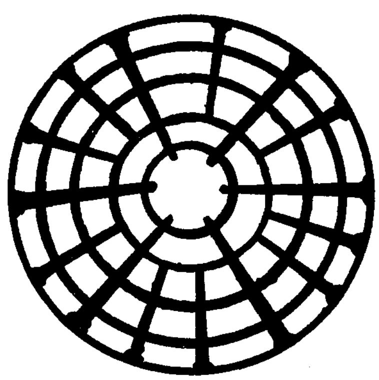
FIG.1 SCHEMATIC DIAGRAM OF THE APPARATUS



1) SINGLE FINNED TUBE



2) SINGLE FINNED TUBE



3) QUINTUPLEX FINNED TUBE

FIG. 2 SCHEMATIC DIAGRAMS OF THE CROSS-SECTIONS OF THE FINNED TUBES

TABLE I. DIMENSIONS OF TEST TUBES

TUBE	SINGLE FINNED 1	SINGLE FINNED 2	QUINTUPLEX FINNED				
			1	2	3	4	5
NUMBER & TYPE OF FINS	16 SPIRAL	16 SPIRAL	16 SPIRAL	16 SPIRAL	16 SPIRAL	6 SPIRAL	6 SPIRAL
OUTSIDE DIMETER, in.	0.781	1.05	0.95	0.80	0.60	0.43	0.28
WALL THICKNESS, in.	0.039	0.03	0.03	0.03	0.025	0.025	0.03
FIN HEIGHT, in.	0.090	0.084	0.045	0.07	0.06	0.05	0.05
FIN PITCH, in.	8.75	6	6	8	8	10	49
OVERALL FLOW AREA, sq in	0.377	0.721	0.359				

TABLE 2. PHYSICAL PROPERTIES OF TEST OIL

TEMPERATURE, F	100	150	200
DENSITY, lb_m/ft^3	53.7	52.7	51.4
VISCOSITY, $lb_m/(ft)(hr)$	118.6	53.5	23.5
HEAT CAPACITY, Btu/(lb)(F)	0.465	0.493	0.528
THERMAL CONDUCTIVITY, Btu/(hr)(ft)(F)	0.0807	0.0819	0.0833
PRANDTL NUMBER	683	322	149

The oil is stored in an oil tank having dimensions of 20" X 30" X 48", with a capacity of 100 U.S. gallons. The oil is heated up to the desired temperature by three heaters which are installed internally on the tank end wall. High oil temperatures of up to 212 F are maintained in the system using three 3.75 KW each, 115V, single phase chromalox electric immersion heaters. The temperature of the oil in the tank is maintained at the desired operating temperature by a Leeds and Northup Electromax temperature controller and a set of three contactors. The input power to one heater is controlled by a manual power variac, while the other two heaters are equipped with on/off switches, thus making it possible to obtain variable heating powers anywhere between 0 and 11.25 KW. Such an arrangement is necessary in order to maintain a fairly constant oil temperature at the inlet of the test section for any oil or water flow rates. As a precautionary measure, an on/off temperature control is installed in the electric circuit to cut off the power when the oil temperature exceeds the desired setting value. The power switches back on automatically when the oil cools down to the desired temperature. The diagrams of the heaters and Electromax Temperature controller are shown in Figs. 46 and 47. From Fig. 47 it is seen that the temperature required is adjusted by a 4 digit digital setter, the last digit being the decimal scale.

The oil from the storage tank is pumped by means of a rotary gear pump through the filter and flowmeters to a horizontal 5-ft long test section, where it is cooled by chilled water running over the test tube in a single-pass shell-and-tube type heat exchanger. The oil, before entering the pump, is filtered through a sump-type oil filter installed in the storage tank on the suction side, to prevent small hard rust particles from entering the pump and the system. The oil filter can be removed easily for cleaning, and this should be done whenever the need arises. It is always better to drive the oil pump at its maximum capacity. The flow of oil through the test section is controlled with the help of a by-pass valve and two oil flowmeters. Figure 48 shows the two oil flowmeters and the water flowmeter. The floats of oil flowmeters are so designed that the meter reading is independent of viscosity. Either rotameter 1, or rotameter 2 is used in our apparatus separately or together, according to the rate of flow. Computer calibrated charts for the rotameters are normally supplied by the factory with an accuracy of $\pm$ 2%. Figures 3, 4 and 5 are the calibration charts for these flowmeters and are taken from the M. Eng. Project of Y. Burnukoglu (8).

Inlet and outlet temperature of oil and water are the factors to be measured accurately. Oil temperatures are

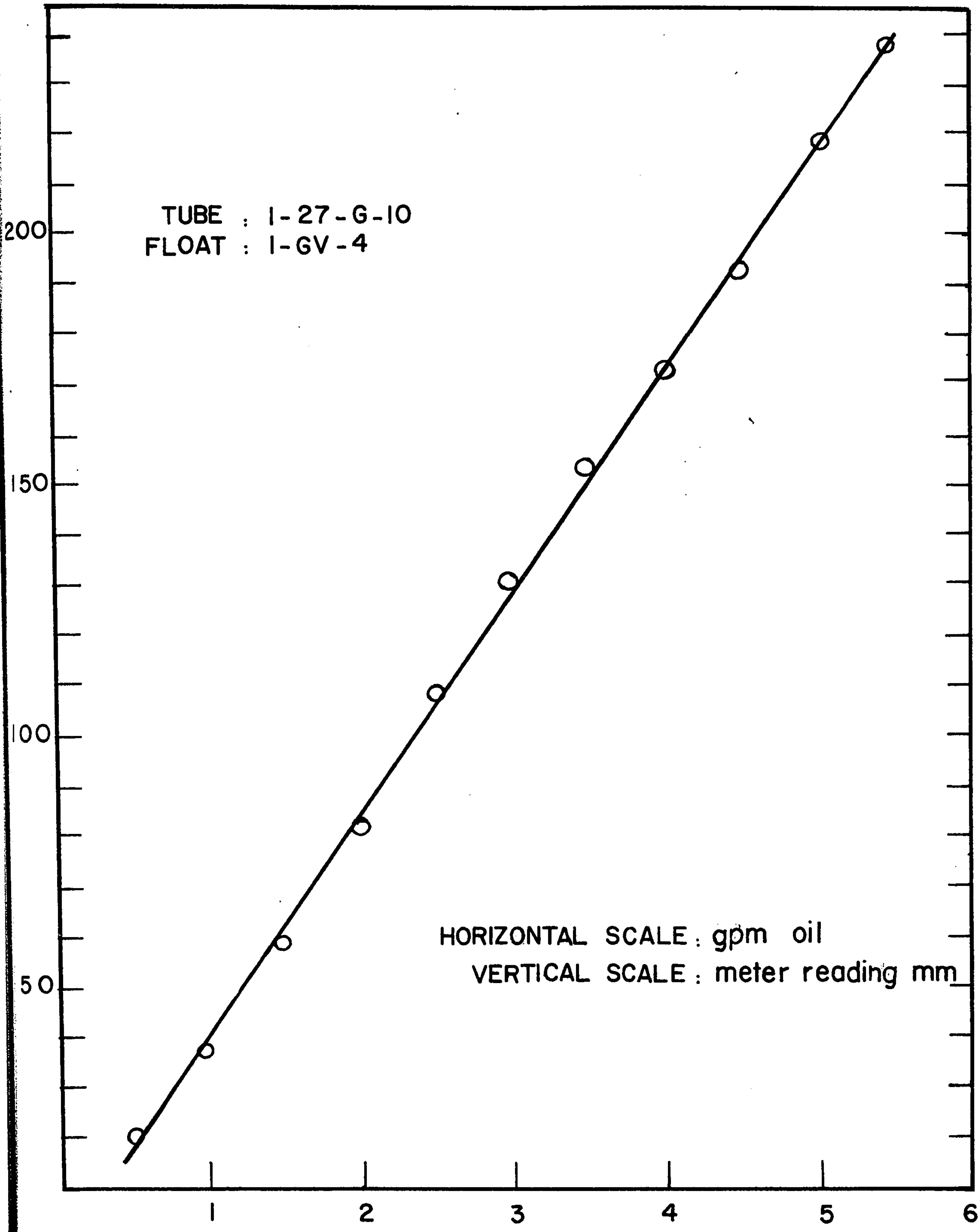


FIG.3 CALIBRATION CHART FOR ROTAMETER I

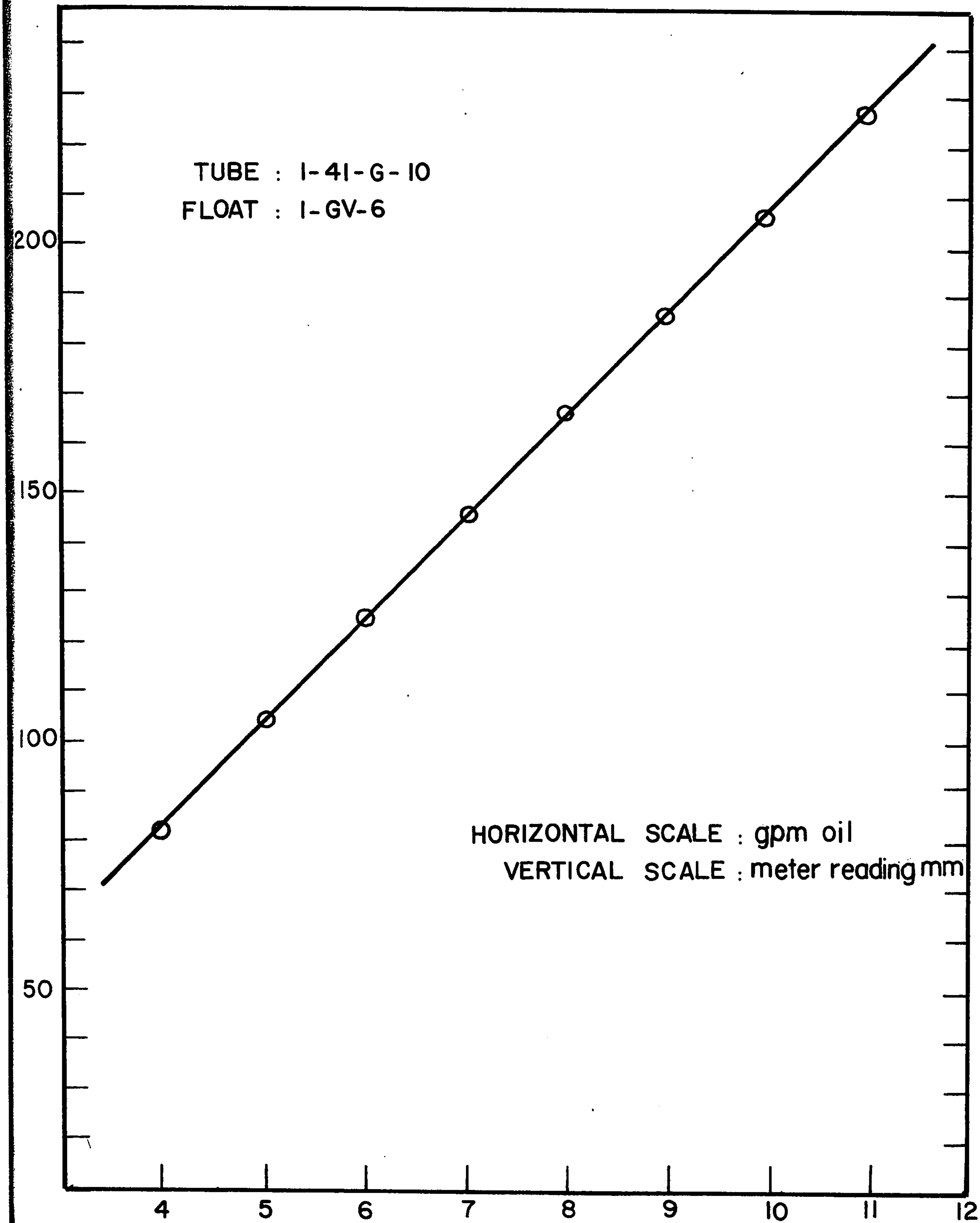


FIG. 4 CALIBRATION CHART FOR ROTAMETER 2

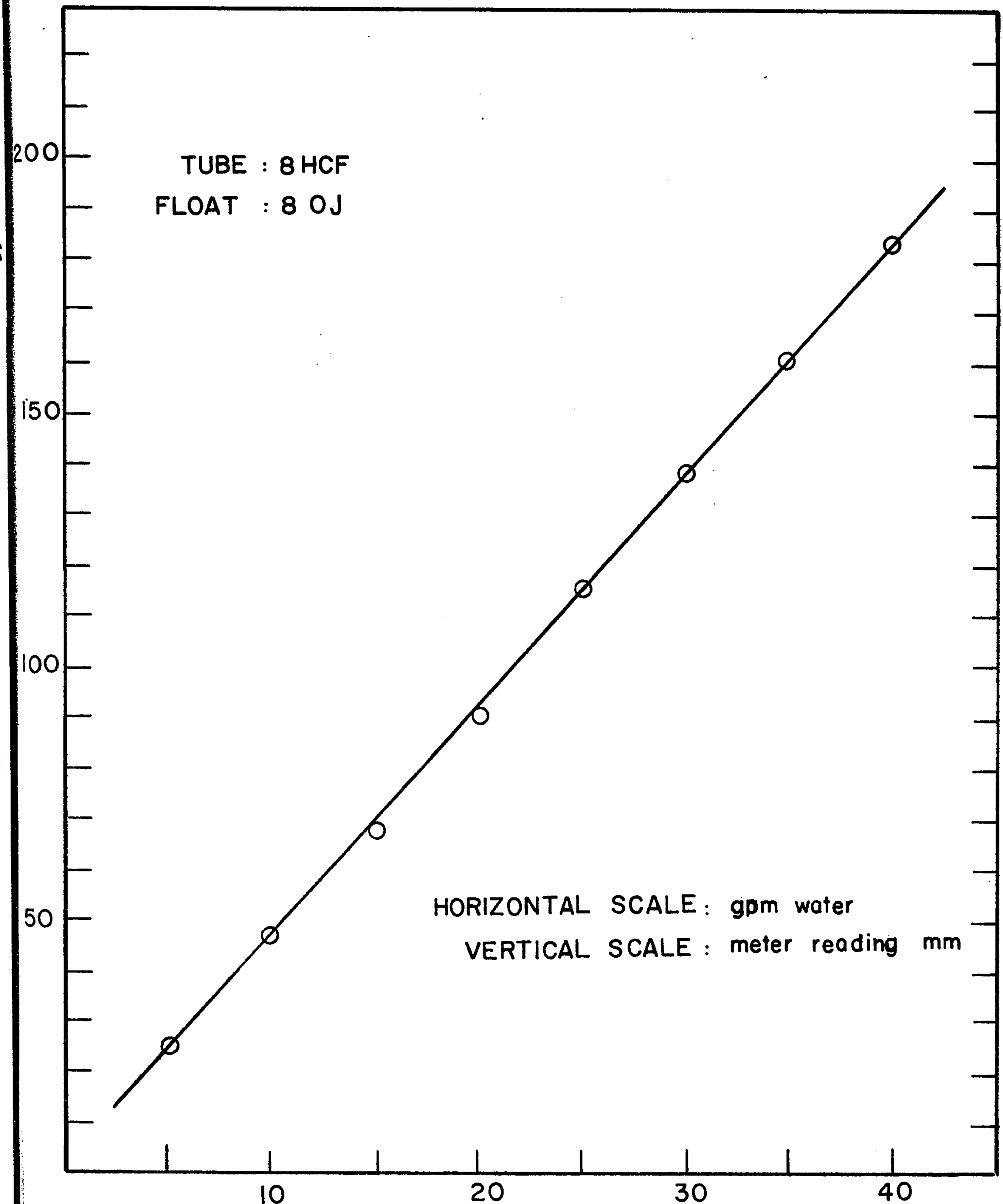


FIG.5 CALIBRATION CHART FOR ROTAMETER 3

measured with both mercury expansion thermometers and precalibrated copper-constantan thermocouples. These oil temperatures are recorded at two mixing sections, where the oil is forced to mix in order to enable us to measure the fluid bulk temperatures near the entrance and the exit of 5 - ft long test section. Three thermocouples are installed in each mixing section, one at the center, other 1/3 r distance from the center, and the third one 2/3 r distance from the center. The thermometers are installed in a thermowell embedded in each mixing section. As a precautionary measure, the thermowells are checked and should contain oil before taking the readings on thermometer. All thermocouples e.m.f. readings are displayed on a Leeds and Northrup Numatron Numeric display directly in degrees Fahrenheit, having an accuracy of ± 0.5F. This unit has originally been calibrated for iron constantan thermocouple wires. When this type of thermocouples is used, the temperature is directly displayed in degrees Fahrenheit, otherwise the unit should be re-calibrated. I have employed copper constantan thermocouples and have performed the requisite calibrations. The resulting linear equation is

$$T = 1.229t - 26.99 \qquad (3)$$

where, t is the temperature displayed on the unit and

T is the true temperature.

The test tube is symmetrically inserted in the 2"O.D. cooling jacket through flanges. In the cooling section of test tube, copper-constantan thermocouple wires are soldered into grooves on the outside wall at seven axial locations 4in. apart. The 2-ft long cooling section is positioned with the thermocouples at the top. The water inlet and oulet temperatures are recorded with the help of two thermocouples fixed radially, one at the center and the other $\frac{1}{2}$r distance from the center.

The schematic diagram of the water loop system is shown in Fig. 6. The system is designed so that water at the inlet to the test section is maintained constant. The water from the tank is pumped through the 2-ft long cooling section and is brought back into the water tank where it is further mixed with the cold water coming from the tap. The returning hot water from the test section and the cold tap water are properly mixed, so that by further manipulation of the valves, it is easier to get the desired temperature at the inlet to the test section. The flow of water is regulated by water flowmeter and a by-pass valve. Figure 49 shows the cooling water system discussed above. Drainage valve for cleaning is provided at the base of the water tank. Also, a pipe leeds the overflow of the tank to the drain. Fibre glass insulation of 1.5in. thickness is provided as required in order to minimize the heat loss.

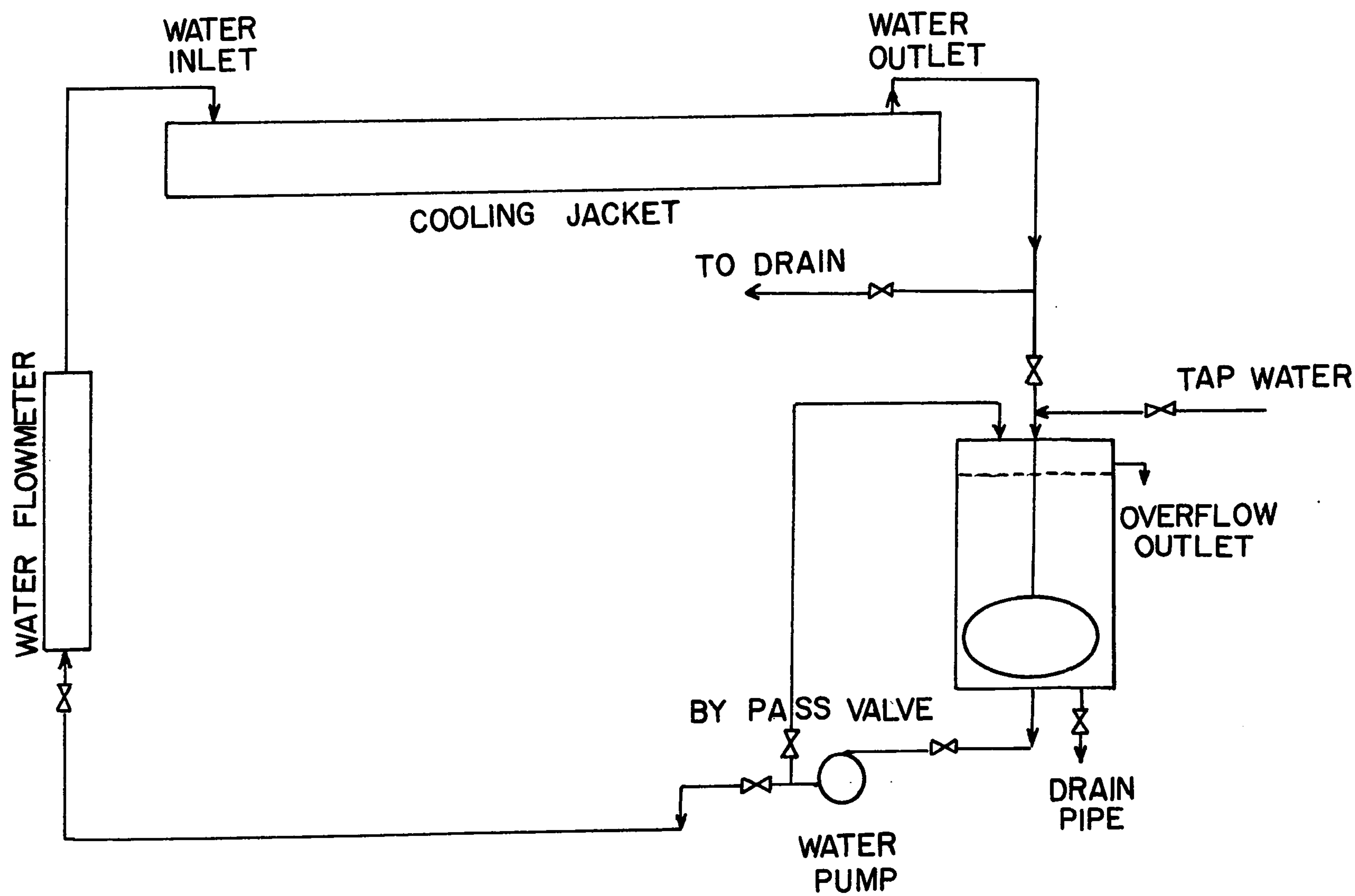


FIG.6 SCHEMATIC DIAGRAM OF COOLING WATER SYSTEM

The isothermal pressure drop data are collected and two pressure points are located 5 feet apart. The pressure drop across the test section is measured by three transducers with ranges 0-0.4; 0-3; and 0-25 psi. A mercury manometer is also installed which guides us in the choice of appropriate transducer. The transducers are shown in Fig. 50, and the pressure drop data is recorded from a milli-voltmeter. Before taking pressure drop data, certain precautions are taken in order to avoid the damage to the milli-voltmeter. Power connections between milli-voltmeter and the transducers to the D.C. power supply shown in Figs. 7, 8 and 9 are checked. The neutral valve between high and low pressure sides of manometer and transducers is then completely opened in order to equalize the initial pressure between the manometer and the transducers.

2.2 Water Tank System

The water tank is installed in order to investigate the behaviour of the tubes almost under the same operating conditions. Previously, Soliman and Feingold could not evaluate the performance of these tubes at constant water inlet temperature because they were using tap water. It was therefore difficult for them to repeat the same run on another day when water supply came at a different temperature.

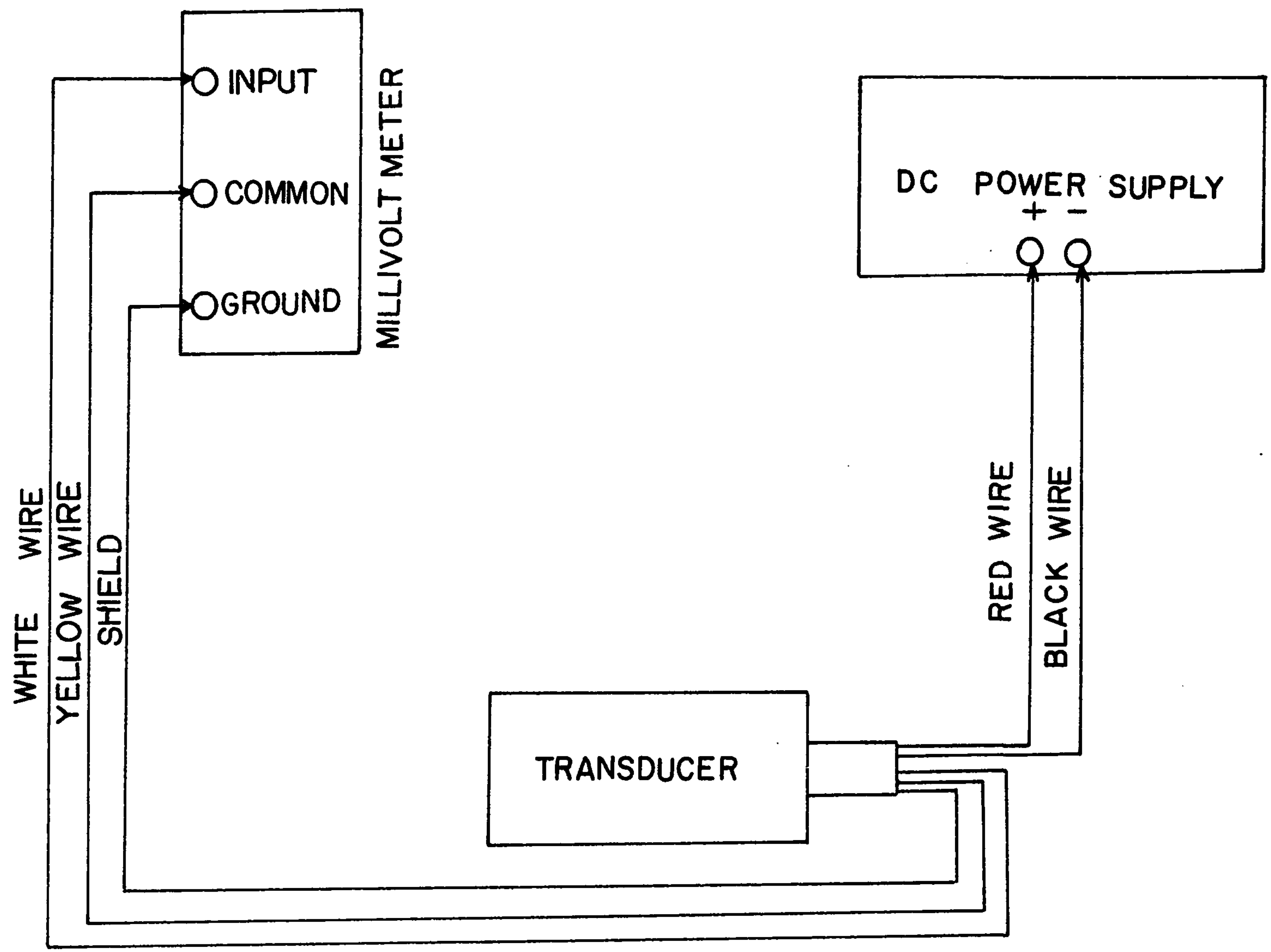


FIG. 7 POWER CONNECTION FOR VALIDYNE TRANSDUCER
RANGE 0.4 psi

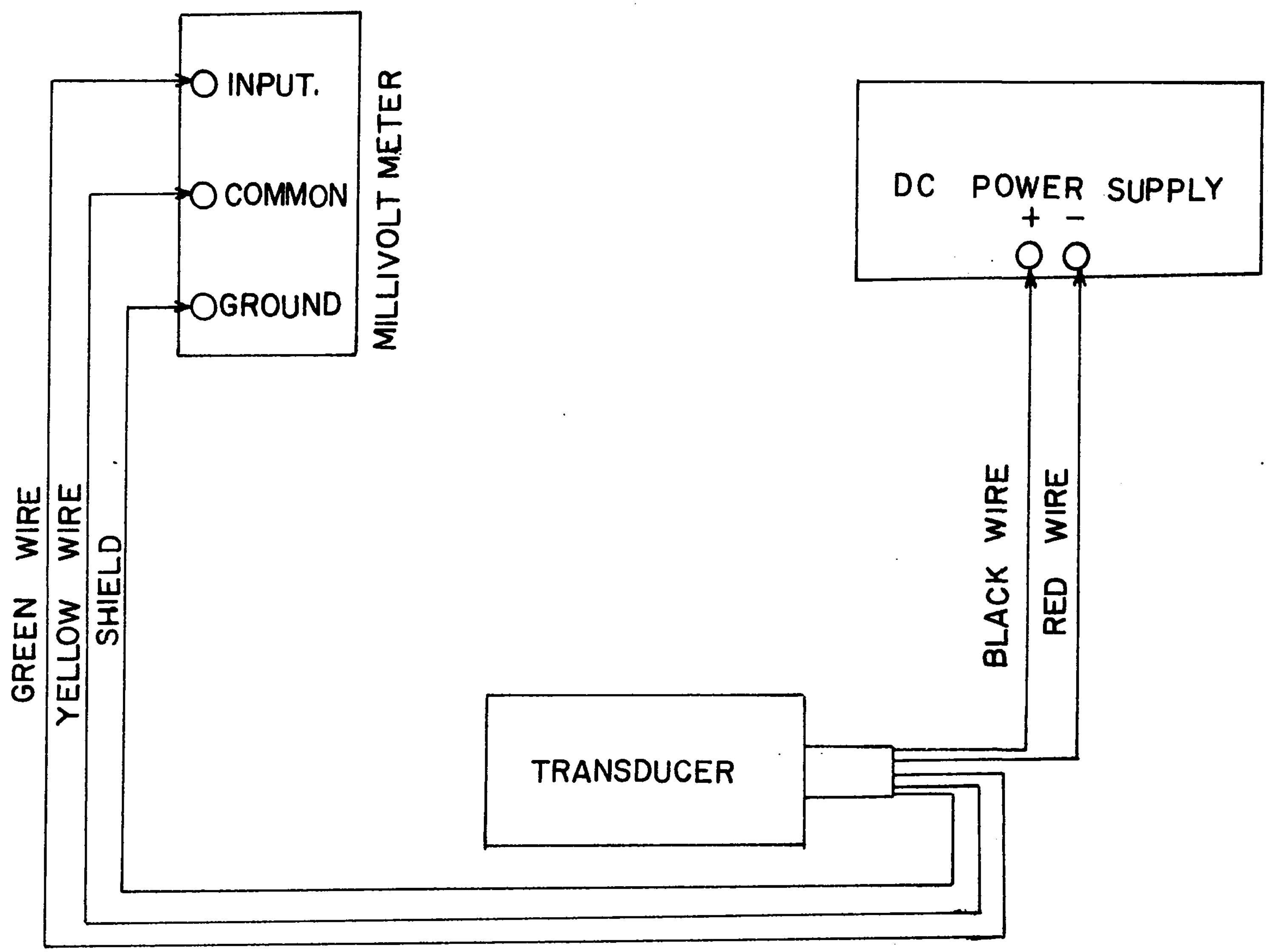


FIG. 8 POWER CONNECTION FOR THE PD-200 GA TRANSDUCER RANGE 0.2 kg/cm^2

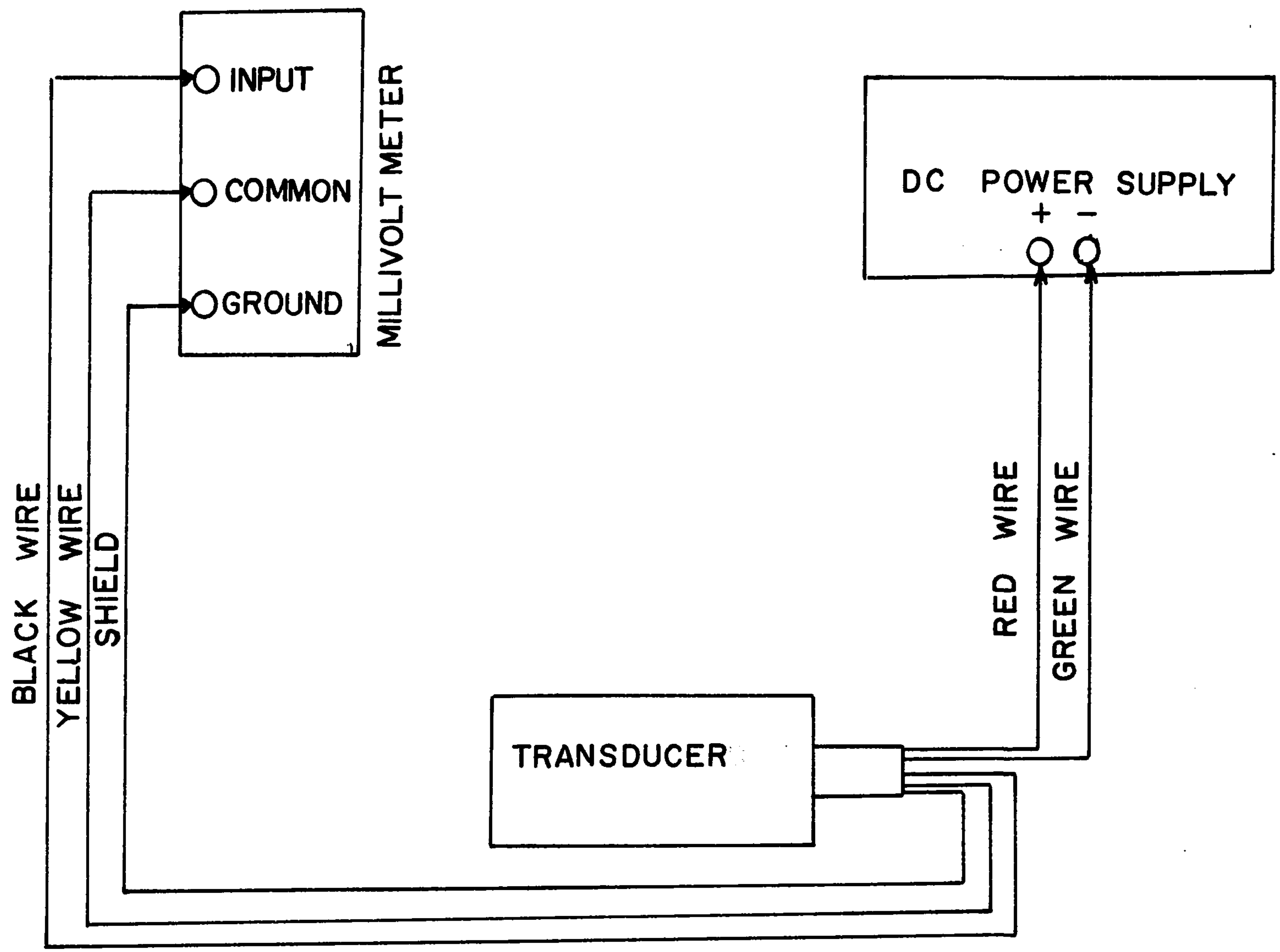


FIG. 9 POWER CONNECTION FOR THE PM 60TC ±25-350 TRANSDUCER RANGE ±25 psi.

Therefore, it was suggested that I design a system which could be able to control the constant water inlet temperature to the test section. The water tank system employed here is fairly simple. The hot water coming out of the test section is brought back into the water tank where it is mixed with the cold tap water. So by proper manipulation of valves and mixing of tap water with returning hot water, a constant water inlet temperature to the test section can be obtained. The valves used for controlling tap water and returning hot water to the tank must have the ability to exercise their function very accurately. Needle type valves were thought to be most appropriate and were choosen accordingly. The subsequent operating experience has shown this choice to have been a good one. Previously, the water inlet temperature to the test section was not subject to regulation, but after the installation of this new system the inlet water temperature was regulated within ± 1 degree Fahrenheit, always provided we had tap water of sufficiently low temperature.

Figure 49 shows the photographic view of the apparatus discussed above.

2.3 Test Section Design and Entrance Length

A shorter test section is designed in order to investigate the nature of the tubes at higher mass flow rate of oil, particularly the quintuplex finned tube, which Soliman and Feingold (4) could not evaluate due to high pumping power requirement. Moreover, Soliman and Feingold developed the previously discussed concept of "equivalent length". Therefore, I had to design a shorter test section in order to prove the validity of that concept by actual experiments.

Each test tube was tested under three different operating conditions, namely:

(1) High inlet temperature of oil with low mass flow rate of water.

(2) High inlet temperature of oil with high mass flow rate of water.

(3) Low inlet temperature of oil with low mass flow rate of water.

In order to obtain the same heat transfer rate at the same oil mass flow rate of oil, three different quintuplex lengths would result while tested under three above-mentioned operating conditions. The ideal would be to manufacture three separate sections, but limitations of cost, time and materials forced me to choose a length which would provide a reasonable compromise. The tube would be sufficiently long to measure the heat transfer with all but the very large mass rates of oil flow, while limiting the pumping power requirements to the capacity of existing pumping equipment.

Corresponding to Figs. 42, 43 and 44 in Ref.4 a value of 40,000 Btu/hr or ($11.7X10^3$W) is approximately obtained at an oil mass flow rate of 10,000 lb_m/hr or (1.262 kg/sec) if the single-finned tube curves plotted by Soliman and Feingold are extrapolated up to that value of $\dot{m}_o$. Such an extrapolation is, of course, somewhat arbitrary, but we must make an educated guess, and the exact value of predicted heat transfer is not all that important in view of the compromise that will have to be made eventually. The concept of "equivalent length" is now used to calculate the required length of the quintuplex tube in order to achieve the heat transfer rate of 40,000Btu/hr at the oil mass flow rate of 10,000 lb_m/hr.

The relevant calculations under different operating conditions are shown below:

CASE I

High inlet temperature of oil with low mass flow rate of water.

If the curve for the quintuplex tube plotted by Soliman and Feingold in Fig. 43 is extended further, a value of 200,000 Btu/hr or ($58.6X10^3$W) is obtained at an oil mass flow rate of 10,000 lb_m/hr or (1.262 kg/sec), under the following operating conditions:

oil inlet temperature,	$T_{o,i}$ = 218F or (103.3C) (High)
water inlet temperature,	$T_{w,i}$ = 40F or (4.4C)
water mass flow rate,	$\dot{m}_w$ = 1800lb_m/hr or (0.227kg/sec) (Low)

C_{p_o} and C_{p_w} was taken as 0.5 and 1 Btu/lbF

Heat rate lost by oil, $Q_o = \dot{m}_o\, C_{p_o}\, (T_{o,i} - T_{o,o})$ (4)

$$200{,}000 = 10{,}000 \times 0.5\ (218 - T_{o,o})$$

$$T_{o,o} = 178\text{F}$$

Heat rate lost by oil = Heat rate gained by water

$$200{,}000 = \dot{m}_w\, C_{p_w}\, (T_{w,o} - T_{w,i}) \tag{5}$$

$$= 1800 \times 1\ (T_{w,o} - 40)$$

$$T_{w,o} = 151.11\text{F}$$

Log-mean temperature difference would be

$$\Delta T_m = \frac{(T_{o,i} - T_{w,i}) - (T_{o,o} - T_{w,o})}{\ln\left(\dfrac{T_{o,i} - T_{wi}}{T_{o,o} - T_{w,o}}\right)} \tag{6}$$

$$\Delta T_m = \frac{(218-40) - (178-151.11)}{\ln\left(\dfrac{218-40}{178-151.11}\right)} = \frac{178-26.89}{\ln\left(\dfrac{178}{26.89}\right)}$$

$$= 80\text{F}$$

On the other hand,

$$Q_o = UA\,\Delta T_m$$

where U is the overall heat transfer coefficient

$$200{,}000 = U \pi D \times 8 \times 80 \tag{a}$$

The shorter test section is to be tested under the following operating conditions:

oil inlet temperature , $T_{o,i} = 212F$ (High)

water inlet temperature , $T_{w,i} = 40F$

water mass flow rate , $\dot{m}_w = 5000\ lb_m/hr$ (Low)

Heat rate lost by oil, $$Q_o = \dot{m}_o\ C_{p_o}\ (T_{o,i} - T_{o,o})$$

$$40{,}000 = 10{,}000 \times 0.5\ (212 - T_{o,o})$$

$$T_{o,o} = 204F$$

Heat rate lost by oil

$$= \text{Heat rate gained by water}$$

$$40{,}000 = \dot{m}_w\ C_{p_w}\ (T_{w,o} - T_{w,i})$$

$$= 5000 \times 1\ (T_{w,o} - 40)$$

$$T_{w,o} = 48F$$

Log-mean temperature difference would be given by

$$\Delta T_m = \frac{(T_{o,i} - T_{w,i}) - (T_{o,o} - T_{w,o})}{\ln\left(\dfrac{T_{o,i} - T_{w,i}}{T_{o,o} - T_{w,o}}\right)}$$

$$= \frac{(212 - 40) - (204 - 48)}{\ln\left(\dfrac{212 - 40}{204 - 48}\right)}$$

$$= \frac{172 - 156}{\ln\left(\dfrac{172}{156}\right)} = 164.07F$$

$$Q_o = U\ A\ \Delta T_m$$

$$40{,}000 = U\ \pi\ D\ L \times 164.07 \qquad \text{(b)}$$

From equations (a) and (b) we get

$$\frac{40{,}000}{200{,}000} = \frac{U \pi D \times L \times 164.07}{U \pi D \times 8 \times 80}$$

and, assuming that U is the same in both cases, we have

$$L = \frac{40{,}000 \times 8 \times 80}{200{,}000 \times 164.07} = 0.78 \text{ ft}$$

CASE II

High inlet temperature of oil with high mass flow rate of water.

If the quintuplex curve plotted by Soliman and Feingold in Fig. 44 is extended, a value of 200,000 Btu/hr or (58.6×10^3W) is obtained at an oil mass flow rate of 10,000 lb_m/hr or (1.262kg/sec), under the following operating conditions:

oil inlet temperature , $T_{o,i}$ = 218F or (103.3C) (High)

water inlet temperature , $T_{w,i}$ = 40F or (4.4C)

water mass flow rate , $\dot{m}_w$ = 3600 lb_m/hr or (.454kg/sec) (High)

Heat rate lost by oil , $Q_o = \dot{m}_o C_{p_o} (T_{o,i} - T_{o,o})$

$$200{,}000 = 10{,}000 \times 0.5\ (218 - T_{o,o})$$

$$T_{o,o} = 178F$$

Heat rate lost by oil = Heat rate gained by water

$$200{,}000 = \dot{m}_w\, C_{p_w} (T_{w,o} - T_{w,i})$$

$$= 3{,}600 \times 1\,(T_{w,o} - 40)$$

$$T_{w,o} = 95.55\text{F}$$

Log-mean temperature difference is given by

$$\Delta T_m = \frac{(T_{o,i} - T_{w,i}) - (T_{o,o} - T_{w,o})}{\ln\left(\dfrac{T_{o,i} - T_{w,i}}{T_{o,o} - T_{w,o}}\right)}$$

$$\Delta T_m = \frac{(218 - 40) - (178 - 95.55)}{\ln\left(\dfrac{218 - 40}{178 - 95.55}\right)}$$

$$= \frac{178 - 82.45}{\ln\left(\dfrac{178}{82.45}\right)} = \frac{95.55}{0.7682} = 124.38\text{F}$$

On the other hand,

$$Q_o = U A \Delta T_m$$

$$200{,}000 = U \pi D \times 8 \times 124.38 \qquad \text{(a)}$$

The shorter test section is to be tested under the following operating conditions:

oil inlet temperature , $T_{o,i} = 212\text{F}$ (High)

water inlet temperature , $T_{w,i} = 40\text{F}$

water mass flow rate , $\dot{m}_w = 10{,}000\ \text{lb}_m/\text{hr}$ (High)

Heat rate lost by oil, $Q_o = \dot{m}_o \, C_{p_o} \, (T_{o,i} - T_{o,o})$

$$40{,}000 = 10{,}000 \times 0.5 \, (212 - T_{o,o})$$

$$T_{o,o} = 204F$$

Heat rate lost by oil = Heat rate gained by water

$$40{,}000 = \dot{m}_w \, C_{p_w} \, (T_{w,o} - T_{w,i})$$

$$40{,}000 = 10{,}000 \times 1 \, (T_{w,o} - 40)$$

$$T_{w,o} = 44F$$

Log-mean temperature difference is given by

$$\Delta T_m = \frac{(T_{o,i} - T_{w,i}) - (T_{o,o} - T_{w,o})}{\ln\left(\dfrac{T_{o,i} - T_{w,o}}{T_{o,o} - T_{w,o}}\right)}$$

$$= \frac{(212 - 40) - (204 - 44)}{\ln\left(\dfrac{212 - 40}{204 - 44}\right)}$$

$$= \frac{172 - 160}{\ln\left(\dfrac{172}{160}\right)} = \frac{12}{.06647} = 180.53F$$

$$Q_o = U \, A \, \Delta T_m$$

$$40{,}000 = U \, \pi \, D \, L \times 180.53 \qquad \text{(b)}$$

From equations (a) and (b), we get

$$\frac{40{,}000}{200{,}000} = \frac{U \pi D \, L \times 180.53}{U \pi D \times 8 \times 124.38}$$

$$L = \frac{40{,}000 \times 8 \times 124.38}{200{,}000 \times 180.53} = 1.10\text{ft.}$$

CASE III

Low inlet temperature of oil with low mass flow rate of water.

If the quintuplex curve plotted by Soliman and Feingold in Fig. 42 is extended, a value of 100,000Btu/hr or (29.31×10^3W) is obtained at an oil mass flow rate of 10,000 lb_m/hr or (1.262 kg/sec), under the following operating conditions:

oil inlet temperature , $T_{o,i}$ = 176F or (80C) (Low)

water inlet temperature , $T_{w,i}$ = 40F or (4.4C)

water mass flow rate , $\dot{m}_w$ = 1800lb_m/hr or (.227 kg/sec) (Low)

$$\text{Heat rate lost by oil, } Q_o = \dot{m}_o C_{p_o} (T_{o,i} - T_{o,o})$$

$$100{,}000 = 10{,}000 \times 0.5\,(176 - T_{o,o})$$

$$T_{o,o} = 156\text{F}$$

$$\text{Heat rate lost by oil} = \text{Heat rate gained by water}$$

$$100{,}000 = \dot{m}_w C_{p_w} (T_{w,o} - T_{w,i})$$

$$= 1800 \times 1\,(T_{w,o} - 40)$$

$$T_{w,o} = 95.55\text{F}$$

Log-mean temperature difference is given by

$$\Delta T_m = \frac{(T_{o,i} - T_{w,i}) - (T_{o,o} - T_{w,o})}{\ln\left(\dfrac{T_{o,i} - T_{w,i}}{T_{o,o} - T_{w,o}}\right)}$$

$$= \frac{(176-40) - (156-95.55)}{\ln\left(\dfrac{176 - 40}{156 - 95.55}\right)} = 93.34\text{F}$$

On the other hand,

$$Q_o = UA\ \Delta T_m$$

$$100{,}000 = U\ \pi D \times 8 \times 93.34 \qquad \text{(a)}$$

The shorter tube is to be tested under the following operating conditions:

oil inlet temperature, $T_{o,i} = 176F$ (Low)

water inlet temperature, $T_{w,i} = 40F$

water mass flow rate, $\dot{m}_w = 5000\ lb_m/hr$ (Low)

Heat rate lost by oil, $Q_o = \dot{m}_o\ C_{p_o}\ (T_{o,i} - T_{o,o})$

$$40{,}000 = 10{,}000 \times 0.5\ (176 - T_{o,o})$$

$$T_{o,o} = 168F$$

Heat rate lost by oil = Heat rate gained by water

$$40{,}000 = \dot{m}_w C_{p_w}\ (T_{w,o} - T_{w,i})$$

$$= 5000 \times 1\ (T_{w,o} - 40)$$

$$T_{w,o} = 48F$$

Log-mean temperature difference would be

$$\Delta T_m = \frac{(T_{o,i} - T_{w,i}) - (T_{o,o} - T_{w,o})}{\ln\left(\dfrac{T_{o,i} - T_{w,i}}{T_{o,o} - T_{w,o}}\right)}$$

$$= \frac{(176 - 40) - (168 - 48)}{\ln\left(\dfrac{176 - 40}{168 - 48}\right)}$$

$$= \frac{136 - 120}{\ln\left(\dfrac{136}{120}\right)} = 128.11F$$

On the other hand,

$$Q_o = UA\Delta T_m$$

$$40{,}000 = U\pi DL \times 128.11 \qquad \text{(b)}$$

Therefore, from equations (a) and (b) we have

$$\frac{40{,}000}{100{,}000} = \frac{U\pi D\ L \times 128.11}{U\pi D \times 8 \times 93.34}$$

$$L = 2.33 \text{ ft}$$

The results of these three calculations are summarized in table 3.

I selected the cooling length of 2ft, hoping that this compromise will enable me to perform a sufficiently broad spectrum of experiments to prove or disprove the heat transfer behaviour of quintuplex tubes predicted by Soliman and Feingold for conditions lying behond the range of their own tests.

In fact, the test section ought to consist of the actual 2-ft cooling section between two measuring stations as well as appropriate additional lengths at the entry and at the exit in order to prevent the distortions of flow pattern.

While it is obvious that a longer additional length is required at the entry as compared to the exit, there was no particular reason to dwell on that subject too long and it was simply decided that 1.5ft on each side of the cooling section will be ample to provide the required protection against any local flow pattern distortion due to causes extraneous to our tests.

TABLE 3. SUMMARY OF THE CALCULATED VALUES OF EQUIVALENT LENGTHS, L (ft), FOR $Q = 40{,}000$ Btu/hr AND $\dot{m}_o = 10{,}000\ lb_m/hr$

CONDITIONS / TUBE	$T_{o,i} = 212F$ LOW $\dot{m}_w$	$T_{o,i} = 212F$ HIGH $\dot{m}_w$	$T_{o,i} = 176F$ LOW $\dot{m}_w$
QUINTUPLEX	0.78	1.10	2.33

2.4 Test Procedure

The electric power is switched on and the temperature controller is adjusted to the required value. The oil in the tank is heated by operating all the heaters in order to achieve the desired temperature at the inlet to the test section. The temperature recorder should be plugged in at least half an hour before taking readings on it. The oil and water pumps are activated and hot oil is circulated through 5 ft long test section, where it is cooled by water running over the test tube in a single pass shell-and-tube-type heat exchanger. After passing through the test section the cooled oil is returned to the storage tank and hot water is brought back to water tank, where it is further mixed with the tap water. The flow of oil is regulated by two flowmeters and with the help of a by-pass pump valve. The constant temperature of the oil at the inlet to the test section is controlled by controlling the amount of input power, and this is done by powerstat which regulates the power to the heaters. Continious adjustement of by-pass valve across the pump, the valves of oil and water flowmeters and the amount of input power is done so as to achieve steady state conditions.

This state is deemed to have been reached when all the readings to be recorded, such as oil and water inlet and outlet temperatures, oil and water mass flow rates and the thermocouple readings, do not change for at least half an hour. When the collection of heat transfer data for a particular tube is completed, the waterflow to the heat exchanger is stopped and isothermal pressure drop data are recorded. The isothermal pressure drop is first recorded on mercury manometer which helps in selecting a transducer of a suitable range. The final reading is taken from this transducer.

As was to be expected, test runs on the quintuplex finned tube have shown that the hydrodynamic resistance offered by this tube is much higher than those of single finned tubes. At the start of any test run, the quintuplex tube is full of oil, which being at room temperature has a relatively high viscosity. That cold oil, already filling the tiny ports of quintuplex tube,offers excessively high resistance to the incoming hot oil, and it takes at least 15 minutes to establish a constant rate of oil flow, after which cooling water is circulated. This difficulty is not experienced with the other test tubes.

3. RESULTS AND DISCUSSIONS

The heat transfer and isothermal pressure drop data were collected for three tubes, namely, quintuplex and two single-finned tubes, tested almost under the same operating conditions. No difficulty has been experienced in maintaining the water temperature constant at the inlet to the test section. The quintuplex finned tube was tested first, using water inlet temperature of 40F. That was done during the month of February with very cold tap water available. It was thought that other experiments will follow soon after. For various reasons, these experiments were delayed and by that time the tap water temperature rose to above 40F. In these circumstances it was decided to perform the tests on the two single-finned tubes at 50F.

The effect of oil inlet temperature and water mass flow rate have been investigated and the results are compared with the work of Soliman and Feingold. Furthermore, effect of cooling length and cooling water flow rate on heat transfer is also studied. Each test tube is tested for two different oil temperatures at the inlet and at two different water mass flow rates. A total of nine sets of experiments were conducted for the three test sections, three sets for each section.

Ranges of operating conditions are listed in Table 4. Each test tube was tested at two different oil inlet temperatures with the same water mass flow rate, and at two different water mass flow rates with the same oil inlet temperature. In each case a series of tests was conducted varying the mass flow rate of oil.

3.1 Heat Transfer Results

Care was taken to insulate the test section to minimize the heat transfer between the cooling water and outside air. This notwithstanding, some heat transfer is bound to occur across the insulation unless accidentally the room temperature is equal to that of the water. This fact as well as necessarily limited accuracy of our measurements will cause Q_w, the heat rate gained by water, to differ somewhat from Q_o, the heat lost by oil. In general, air temperature was higher than that of the water, thus Q_w should, in principle, come out larger than Q_o. A combination of measurement errors, however, did occasionally produce $Q_w < Q_o$.

A quantity which we shall call the "heat balance error" was defined as follows:

$$e = \frac{Q_w - Q_o}{Q_o}$$

TABLE 4 RANGES OF OPERATING CONDITIONS

TEST TUBES	SET NO.	WATER FLOW RATE, lb_m/hr	OIL INLET TEMPERATURE, F	OIL OUTLET TEMPERATURE, F	OIL FLOW RATE, lb_m/hr	RATE OF HEAT TRANSFER, $Btu/hr \times 10^{-4}$
SINGLE FINNED 1	1	5000	212 ± 1	205.9–207.9	1114–7529	0.23–1.93
	2	10,000	212 ± 1	206.1–210.1	1406–7529	0.19–1.83
	3	5000	176 ± 1	170.5–173.6	1924–7641	0.15–1.28
SINGLE FINNED 2	4	5000	212 ± 1	205.8–207.5	1016–7529	0.27–1.94
	5	10,000	212 ± 1	206.3–209.5	1406–7527	0.24–1.82
	6	5000	176 ± 1	171.0–173.8	1229–7629	0.18–1.37
QUINTUPLEX FINNED	7	5000	212 ± 1	146.3–191.5	185–2741	0.61–2.93
	8	10,000	212 ± 1	134.2–177.6	185–1407	0.72–2.51
	9	5000	176 ± 1	115.1–166.7	187–5100	0.55–2.71

It was decided to accept as valid only those experimental runs for which the absolute value of e was smaller than 10 percent.

The heat transfer curves, plotted for one quintuplex and two single-finned tubes at different oil inlet temperatures with the same mass flow rates of water and at different mass flow rates of water with the same oil inlet temperature, are shown in Figs. 10 to 18. Further, the comparison between the quintuplex and two single-finned tubes are made under almost the same operating conditions and are shown in Figs. 19, 20 and 21. Data presented in these figures show that quintuplex tube provides a sizable boost in the heat transfer performance over the single-finned tubes at almost the same operating conditions. A sharp increase in the slopes of Q versus $\dot{m}_o$ curves is observed in the data of two single-finned tubes. This marked change in the slopes is attributed to a change of regime associated with transition from laminar to turbulent. Data of quintuplex tube did not exhibit this behaviour and I conclude, therefore, that no change of regime occured here within the range of our experiments.

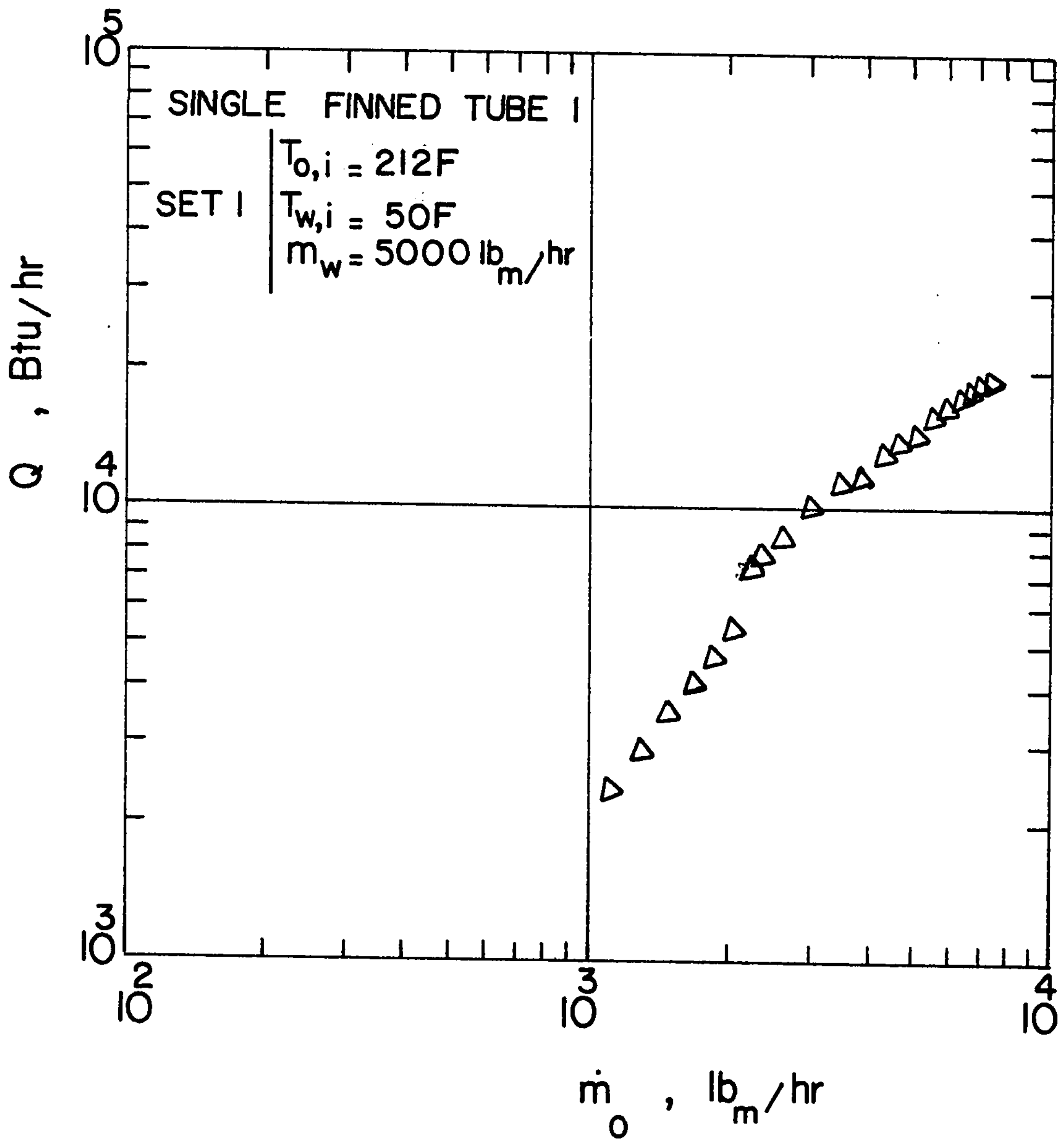


FIG. 10 HEAT TRANSFER DATA AT AN OIL INLET TEMPERATURE OF 212F FOR SINGLE FINNED TUBE I

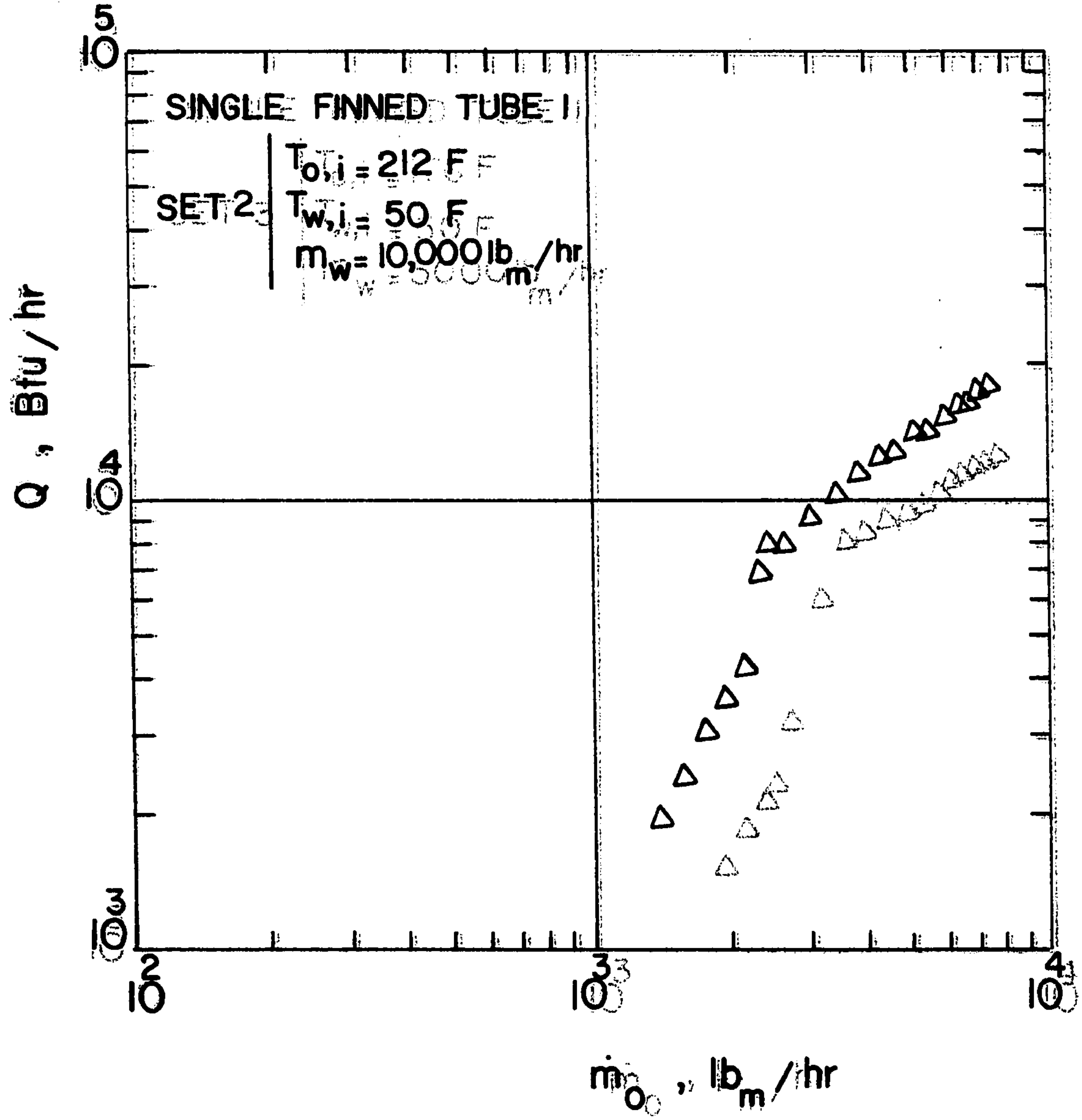


FIG. 12 HEAT TRANSFER DATA AT AN OIL INLET TEMPERATURE OF 212 F FOR SINGLE FINNED TUBE I

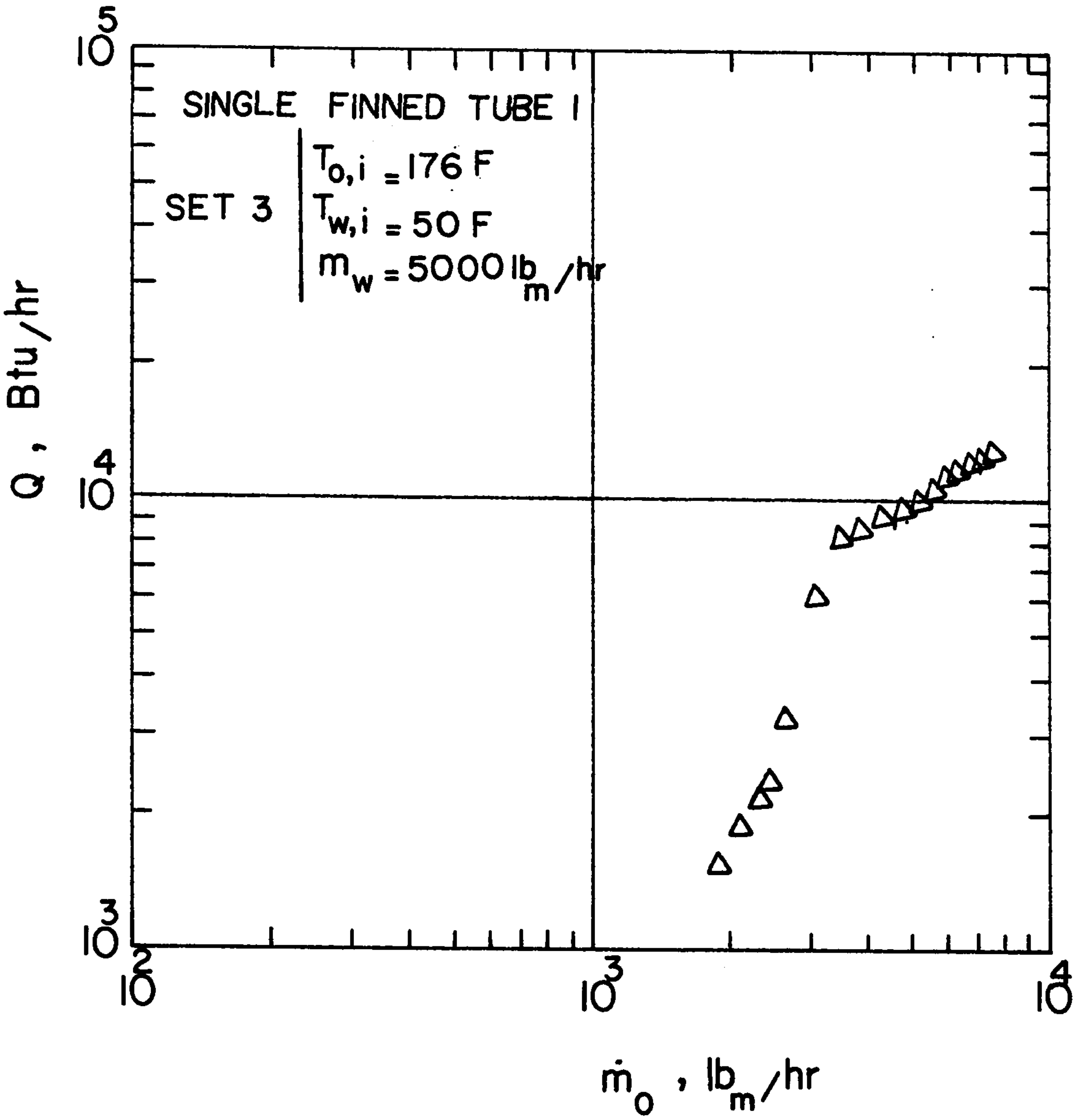


FIG. 12 HEAT TRANSFER DATA AT AN OIL INLET TEMPERATURE OF 176 F FOR SINGLE FINNED TUBE I

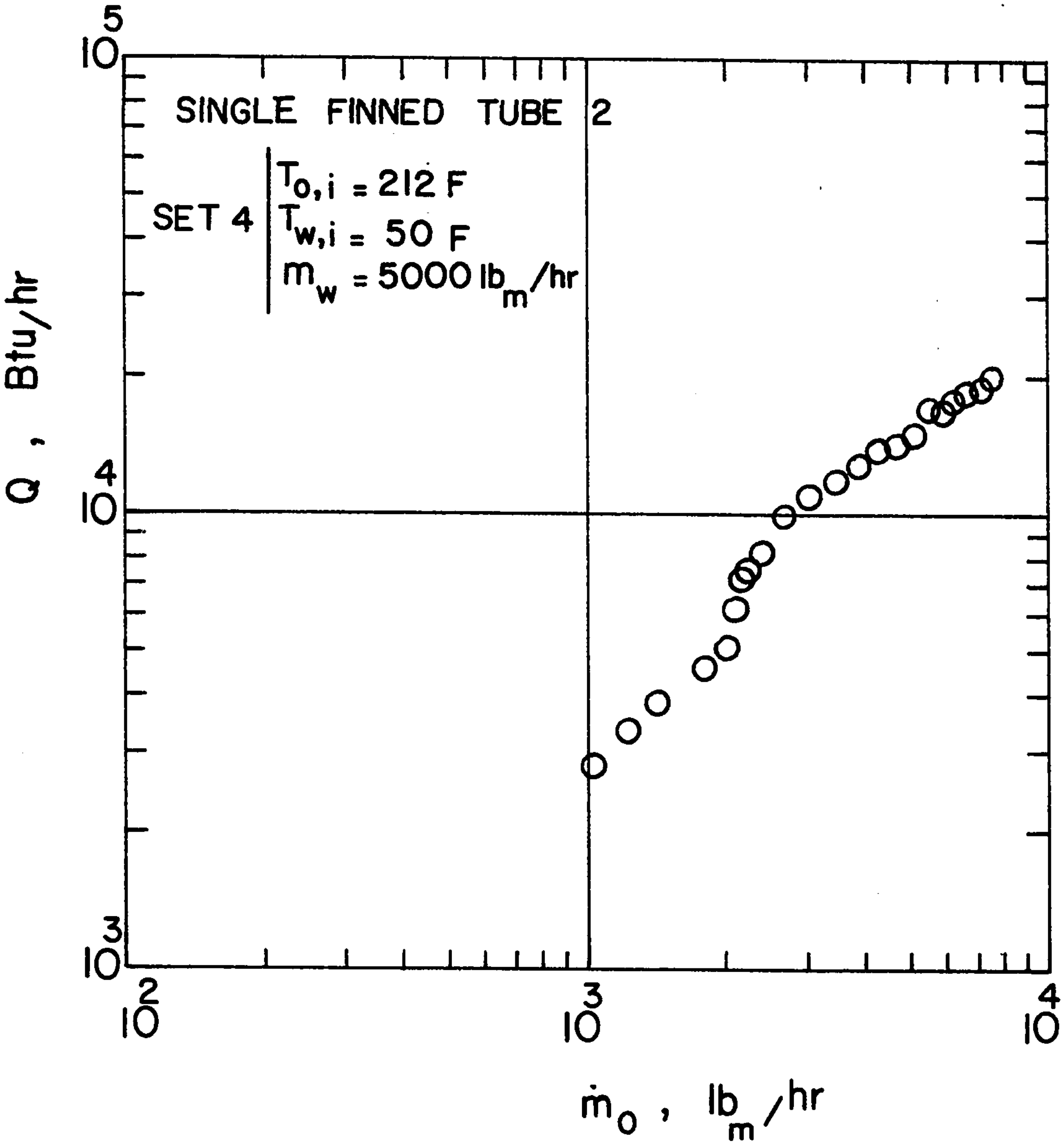


FIG. 13 HEAT TRANSFER DATA AT AN OIL INLET TEMPERATURE OF 212 F FOR SINGLE FINNED TUBE 2

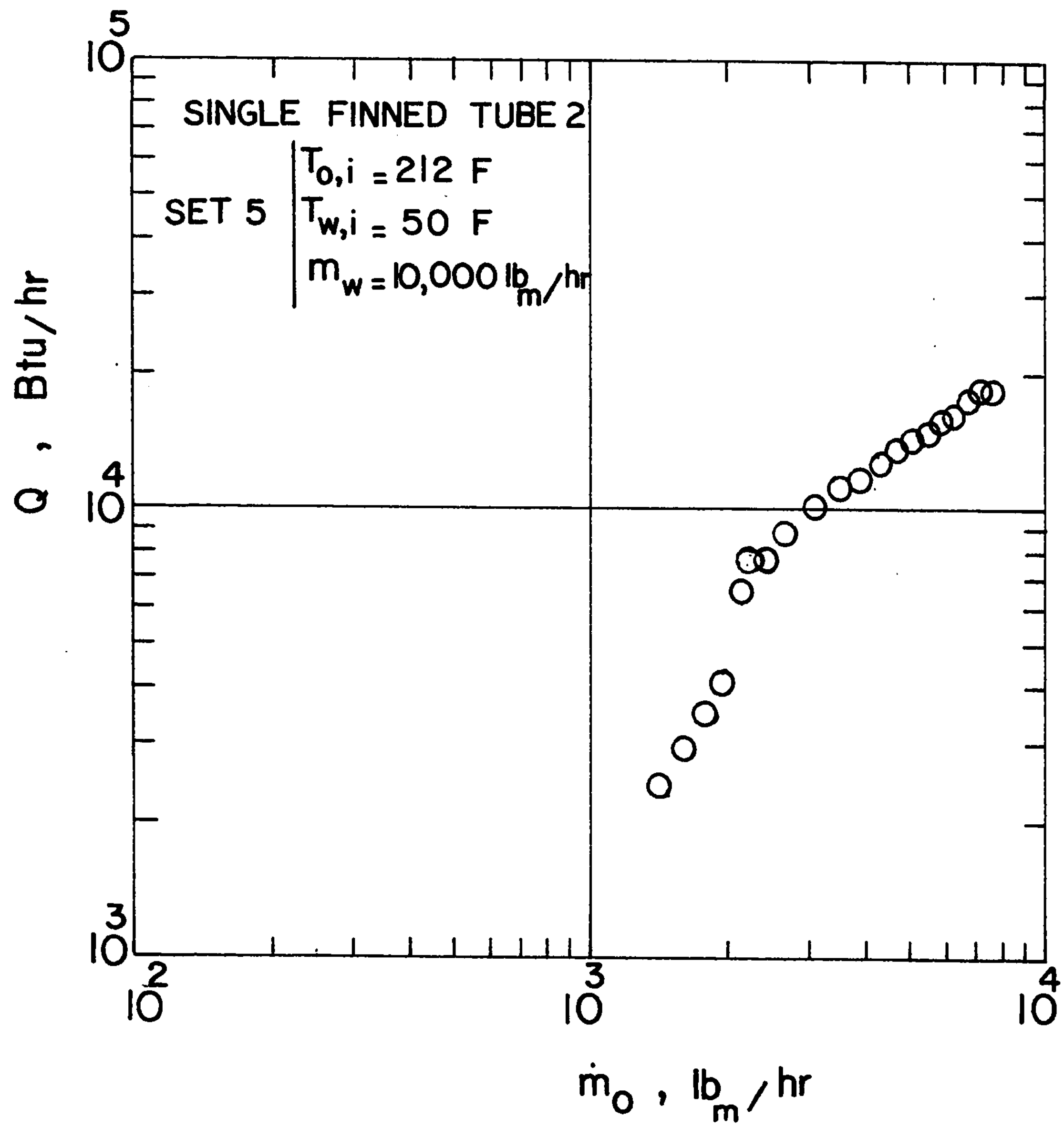


FIG. 14 HEAT TRANSFER DATA AT AN OIL INLET TEMPERATURE OF 212F FOR SINGLE FINNED TUBE 2

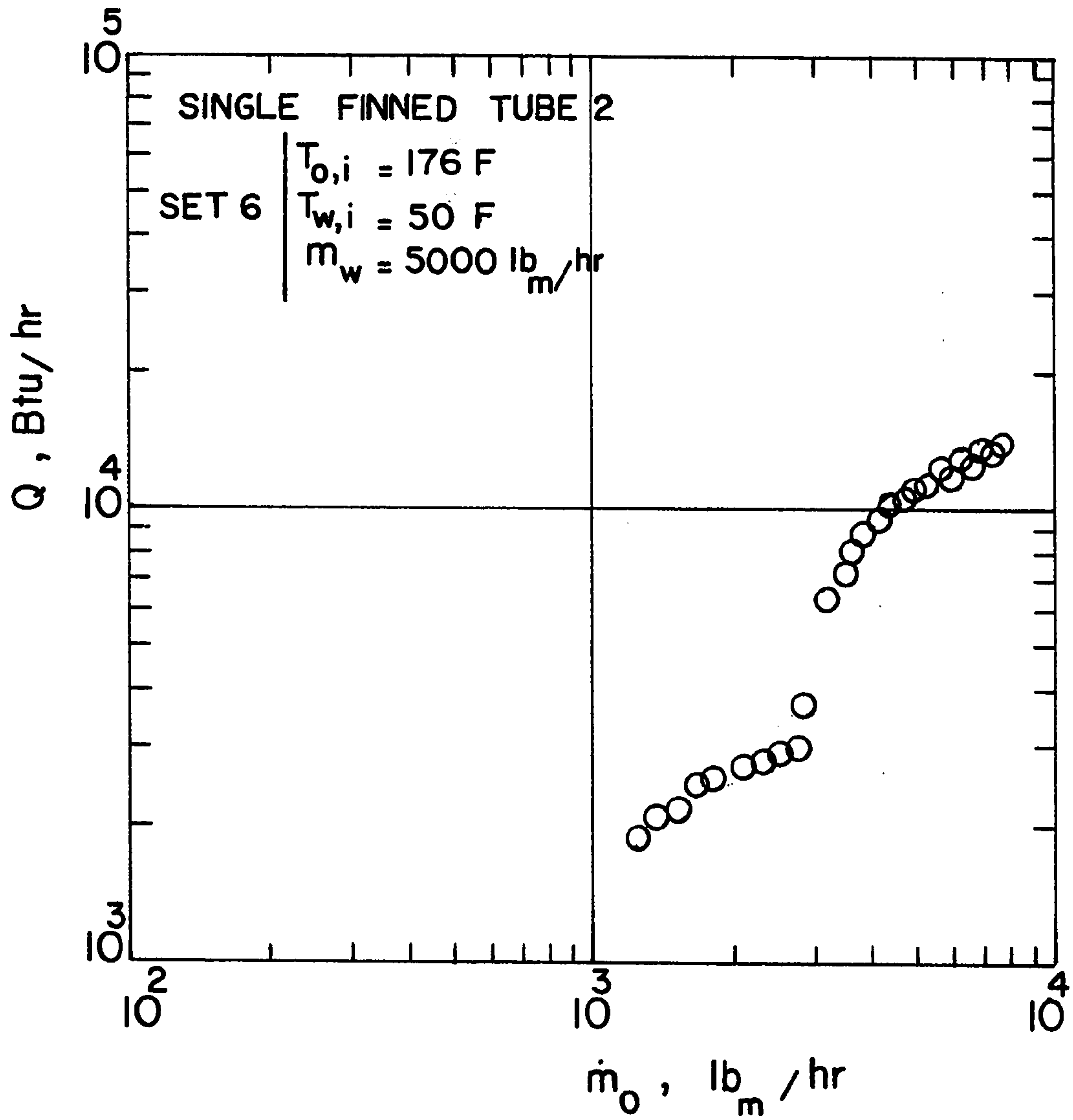


FIG. 15 HEAT TRANSFER DATA AT AN OIL INLET TEMPERATURE OF 176 F FOR SINGLE FINNED TUBE 2

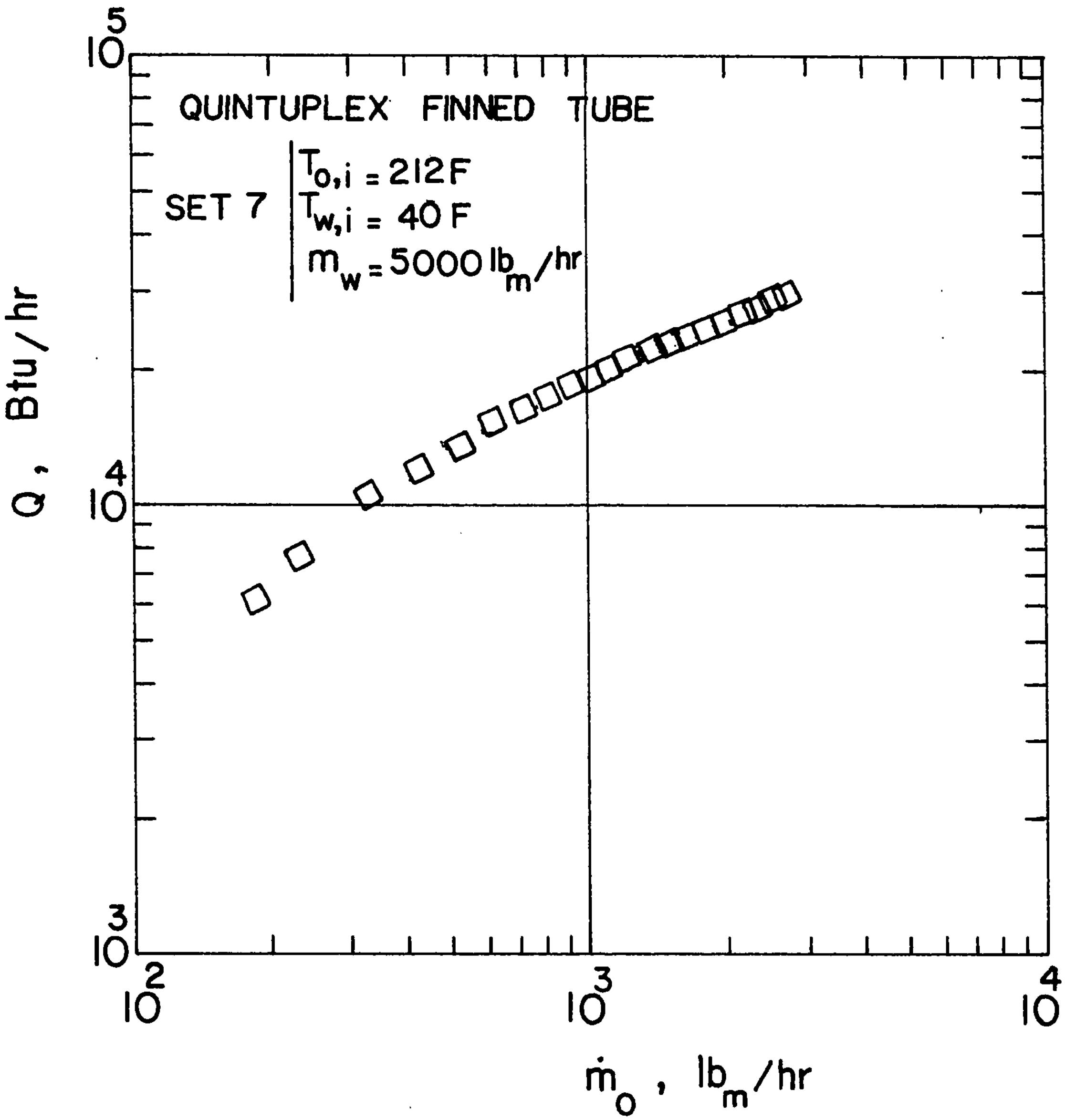


FIG. 16 HEAT TRANSFER DATA AT AN OIL INLET TEMPERATURE OF 212F FOR QUINTUPLEX FINNED TUBE

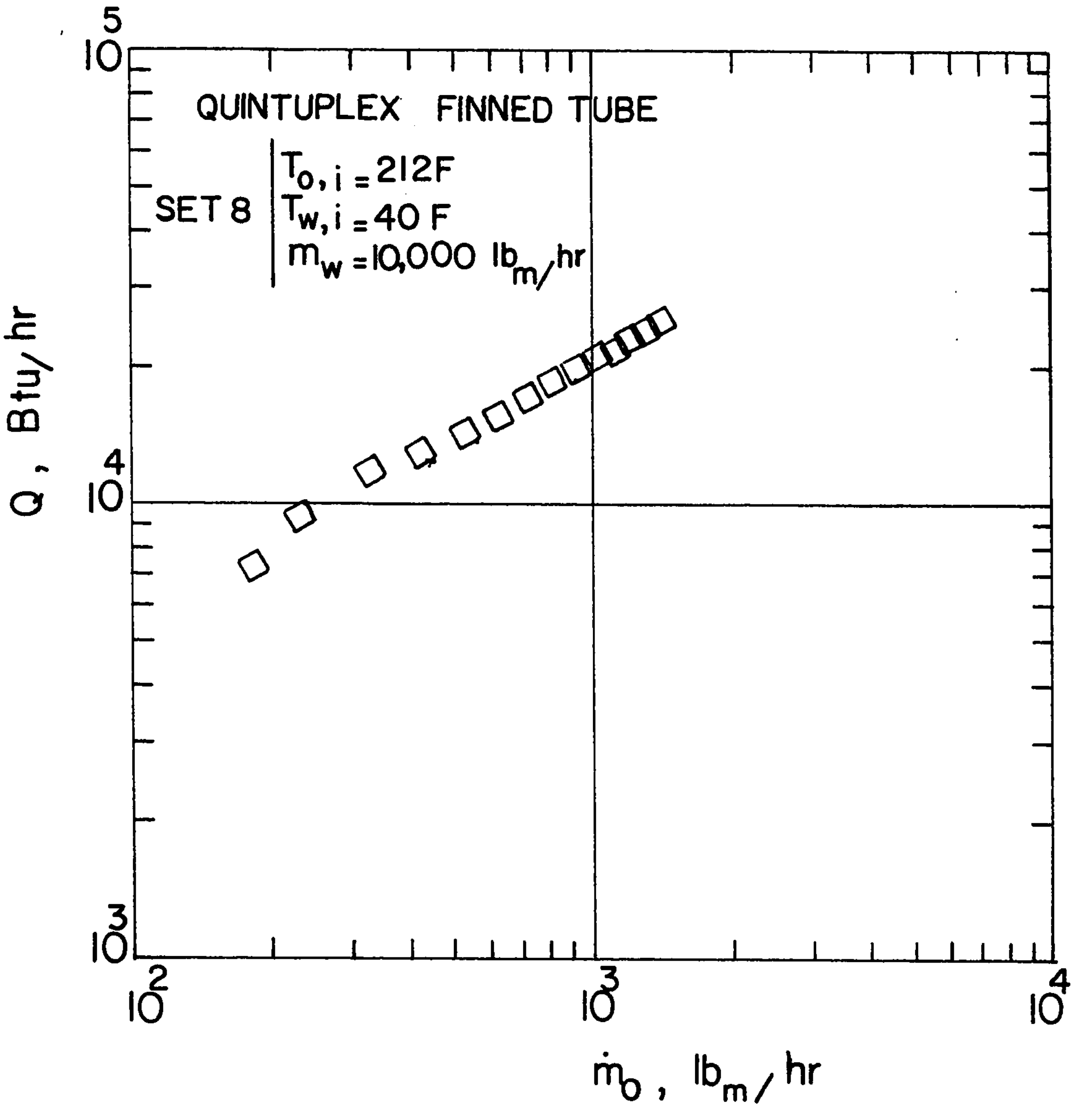


FIG. 17 HEAT TRANSFER DATA AT AN OIL INLET TEMPERATURE OF 212 F FOR QUINTUPLEX FINNED TUBE

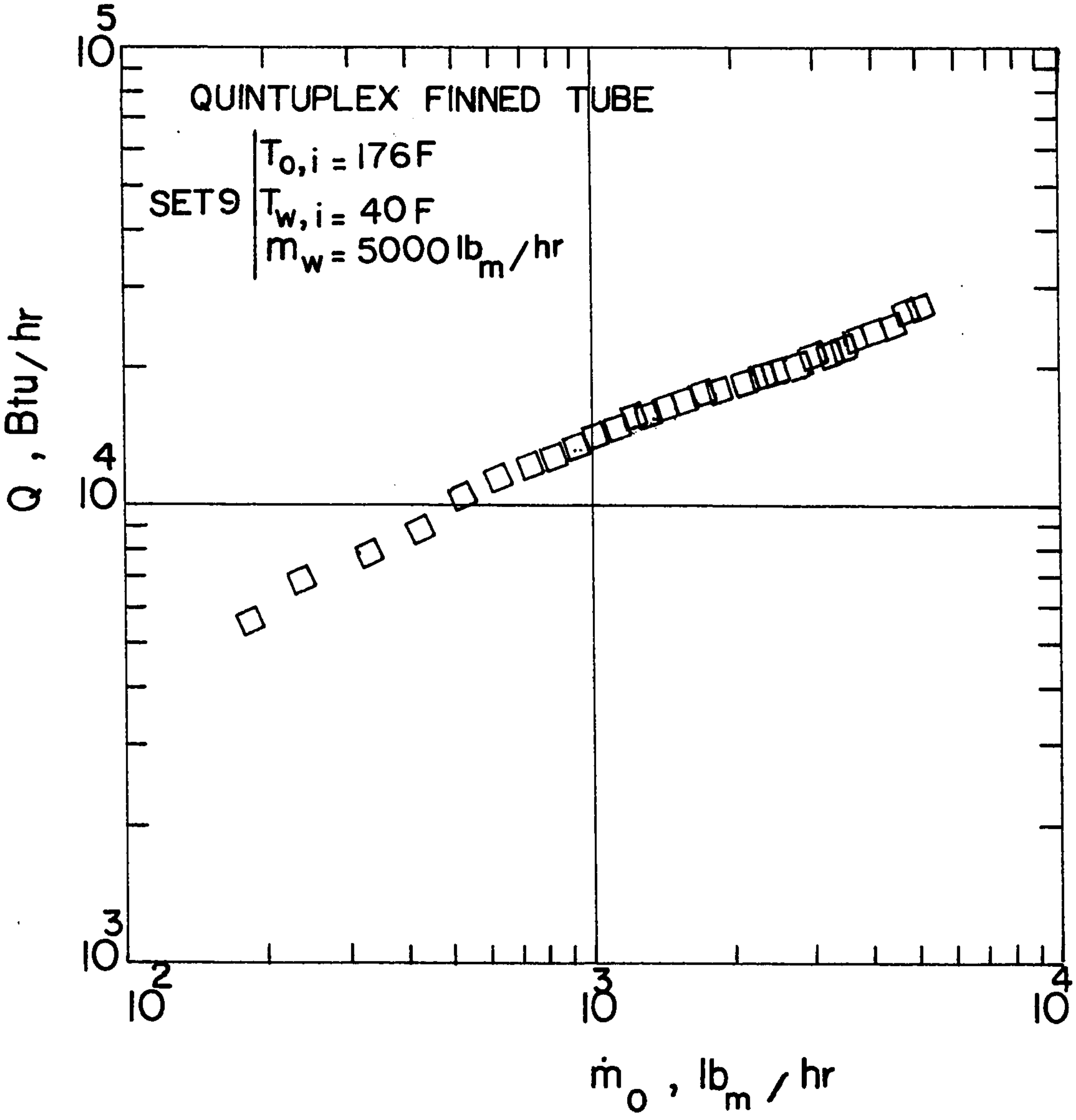


FIG. 18 HEAT TRANSFER DATA AT AN OIL INLET TEMPERATURE OF 176 F FOR QUINTUPLEX FINNED TUBE

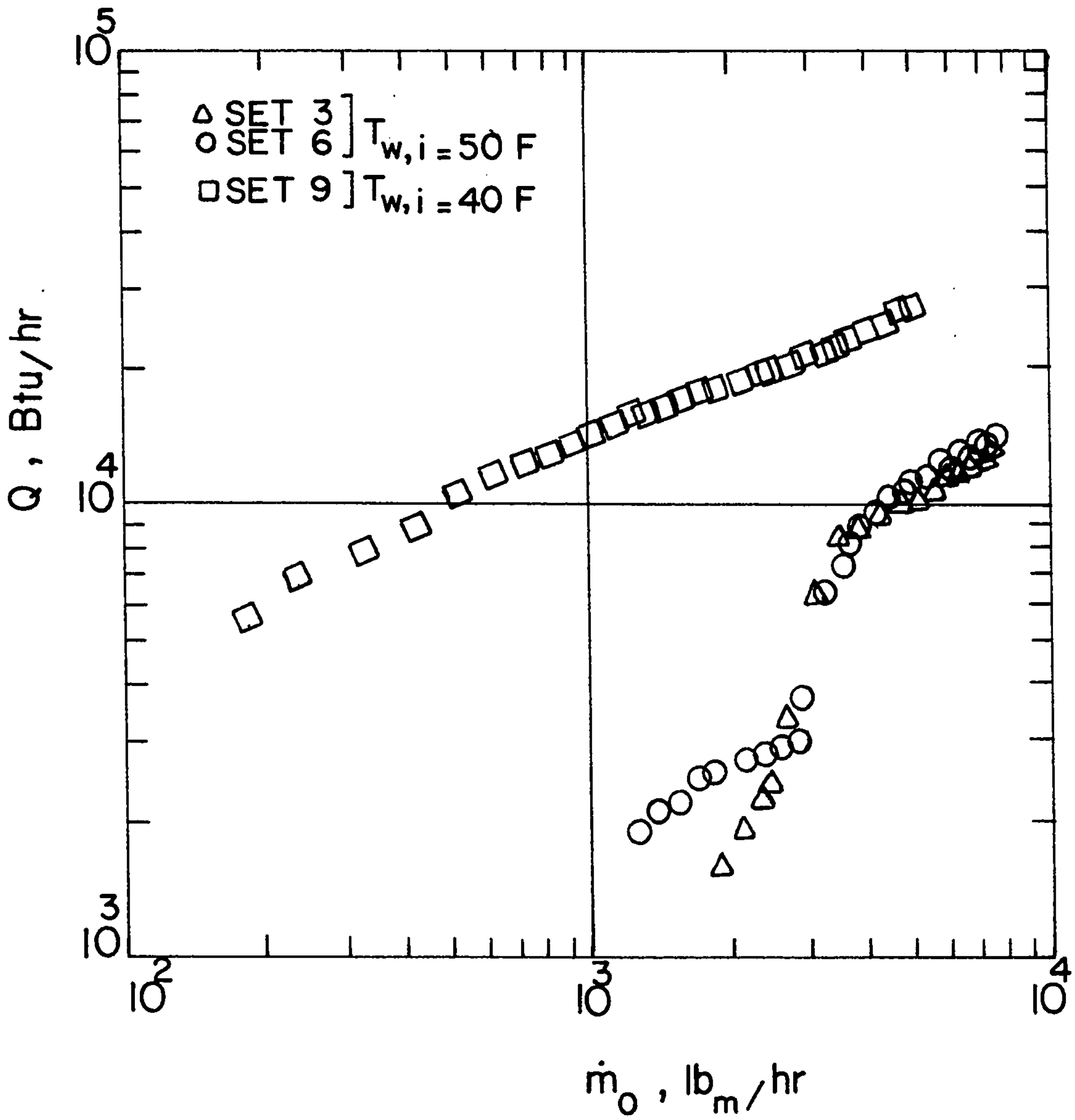


FIG. 19 HEAT TRANSFER DATA AT AN OIL INLET TEMPERATURE OF 176 F FOR COMPARISON BETWEEN THE TUBES

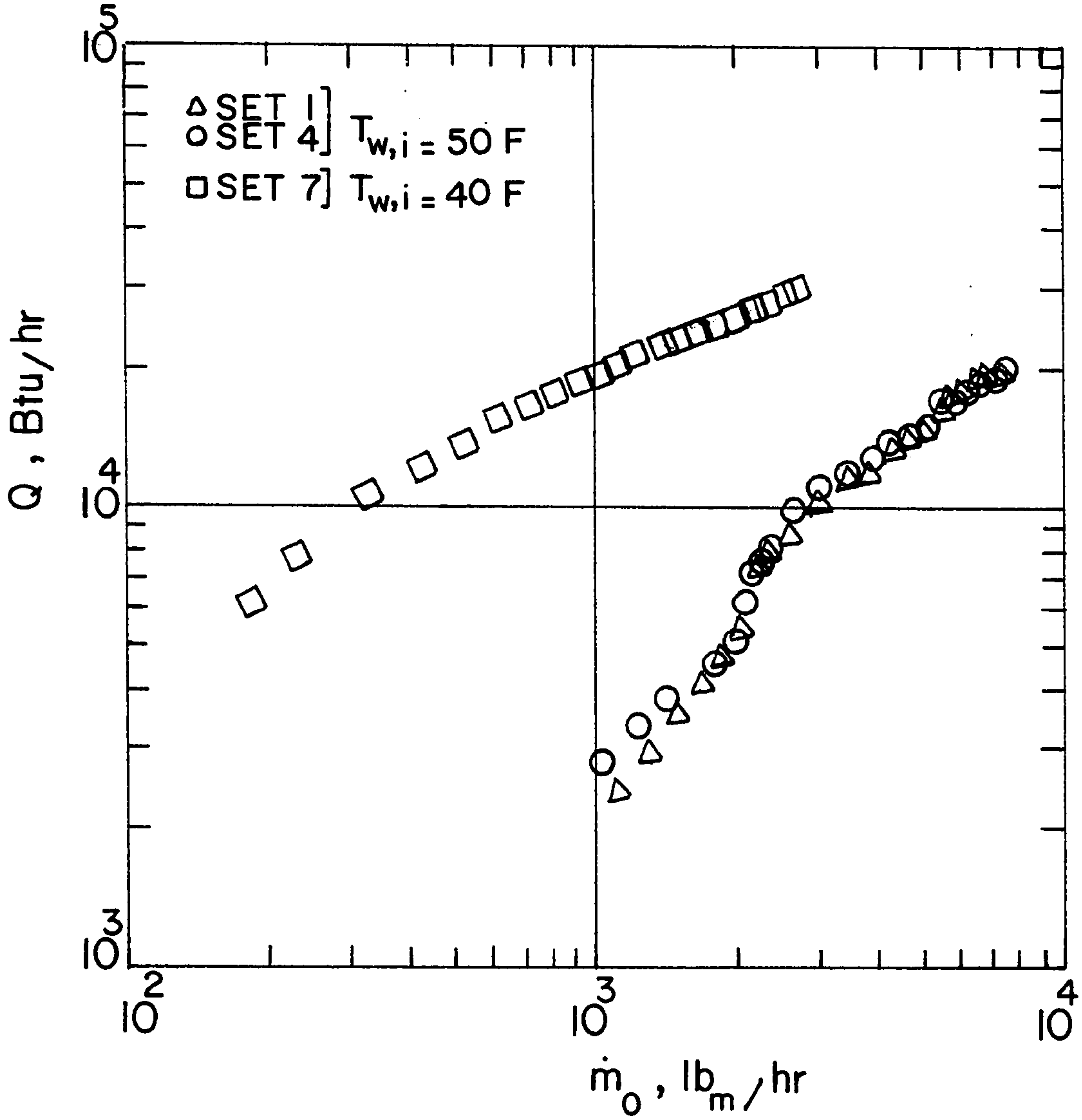


FIG. 20 HEAT TRANSFER DATA AT AN OIL INLET TEMPERATURE OF 212F FOR COMPARISON BETWEEN THE TUBES

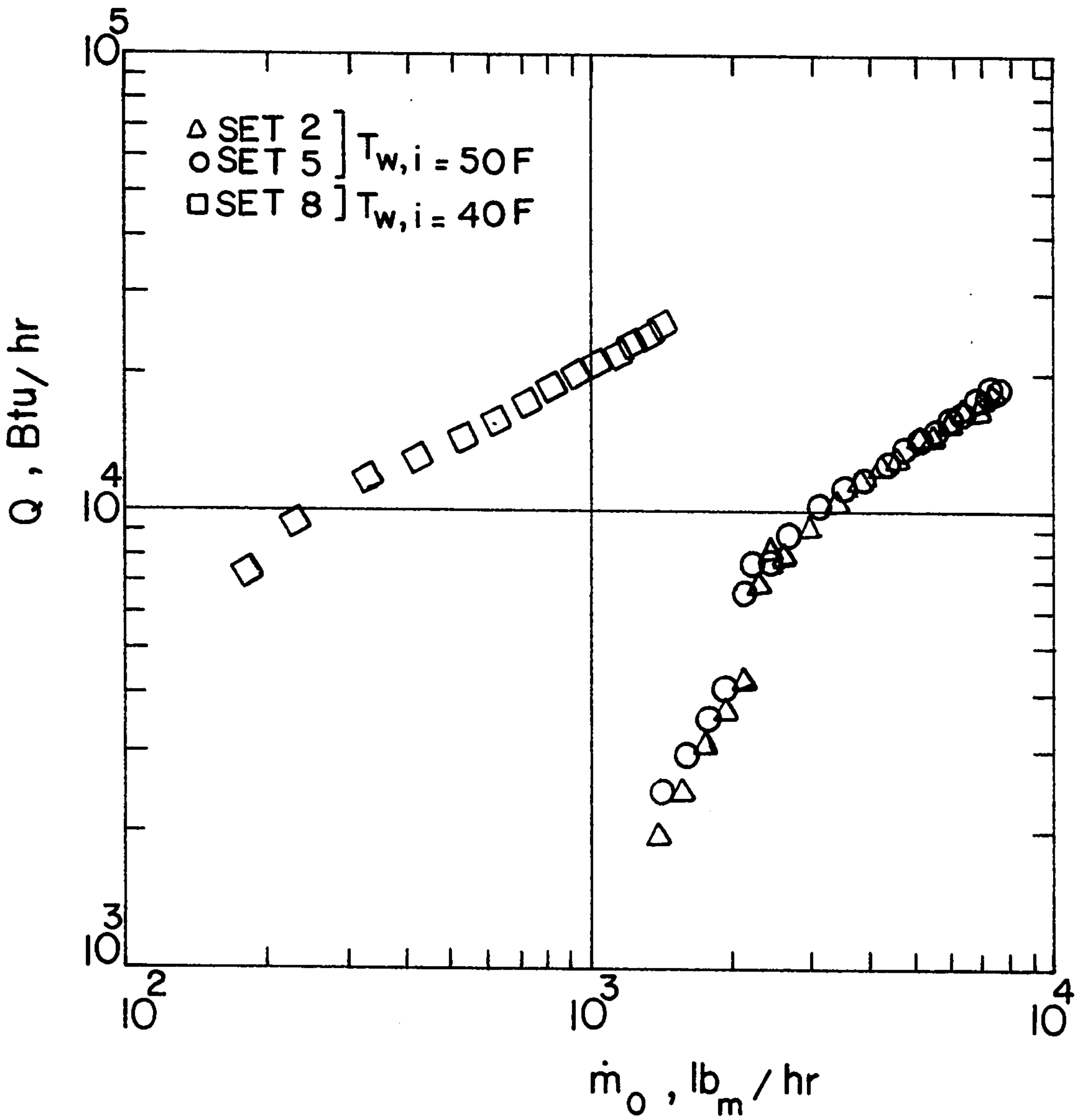


FIG. 21 HEAT TRANSFER DATA AT AN OIL INLET TEMPERATURE OF 212F FOR COMPARISON BETWEEN THE TUBES

The effect of the cooling water flow rate on heat transfer performance of the three test tubes is investigated, and the results are shown in Figs. 22 and 23. All the data points presented in Figs. 22 and 23 correspond to the same oil inlet temperature, however for every test section two sets of data points corresponding to two different water flow rates are presented. According to the results shown in Figs. 22 and 23, the water flow rate had little effect on the critical conditions at which change of regime in single finned tubes occured. In addition, the effect of water flow rate on the values of Q also does not appear to be significant.

3.2 Pressure Drop Results

Isothermal pressure drop data are collected for three tubes and the results are shown in Figs. 24 to 29. The isothermal pressure drop is recorded while the flow of water is completely stopped. Therefore, the temperature and viscosity of oil is almost constant over the entire length of the tube. A slight temperature drop is observed at the exit of the test section because of heat loss to the atmosphere. Thus, e.g., when the inlet oil temperature was 176F, the exit was 172F. The pressure drop measured under these conditions may be considered isothermal at 174F.

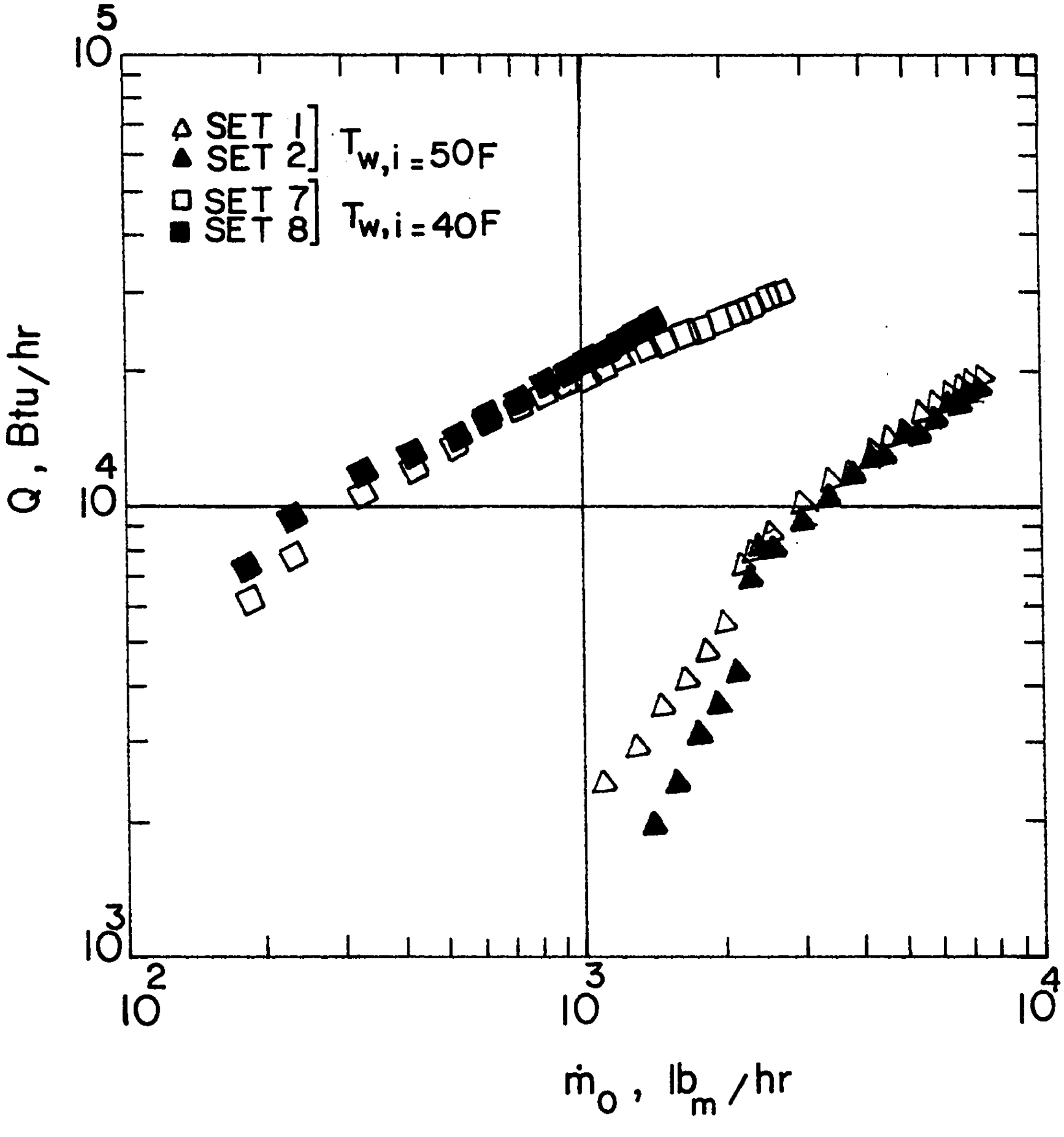


FIG. 22 EFFECT OF COOLING WATER FLOW RATE ON HEAT TRANSFER AT AN OIL INLET TEMPERATURE OF 212F

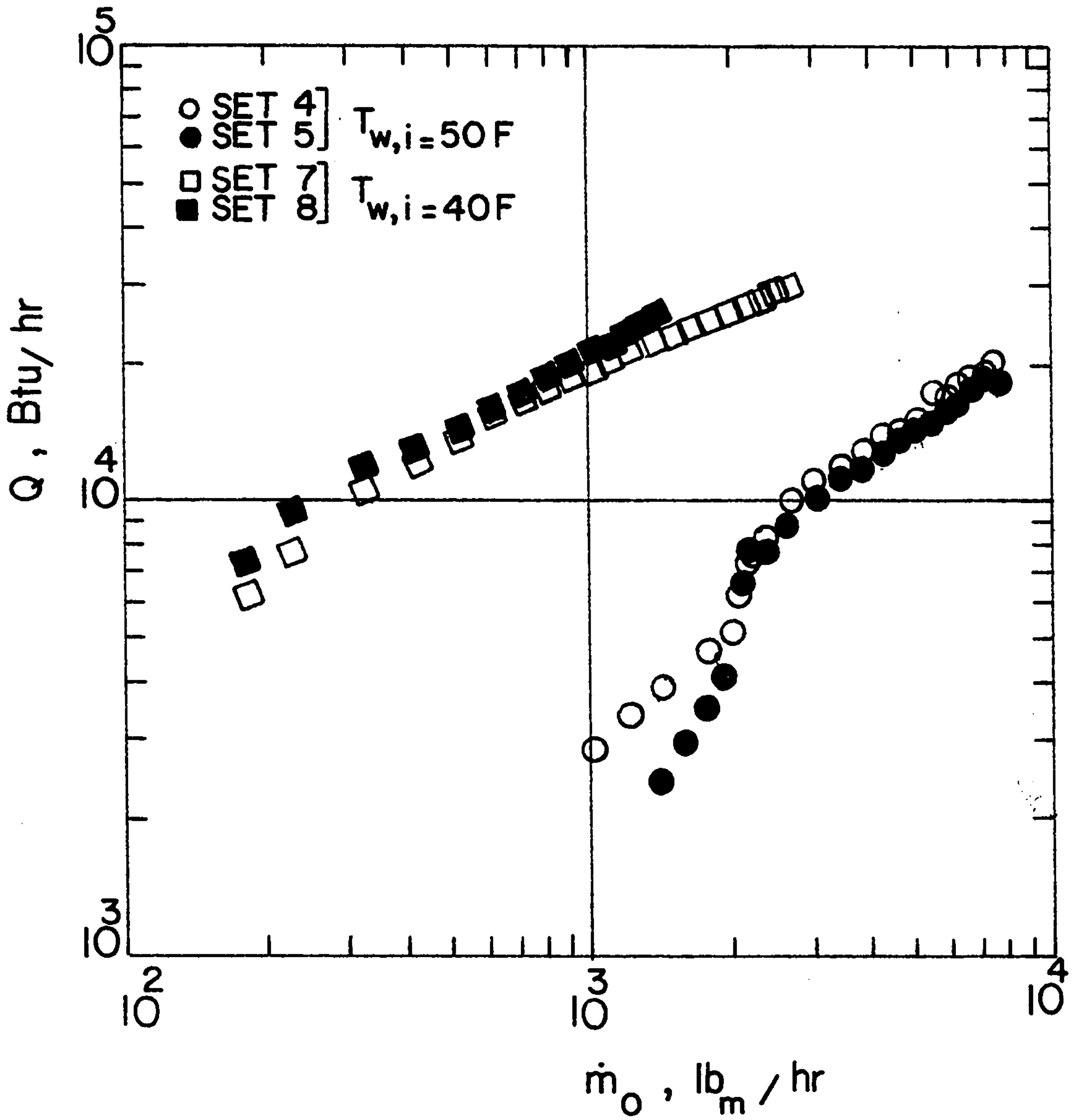


FIG. 23 EFFECT OF COOLING WATER FLOW RATE ON HEAT TRANSFER AT AN OIL INLET TEMPERATURE OF 212 F

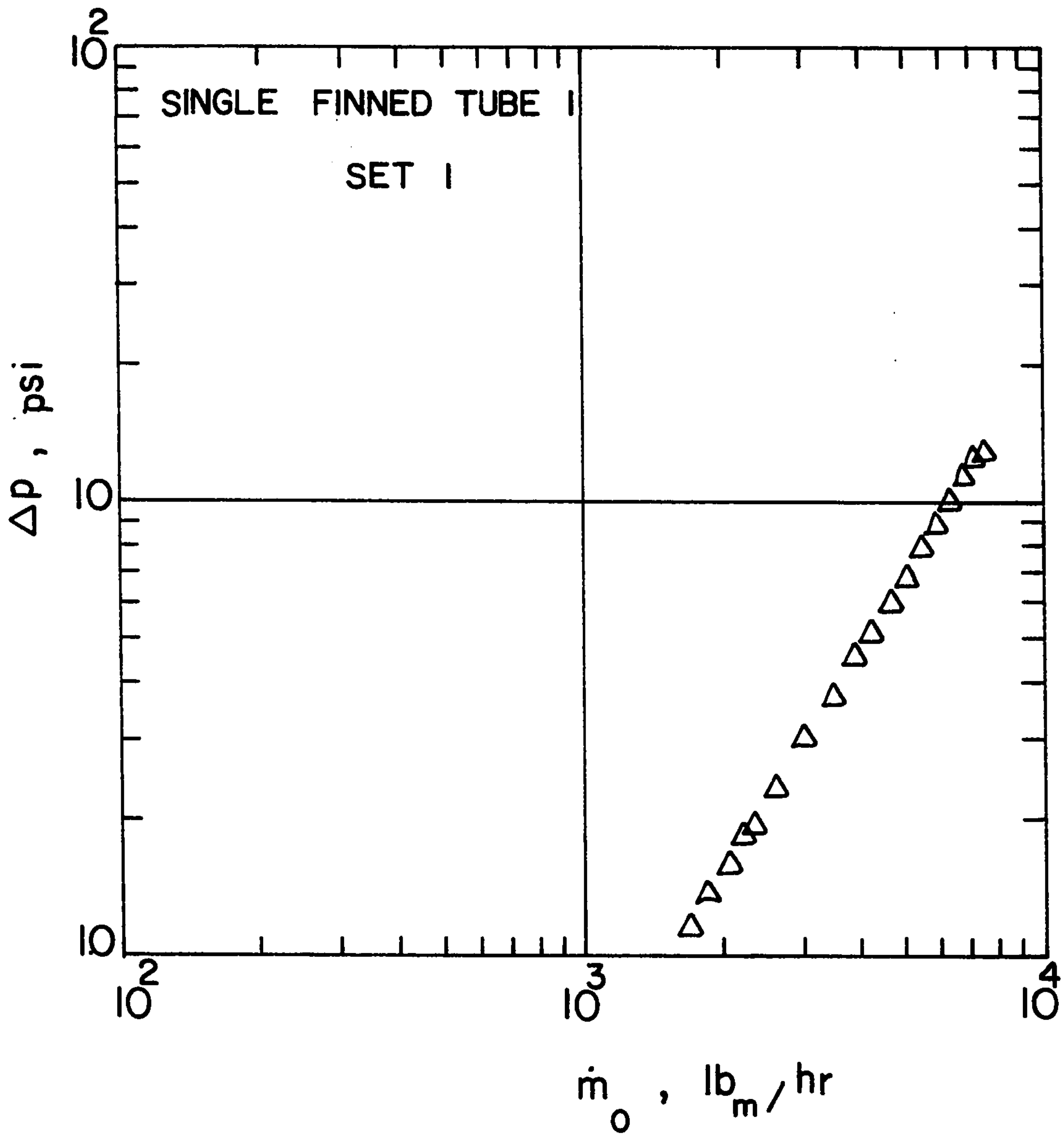


FIG. 24 ISOTHERMAL PRESSURE DROP DATA AT AN OIL INLET TEMPERATURE OF 210F FOR SINGLE FINNED TUBE I

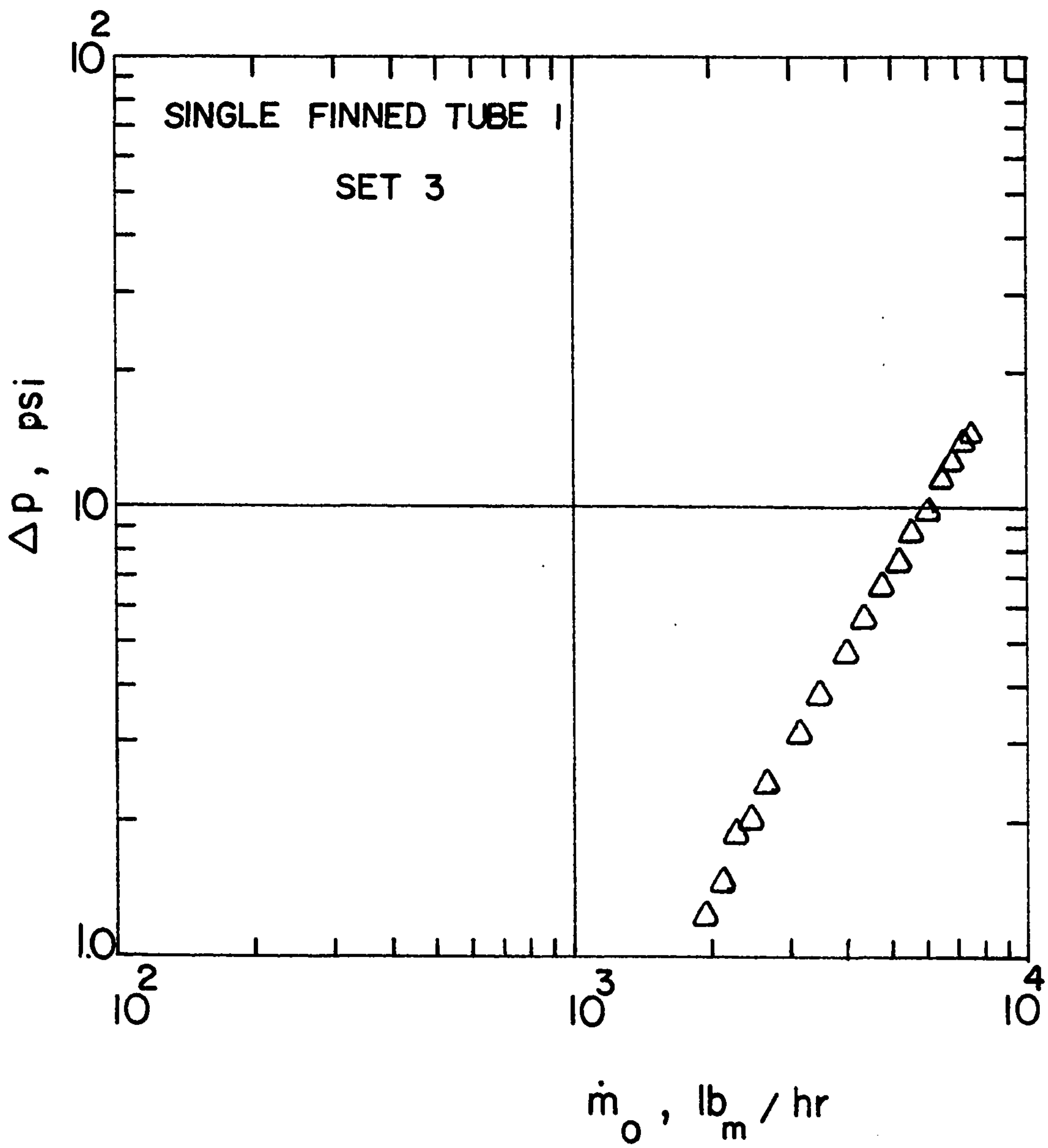


FIG. 25 ISOTHERMAL PRESSURE DROP DATA AT AN OIL INLET TEMPERATURE OF 174F FOR SINGLE FINNED TUBE I

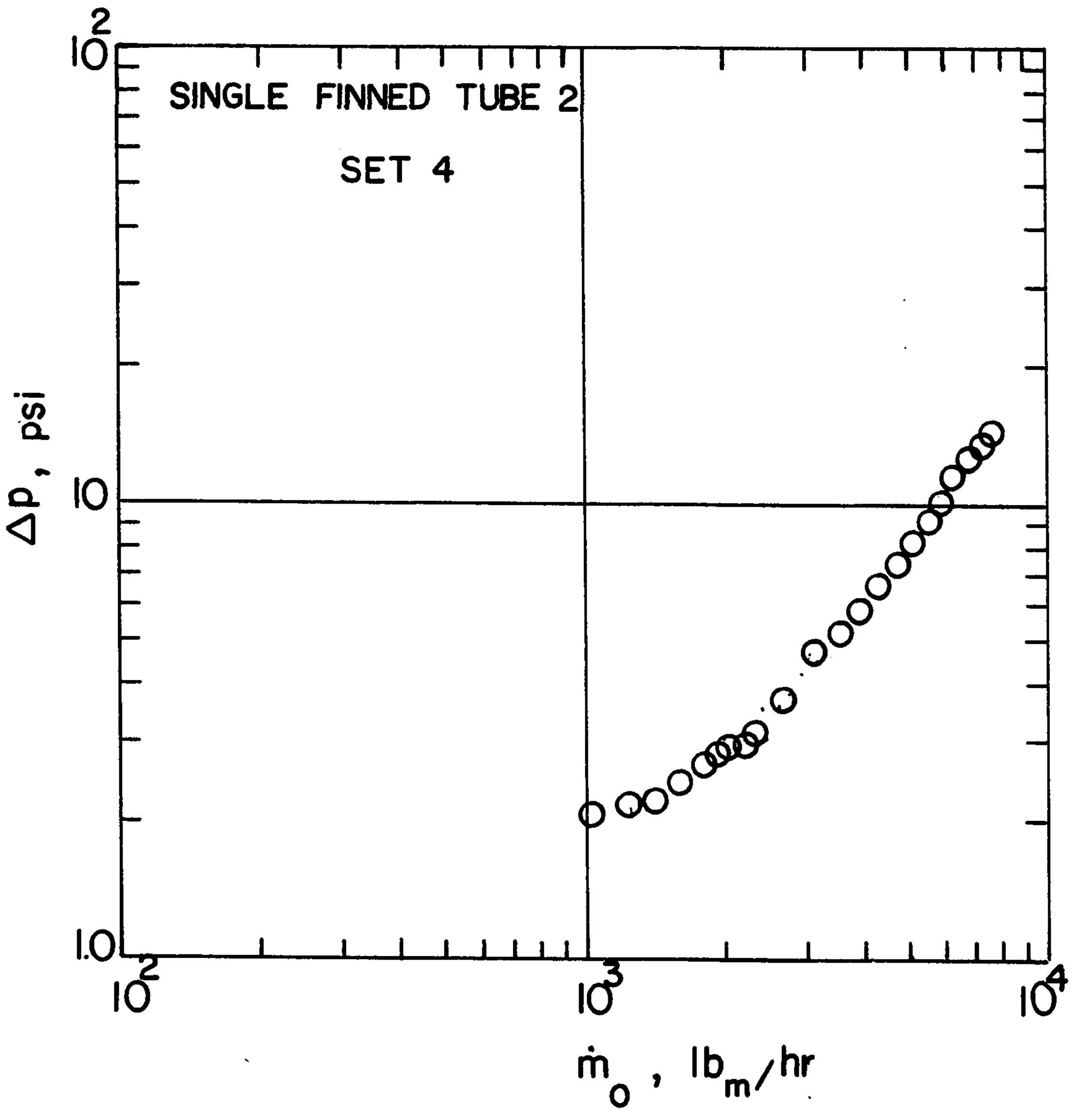


FIG. 26 ISOTHERMAL PRESSURE DROP DATA AT AN OIL INLET TEMPERATURE OF 210F FOR SINGLE FINNED TUBE 2

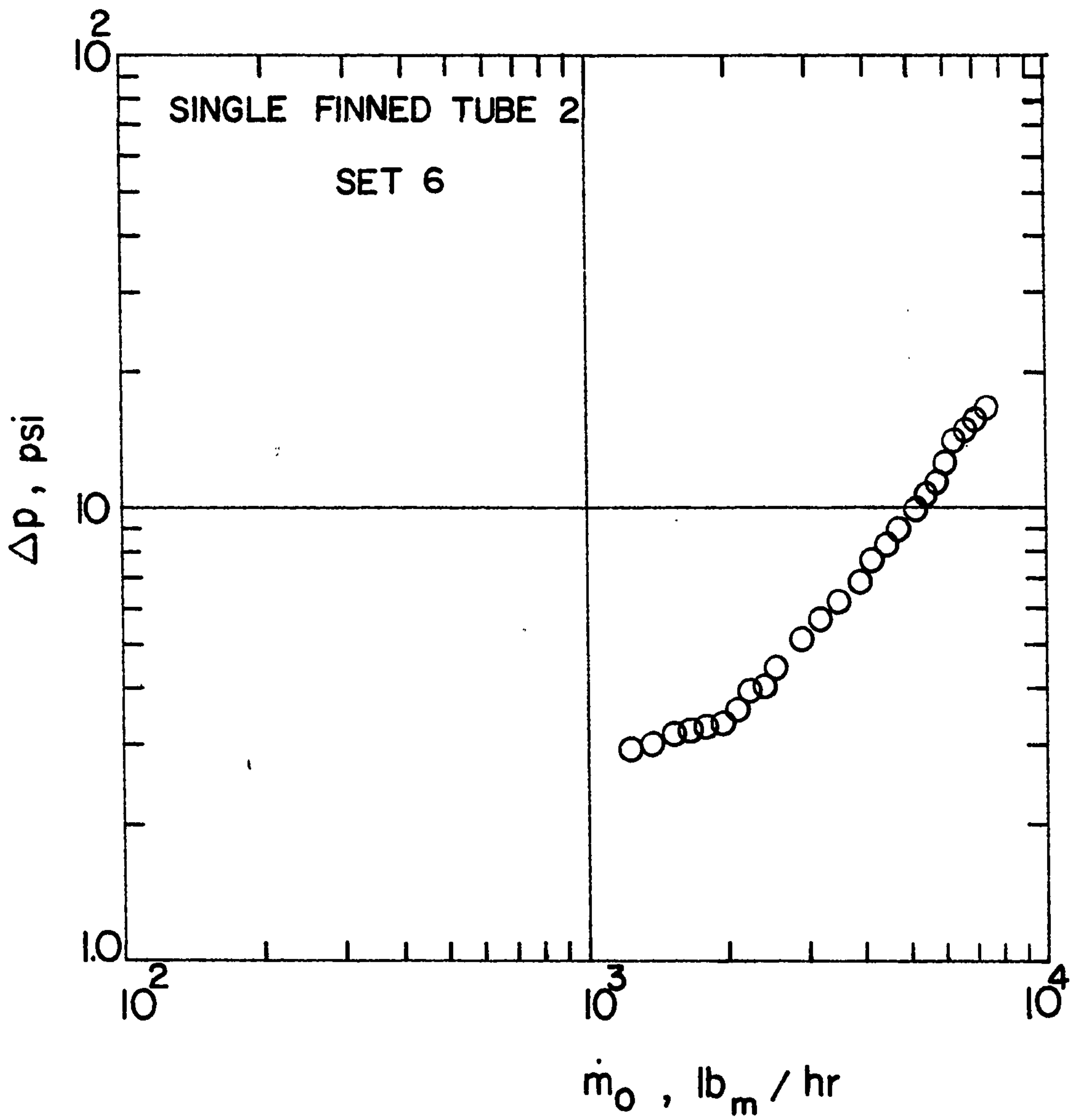


FIG. 27 ISOTHERMAL PRESSURE DROP DATA AT AN OIL INLET TEMPERATURE OF 174F FOR SINGLE FINNED TUBE 2

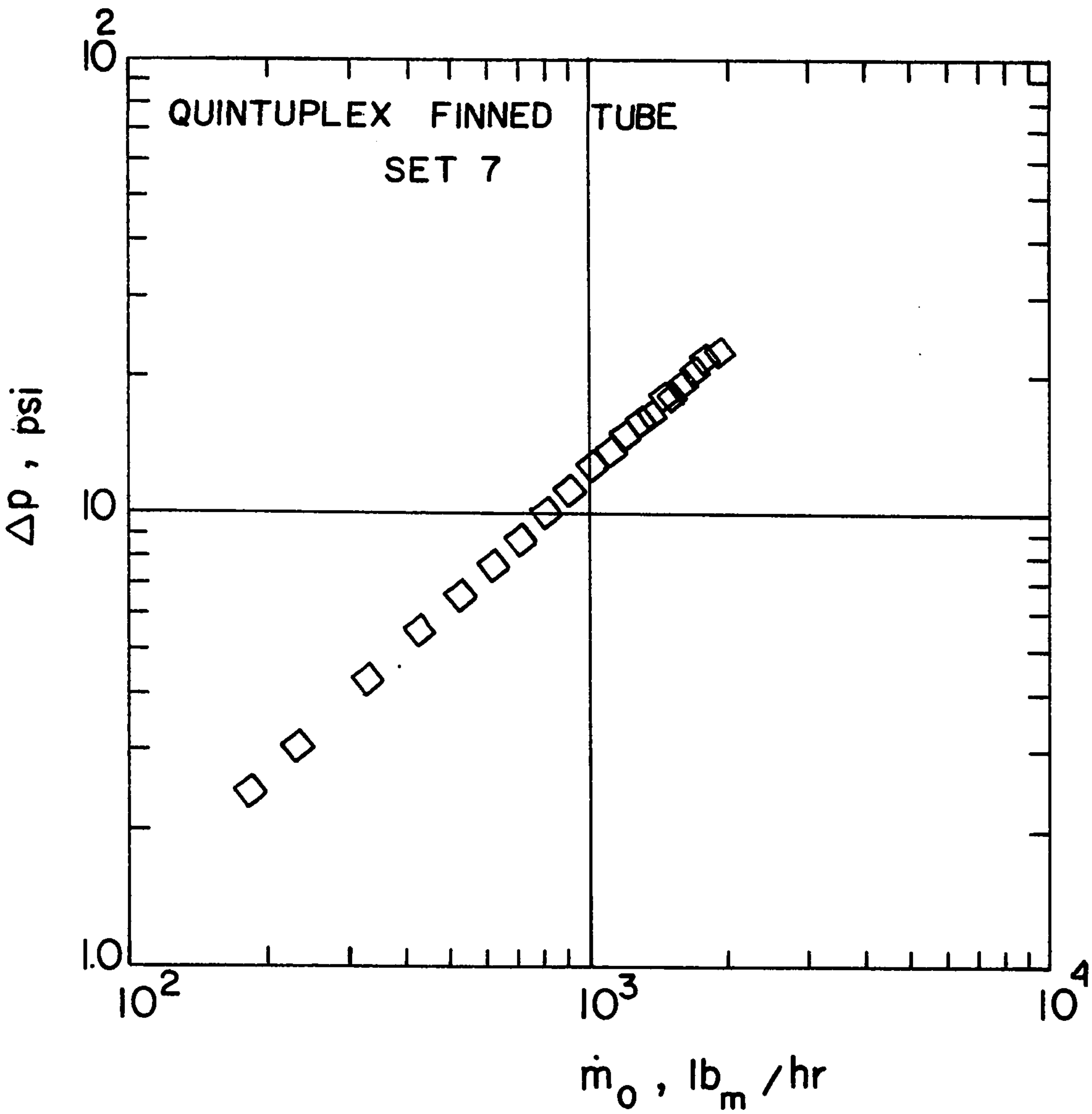


FIG. 28 ISOTHERMAL PRESSURE DROP DATA AT AN OIL INLET TEMPERATURE OF 210 F FOR QUINTUPLEX FINNED TUBE

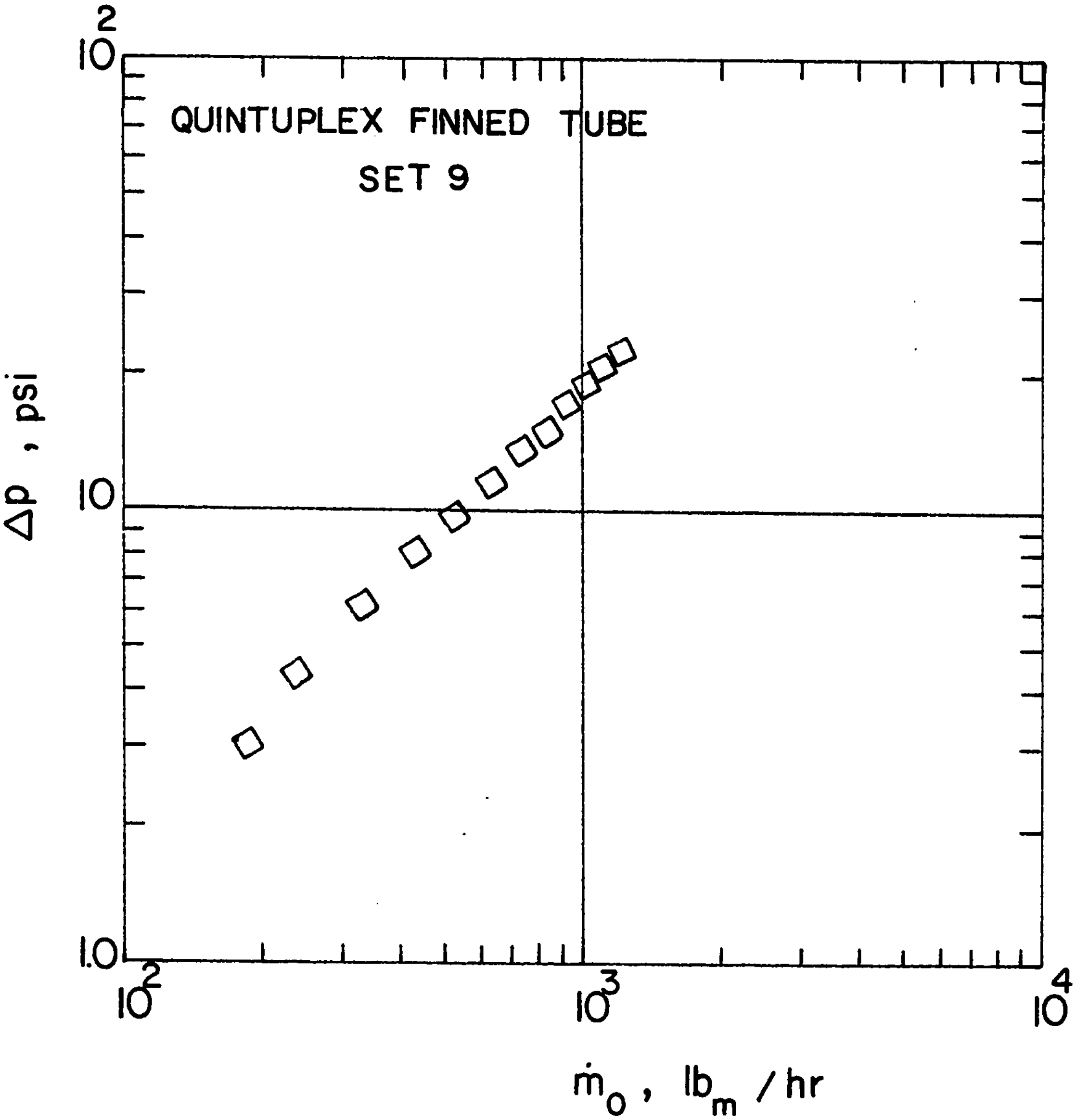


FIG. 29 ISOTHERMAL PRESSURE DROP DATA AT AN OIL INLET TEMPERATURE OF 174F FOR QUINTUPLEX FINNED TUBE

The advantage of taking isothermal pressure drop is to be able to collect more data at higher mass flow rate of oil, otherwise in the presence of cooling water, viscosity progressively increases over the entire length and thus the total pressure drop increases. We are limited by the fact that the maximum range of our largest transducer does not permit to record the pressure drop over 25 psi. Our three transducers have ranges of 0-0.4; 0-3; and 0 - 25 psi. Isothermal pressure drop is recorded when the temperatures at the inlet and the exit of the test section are observed to be constant for at least half an hour. Pressure drop comparison between quintuplex and single-finned tubes are shown in Figs. 30 & 31. The enhancement in heat transfer outlined earlier for the quintuplex finned tube is associated with high pressure drops as compared with the single-finned tubes.

When the designer wishes to use these isothermal pressure drop curves for an estimation of pumping power in actual operation of his planned heat exchanger he needs to consider an average between his inlet and outlet temperatures. This, of course, will only provide an approximation. To do it more exactly, one would need to know the relationship between the temperature and viscosity throughout the entire range of operating conditions.

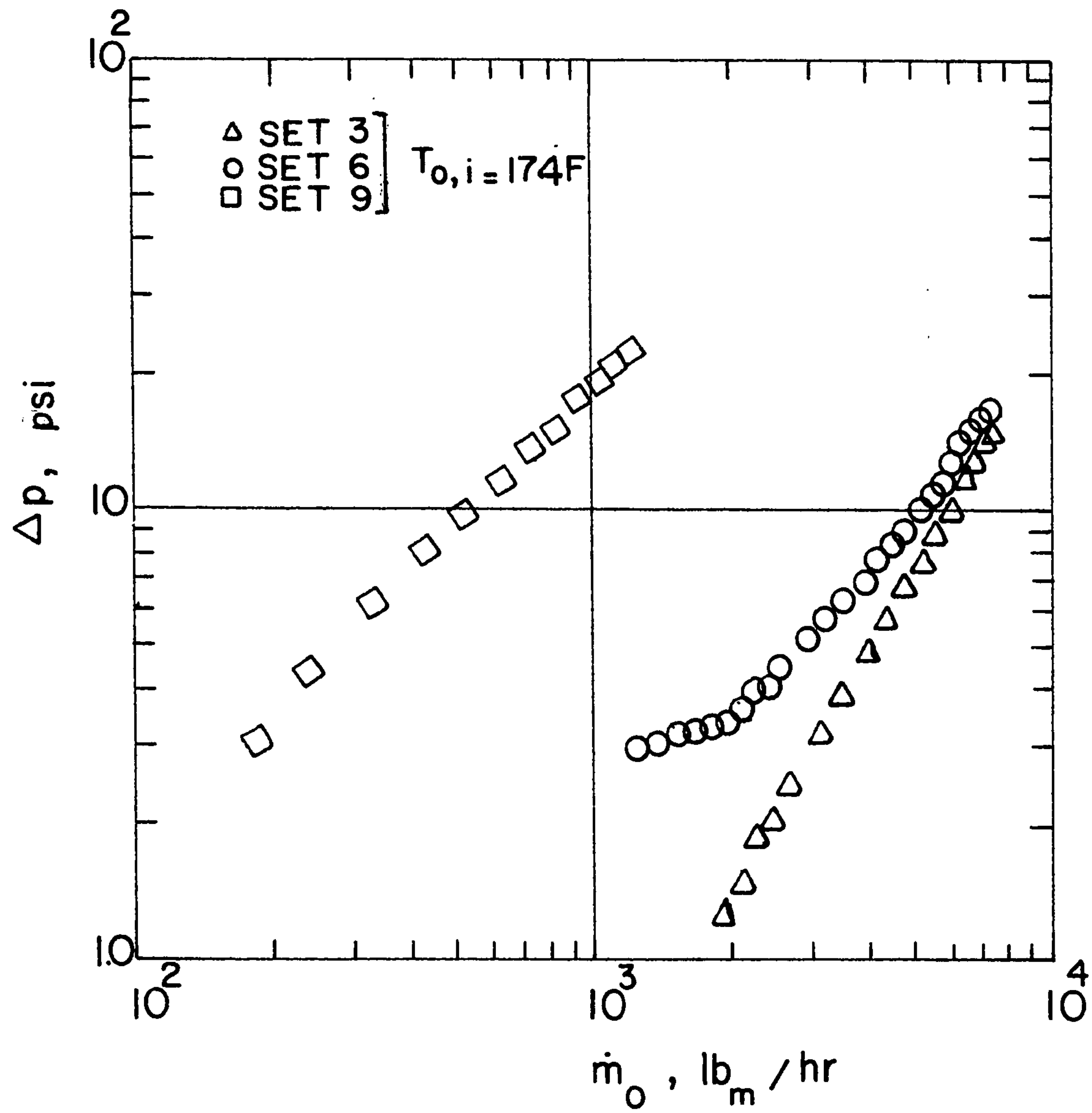


FIG. 30 ISOTHERMAL PRESSURE DROP DATA AT AN OIL INLET TEMPERATURE OF 174F FOR COMPARISON BETWEEN THE TUBES

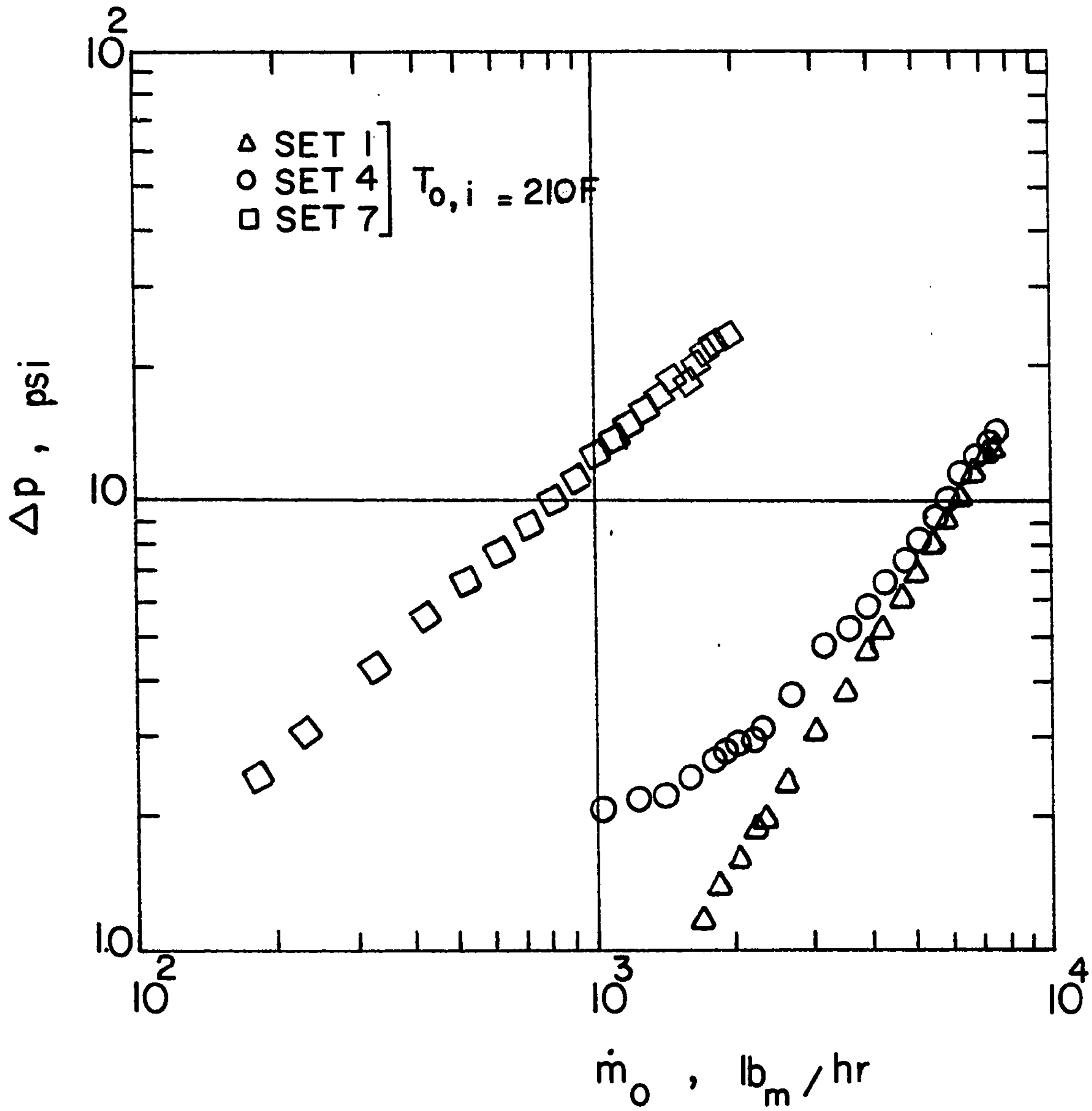


FIG. 31 ISOTHERMAL PRESSURE DROP DATA AT AN OIL INLET TEMPERATURE OF 210F FOR COMPARISON BETWEEN THE TUBES

3.3 Effect of Oil Inlet Temperature on Heat Transfer, Pressure Drop and Pumping Power

Different curves are plotted in Figs. 32 to 35, representing the effect of oil inlet temperature on heat transfer, pressure drop and pumping power. The quintuplex finned tube has been tested at two different oil inlet tempratures and the results are shown in Fig. 34. It is observed that at the same oil mass flow rate, a higher value of Q is obtained with less pressure drop when the oil inlet temperature to the test section is higher, and hence less pumping power is required.

Figure 35 presents the comparison of heat transfer, pressure drop and pumping power for the quintuplex and single-finned tube. The oil inlet temperature in both cases is 212F.

The values of pumping power for all the test runs are calculated by using the following relation:

$$P = \frac{144\dot{m}_o \Delta P}{60 \rho_o} \quad \text{lb ft/min} \qquad (9)$$

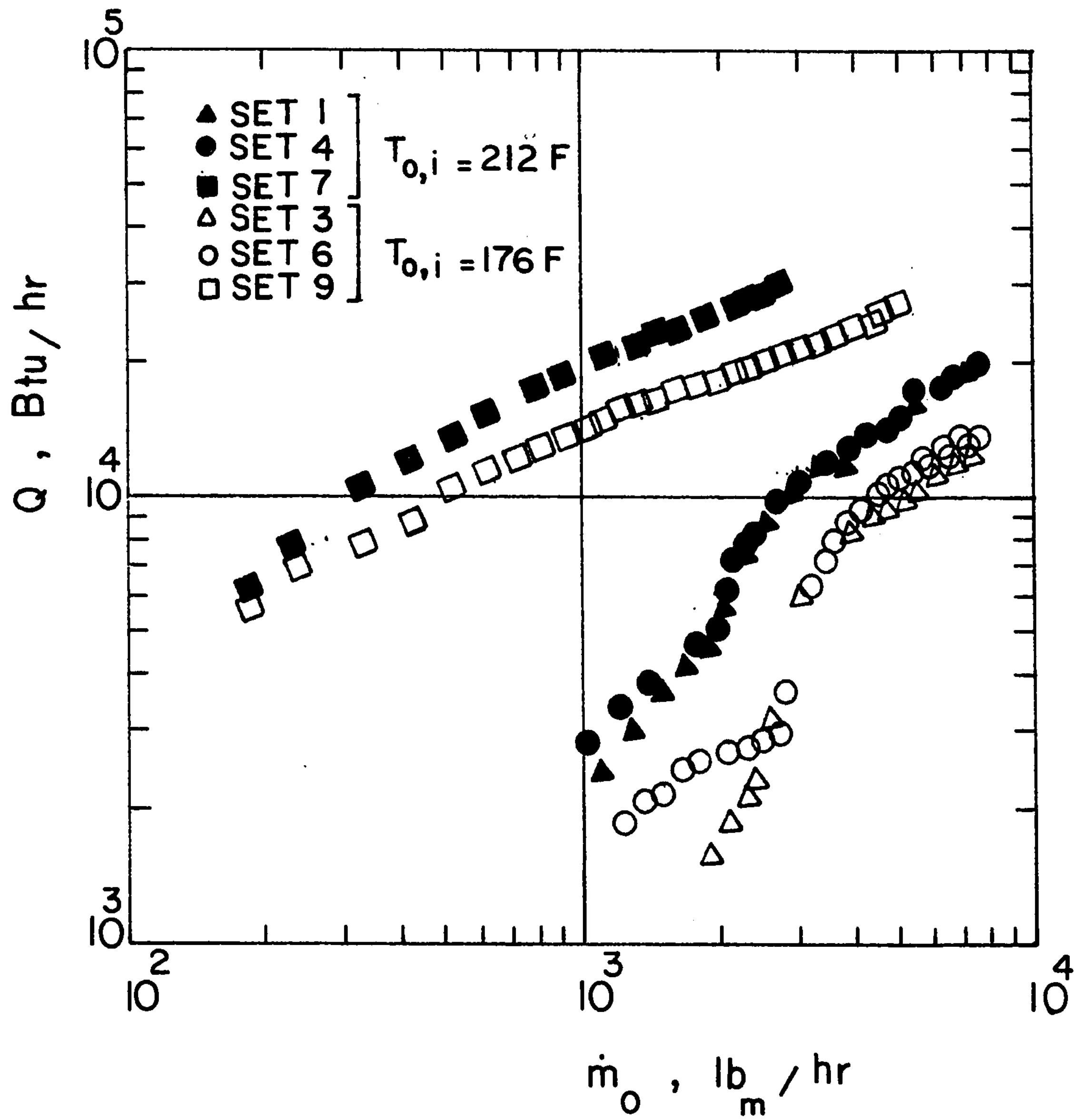


FIG. 32 EFFECT OF OIL INLET TEMPERATURE ON HEAT TRANSFER

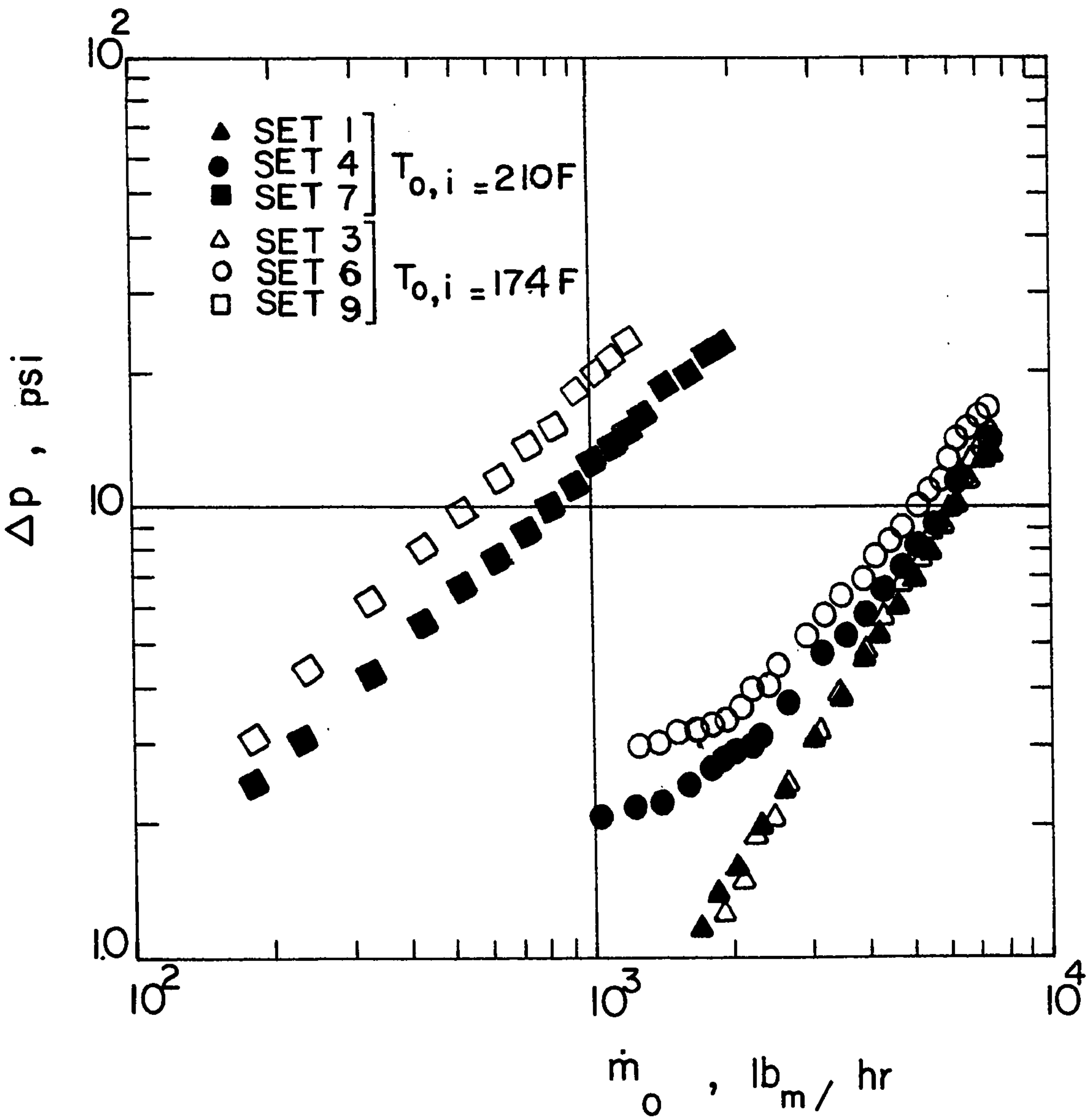


FIG.33 EFFECT OF OIL INLET TEMPERATURE ON ISOTHERMAL PRESSURE DROP

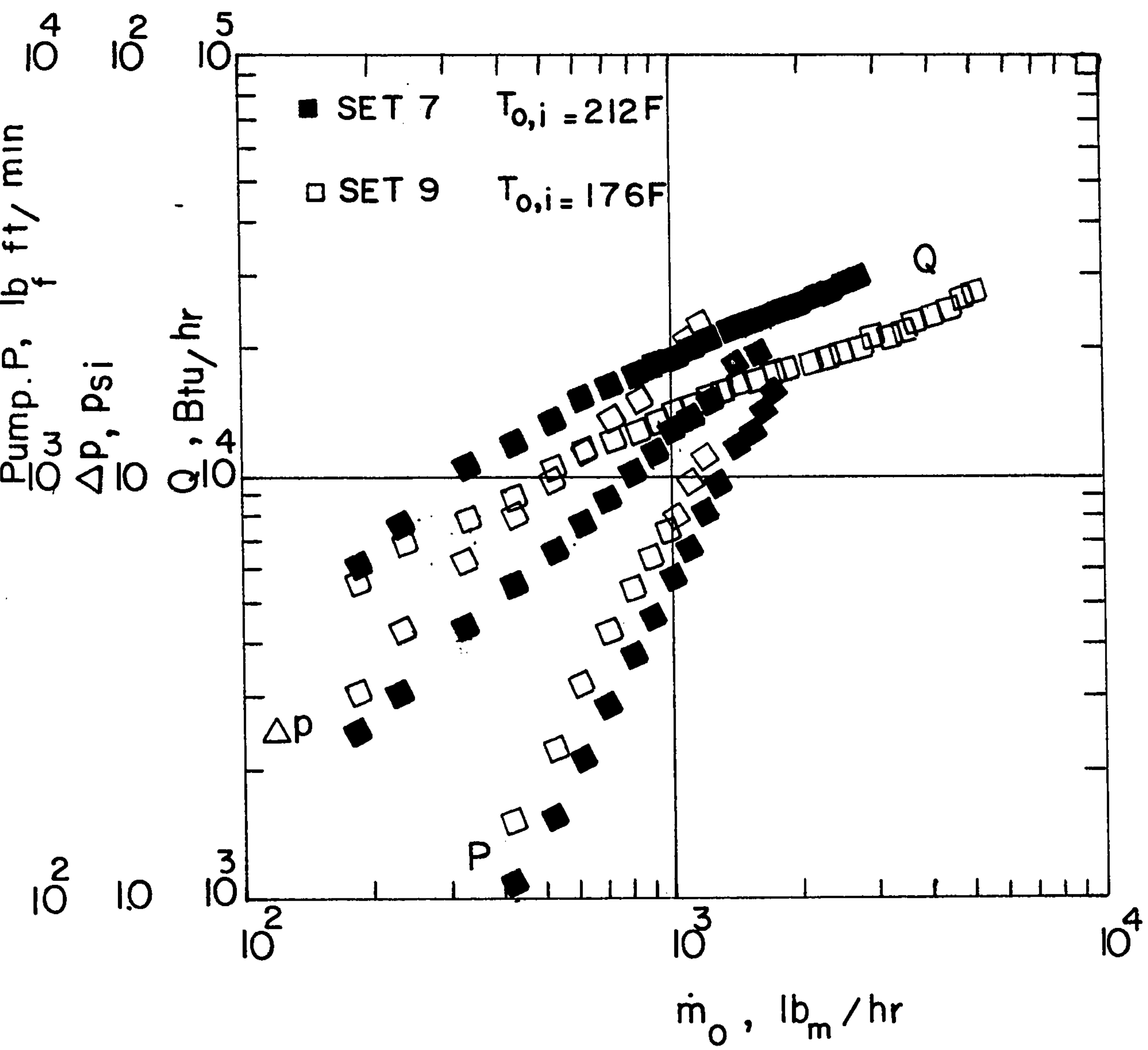


FIG.34 EFFECT OF OIL INLET TEMPERATURE ON HEAT TRANSFER, ISOTHERMAL PRESSURE DROP, AND PUMPING POWER FOR QUINTUPLEX FINNED TUBE

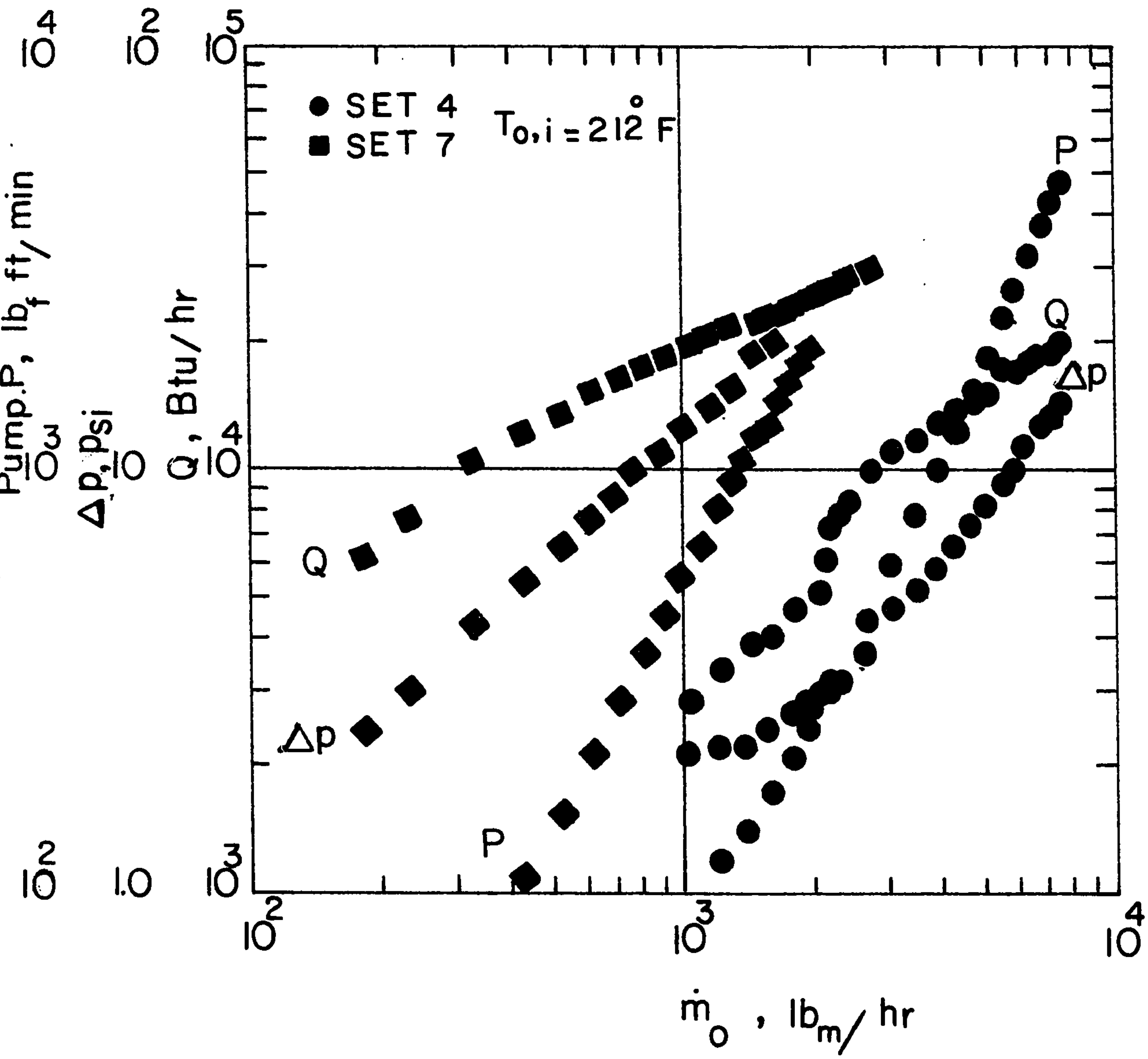


FIG. 35 HEAT TRANSFER, ISOTHERMAL PRESSURE DROP DATA AND PUMPING POWER AT AN OIL INLET TEMPERATURE OF 212°F FOR COMPARISON BETWEEN THE TUBES

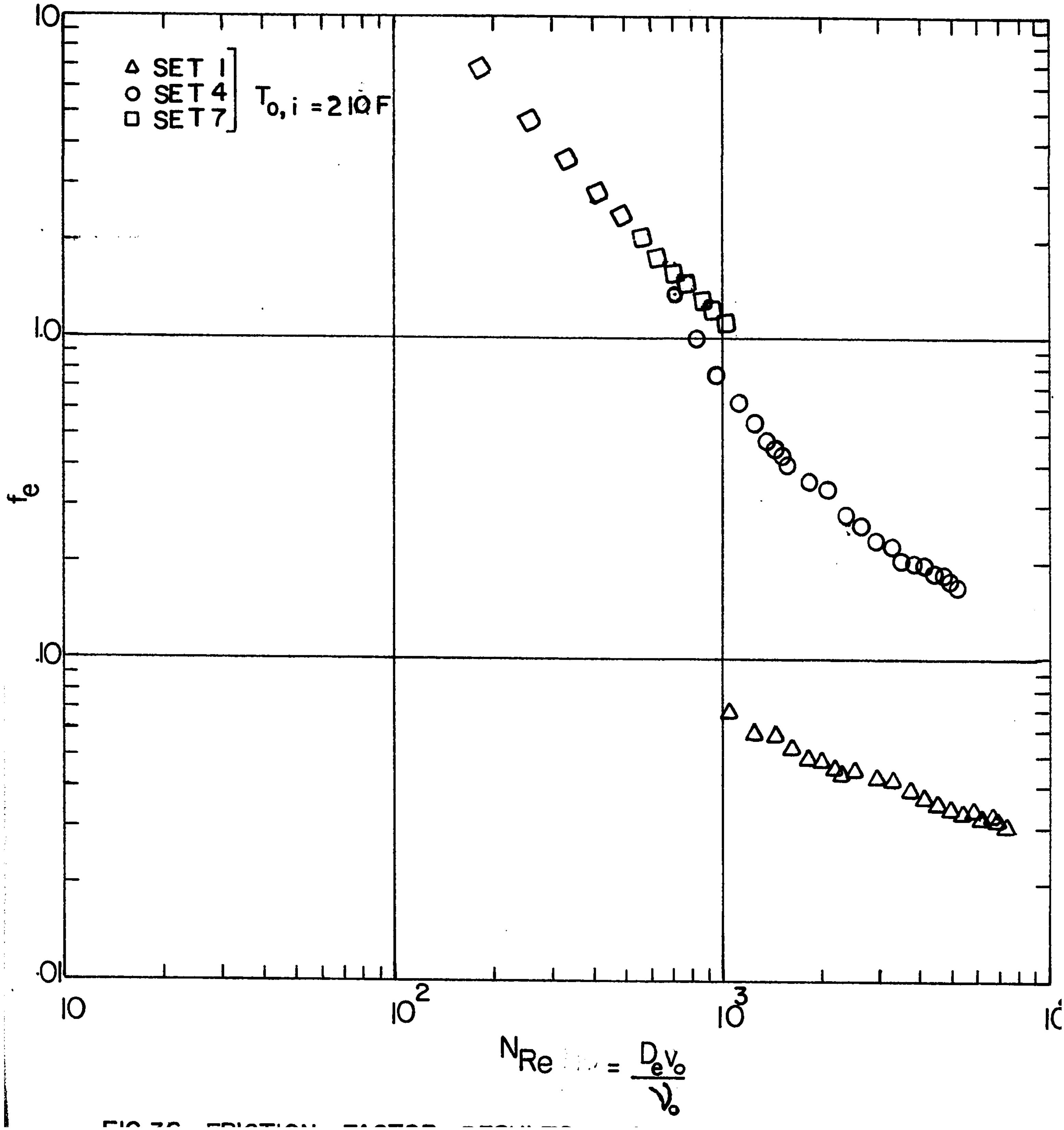
3.4 Friction Factor Results

Friction factors were calculated twice for each test: (a) based on the inside diameter, (b) based on the previously defined equivalent diameter. The results are plotted in Figs. 36 to 41 as a function of Reynolds number. The relations employed were as follows:

$$f = \frac{D \, \Delta P \, g_c}{2L \, \rho_o v^2_o} \tag{10}$$

$$N_{Re} = \frac{D v_o}{\nu_o} \tag{11}$$

These results are reported for the convenience of designers who may wish to employ the particular tubes. Thus, these graphs are in essence "product oriented" and no generalizations are, or should be, made from them.



10
1.0
.10
.01
f_e
△ SET 1
○ SET 4
□ SET 7
$T_{o,i} = 210F$
10
10^2
10^3
$N_{Re} = \frac{D_e V_o}{\nu_o}$

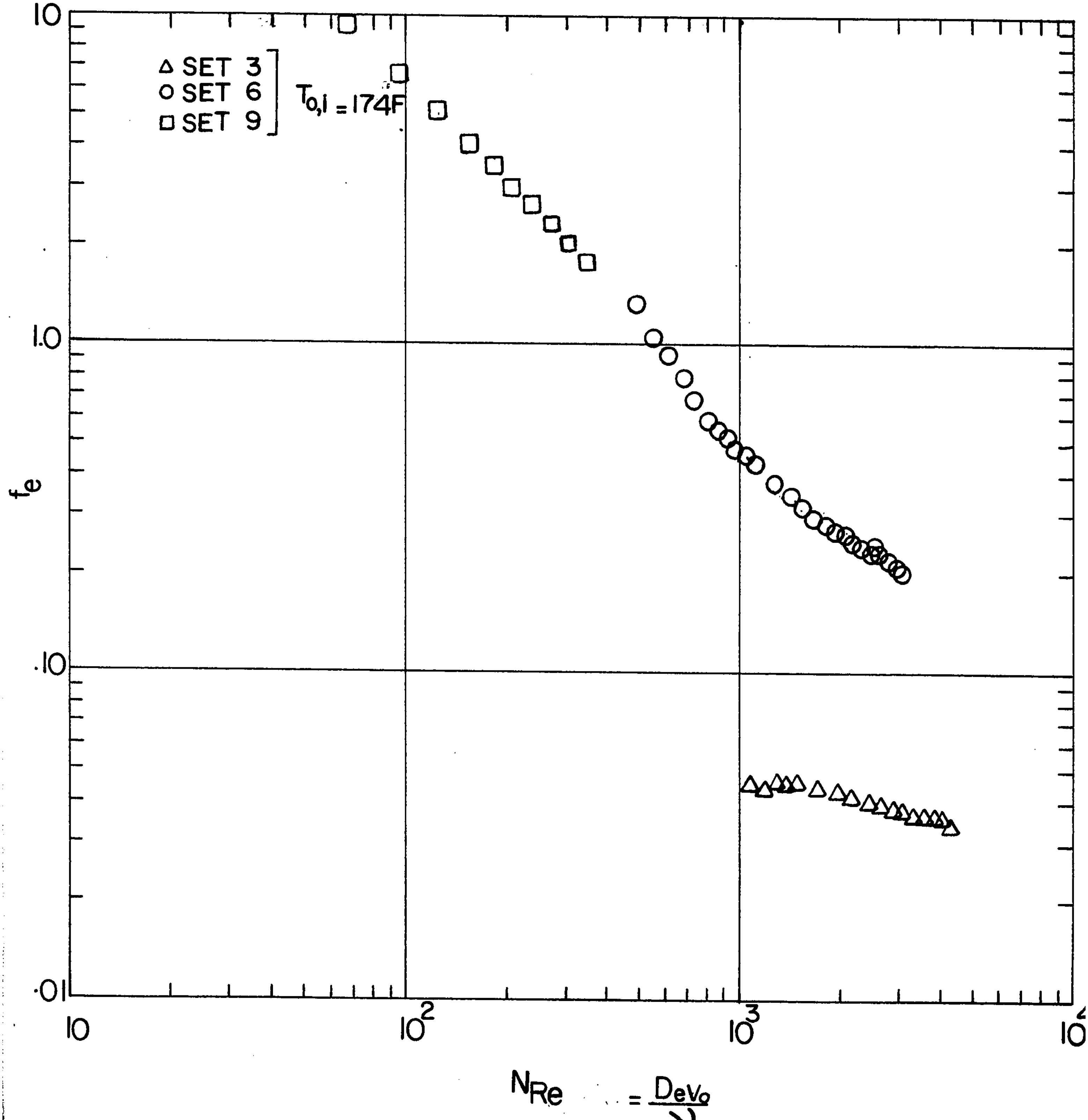


FIG. 37 FRICTION FACTOR RESULTS BASED ON EQUIVALENT DIAMETER

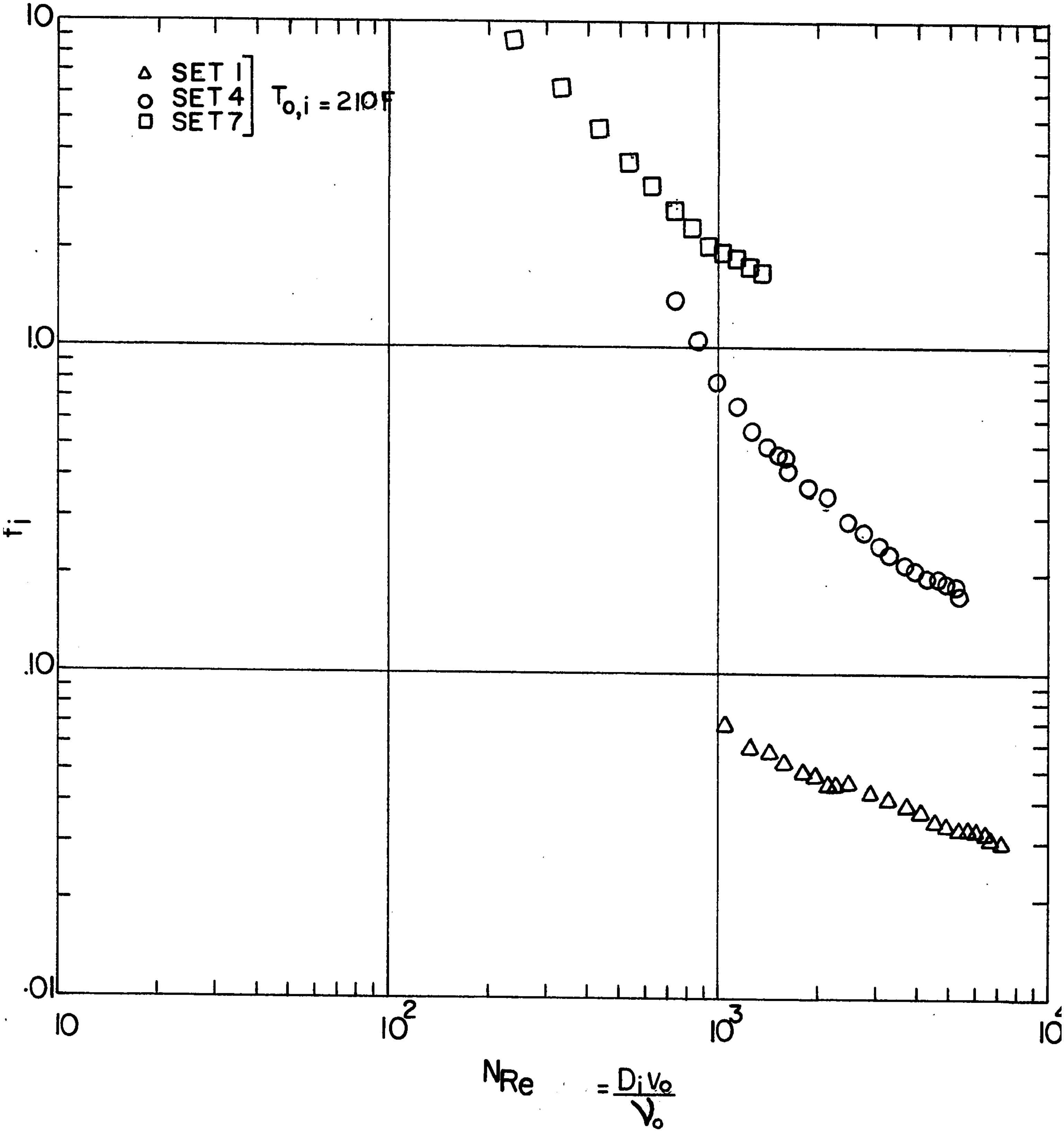


FIG.38 FRICTION FACTOR RESULTS BASED ON INSIDE DIAMETER

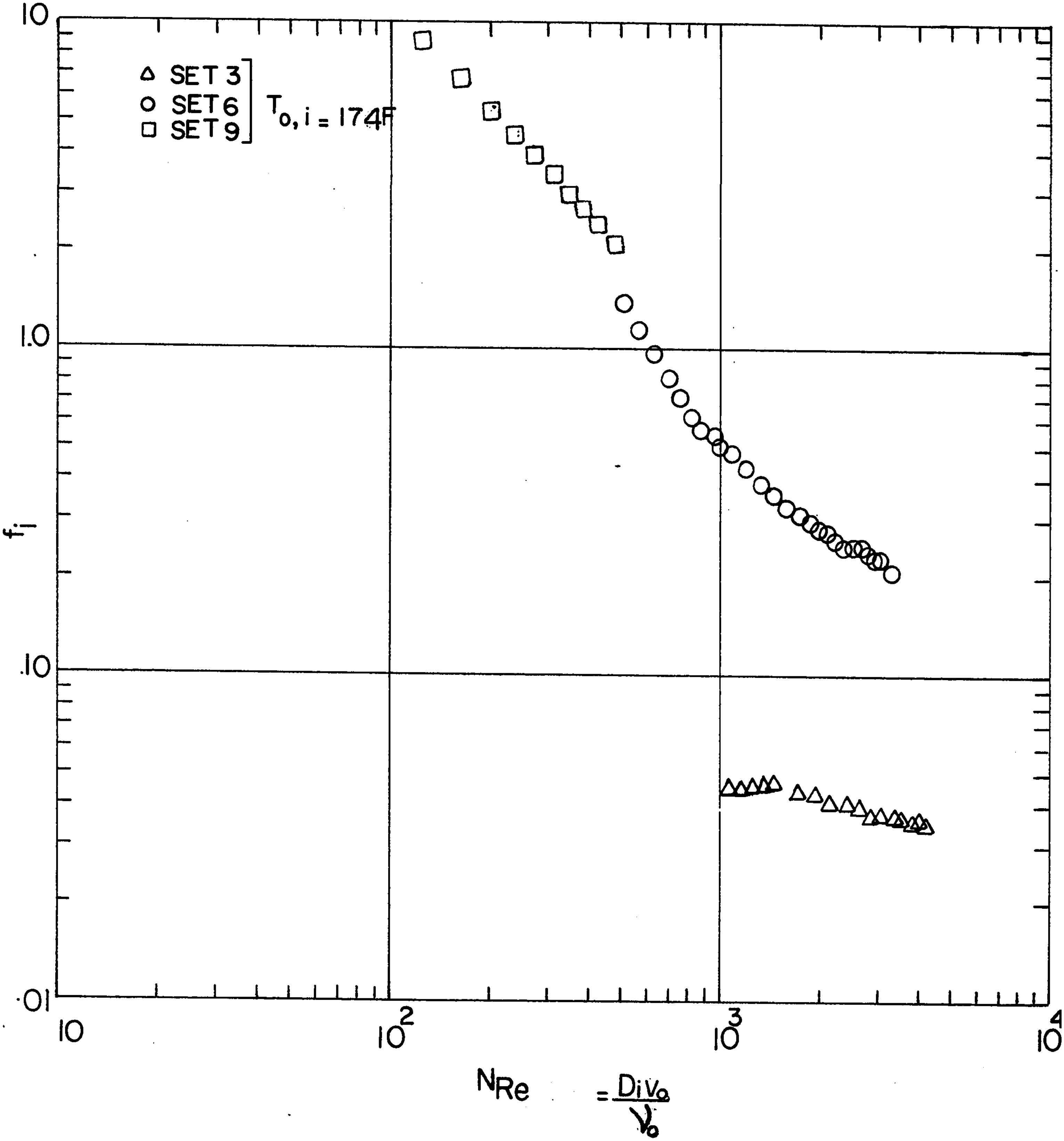


FIG. 39 FRICTION FACTOR RESULTS BASED ON INSIDE DIAMETER

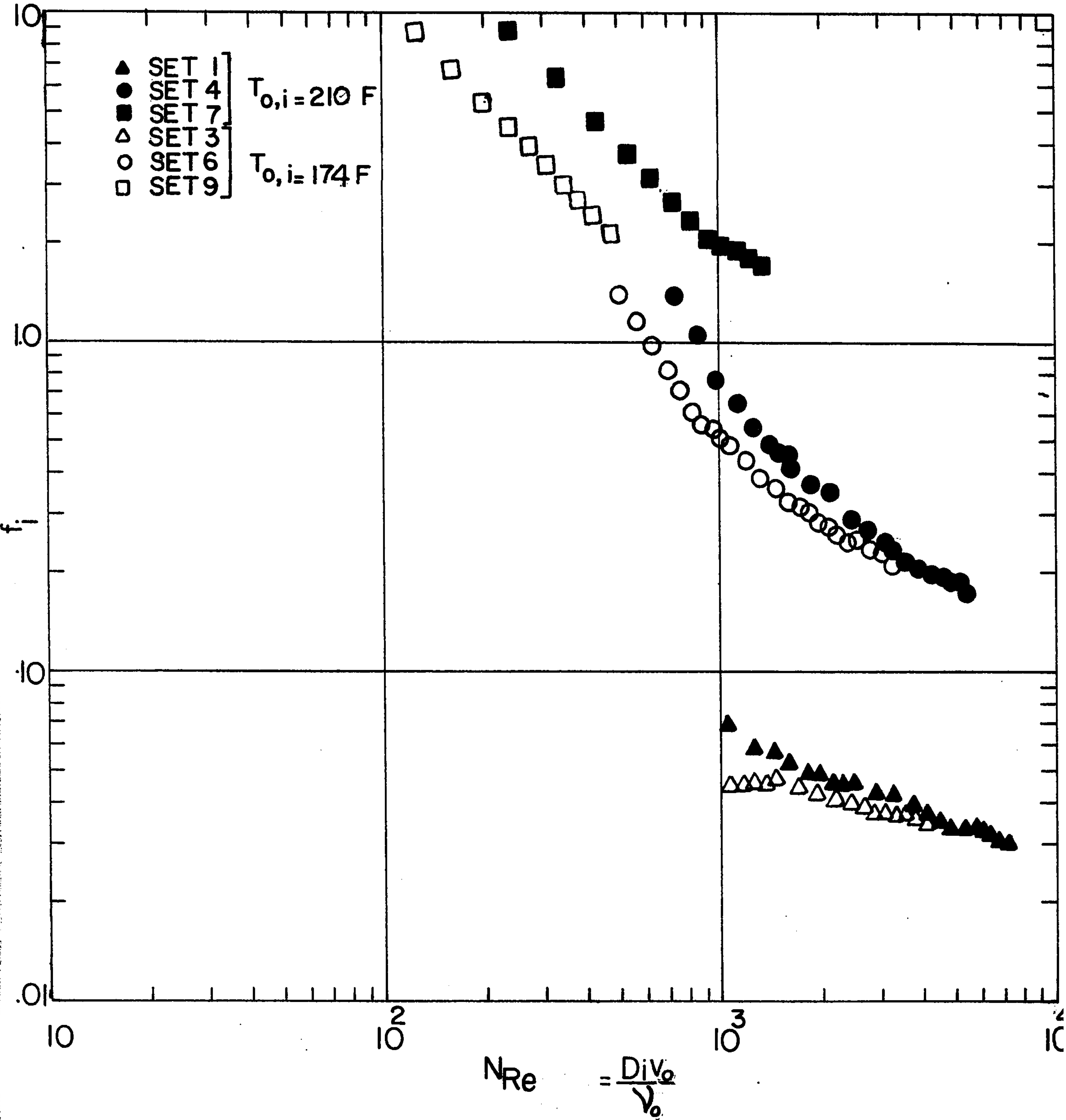


FIG. 40 EFFECT OF OIL INLET TEMPERATURE ON FRICTION FACTOR

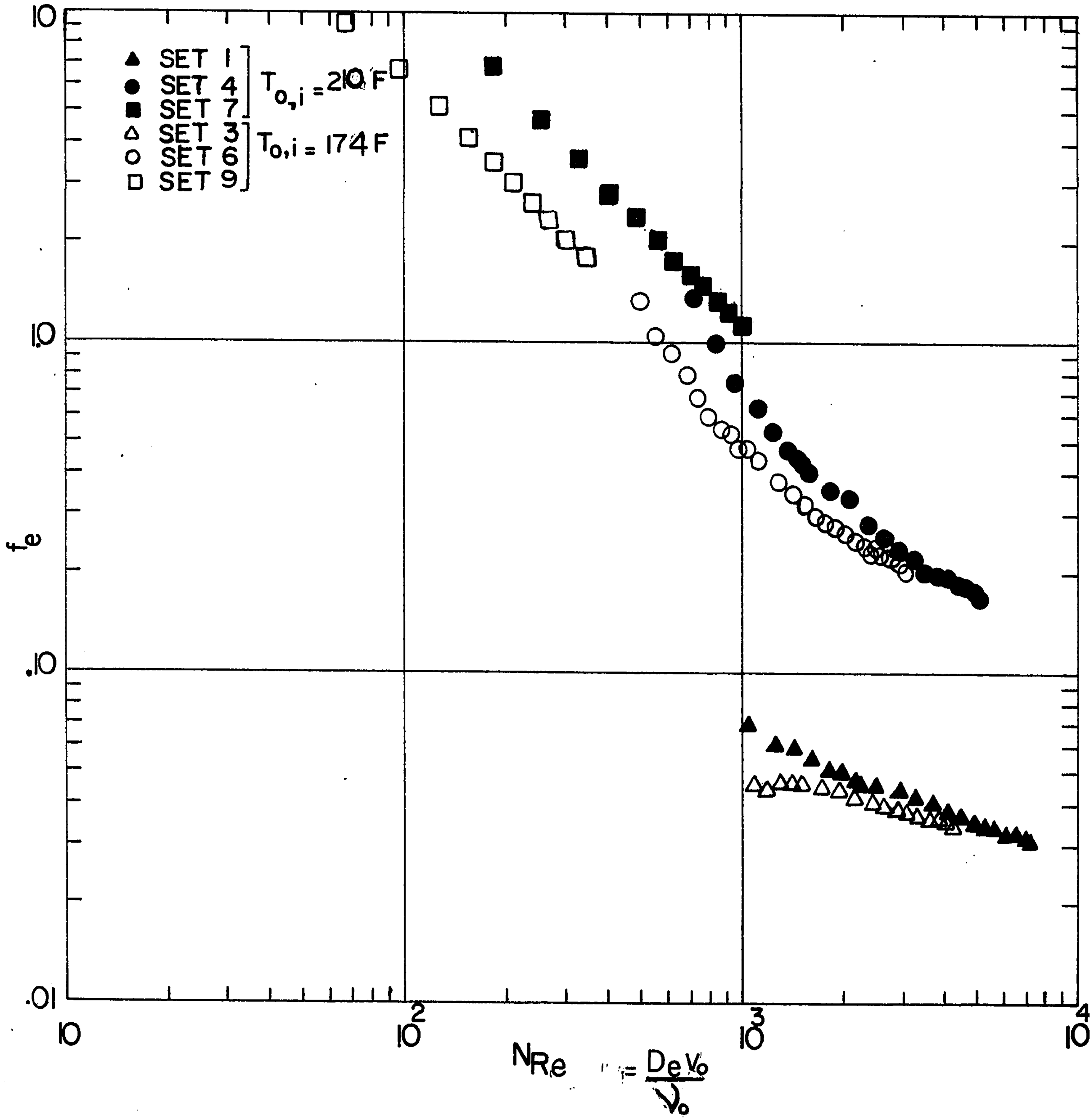


FIG. 41 EFFECT OF OIL INLET TEMPERATURE ON FRICTION FACTOR

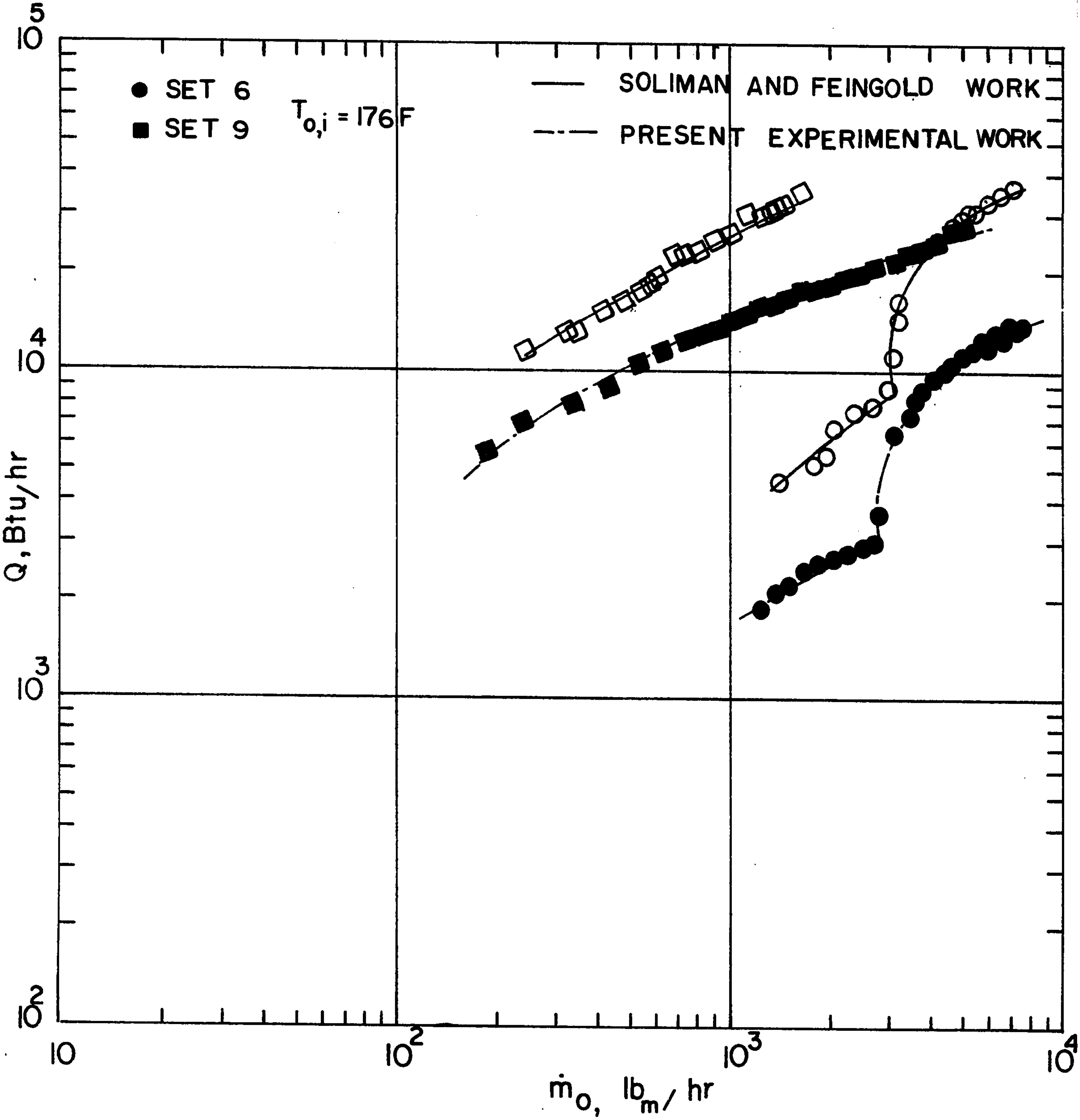


FIG. 42 EFFECT OF COOLING LENGTH ON HEAT TRANSFER

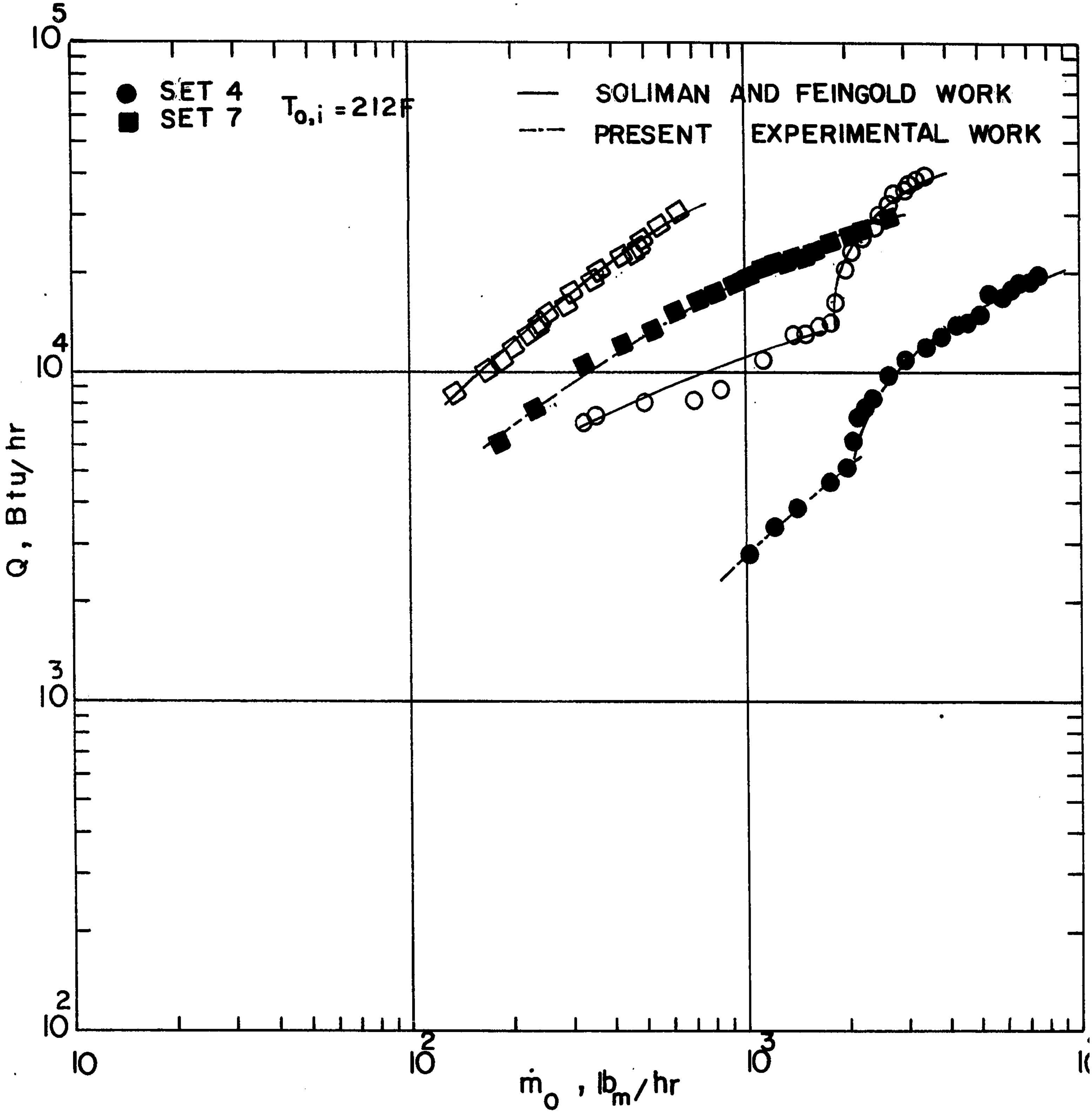


FIG. 43 EFFECT OF COOLING LENGTH ON HEAT TRANSFER

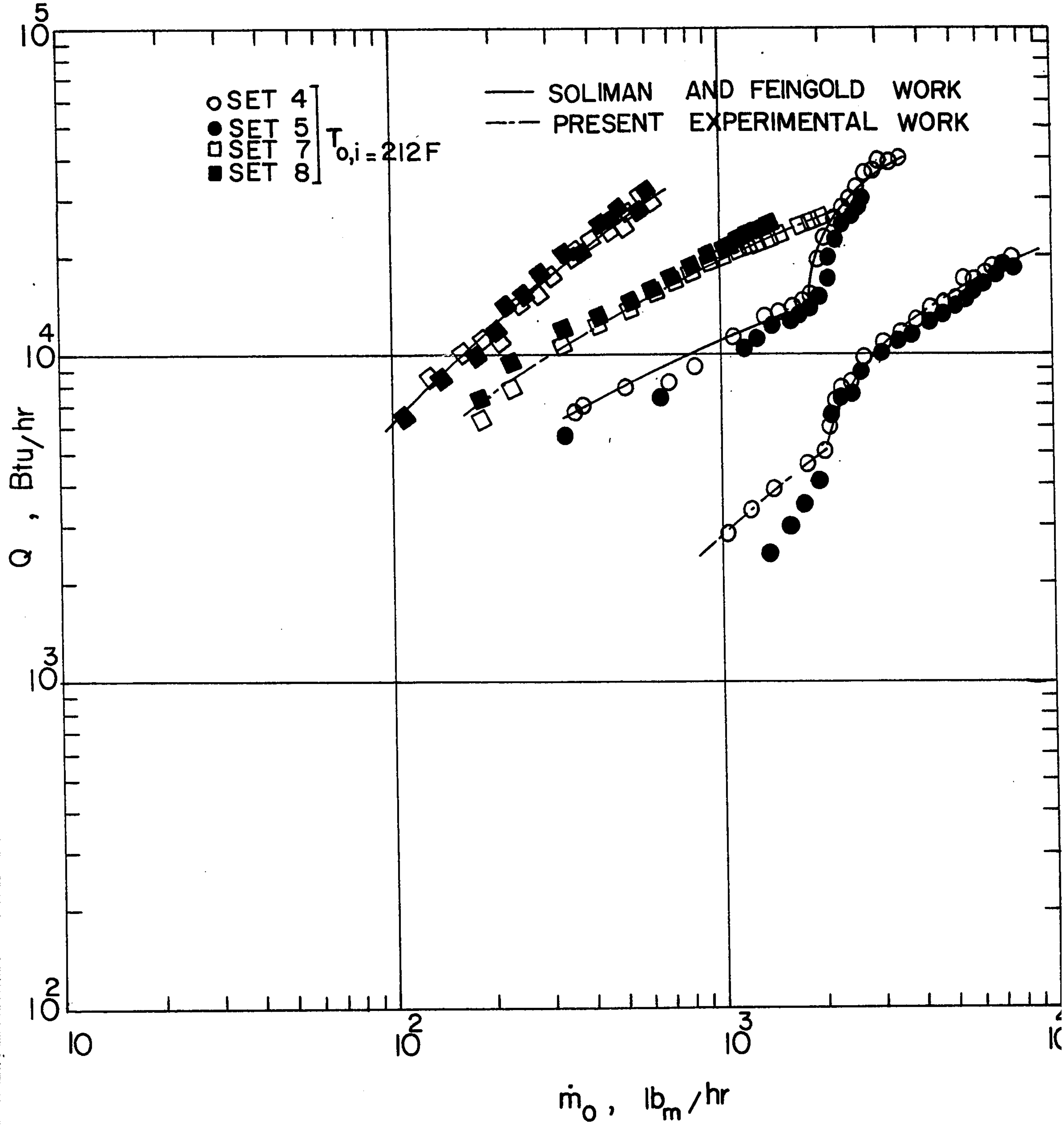


FIG. 44 EFFECT OF COOLING WATER FLOW RATE ON HEAT TRANSFER

4. SUMMARY AND CONCLUSIONS

Heat transfer and isothermal pressure drop measurements were taken for three test tubes, namely quintuplex and two single-finned tubes, almost under the same operating conditions. Nine sets were collected for three test sections, three sets for each section. All tubes were tested at two different oil inlet temperatures with the same mass flow rate of water and also at two different mass flow rates of water at the same oil inlet temperature. Heat transfer performance of quintuplex tube was found superior to that of other tubes, but at the expense of higher pumping power.

The shorter tube having permitted a wide range of flow rates, produced interesting results. First of all, when the present shorter quintuplex tube is compared with previous longer single-finned tube, the heat transfer enhancement is still observed at low mass flow rates of oil. But then, at higher rates the advantage begins to disappear. This is because of the fact that, in the case of a single-finned tube, a jump in the value of Q occurs at higher mass flow rate of oil, while no such behaviour is observed in the quintuplex tube. Secondly, it confirmed previous workers findings regarding the enhancement of heat transfer in the quintuplex as compared with the single-finned tube.

So, quintuplex tubes are recommended for applications where the size of heat transfer equipment is the major concern. On the other hand, for applications where a certain amount of heat is to be transferred with minimum pumping power expenditure irrespective of the flow rate of the circulated fluid, the single-finned tube proved to be the best choice.

This study proves experimentally the validity of the concept of equivalent length as used by Soliman and Feingold, but its applicability is limited to relatively low flow rates.

5. REFERENCES

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F I G U R E S

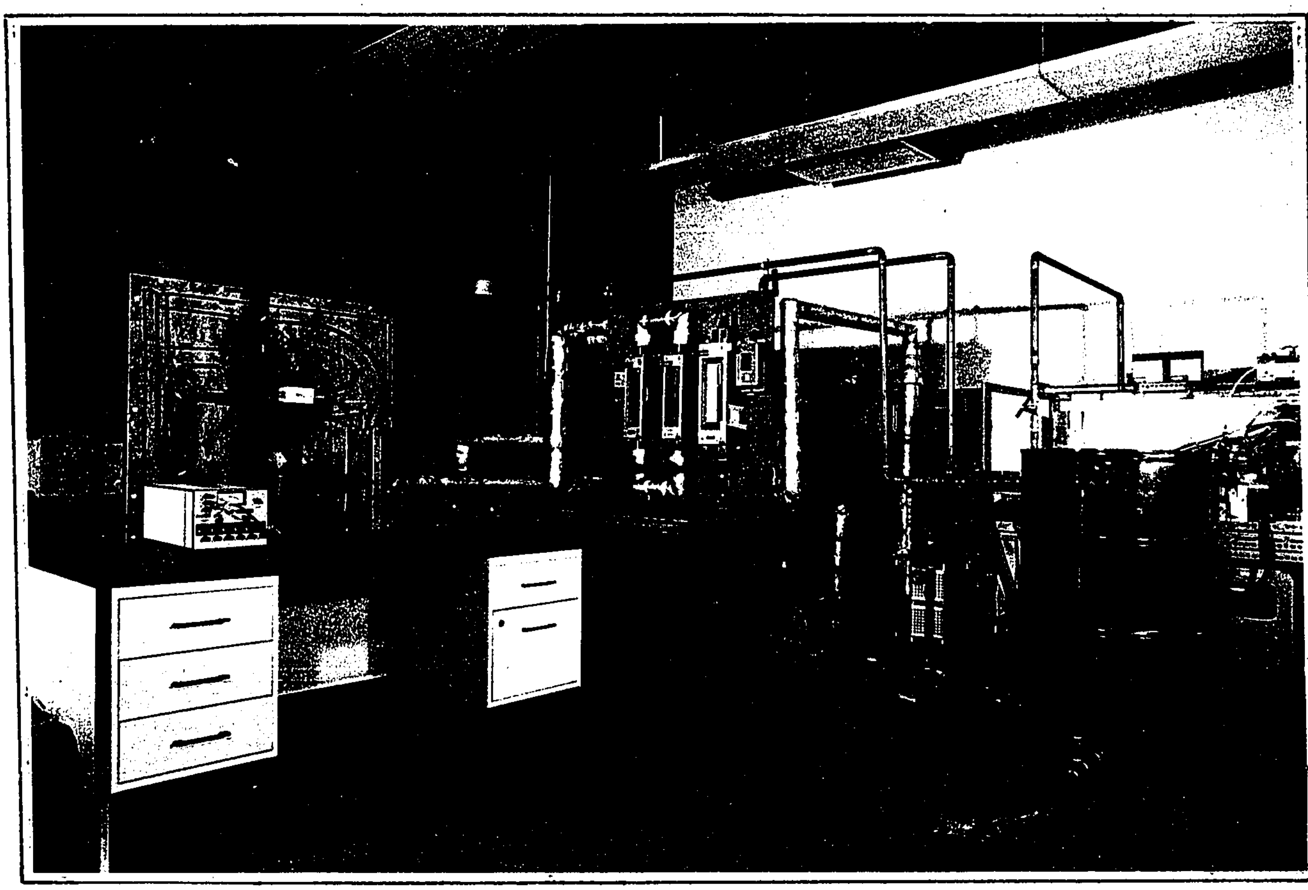

FIG. 45 GENERAL VIEW OF THE APPARATUS

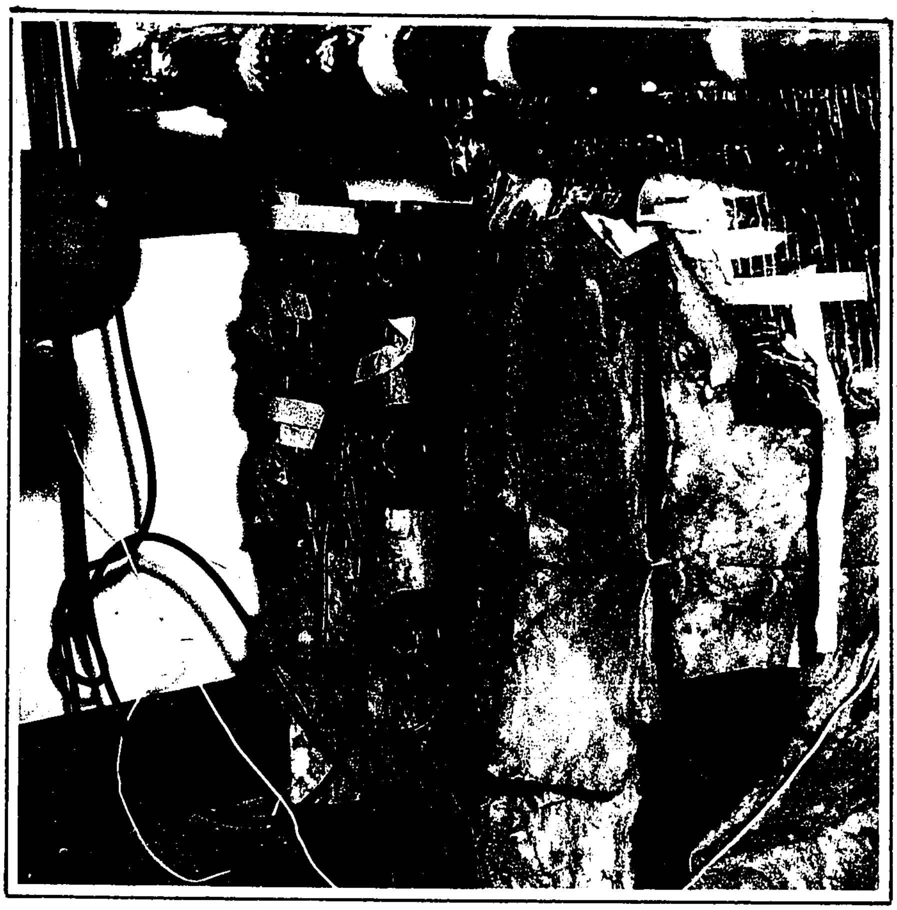

FIG. 46 PHOTOGRAPHIC VIEW OF THE HEATERS

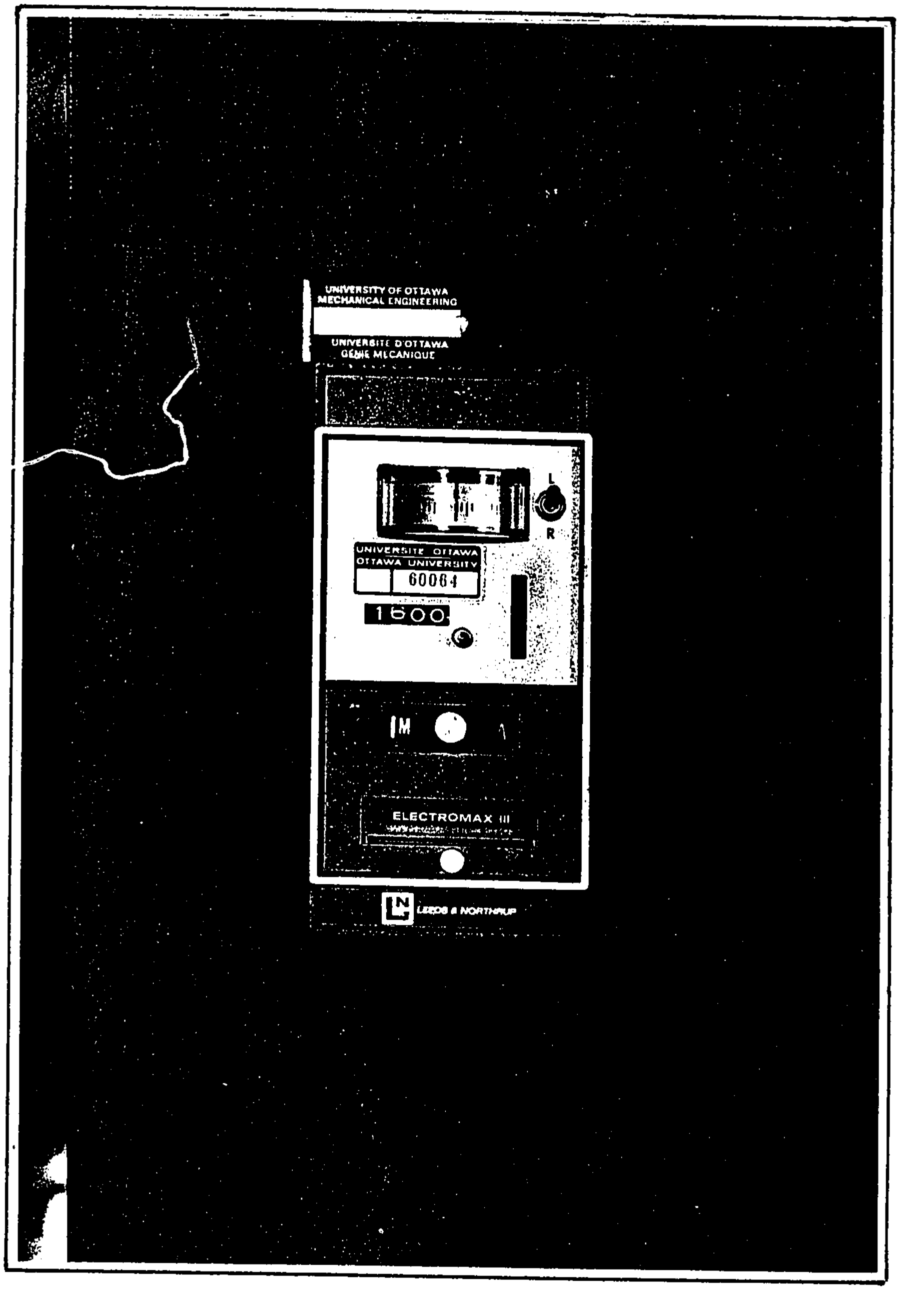

FIG. 47 PHOTOGRAPHIC VIEW OF THE ELECTROMAX TEMPERATURE CONTROLLER

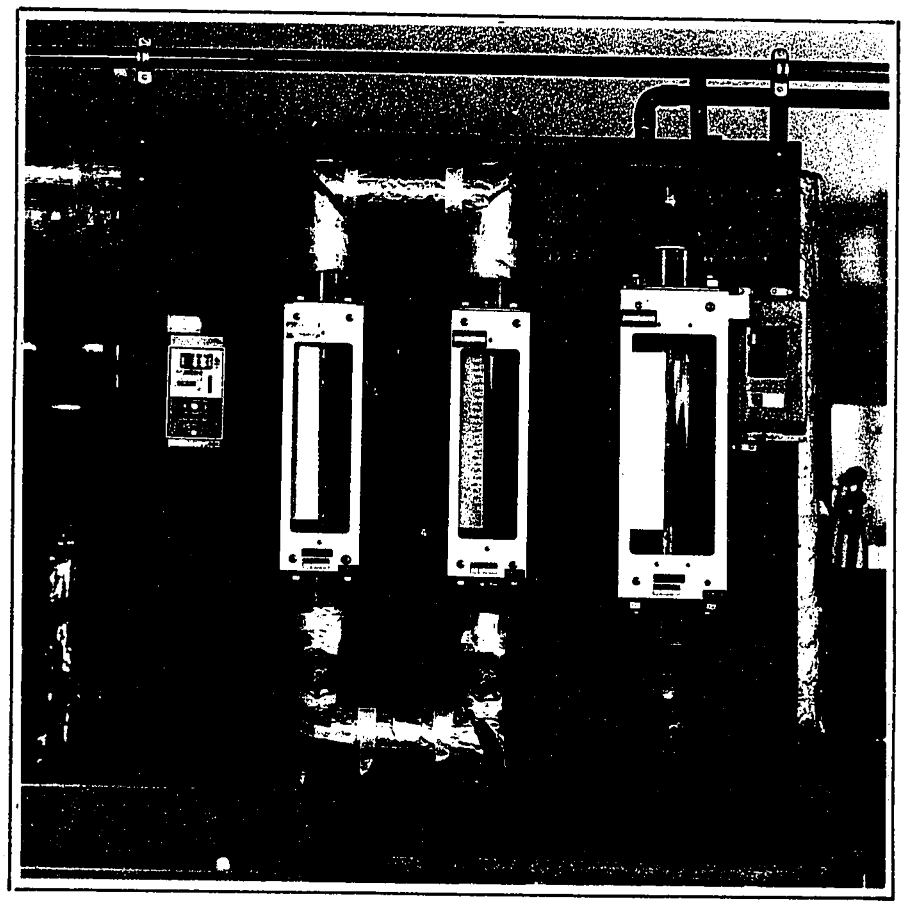

FIG. 48 PHOTOGRAPHIC VIEW OF OIL AND WATER FLOWMETERS

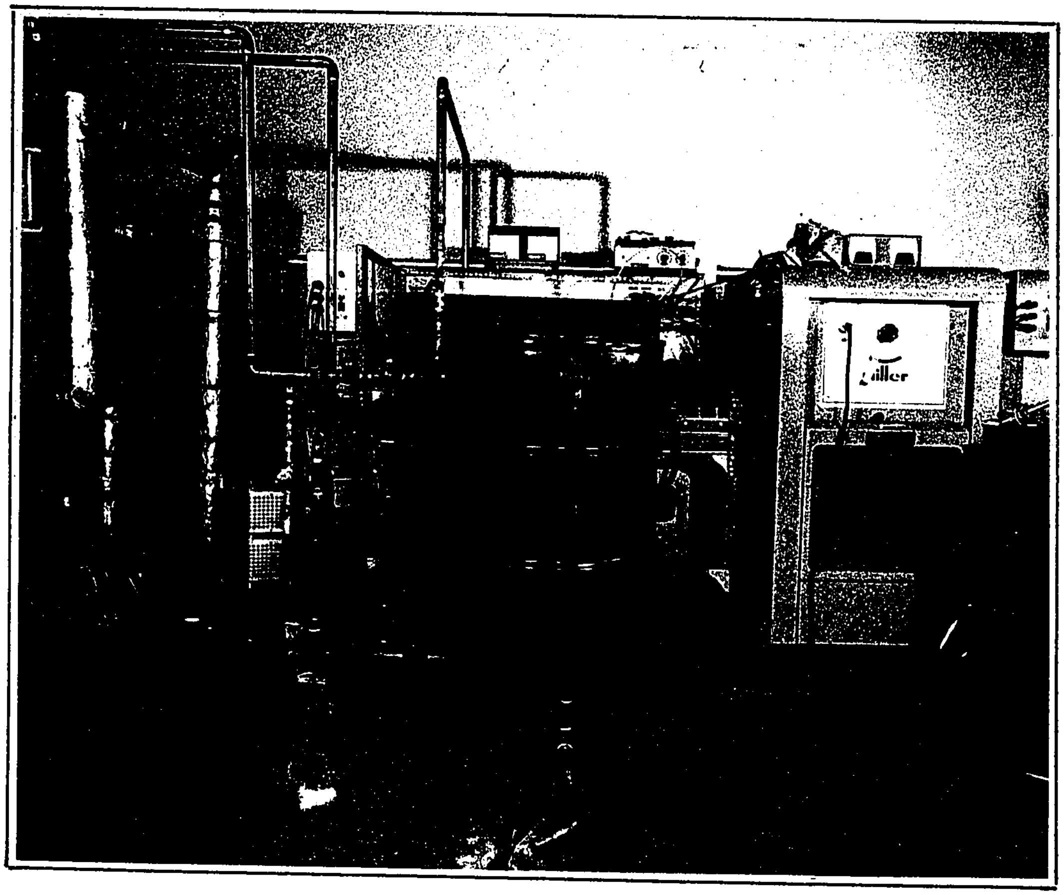

FIG. 49 PHOTOGRAPHIC VIEW OF COOLING WATER SYSTEM

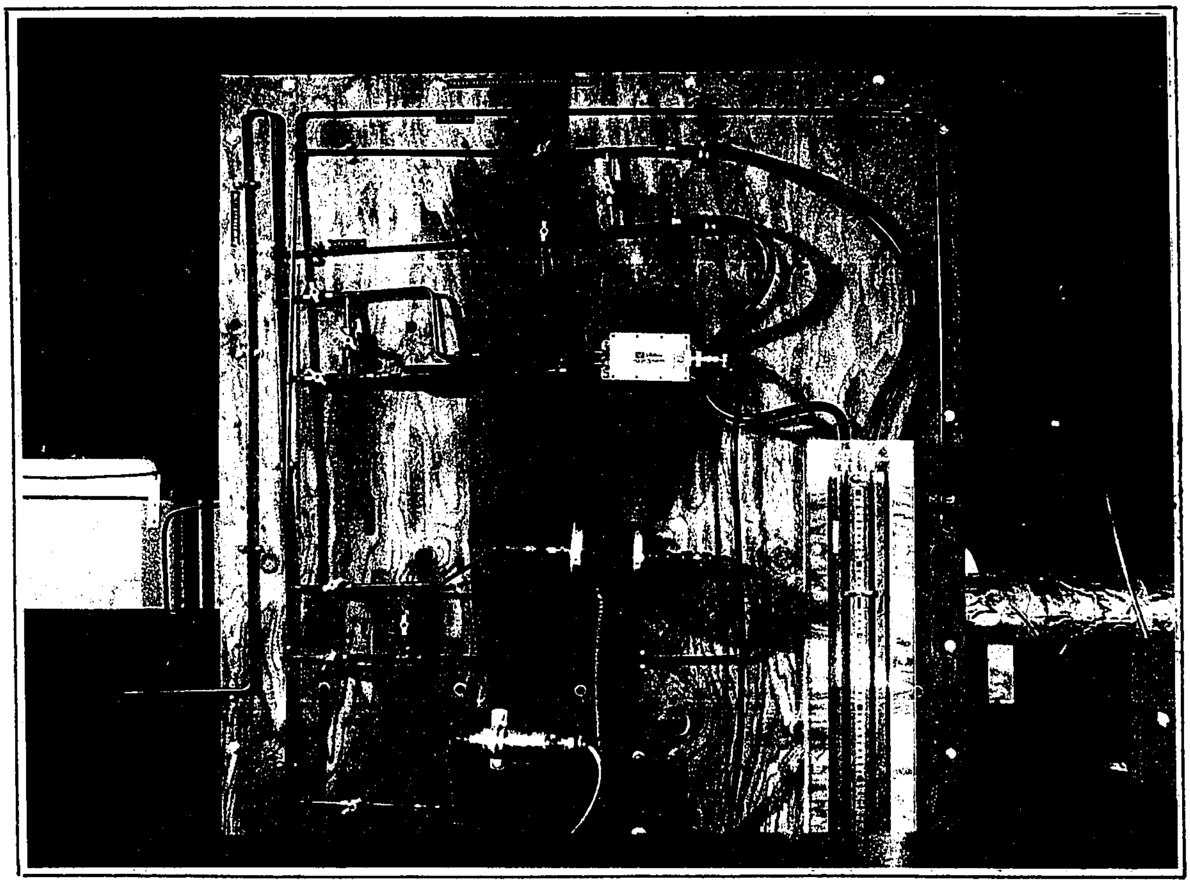

FIG.50 PHOTOGRAPHIC VIEW OF PRESSURE DROP SYSTEM

7. APPENDIX

The equations which have been used in previous sections are summarized below.

The cross-sectional flow area of the tubes was calculated from the relation

$$A_{xs} = \frac{\pi}{4} D_o^2 - \frac{W}{\gamma L} \tag{1}$$

The equivalent diameter was defined as

$$D_e = \sqrt{\frac{4\ A_{xs}}{\pi}} \tag{2}$$

The calibration equation for digital type temperature measuring recorder used in this investigation is

$$T = 1.229t - 26.99 \tag{3}$$

Heat rate lost by oil is

$$Q_o = \dot{m}_o\ C_{p_o}\ (T_{o,i} - T_{o,o}) \tag{4}$$

Heat rate gained by water is

$$Q_w = \dot{m}_w\ C_{p_w}\ (T_{w,o} - T_{w,i}) \tag{5}$$

Log-mean temperature difference is

$$\Delta T_m = \frac{(T_{o,i} - T_{w,i}) - (T_{o,o} - T_{w,o})}{\ln\left(\frac{T_{o,i} - T_{w,i}}{T_{o,o} - T_{w,o}}\right)} \qquad (6)$$

The minimum requirement for the entrance length of the tube was calculated from

$$\frac{x}{D_h} \geqslant \frac{10^5}{N_{Re}} \qquad (7)$$

Heat balance error was defined as

$$e = \frac{Q_w - Q_o}{Q_o} \qquad (8)$$

The values of pumping power for all the test runs were calculated by using the following relation

$$P = \frac{144\ \dot{m}_o \Delta P}{60\ \rho_o} \quad \text{lb ft/min} \qquad (9)$$

Friction factors and Reynolds number were calculated from

$$f = \frac{D \Delta P\ g_c}{2L\ \rho_o v_o^2} \qquad (10)$$

$$N_{Re} = \frac{D_{vo}}{\nu_o} \qquad (11)$$