AUGMENTATION OF SINGLE PHASE HEATING AND SUBCOOLED

BOILING BY INTERNALLY FINNED TUBES

by

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CHAPTER I

INTRODUCTION

The demand for augmenting the heat transfer of heat exchangers has required a large research effort, especially in those fields concerned with nuclear and conventional power plants, space ships, refrigeration, air conditionaing, and other modern technological equipment. The direct benefit of augmenting the heat transfer in heat exchangers is the reduction of their sizes, and thereby the reduction of their costs. In a report published in 1975 by Bergles [4], it was estimated that the investment in heat exchanger equipment in the U.S. was then approaching one billion dollars yearly. A 10 to 20% reduction in costs, due to size reduction, could result in savings in the 100 million dollar range.

The techniques that have been used to improve the heat transfer in the past several years have been finned surfaces, turbulence promotors, vibration of the heat transfer surface or of the fluid near the surface, electrostatic fields, and fluid additives.

Setting longitudinal fins on surfaces is a widely used technique. Surface promoters, which can be of either the overlapped or integral type, are fitted on the surface with the purpose of disturbing the fluid layer close to the wall. Extensive knowledge has been accumulated so far on the effects of the variously shaped surface promoters such as transverse ribs, shallow grooves, screw-threads, sand grain roughness, wound wires, and others.

Integral ribs, which are the most reliable in regard to endurance and effectiveness at present are also produced inside tubes by means of special techniques such as roll forming and electro and chemical erosion. Displaced promoters are suitably shaped bodies inserted within a channel in order to alternately increase and diminish the fluid speed. They allow only moderate performances to be achieved. Vortex generators chiefly consist of twisted tapes placed within a channel with the aim of generating a spiral flow in the fluid, thus increasing, at the same flow rate, the fluid speed, and furthermore establishing a centrifugal convective effect. Twisted tapes have been extensively used in augmenting forced boiling heat transfer and were found to delay burnout and improve post-burnout heat transfer. They were also found to improve the single phase heat transfer.

The main concern of the present study was to investigate the augmentation of single phase flow and subcooled boiling heat transfer by internally finned tubes.

CHAPTER II

LITERATURE SURVEY

In 1970, Bergles and Webb [6] published a bibliography on augmentation of convective heat and mass transfer. Their bibliography included 472 listings of the world literature on augmentation. In 1979 a similar bibliography was published by Bergles et al. [7]. Over 1900 references were cited in this bibliography.

Due to the fact that the scope of previous studies on augmentation is too extensive to be fully reviewed in the present work, the literature survey is limited to single phase flow and two phase flow heat transfer augmentation by internally finned tubes which are relevant to the present study.

Internally Finned Tubes

In 1964, Bernstein et al. [2] studied two-phase water flow in six different tube shapes. They reported that those tubes with some device to turbulate or baffle the fluid displayed a remarkable improvement in heat transfer over straight round tubes.

In 1969 Kidd [14], studied the heat transfer and pressure drop characteristics of gas flow inside spirally corrugated tubes. His research showed that tubes of this geometry were found to be very effective in enhancing the heat transfer. On an equal pumping power basis, for example, a tube with a ratio of the spacing of a corrugation to its depth of 22 had a heat transfer coefficient 22 per cent greater than did a smooth tube. Friction factors, measured with a diabatic air flow, were found to be up to 1.7 times that for smooth tubes.

In the same year Lipets [20] showed that internal longitudinal fins appreciably increased the heat transfer coefficients over those of smooth tubes. Sauer et al. [29] published their study on heat transfer coefficients and friction factors for longitudinally grooved tubes. The purpose of their study was to investigate longitudinal grooves of various depths on the convective heat transfer coefficient and on the friction factor. Their conclusion was that the roughened sections yielded definite gains in heat transfer over the smooth sections, but with much greater increase in the friction factor. Kuntysh et al. [17] reported that the heat transfer coefficients of staggered finned bundles are higher than those of in-line bundles with the same type of fins.

Homman [12] discussed the various expressions obtained by different investigators for predicting the boiling heat transfer with smooth and internally finned tubes using freons 11, 12, and 22. He suggested a new correlation which he claimed that it can be used for obtaining highly accurate heat transfer coefficients for specific sets of conditions. Danilova et al. [11] found that the effect of the heat flux and boiling temperature on the heat transfer differs with each of freons 12 and 22 they tested, and also that the oil contamination had an adverse effect on the transfer of heat. Ornatskiy et al. [26] presented a method for calculating the temperature of a pipe with internal fins trapezoidal or rectangular. This method also helped in rationally selecting the geometry of the finned surfaces.

In a study by Bergles et al. [3] it was reported that short spiraled fins produced the greatest increase in heat transfer above the smooth tube values, over 20% at equal flow conditions, and up to 170% at equal pumping power. Watkinson et al. [34, 35] did a series of studies on turbulent heat transfer and pressure drop in internally finned tubes. Their study [34] reported that, based on inside tube diameter and nominal area, heat transfer was enhanced over smooth tube values up to 170% at a constant Reynolds number, and up to 80% at a constant pumping power. Watkinson et al. [35] reported the results of heat transfer and pressure drops of turbulent air flow inside internally finned tubes. Their report indicates that heat transfer enhancement over smooth tube values, at a Reynolds number equal to 50,000, based on the inside diameter, varied from 17 to 95%, depending on the geometry of the finned tube. At constant pumping power, for Re = 50,000, the heat transfer performance for air varied from -11% to +47% of smooth tube values.

Soliman and Feingold [30, 31, 32] did several studies on heat transfer and pressure drops of internally finned tubes. Soliman and Feingold [32] studied the heat transfer, pressure drop and performance of a quintuplex finned tube, single finned and smooth tubes. Lubricating oil was used as the test fluid. Results showed that with the use of quintuplex tubes, remarkable compact heat transfer equipment could be produced, in comparison with those possible using smooth tubes, at the expense of increased pumping power. Soliman and Feingold [31] analytically investigated the fully developed laminar flow in internally finned tubes, with fin shapes approximating real fin configurations as closely as possible. Velocity and friction factor functions were determined for a wide range of fin heights, number, and thicknesses. Soliman [30] also analytically investigated the effect of the fins material on laminar heat transfer characteristics of internally finned tubes. He found for any tube geometry, heat transfer characteristics are influenced by a single paramter K, defined as ($\beta K_s/K_f$, K_f = thermal conductivity of the fluid, K_s = thermal conductivity of the fin material, β = half the angle subtended by one fin).

Masliyah and Nandakumar [22,23,24] analytically studied the heat transfer characteristics for laminar forced convection, fully developed flow in an internally finned circular tube, with uniform axial heat flux and with peripherally uniform temperature, using finite element method. The Nusselt number based on inside tube diameter was higher than that for a smooth tube. They reported that, for maximum heat transfer, there exists an optimum fin number for a given fin configuration.

Hu and Chang [13] did an analytical study on the heat transfer of fully developed laminar flow in internally finned tubes. They concluded that if there is no heat generation in the fluid, the highest Nusselt number is obtained with a tube with 22 fins extended to about 80% of the tube's radius. The heat transfer was almost 20 times that for the finless tube. When there is heat generation at a sufficiently high rate, the number of fins was reduced from 22 to 16 in order to obtain the highest Nusselt number.

Van Rooyen and Kröger [33] investigated the heat transfer and pressure drop characteristics for laminar flow of oil in smooth and internally finned tubes with twisted tape inserts. The heat transfer coefficients in finned tubes with twisted tape inserts were found to be as much as four times the smooth tube values, when cooling at the same pumping power, and three times in the case of heating.

Michenko and Shvartsman [25] presented the results of an experimental investigation of the optimum geometry of internally helical finned steam generator tubes. They demonstrated the high thermal efficiency of the finned tubes in sub and super critical pressure steam generators. The study showed that the placing of helical fins with optimum geometries in steam generating tubes allowed the designer to reduce the mass flow rate by approximately a factor of two, with corresponding reduction in pumping losses.

Kubanek and Miletti [16] conducted heat transfer and pressure drop studies on three spiral finned tubes with two phase flow of R-22 under evaporating conditions. They compared the results with the results of smooth tubes with and without a star shaped insert. Heat transfer enhancements for internally finned tubes ranged from 30 to 760% over those for the smooth tubes and increased with mass velocity. Pressure drop increases ranged from 10 to 290%. It was concluded that internally finned tubes would be beneficial in the design of compact direct expansion water chillers and other equipment in which the refrigerant is evaporated inside the tube to cool a fluid on the outside.

Carnavos [8, 9, 10] has published several papers on heat transfer coefficients and pressure drop of internally finned tubes in single phase flow. In reference, [10], he reported on the air cooling by internally finned tubes and concluded that the heat transfer performance of longitudinal inner fin tubes exhibited the same slope as smooth tube performance curves. Heat transfer enhancement was roughly equal to the heat transfer area increase. The increases in friction factors were in the range of 80 to 100% of the square of the heat transfer area increase. Carnavos [8] studied the heat transfer performance of single phase turbulent flow inside internally finned tubes. He developed heat transfer and fanning friction factors correlations, which correlated air, water, and ethylene glycol/water data to within ± 10%. His correlations will be discussed in a later chapter. Carnavos [9] also investigated the cooling of air in turbulent flow inside 21 internally finned tubes having integral internal spiral and longitudinal fins. The performance of the tubes was compared to the performance of smooth tubes at constant pumping power. The finned tubes were found to perform better by factors of 1.2 - 2. Correlating equations were also presented for heat transfer and friction factors that describe the data to within ± 6 and ± 7%, respectively.

Marner and Bergles [21] experimentally studied the augmentation of heat transfer inside horizontal tubes by means of twisted-tape inserts, static mixer inserts, and internally finned tubes under laminar flow conditions. In this study isothermal pressure drops and local heat transfer coefficients were measured with water and ethylene glycol as test fluids. Increases in pressure drop, especially for the static-mixer assemblies, were observed, along with increases in the heat transfer for all seven augmented tubes tested. Based on a constant pumping power performance analysis, the internally finned tube used in the large-scale tests was reported to look especially promising, for heating as well as for cooling.

Patankar et al. [27] analysed the turbulent flow and heat transfer characteristics of circular tubes and annuli with longitudinal internal fins. The analysis was based on the differential equations for momentum and energy conservation in the flowing fluid supplemented by a turbulence model having an adjustable constant. Average Nusselt numbers and friction factors were evaluated for a range of Reynolds numbers, fin height and number of fins. The results were found to be quite insensitive both to fin height and to the number of fins.

CHAPTER III

EXPERIMENTAL INVESTIGATION

The objectives of this investigation were:

1) To collect different sets of experimental heat transfer and pressure drop data for single phase heating and sub-cooled boiling inside vertical smooth and internally finned tubes.

2) To compare the heat transfer and pressure drop of the internally finned tubes with the smooth tube and furthermore, to develop correlations for predicting the heat transfer and pressure drop for single phase heating for this augmentation technique. R-113 was used as test fluid.

3.1 TEST FACILITY

Figure 3.1 shows a schematic diagram of the R-113 flow loop. It included the following main components:

i) Gear pump for circulating the liquid.

ii) Precooler

iii) Observation section located at inlet to test section.

iv) Four, parralell, vertically mounted, electrically heated tubes. One tube was smooth, the second had straight fins on the inside, the third and fourth tubes had spiral fins.

v) Cooler-condenser.

vi) Liquid receiver.

vii) Liquid flow meters.

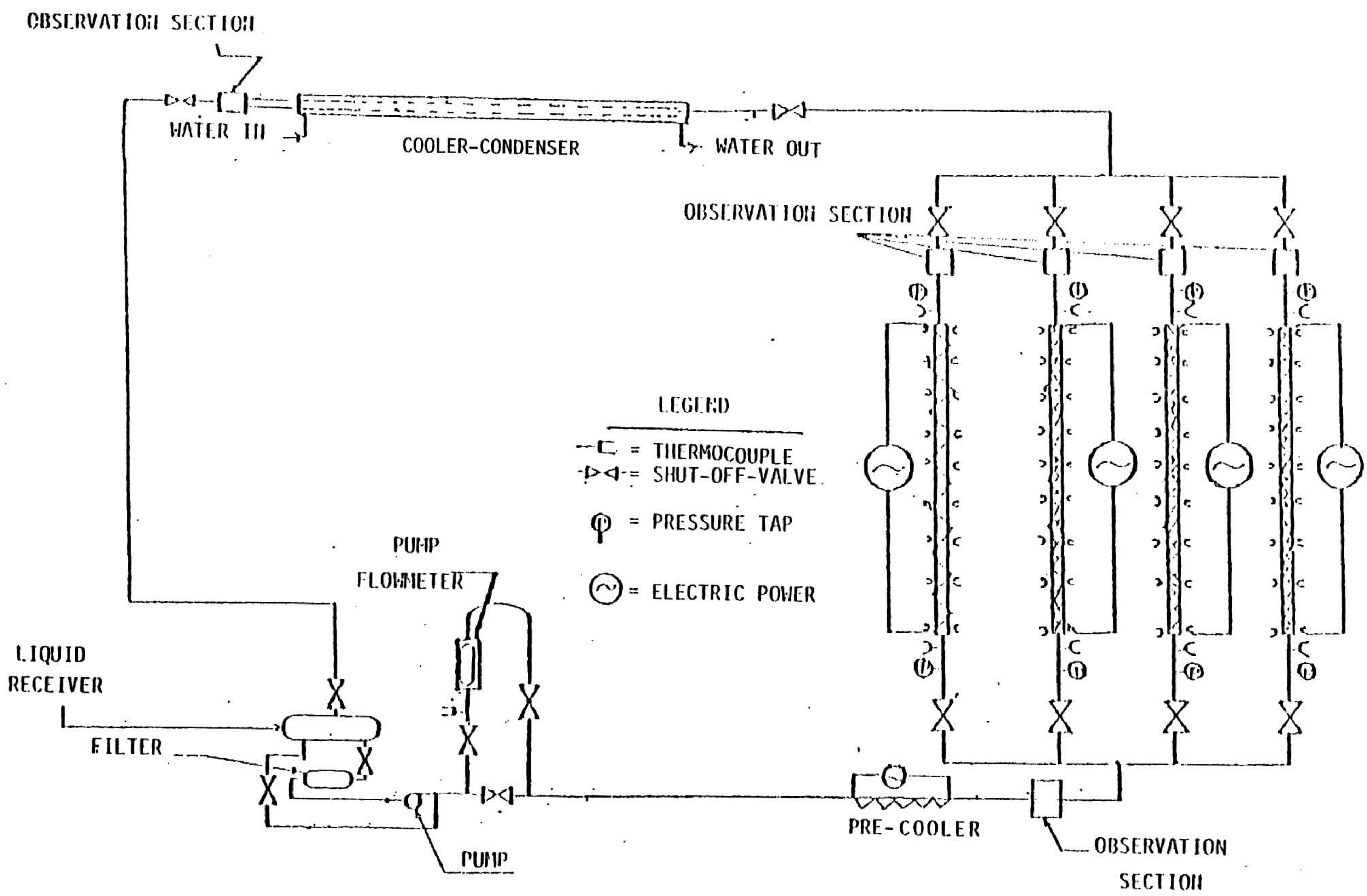

Fig. 3-1 A Schematic Diagram for the R-113 Flow Circuit.

R-113 was pumped from the liquid receiver to the test section through the precooler. The test sections were electrically heated, well instrumented to measure the outside wall temperatures, at different locations along the test section. The flow meter was used to measure the total flow rate of the R-113 entering the test section. All temperature were measured by copper-constantan thermocouples of type RIP-24. All wall thermocouples were connected to a data acquisition system which gave a printout of the readings along the test section. A Honeywell Electronic-18, multichannel potentiometer was also used to measure separately the inlet and exit temperatures of R-113 to each test section.

The pressure drop across the test section was measured by a pressure transducer along with a digital voltmeter. A photographic view of the entire tests facility is given in Fig. 3.2.

3.2 TEST SECTION CONSTRUCTION AND INSTRUMENTATION

The following tubes were tested: a smooth tube, tube 1, a straight finned tube, tube 2, and two spiral finned tubes with different helix angles and outside diameters, tubes 3 and 4. The geometric parameters of these tubes are given in Table 3.1. A photographic view of all the tubes tested is given in Fig. 3.3.

The four tubes were mounted vertically, in parallel, in the test facility as shown in Fig. 3.1. Each test tube was instrumented to measure the wall surface temperatures at equidistant axial locations. At each location two thermocouples were silver-brazed to the surface opposite each other. The distance between the two thermocouple stations at the top and the bottom was half the distance between

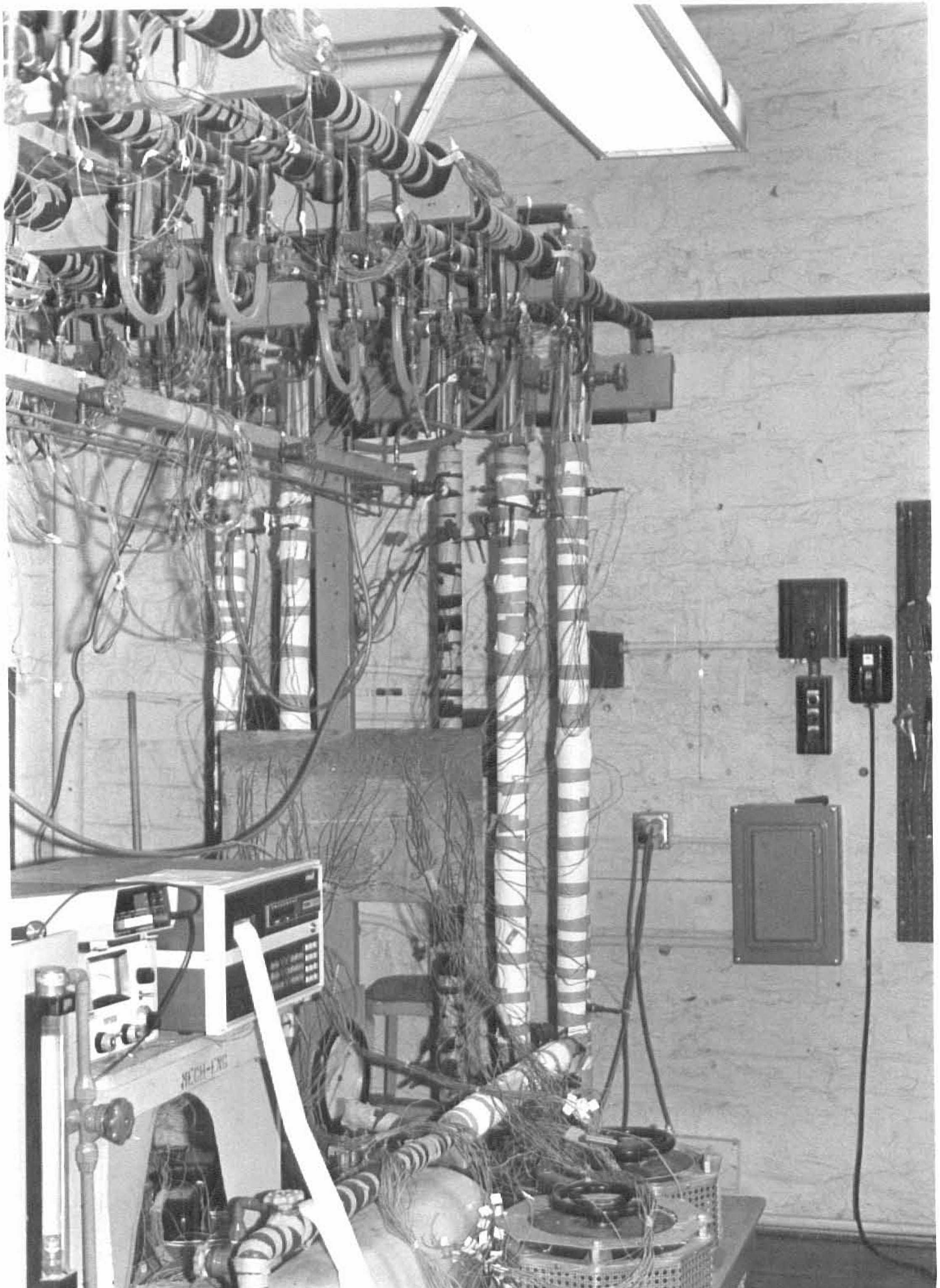

Fig. 3-2 A Photographic view of the Test Facility

TABLE 3.1. Geometric Parameters of the Tubes Under Study

(All Values in Cm.)

Tube No.	1	2	3	4
Type	Smooth	Straight Finmed	Spiral Finned	Spiral Finned
Material	Cu	Cu	Cu	Cu
Length	133.5	135.89	134.62	133.5
No. of fins, n	-	10	32	16
Outside diameter, D_o	1.5875	1.5850	1.5875	2.2162
Inside diamter, D_i	1.3843	1.4199	1.4707	2.0384
Equivalent diameter, D_e	1.3843	1.3600	1.4028	1.9870
Hydraulic diameter, D_h	1.3843	0.8530	0.6772	1.1300
Fin Height, b	-	0.1575	0.0686	0.1981
Wall thickness	0.1016	0.0826	0.0463	0.0889
Fin Height/Inside diameter	-	0.1109	0.0466	0.0972
Actual flow area, A_{fa}**	1.5050	1.4527	1.5455	3.1009
Nominal flow area, A_{fn}**	1.5050	1.5835	1.6988	3.2634
Core flow area, A_{fc}**	-	0.9588	1.3966	2.1181
Actual area, A_a**	4.3489	6.6800	9.1292	11.300
Nominal Area, A_n**	4.3489	4.4607	4.6203	6.4038
Inter-fin spacing, W	-	0.297	0.102	0.305
Helix Angle, α	-	0^o	16.2^o	12.34^o
Pitch, cm/360^o	-	ST	30.48	20.3

* all lengths and areas are in cm and cm^2 respectively, as appropriate.

** area in cm^2

** area in cm^2/cm

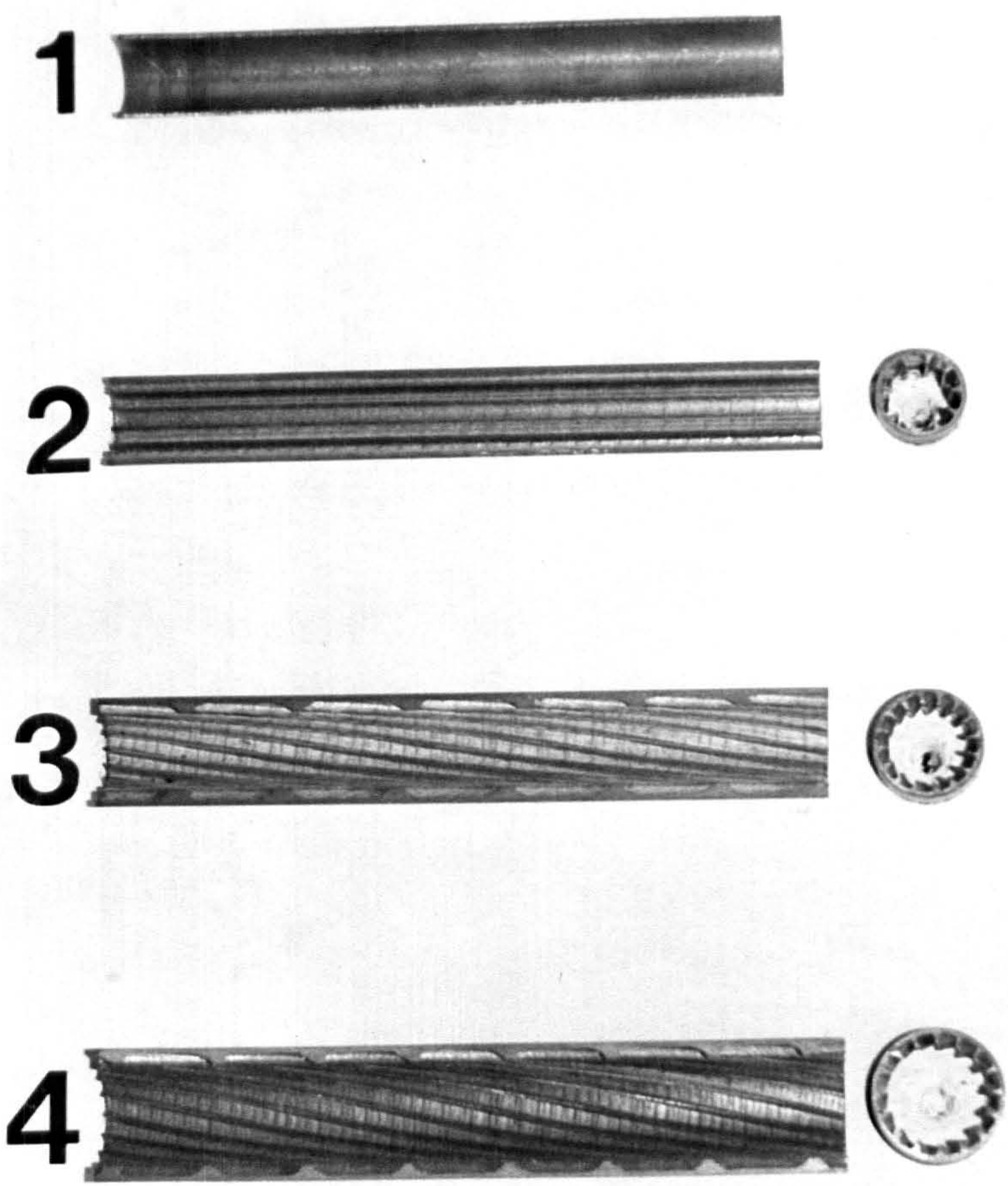

Fig. 3-3 A Photographic view of the Test Tubes

the intermediate stations. The test tube was also instrumented to measure the inlet and outlet temperature of R-113 and the pressure drop along its length. Figure 3.4 gives additional details about the construction of each test tube. After attaching the thermocouples each tube was wrapped with teflon tape on the outside, in order to insulate the heating element from the surface of the tube. Heating was accomplished by a ribbon-type chromel heating element, which was wound uniformly on the teflon tape. The heating element was then coated with a layer of epoxy resin to hold it onto the tube, and a second layer of teflon tape was again wound over the resin. The entire test section was then insulated with 6.25 cm thick fiberglass insulation to prevent any heat losses to the atmosphere. The exit of each test section had a transparent glass section attached to it. There was also another transparent observation section at the inlet to the test sections, which was used in observing the flow regime of R-113 as it entered the test section. The transparent sections were made of standard, clear, high pressure glass tubing. The construction details of the transparent section are shown in Fig. 3.5.

The construction details of the pressure taps are shown in Fig. 3.6. Four holes 1/16 in (.002 m) diameter each and spaced 90^{o} apart around the circumference of the tube, were drilled in the test section. The holes were then covered with a copper sleeve, whose inner diameter was approximately equal to the outside diameter of the boiler tube plus 3/16 in (.005 m). The copper sleeve was silver brazed to the tube and a hole was drilled in it to accomodate a short copper tube 1/8 in (.003 m) I.D. The two pressure taps at the top

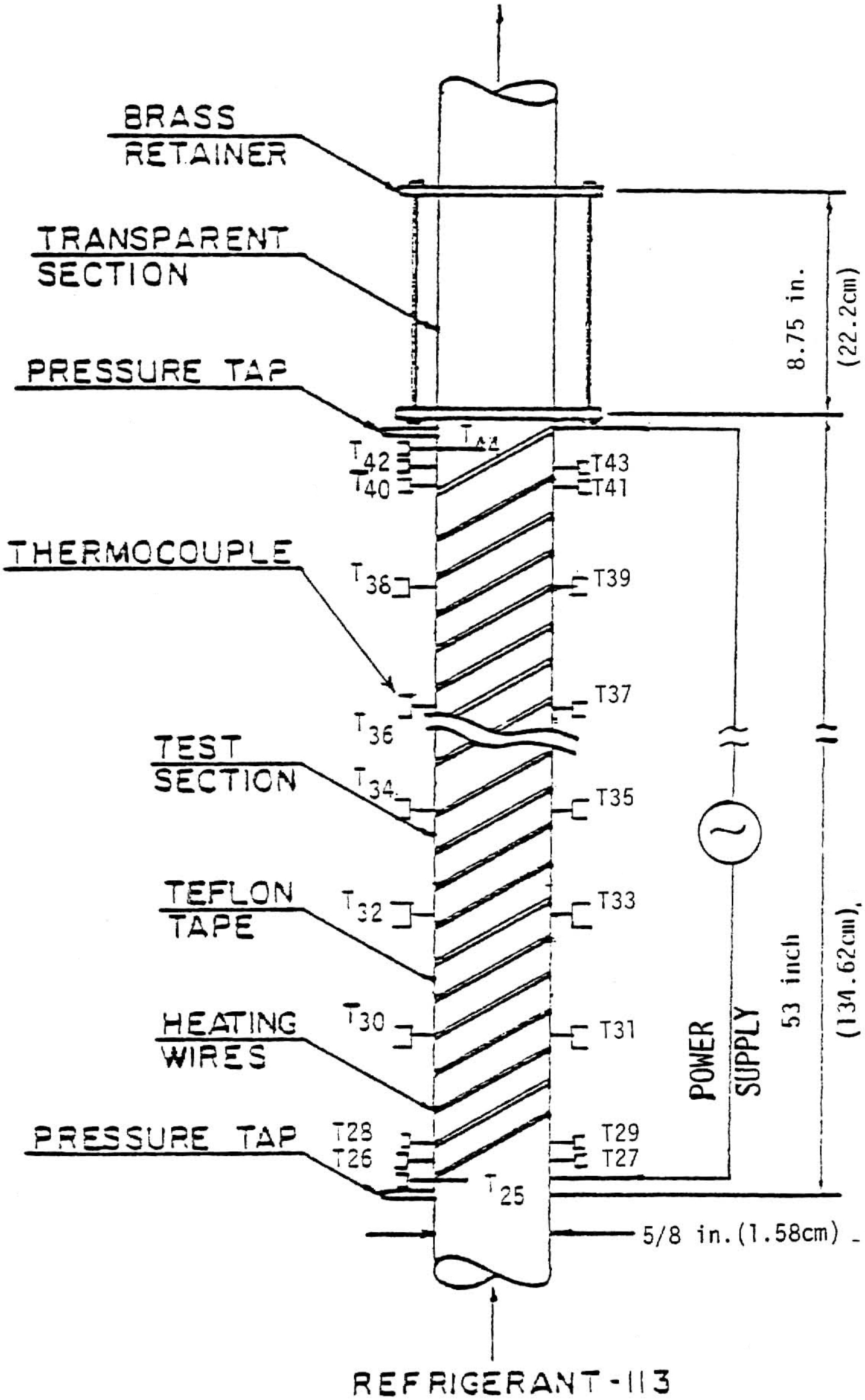

Fig. 3.4 A Schematic Diagram of the Heating and Boiling Test Section.

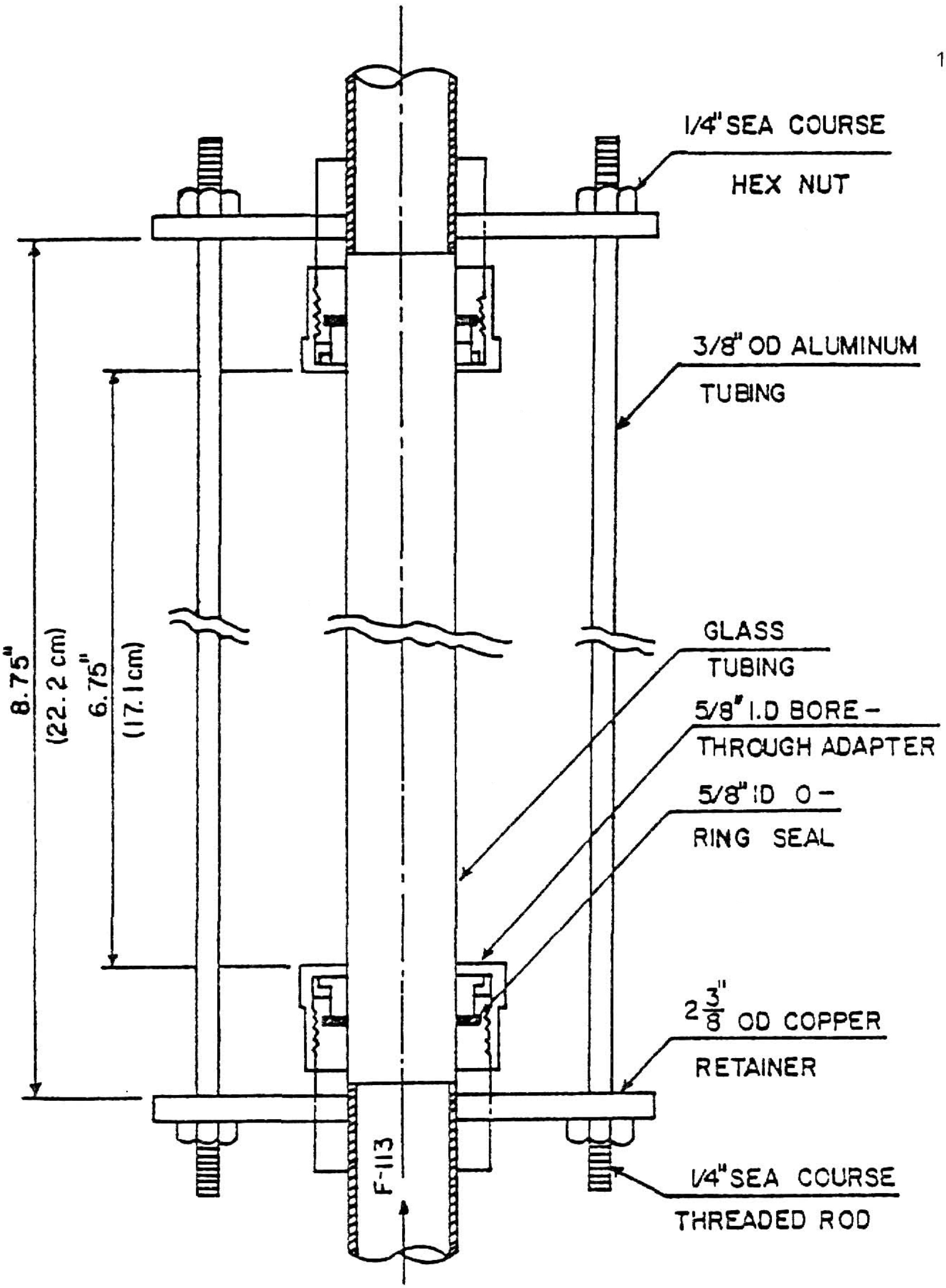

Fig. 3.5. Construction Details of One Transparent Section.

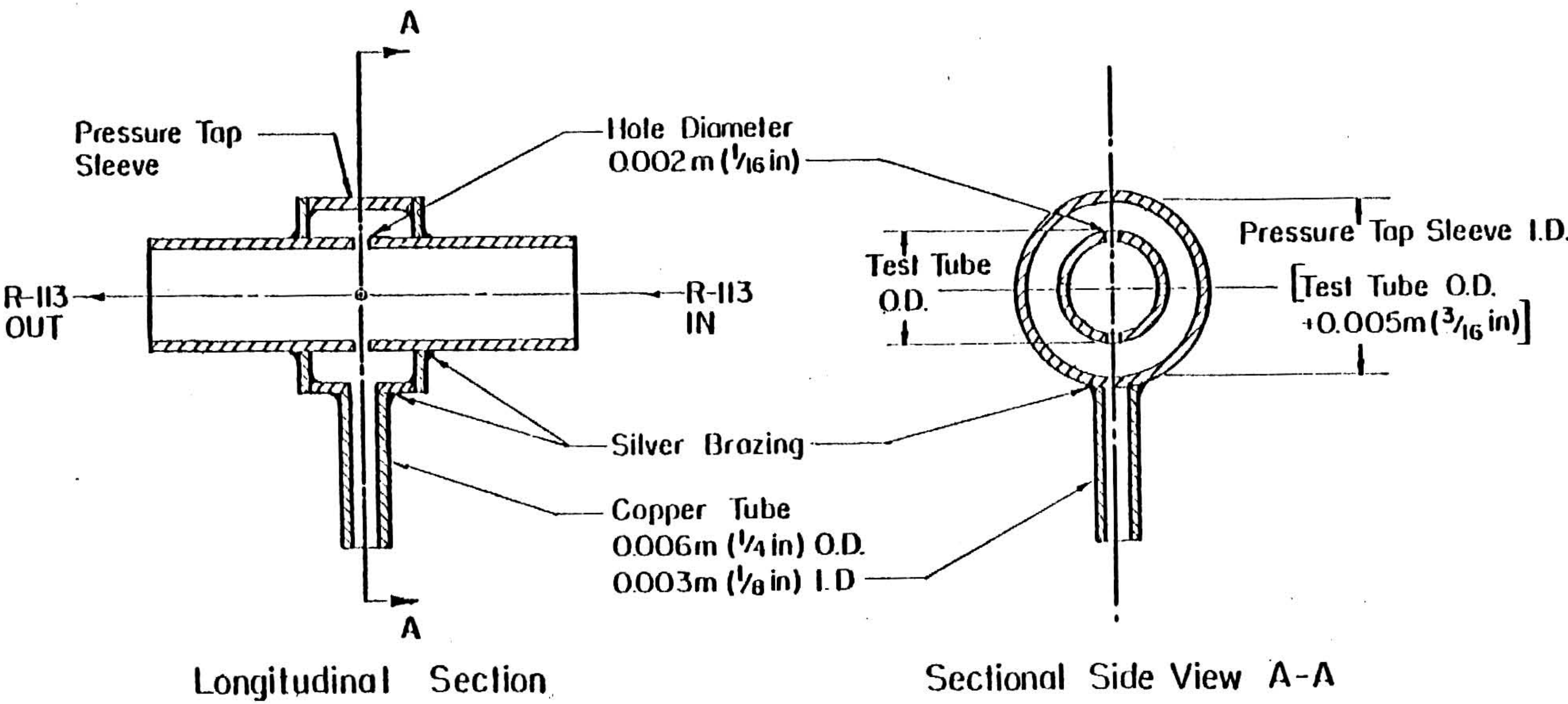

Fig. 3.6. **Pressure Tap Construction Details**

and the bottom, were connected to the pressure transducer by means of two separate pieces of tubing which were connected to the short copper tubes. There existed the possibility that the hole drilled for the pressure tap either went through a fin or the space between the fins. The above arrangement minimized this possibility and produced an averaging effect at the pressure taps.

The pressure drop across the pressure taps was measured by a Pace Wianco pressure transducer model KP15, which was connected to a Pace Wianko digital indicator, model CD25. It was calibrated using a dead weight tester. The flow rate of the refrigerant was measured by two Fisher and Porter variable area flow meters, which were mounted vertically in parallel, to be used individually or simultaneously.

The power input to the test section was controlled by means of a variable transformer and the voltage and current across each heating circuit were measured by a digital voltmeter and a digital clamp-on ampmeter, respectively. Additional details about the instrumentation used is given in appendix E.

3.3 OPERATION OF THE SYSTEM AND DATA ACQUISITION

After the construction and instrumentation of the system, the entire system was checked for leaks under pressurized and evacuated conditions. It was then evacuated down to 1000 microns of Hg by a vacuum pump. The liquid receiver was then filled to the top with R-113 before the final evacuation process was started. It was then isolated from the rest of the system. The system was then evacuated down again to 1000 microns of Hg.

Each experimental run was started by filling the cooling water tank; then the circulating water pump was put into operation. The freon pump was started. After the liquid freon appeared in the observation glass section at the entrance and exit of the test tube and the exit of the horizontal cooler section, water from the tank was allowed to circulate through the precooler section. Electric power was then gradually applied to the test tube to the desired level. In single phase heating experiments electrical energy was added gradually while making sure that R-113 leaving the test section was still liquid. In subcooled boiling experiments, electrical energy was added to the test section such that the R-113 leaving the test section was in a boiling state.

The following measurements were taken during each experimental run:

1) The refrigerant's flow rate, in kg/s (G.P.M.).
2) The inlet gauge pressure of R-113 to the test tube, in bar (PSI).
3) The pressure drop across the inlet and outlet of each test tube, in bar (PSI).
4) The inlet and outlet temperature of R-113 in each test tube, in ^{o}C (^{o}F).
5) Room temperature, in ^{o}C (^{o}F).
6) The temperature of the eighteen thermocouples attached to the outer wall of the test tube in ^{o}C (^{o}F).
7) The electrical power supply to the test section.

The variables that were controlled in the present study were the liquid R-113 flow rate, the electrical power input, and the initial subcooling. Also, after the completion of each experimental run, its collected data were checked for acceptability. The criterion selected for acceptability was ± 7% in the heat balance error. The heat balance error was calculated from the ratio of the difference between electrical power supply and heat gain of R-113 to the electrical power supply. The heat gain rate of the refrigerants was calculated from the freon's flow rate and its enthalpy change between its inlet and outlet. The electrical power input was corrected for conduction losses at the top and bottom of the test section.

For each fixed flow rate and fixed inlet subcooling different single phase heat transfer data were taken by varying the electrical power input. Subcooled boiling data for the same flow rate and same inlet subcooling were obtained by increasing the electrical power input to the point were boiling could be observed at the exit of the test section. Because of the difficulty estimating the exit quality of R-113 in the test section during subcooled boiling experiments it was assumed that the heat error balance in these experiments was also within ± 7%. The effect of such assumption on the calculation heat transfer coefficient will be given later in Appendix F.

CHAPTER IV

EXPERIMENTAL RESULTS AND CORRELATIONS

4.1 INTRODUCTION

Augmentation of single phase heat transfer by internally finned tubes has been studied by several investigators. Their results were discussed earlier in chapter II. It is important to point out that the range of Reynolds number covered in these studies was either in the laminar or the turbulent range. The Reynold's number covered in the present study was in the transition zone, 2400 < Re < 5500. In the present chapter the heat transfer and pressure drop results of single phase flow are reported. A review of existing heat transfer and pressure drop correlations for finned tubes is made, and new correlations based on the results of the present study are proposed. In addition, heat transfer and pressure data of subcooled boiling are reported. However, because of the limited data taken, which was dictated by the limitations of the test facility, no attempt was made to develop design correlation for subcooled boiling.

The overall average heat transfer coefficient for single phase or subcooled boiling for the smooth and finned tubes was calculated as follows:

$$\bar{h} = \frac{Q}{\pi\, D_i L (T_w - T_f)} \qquad (4\text{-}1)$$

Where Q was the electric power input corrected for conduction losses at the tube's ends. D_i was the inside nominal diameter of the tube, smooth or finned. L was the length of the test section. T_w was the average inside wall temperature. It was obtained from the average outside surface temperature of the test tube (= arithematic mean of all 18 surface thermocouples) and the temperature drop across the tube wall. This temperature drop was obtained from the one dimensional heat conduction equation for a cylinder. T_f was the arithmatic mean of the inlet and outlet temperatures of R-113. A sample of the calculation procedure of the film coefficient of heat transfer is given in Appendix A. The ranges of experimental parameters for single phase heating and subcooled boiling are given in Tables 4-1 and 4-2 respectively.

4.2 SINGLE PHASE HEATING

4.2.1 Heat Transfer Results of the Smooth and Finned Tubes

Due to the fact that the Reynolds number range of the present study was in the transition range of Reynolds number; and the fact that none of the existing heat transfer correlations for laminar or turbulent flow could satisfactorily correlate the data of the present study, a new correlation relating Nusselt number Nu, Prandtl number Pr, and Reynolds number Re was sought. Using the least square regression analysis the following correlation was obtained.

$$Nu = 0.0032\ Re^{1.07}\ Pr^{0.4} \qquad (4\text{-}2)$$

Table 4-1 Range of Operating Conditions for Single Phase Heating

Mass Flux (based on nominal inside diameter)	74.93(55248.7) - 262.74(193727) $\frac{kg}{S\text{-}m^2}$ (lbm/hr-ft^2)
Heat Flux Input	498.0(157.9) - 2929.85(928.93) $\frac{w}{m^2}$ (Btu/hr-ft^2)
Overall Heat Transfer coefficient	237.91(41.9) - 1691.48(297.9) $\frac{w}{m^2\text{-}K}$ (Btu/hr-ft^2-^{o}F)
R-113 inlet Temperature	23.03(73.45) - 33.64(92.55) ^{o}C(^{o}F)
R-113 outlet Temperature	24.77(76.59) - 38.25(100.85) ^{o}C(^{o}F)
Inlet Gage Pressure	0.29(4.2) - 0.41(6.0) bar (PSIG)

Table 4-2 Range of Operating Conditions for Subcooled Boiling

Mass Flux (based on nominal inside diameter)	74.3(54781.3) - 259.3(1911967) $\frac{kg}{S\text{-}m^2}$ ($lbm/hr\text{-}ft^2$)
Heat Flux Input	7420.6(2352.76) - 27035.65(8571.86) $\frac{w}{m^2}$ ($Btu/hr\text{-}ft^2$)
Overal Heat Transfer coefficient	567.34(99.92) - 1862.16(327.96) $\frac{w}{m^2\text{-}k}$ ($Btu/hr\text{-}ft^2\text{-}^oF$)
R-113 Inlet Temperature	29.93(85.87) - 111.34(44.08) $^oC(^oF)$
R-113 Outlet Temperature	48.23(118.81) - 61.42(142.56) $^oC(^oF)$
Exit Quality	0.015 - 0.322
Inlet Gage Pressure	0.34(5) - 0.62(9.0) bar(PSIG)

Figure 4-1 shows a plot of $Nu/Pr^{0.4}$ versus Re for the experimental data points of the smooth tube, tube 1. On the same plot Eq. (4-2) is also plotted. As expected Nu increased with the increase in Re. It is to be pointed out that all R-113 thermophysical and transport properties in Eq. (4-2) were evaluated at the average of inlet and outlet temperatures.

Equation (4-2) was taken at the basis for developing the heat transfer correlation for the finned tubes. This was achieved by identifying modifying parameters which are dependent on the geometric parameters of the finned tubes and applying these modifiers to Eq. (4-2) to bring about the best agreement between the predictions and the experimental measurements. Before presenting the new finned tube correlation a review of existing correlations follows.

Heat transfer measurements were made for finned tubes with fins of several designs in turbulent water flow by Watkinson et al. [35]. They correlated the experimental data of eleven spiral fin tubes over the range $5000 \leq Re_e \leq 100{,}000$ by the equation:

$$Nu_e = o.369\ Re_e^{0.63} \left(\frac{p}{D_e}\right)^{0.27} \left(\frac{w}{D_e}\right)^{0.21} Pr^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.1} \qquad (4\text{-}3)$$

where Nu_e and Re_e were based on the equivalent diameter.

Heat transfer data for five straight fin tubes ($0.21 \leq \frac{w}{D_e} \leq 0.49$) were correlated within ± 10% over the range $5000 \leq Re_e \leq 100{,}000$ by the following expression:

$$Nu_e = 0.212\ Re_e^{0.60} \left(\frac{w}{D_e}\right)^{0.34} Pr^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad (4\text{-}4)$$

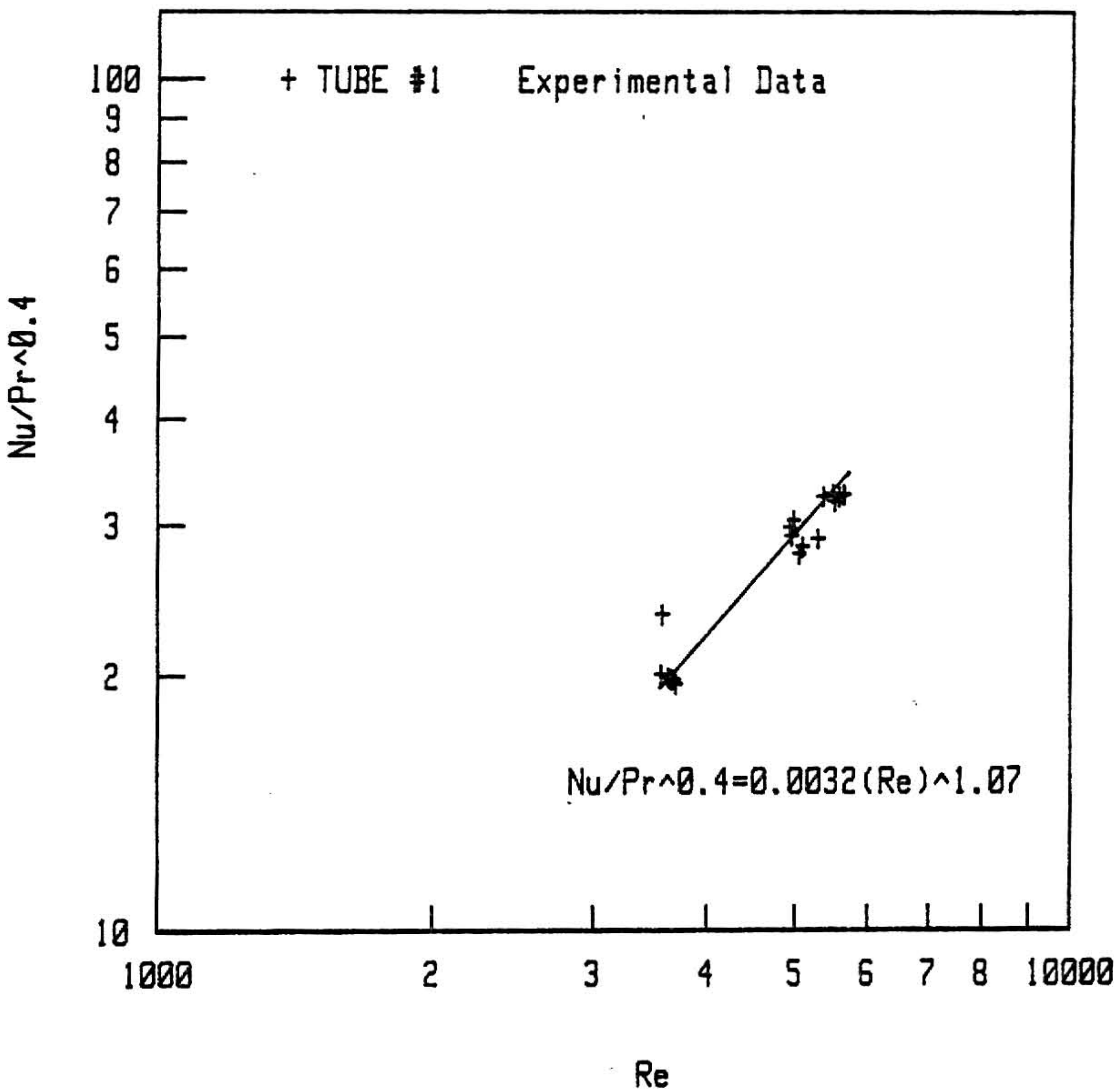

Fig. 4-1 Correlation of Single Phase Heat Transfer Data for Smooth Tube 1

The heat transfer performance for heating water and air was individually determined, in turbulent flow, by Carnavos [8]. He correlated the experimental data using the hydraulic diameter and the average bulk physical properties. All heat transfer data were represented by the following formula.

$$\frac{Nu}{Pr^{0.4}} = 0.023\ Re^{0.8}\ (F) \qquad (4\text{-}5)$$

where

$$F = (F_1)^{0.1}\ (F_2)^{0.5}\ (F_3)^{3} \qquad (4\text{-}5)$$

The parameters in the modifying factor F are defined as follows:

$$F_1 = \frac{A_{fa}}{A_{fc}} \qquad (4\text{-}7)$$

$$F_2 = \frac{A_n}{A_a} \qquad (4\text{-}8)$$

$$F_3 = \mathrm{Sec}\alpha \qquad (4\text{-}9)$$

where

A_{fa} = actual free flow area, $\frac{\pi D_e^2}{4}$, Cm^2,

A_{fc} = Open core free flow area, $\frac{\pi D_c^2}{4}$, Cm^2,

A_n = nominal heat transfer area based on D_i as if fins were not present, Cm^2/Cm,

A_a = actual heat transfer area, Cm^2/Cm,

The equivalent diameter D_e and the core diameter are defined by:

$$\frac{\pi D_e^2}{4} = \frac{\pi D_i^2}{4} - n\ bt/\mathrm{Cos}\ \alpha \tag{4-10}$$

$$\frac{\pi D_c^2}{4} = \frac{\pi}{4}(D_i - 2b)^2 \tag{4-11}$$

The hydraulic diameter was also defined by:

$$D_h = \frac{4A_{fa}}{A_a}$$

Using the above definitions, it can be shown that:

$$F_1 = \frac{A_{fa}}{A_{fc}} = [\frac{D_e}{D_i}]^2/[1 - (2b/D_i)]^2 \tag{4-12}$$

$$= \frac{[1 - (4nbt)/(\pi D_i^2\ \mathrm{Cos}\ \alpha)]}{[1 - 2b/D_i]^2} \tag{4-13}$$

$$F_2 = A_n/A_a = (D_i D_h/D_e^2)$$

$$= \frac{\pi D_i^2}{[\pi D_i + 2nb/\mathrm{Cos}\ \alpha]} \tag{4-14}$$

n = number of fins

α = helix angle of the fin

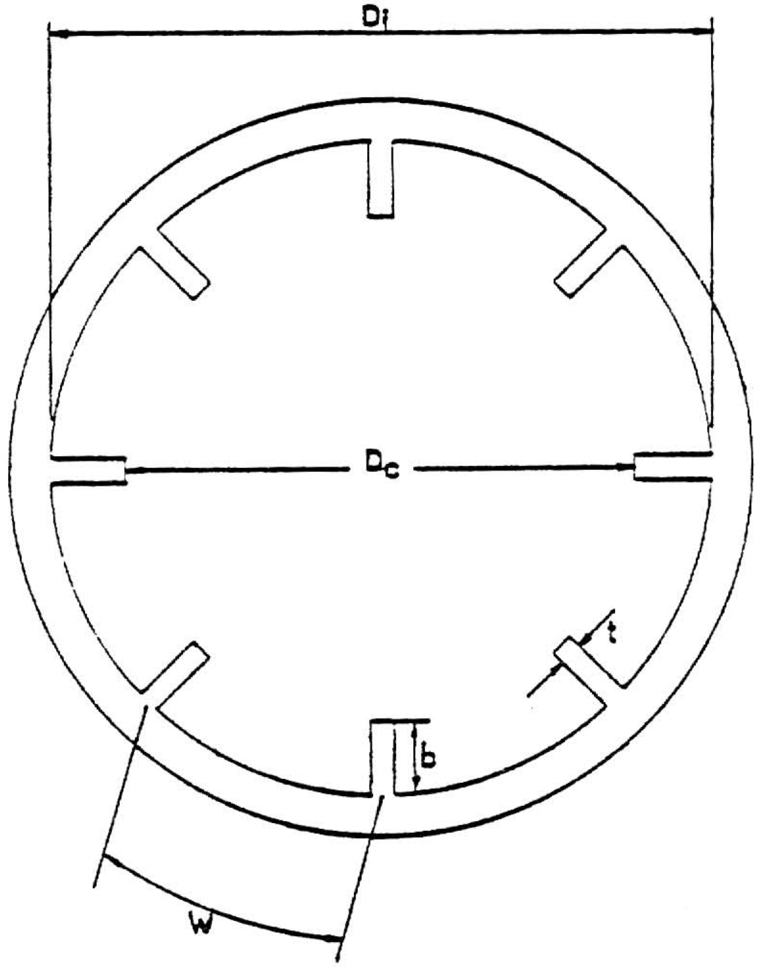

Fig. 4-2 Cross Section of Internally Finned Tube

It is to be noticed that modifying parameters F_1, F_2, and F_3 include all the geometric parameters D_i, w, b, t, n and α of the internally finned tubes. These parameters are defined in Fig. 4-2. Values of F_1, F_2, and F_3 for the finned tubes tested are given in Table 4-3.

Table 4-3 Computed Values of F_1, F_2, and F_3

F value	Tube Tested 2	3	4
F_1	1.515	1.1464	1.464
F_2	0.6678	0.5061	0.5667
F_3	1.000	1.0413	1.0236

After several trials of using different combinations of the parameters F_1, F_2, F_3 and by applying the least regression analysis the following finned tube correlation resulted.

$$Nu = 0.0032\ Re^{1.07}\ Pr^{0.4}\ [1 + 0.0013\ F_1^{5.0}\ F_2^{-9.6}] \qquad (4\text{-}15)$$

Because of the limited range of F_3 of the tubes tested it was not possible to incorporate its effect in the final correlation. According to its definition, Eq. (4-9), it depends on the helix angle α. It is to be pointed out that α is also included in the parameters F_1 and F_2 as shown in Eqs. (4-13) and (4-14) respectively. It is to be pointed out also that the characteristic dimension in Nu and Re in Eq. (4-15)

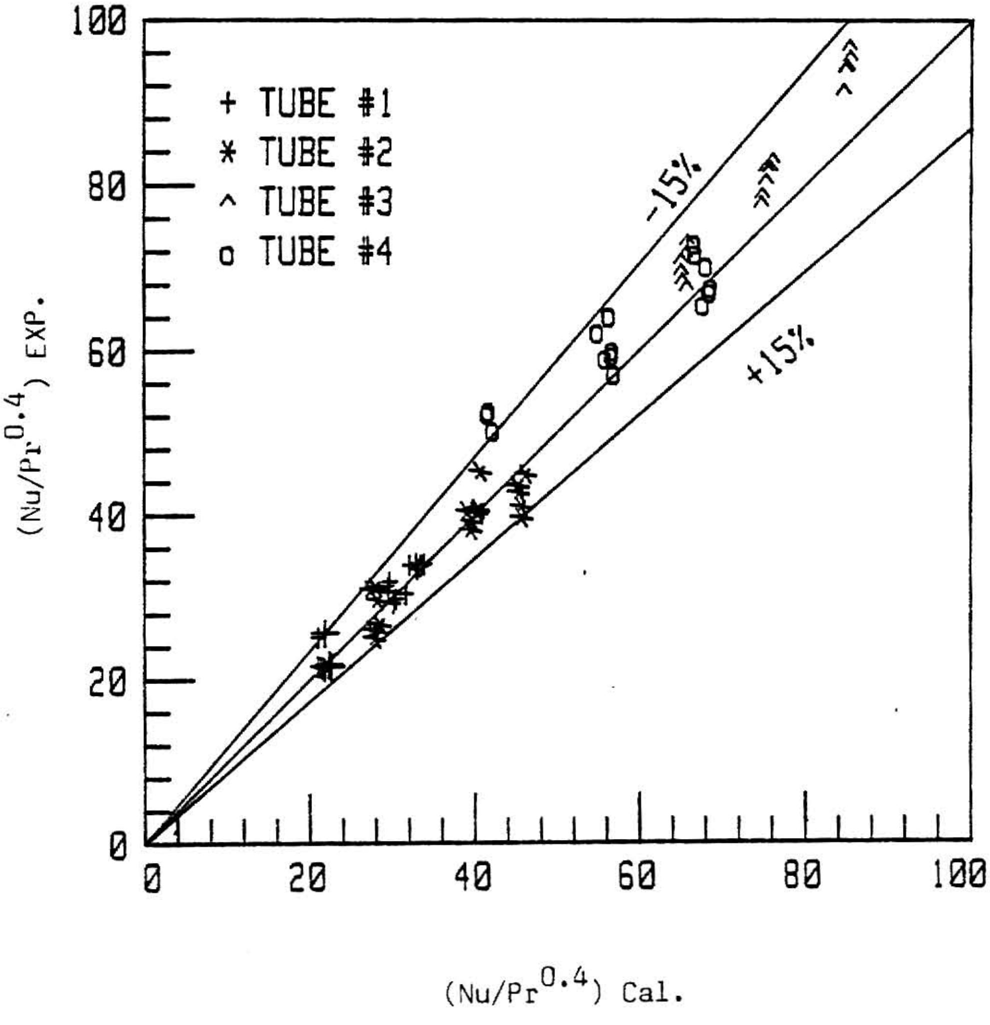

Fig. 4-3 Comparison Between Predicted and Measured $Nu/Pr^{0.4}$ for Single Phase Flow

is the nominal inside diameter D_i of the finned tube rather than the hydraulic diameter D_h. The reason for such a choice was the fact that the heat transfer coefficient $\bar{h}$ was based on D_i, according to Eq. (4-1), and not D_h.

Figure 4-3 shows a comparison between the predictions of $Nu/Pr^{0.4}$ of Eq. (4-15) and the experimental measurements. The results show that the disagreement is in the ± 15% range.

Figure 4-4 shows a plot $Nu/Pr^{0.4}$ of the experimental data points versus Re of all tubes tested. On the same figure the correlation lines of the data points obtained for each tube using the least square regression analysis. Figure 4-5 was reproduced from Fig. 4-4 and it plots the ratio of $Nu/Pr^{0.4}$ of the finned tubes to that of the smooth tube versus Re. The results show that the ratio slightly decreases for tube 2, noticeably decreases for tube 4, and noticeably increases for tube 3, with the increase of Reynolds number. Such results can be explained from the results in Fig. 4-4 which give the behavior of the individual tubes tested. For example, $Nu/Pr^{0.4}$ for tubes 2 and 4 increases at a slower rate, and at a faster rate for tube 3 compared to the smooth tube with the increase of Re. In general it can be stated that, over the range of Re tested, the enhancement in the heat transfer coefficient was 50%, 180%, 140% for tubes 2, 3, and 4 respectively compared to the smooth, on a nominal area basis.

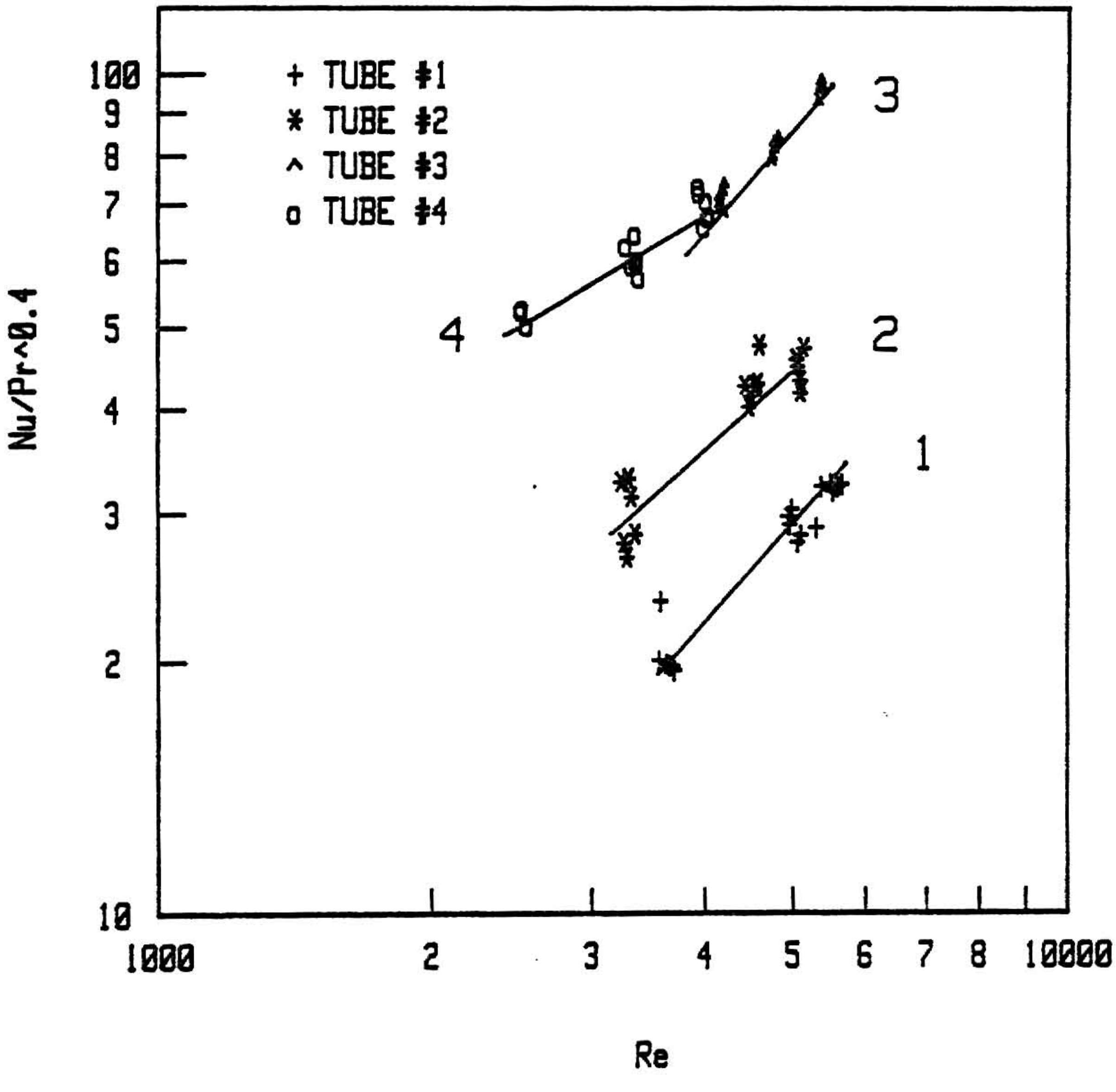

Fig. 4-4 $Nu/Pr^{0.4}$ Versus Re for All Tubes Tested for Single Phase Flow

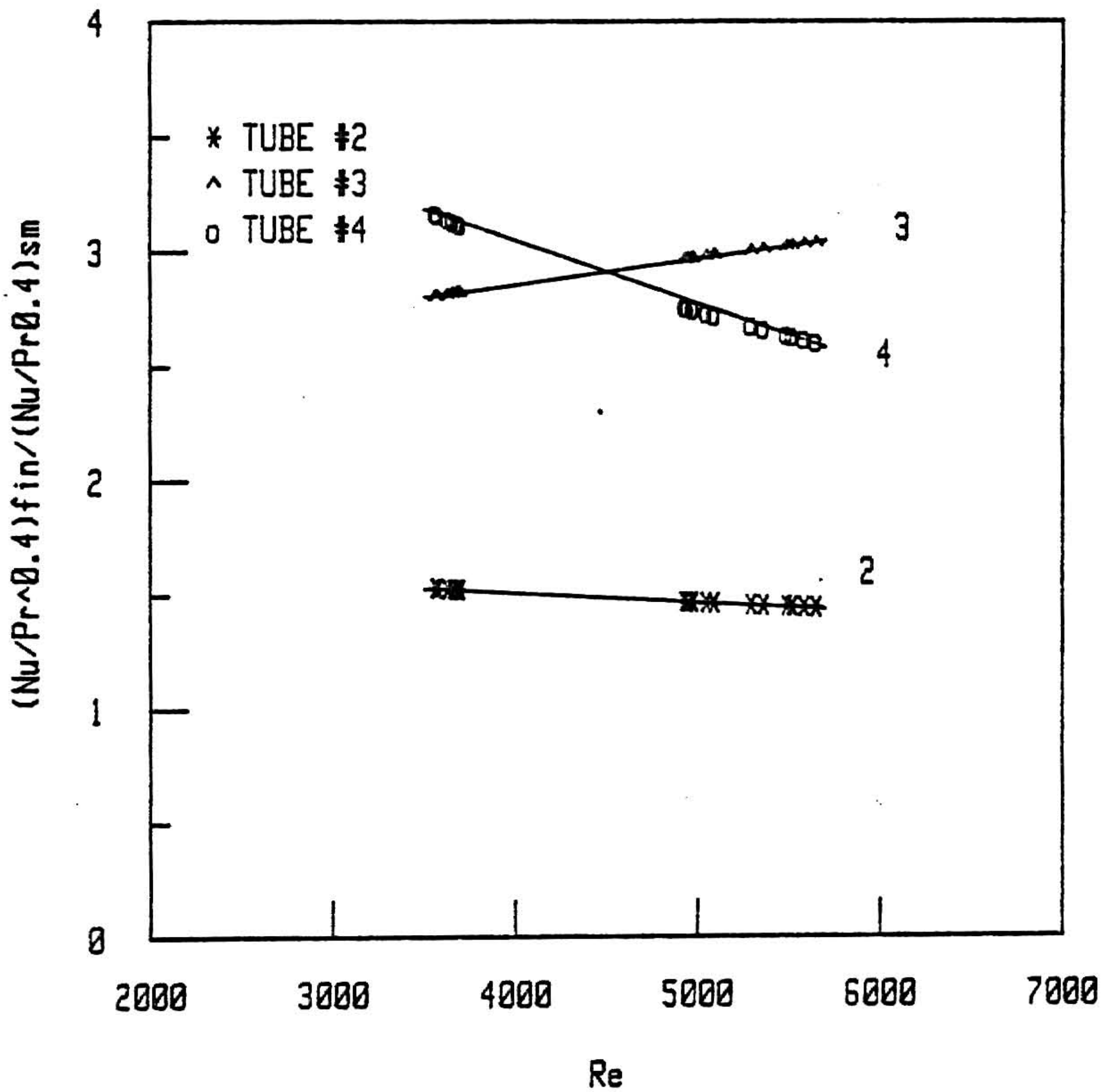

Fig. 4-5 The Ratio of $Nu/Pr^{0.4}$ for Finned to Smooth Tube versus Re

4.2.2. Pressure Drop Results of the Smooth and Finned Tubes.

Figure 4-6 through 4-8 show plots of the pressure drop versus the heat flux for all tubes tested for different flow velocities. The results show that the pressure drop, at a given velocity or flow rate, is insensitive to heat addition for single phase flow. Figures 4-9 through 4-11 are cross plots of Figs. 4-6 to 4-8 to show the effect of increasing the flow rate on the pressure drops for the individual tubes tested. As can be expected, at a fixed heat flux rate the single phase pressure drop increases with the increase of velocity or mass flow rate.

Following the same approach of developing the single phase heat transfer correlation of the finned tubes, a friction coefficient correlation was sought for the smooth tube results. Using the least square regression analysis, the following expression was obtained for the Fanning friction coefficient

$$f = 24.97/Re \tag{4-16}$$

for a range $3300 \leq Re \leq 5500$

Equation (4-16) differs from the friction coefficient for laminar flow ($f = 16/Re$) due to the fact that the Re range to which it is applicable is in the transition zone. Equation (4-16) is shown plotted in Fig. 4-13. In the same figure the experimental data points of f vs. Re for the finned tubes are also shown. The regression equations for the data points of all the tubes are also shown. The results show that at the same Re, the friction coefficient is higher for the finned tubes than for the smooth tube. A performance evaluation of the tubes tested will be given in the next chapter.

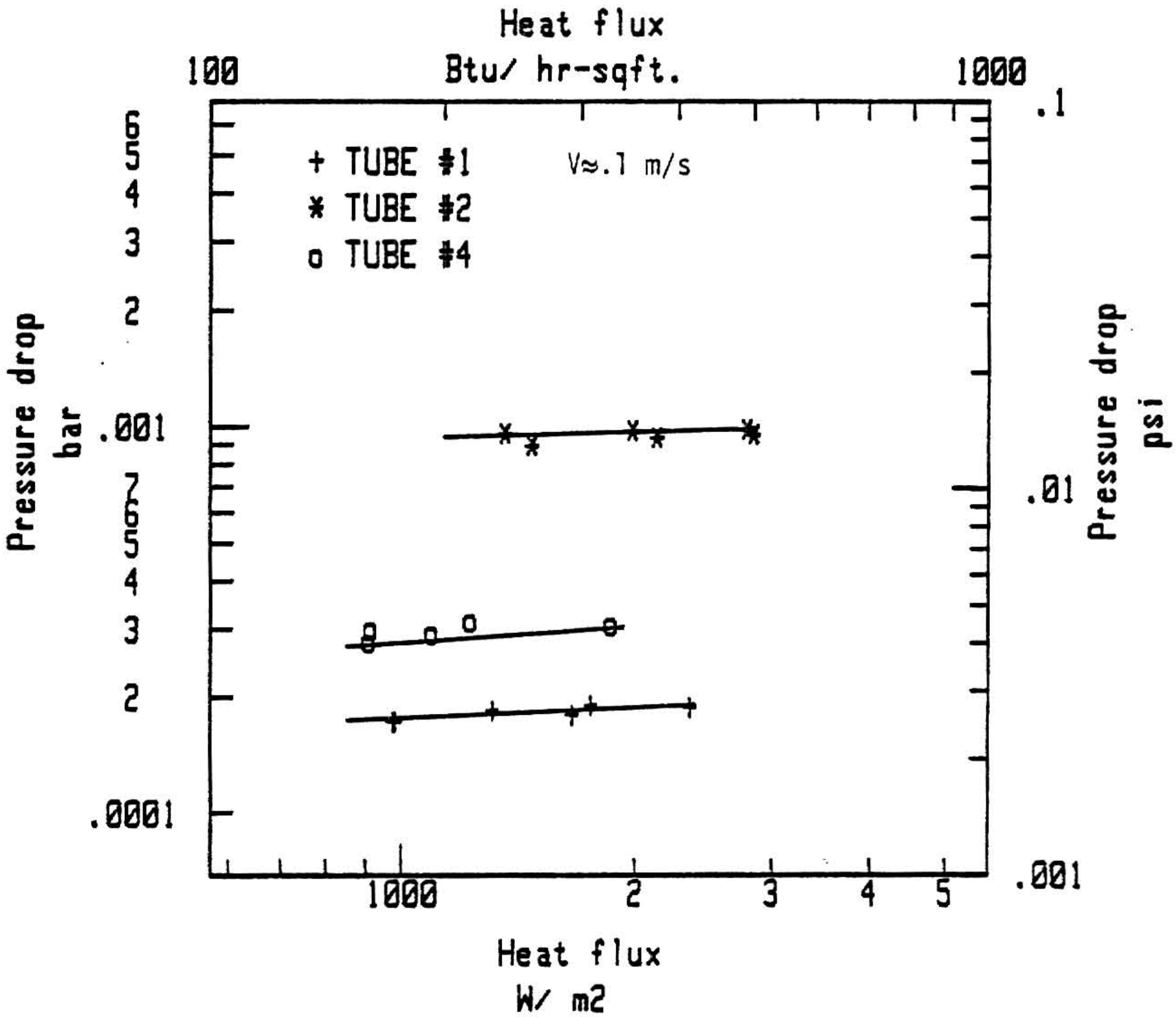

Fig. 4-6 Pressure Drop versus Heat Flux

For Single Phase Flow at V = 0.1 m/s

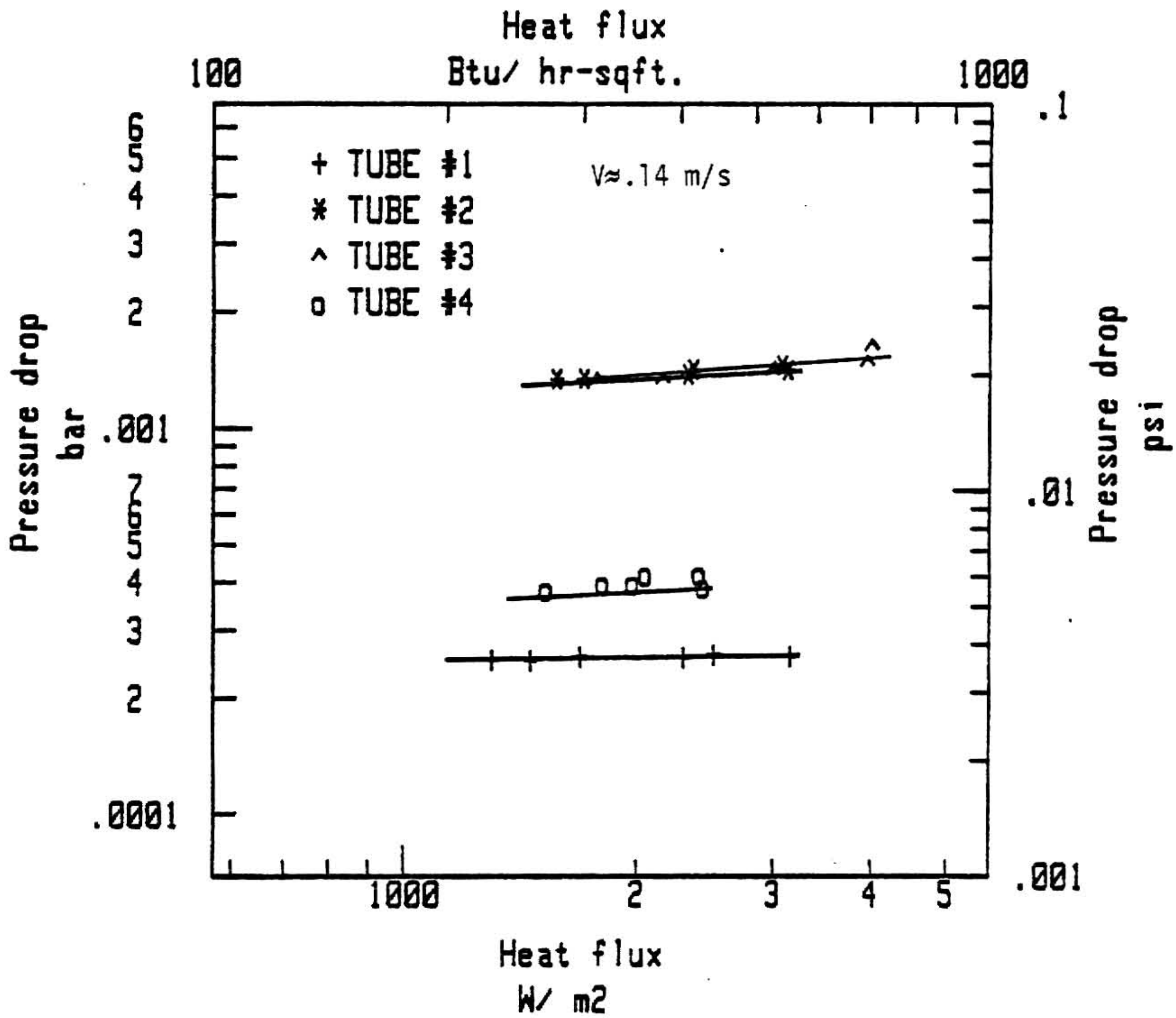

Fig. 4-7 Pressure Drop versus Heat Flux for Single Phase Flow at V = 0.14 m/s

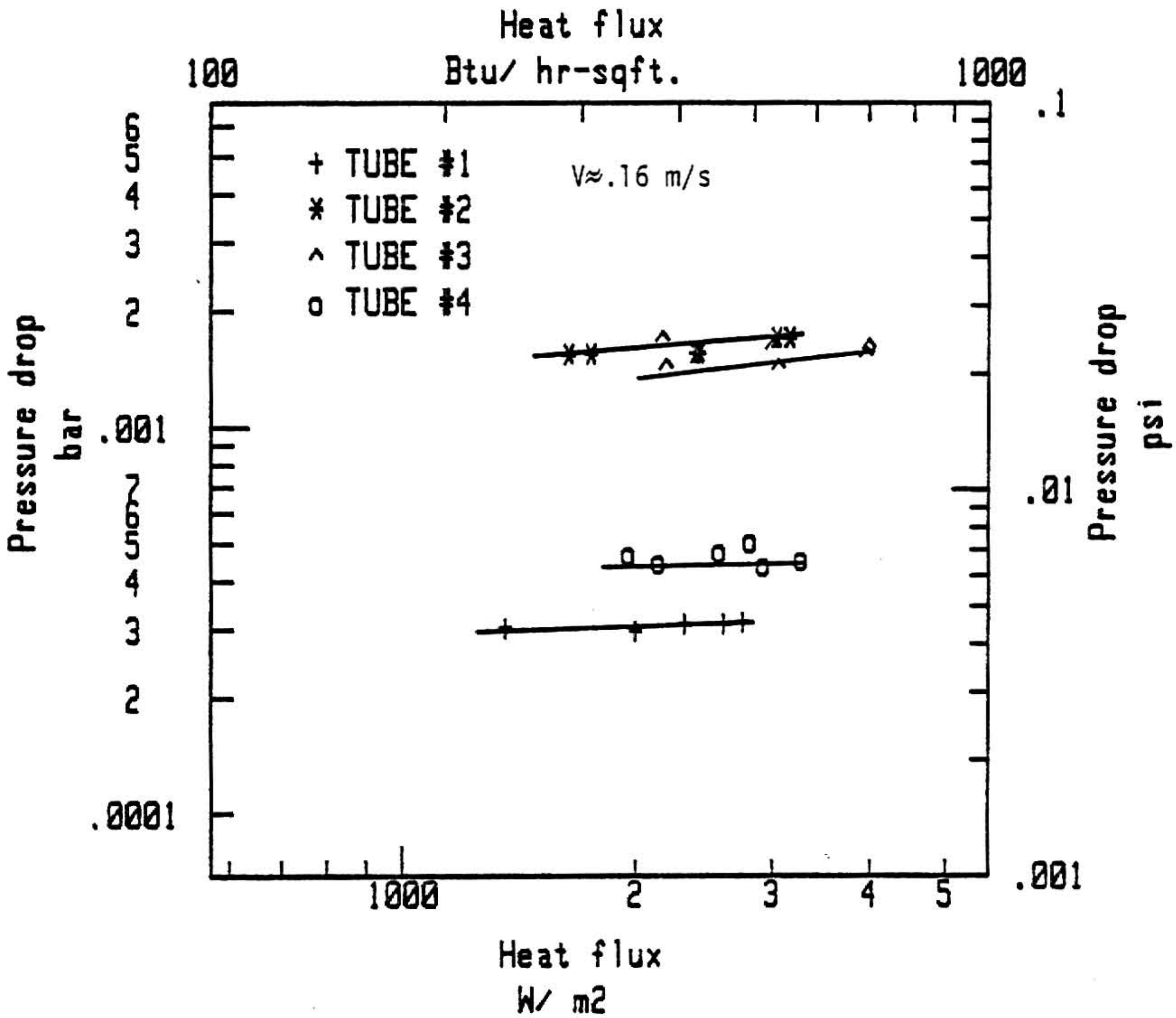

Fig. 4-8 Pressure Drop versus Heat Flux for Single Phase Flow at V = 0.16 m/s

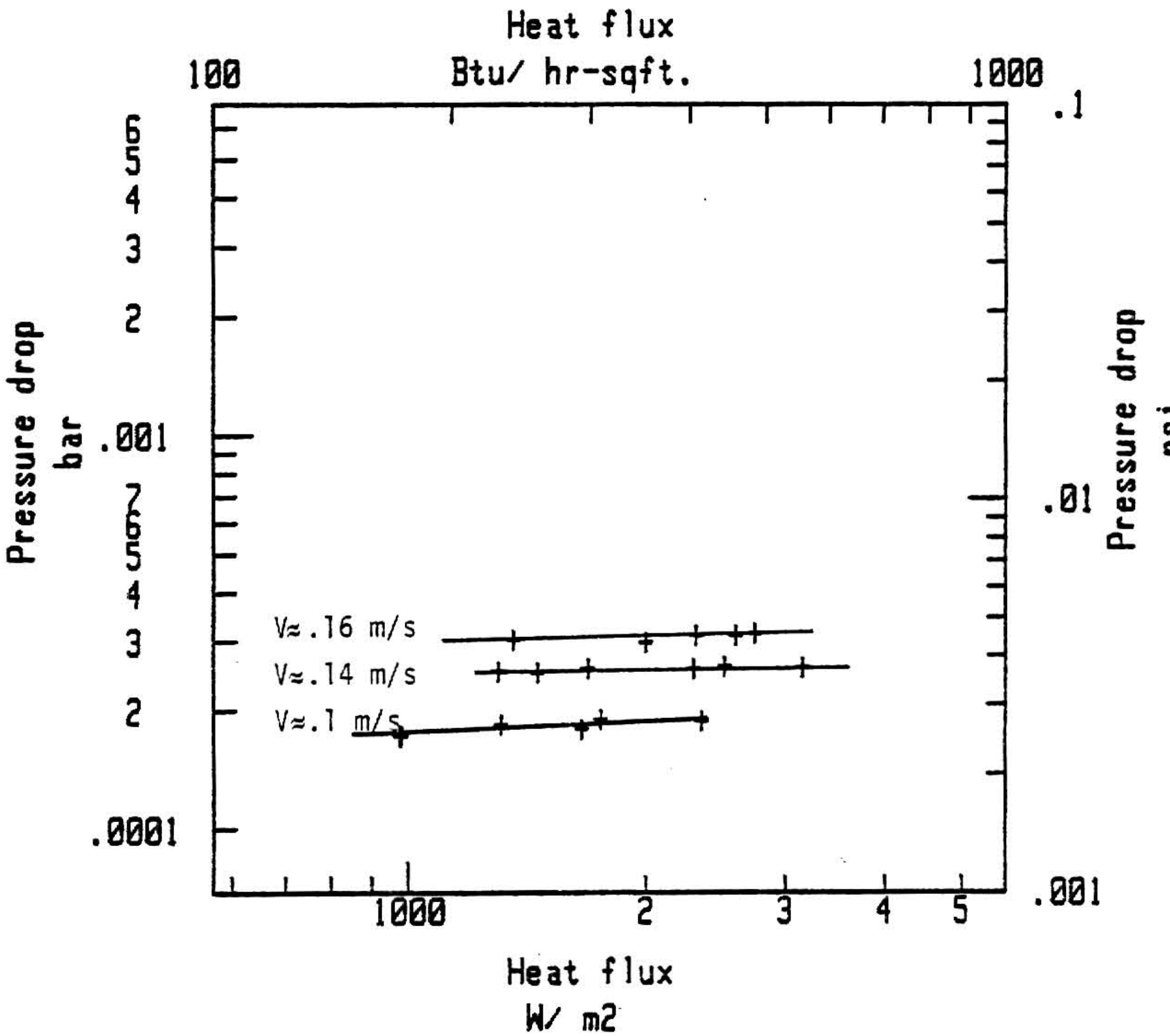

Fig. 4-9 Pressure Drop versus Heat Flux for Single Phase Heating For The Smooth Tube 1

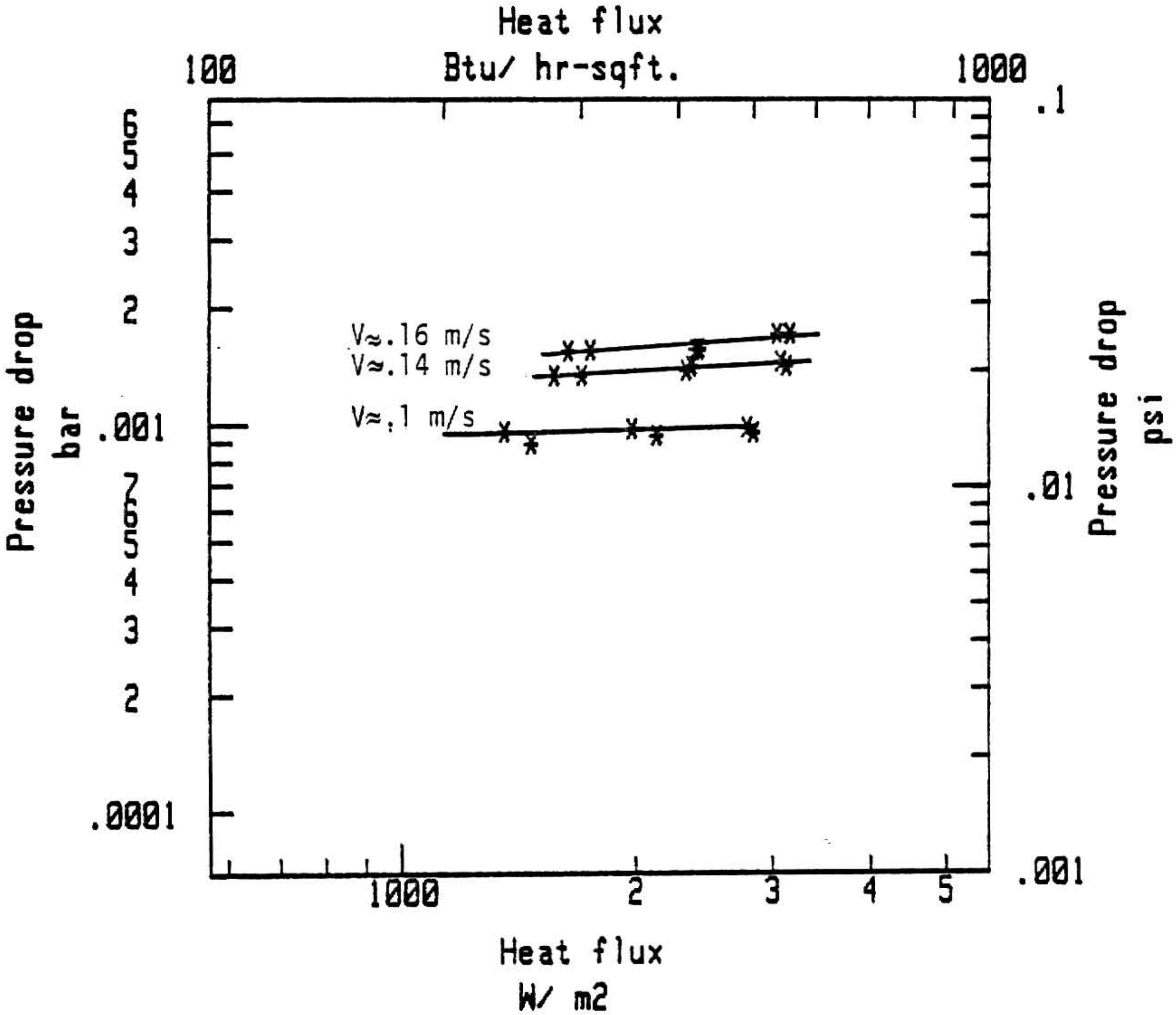

Fig. 4-10 Pressure Drop versus Heat Flux for Single Phase Heating for Finned Tube 2

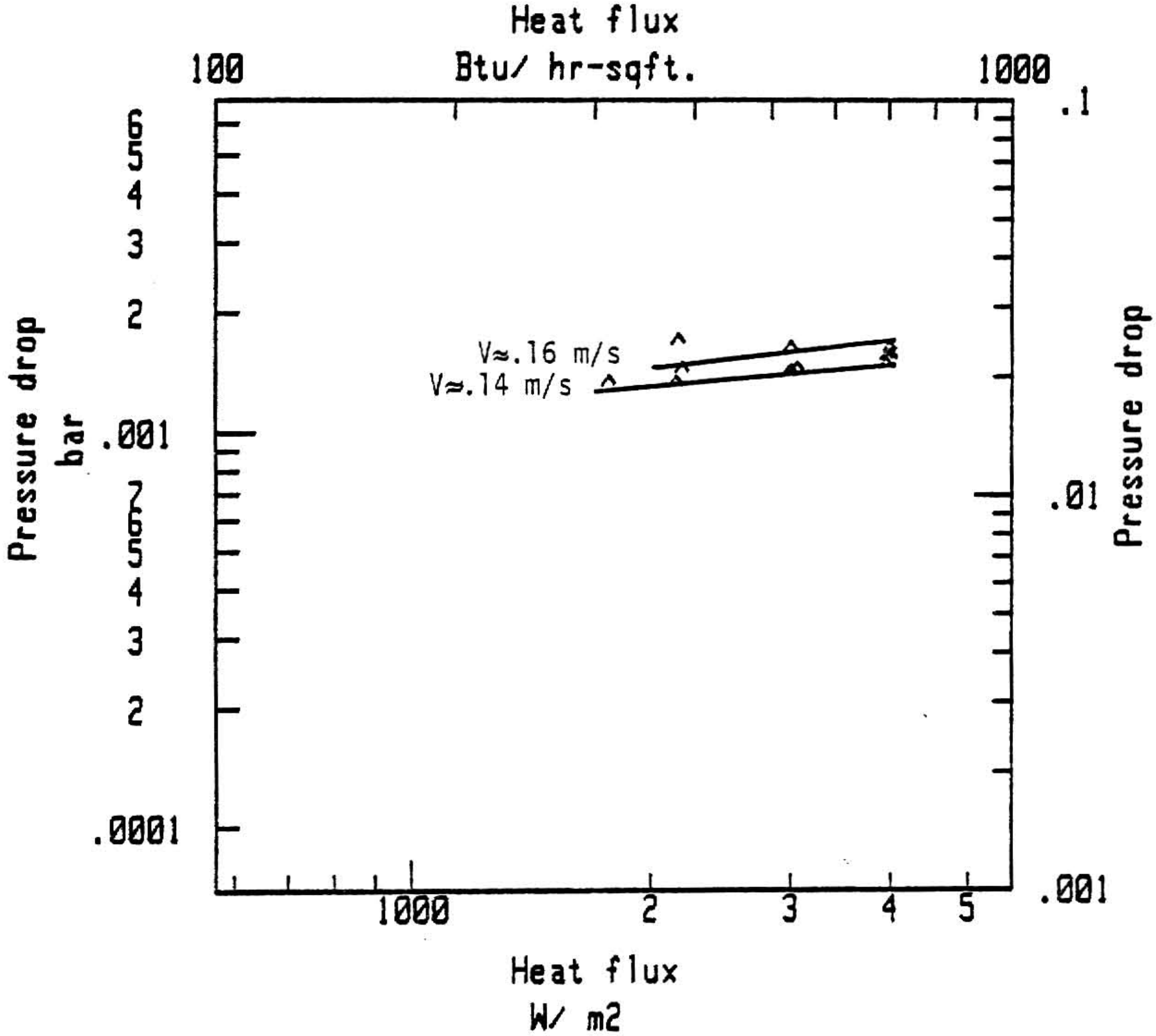

Fig. 4-11 Pressure Drop versus Heat Flux for Single Phase Heating for Finned Tube 3

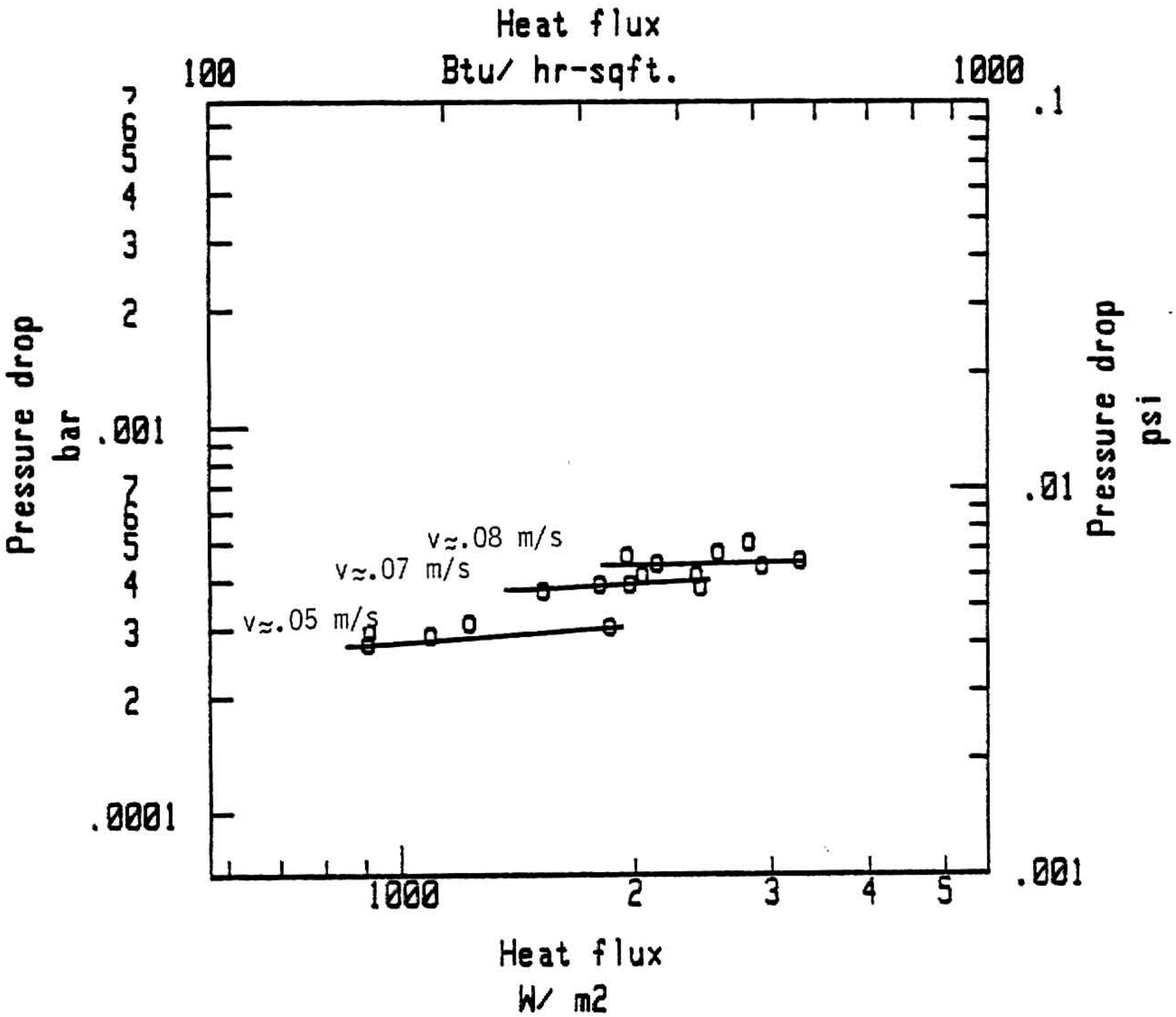

Fig. 4-12 Pressure Drop versus Heat Flux for Single Phase Heating for Finned Tube 4

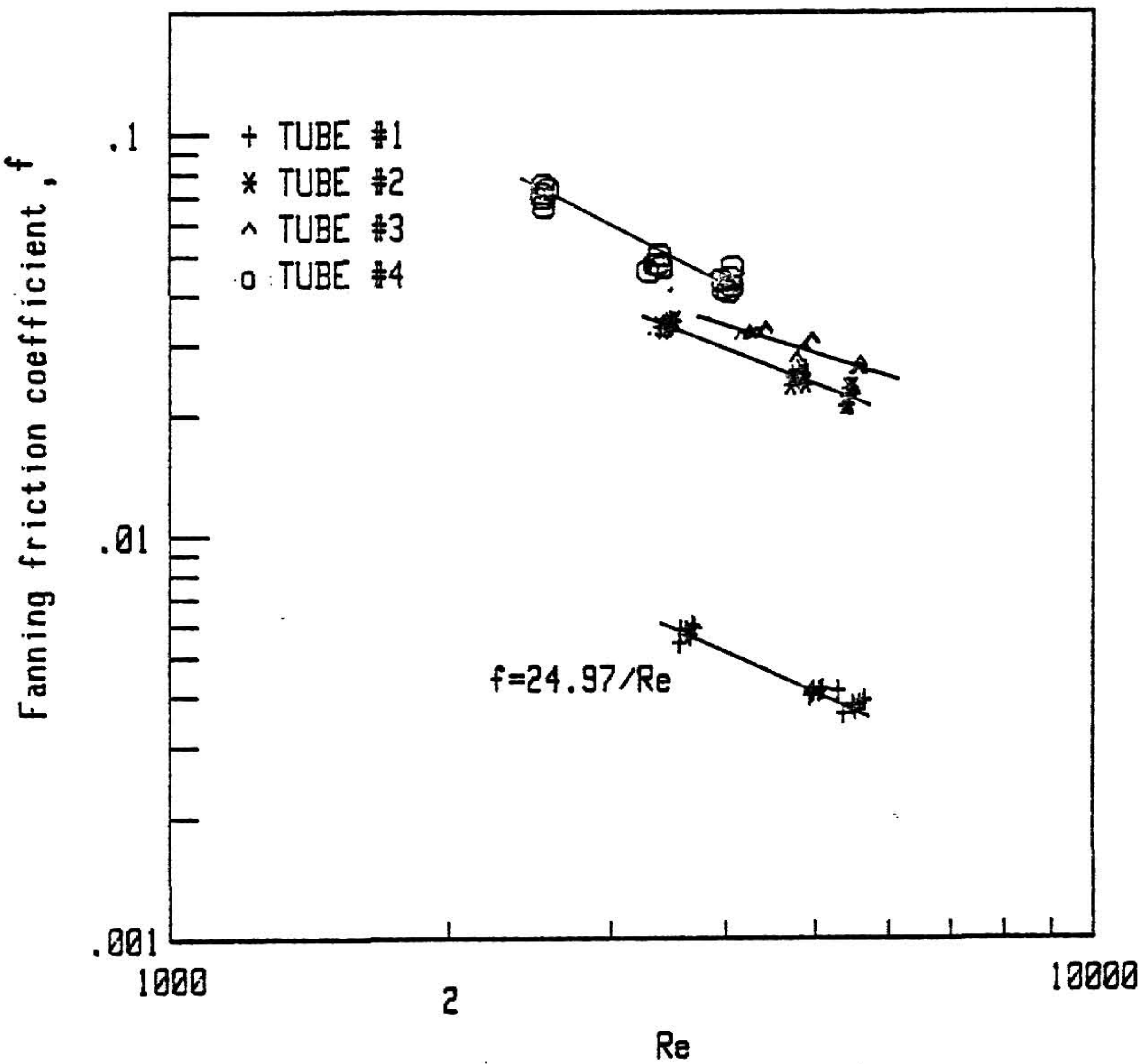

Fig. 4-13 Fanning Friction Coefficient versus Reynolds Number for All Tubes Tested

The attempt was made to develop a friction coefficient for the pressure drop inside the internally finned tubes. The modifying factors originally defined by Carnavos [8] were used as a starting point. Before the final correlation is presented a summary of the work of Carnavos is given in the following.

Carnavos [9] proposed the following correlation for the Fanning friction coefficient for single phase turbulent flow of air inside internally finned tubes.

$$f = \left(\frac{0.046}{Re^{0.20}}\right) F_3^{0.5} F_4^{0.5} \tag{4-17}$$

where, Re is the Reynolds number based on the hydraulic diameter, and

$$F_3 = \cos\alpha \quad , \quad \text{and} \quad F_4 = \frac{A_{fa}}{A_{fn}}$$

A_{fa} was defined earlier as the actual free flow area of the finned tube, and

A_{fn} = nominal flow area based on D_i as if fins were not present,

$$= \frac{\pi D_i^2}{4}$$

It can be shown that

$$F_4 = A_{fa}/A_{fn} = \left[\frac{D_e}{D_i}\right]^2 =$$

$$= [1 - (4nbt)/(\pi D_i^2 \cos\alpha] \tag{4-18}$$

Computed values of F_4 for tubes 2, 3, and 4 of the present study were 0.9174, 0.9376, and 0.9502 respectively. Carnavos [8] also proposed the following correlation for the friction factor based on his work with single phase turbulent flow of water in internally finned tubes.

$$f = \frac{0.046}{Re^{0.2}\ (F^*)} \tag{4-19}$$

where,

$$F^* = (A_{fa}/A_{fn})^{0.5}\ (\text{Sec}\ \alpha)^{0.75} \tag{4-20}$$

None of the correlations of frictional coefficient correlated satisfactorily the experimental measurements of the present study. This is due to the fact that the correlations of Eqs. (4-17) and (4-19) were developed for single phase turbulent flow while the flow was in the transition zone in the present study.

After several trials of using different functional combinations of F_3 and F_4, and using the least square regression analysis the following correlation was obtained for the finned tube.

$$f = (253.94/Re)\ F_4^{10.17} \tag{4-21}$$

where Re was based on D_i rather than D_n as in the heat transfer correlation of Eq. (4-15). Although F_3 (= Cos α) was not included in the final correlation, the effect of the helix angle is included in F_4 given in Eq. (4-18).

Fig. 4-14 shows a comparison between the predicted and experimental Fanning friction coefficient of all tubes tested. The results show that the diagreement is within ± 10%.

4.3 SUBCOOLED BOILING RESULTS

4 .3.1. Heat Transfer

It was mentioned in an earlier section that the subcooled boiling data were obtained, for a given flow rate, by gradually increasing the electric heat input until boiling of R-113 could be observed in the transparent section at the exit of the test section.

Figure 4-15, 4-16, and 4-17 show typical plots of heat flux versus temperature difference for tubes 1, 2, and 3 at velocities 0.1, 0.14, and 0.16 m/s respectively. For the sake of comparison, both single phase heating and subcooled boiling are included in these figures. The curves start at their lower end in pure single phase subcooled liquid and terminate in the subcooled nucleate boiling regime. The results in these figures show that the heat flux increased with the increase in the temperature difference for the smooth tube as well as the finned tubes. They also show that a clear transition point exists between the single phase heating and the subcooled regimes. At the high end of the temperature difference, the curves of the smooth and finned tubes tended to merge into a single curve the so-called fully developed nucleat boiling curve. Such observation suggest that in the nucleate boiling regime the finned tubes had a diminishing effect. Similar observations were reported by Ornatskiy et al. [26], Rohsenow [28] and Azer et al. [1].

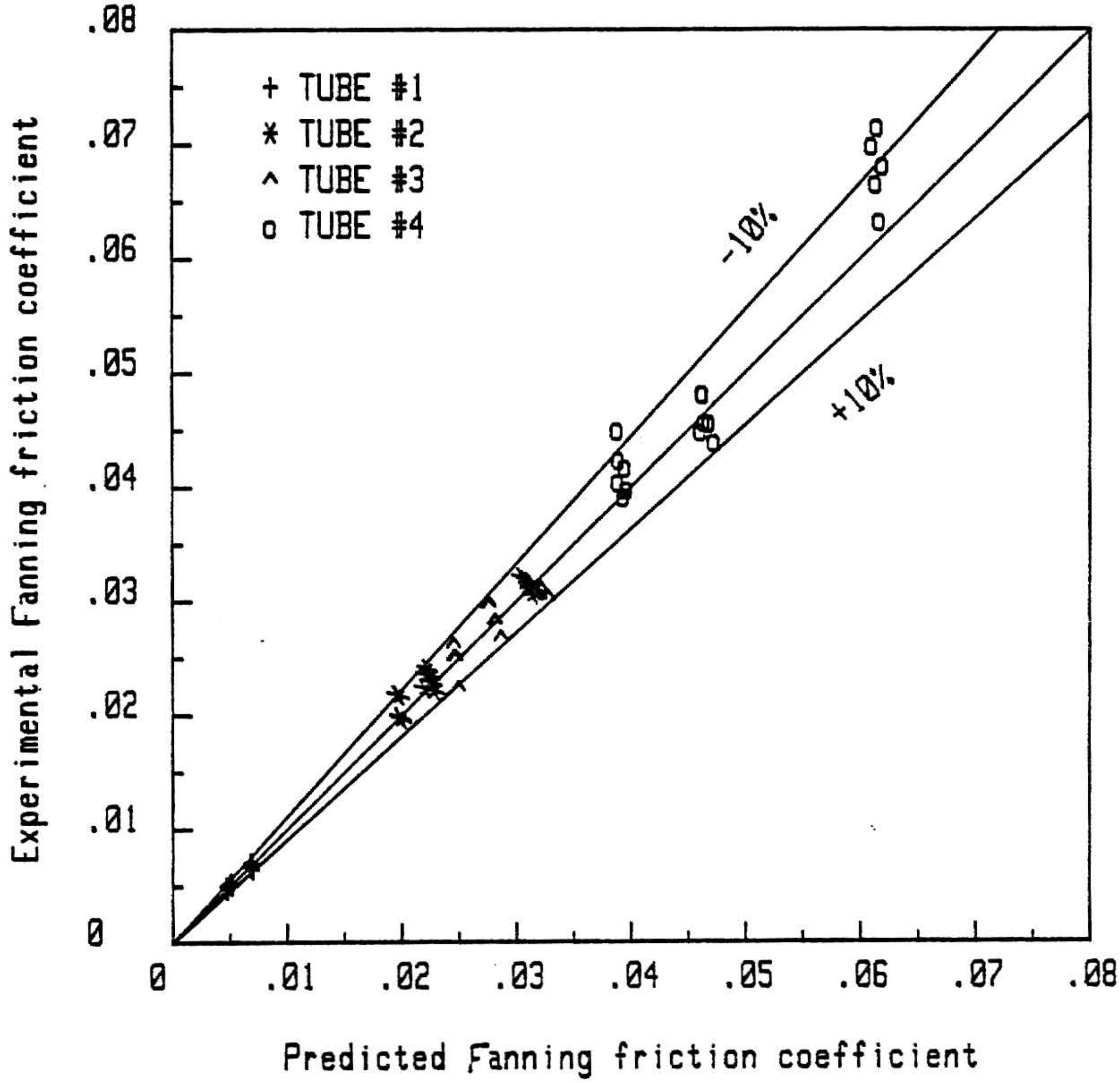

Fig. 4-14 Comparison Between Predicted and Experimental Measurements of the Fanning Friction Coefficient

The heat transfer results of finned tube 4 could not be included in Figs. 4-15, 4-16, and 4-17 due to the fact that the velocities of these figures could not be achieved. Figure 4-18 show a plot of the heat flux versus the temperature difference at three different velocities for tube 4. It is also clear that there is transition point between single phase heating and subcooled boiling regimes. Also, as the velocity increases, at the same temperature difference, the heat flux increases in both regimes.

4.3.2. Pressure Drop

Figure 4-19, 4-20, and 4-21 show plots of pressure drop versus heat flux for tubes 1, 2, and 3 at velocities 0.1, 0.14, and 0.16 respectively. The results indicate for the same velocity and heat flux the pressure drop for the finned tubes 2 and 3 is higher than the smooth tube. The results for tube 4 were not included in these figures because the velocities ranges could not be covered. Figure 4-22 show a plot of the pressure drop versus the heat flux for the finned tube 4 at three different velocities. For the same heat flux as the velocity increases the pressure drop decreases. The reason is due to the fact that as the velocity increases, the mass flow increases and as a result the exit quality decreases if the heat flux remains unchanged. The reduction in exit quality results in a reduction in the pressure drop. Figures 4-23, 4-24, and 4-25 show plots similar to Fig. 4-22 for tubes 1, 2, and 3 respectively. They exhibit the same trend exhibited by Fig. 4-22.

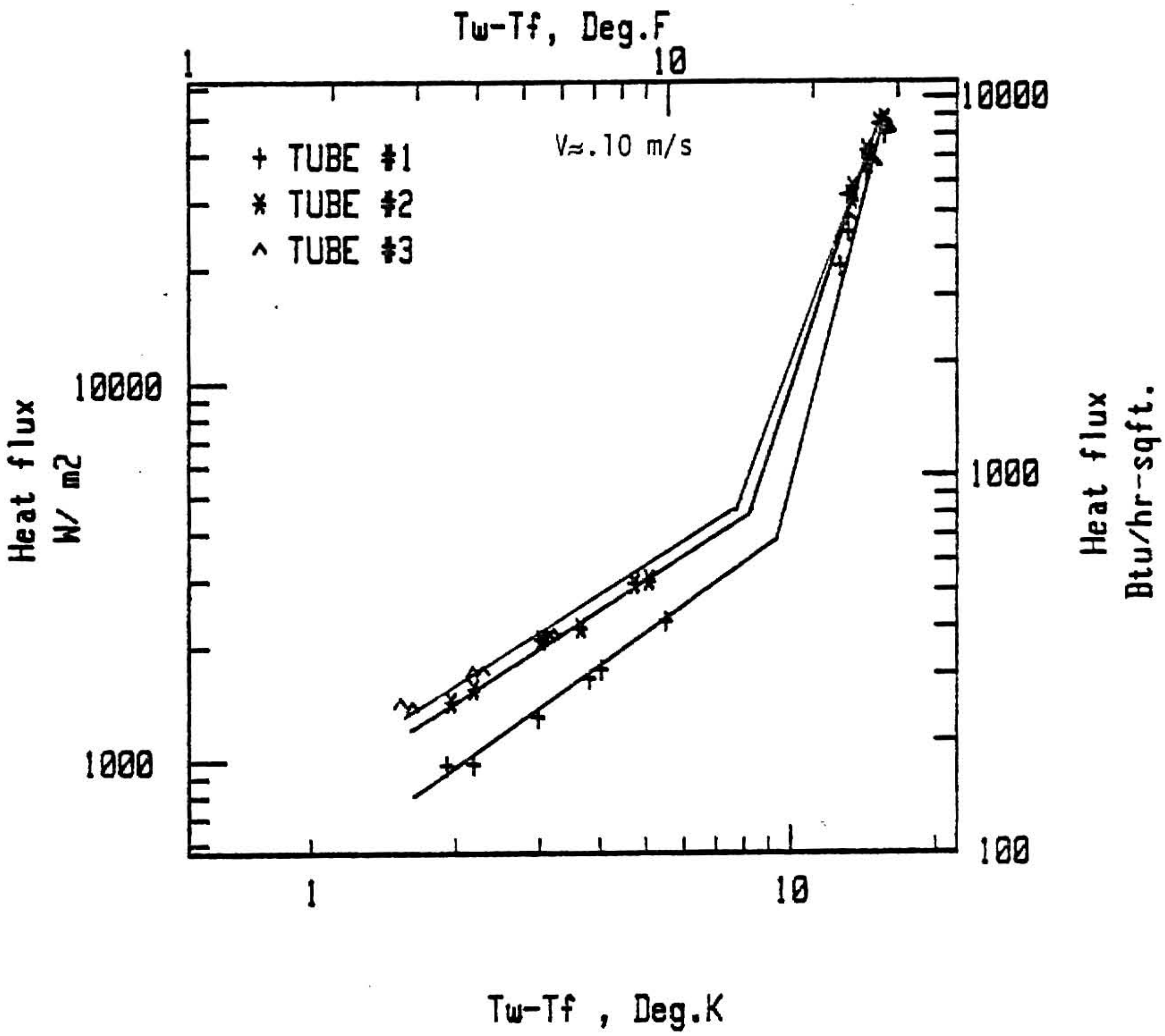

Fig. 4-15 Heat Flux versus Temperature Difference.

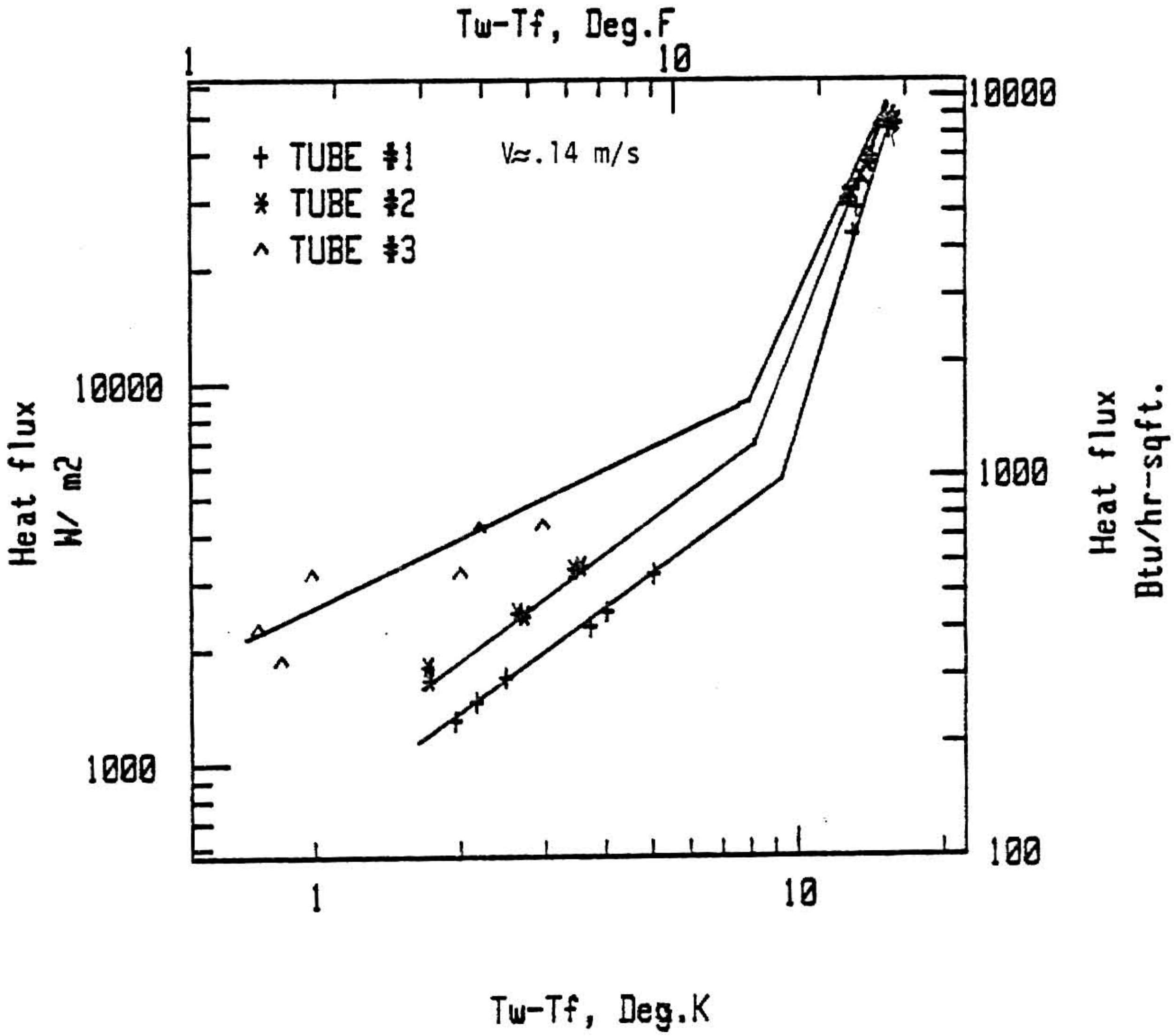

Fig. 4-16 Heat Flux versus Temperature Difference.

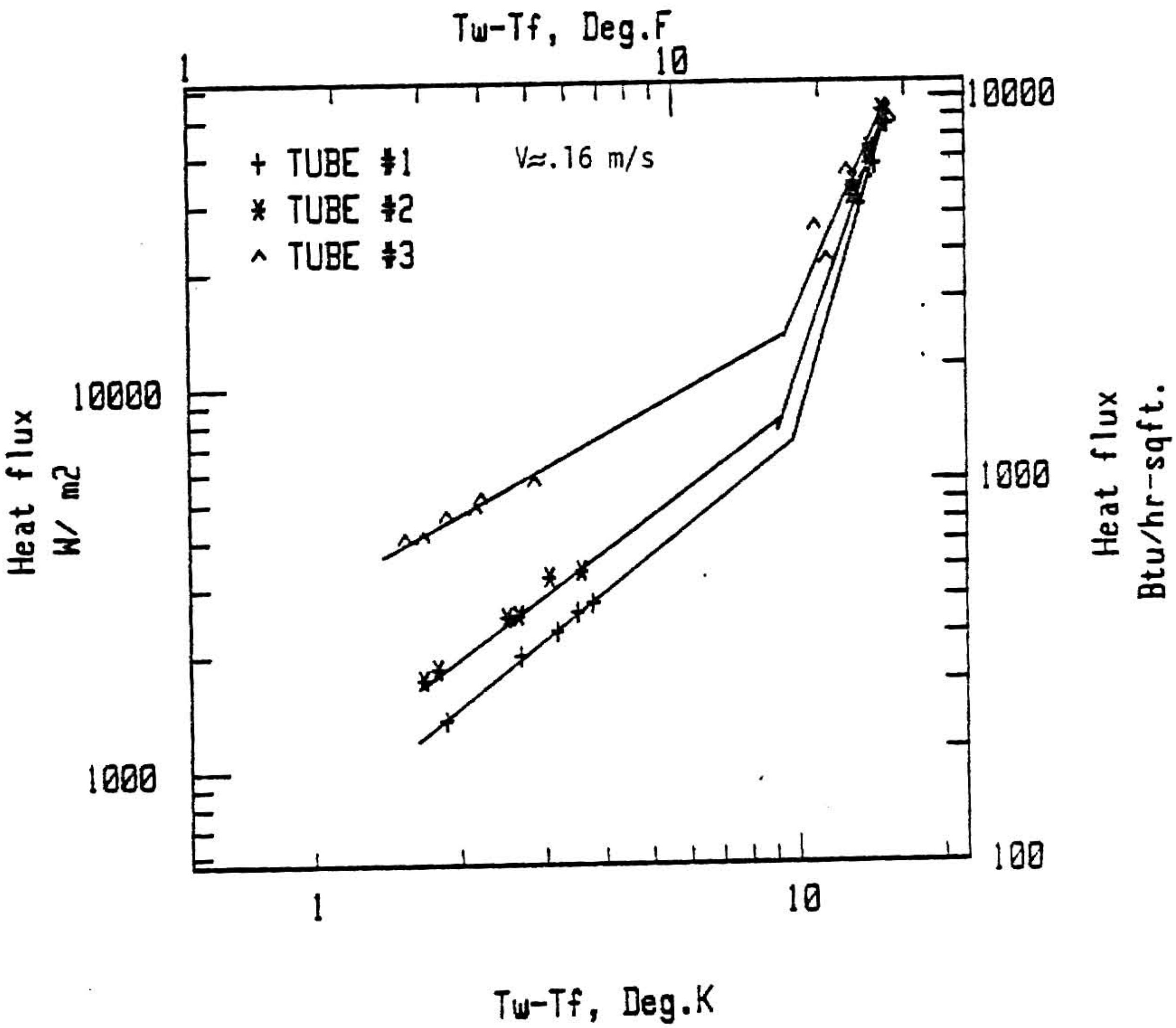

Fig. 4-17 Heat Flux versus Temperature Difference.

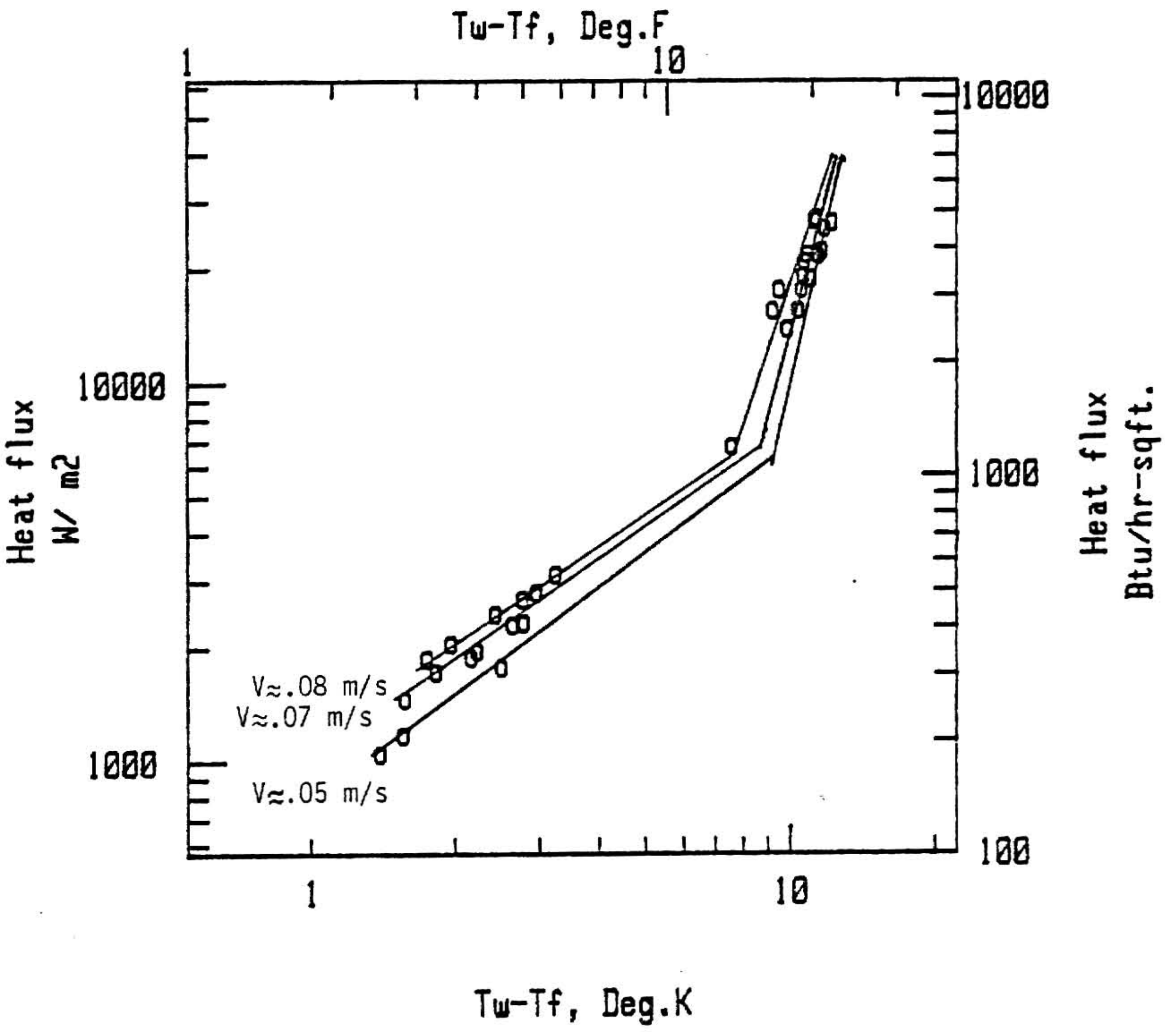

Fig. 4-18 Heat Flux versus Temperature Difference for Tube 4 at Different Velocities.

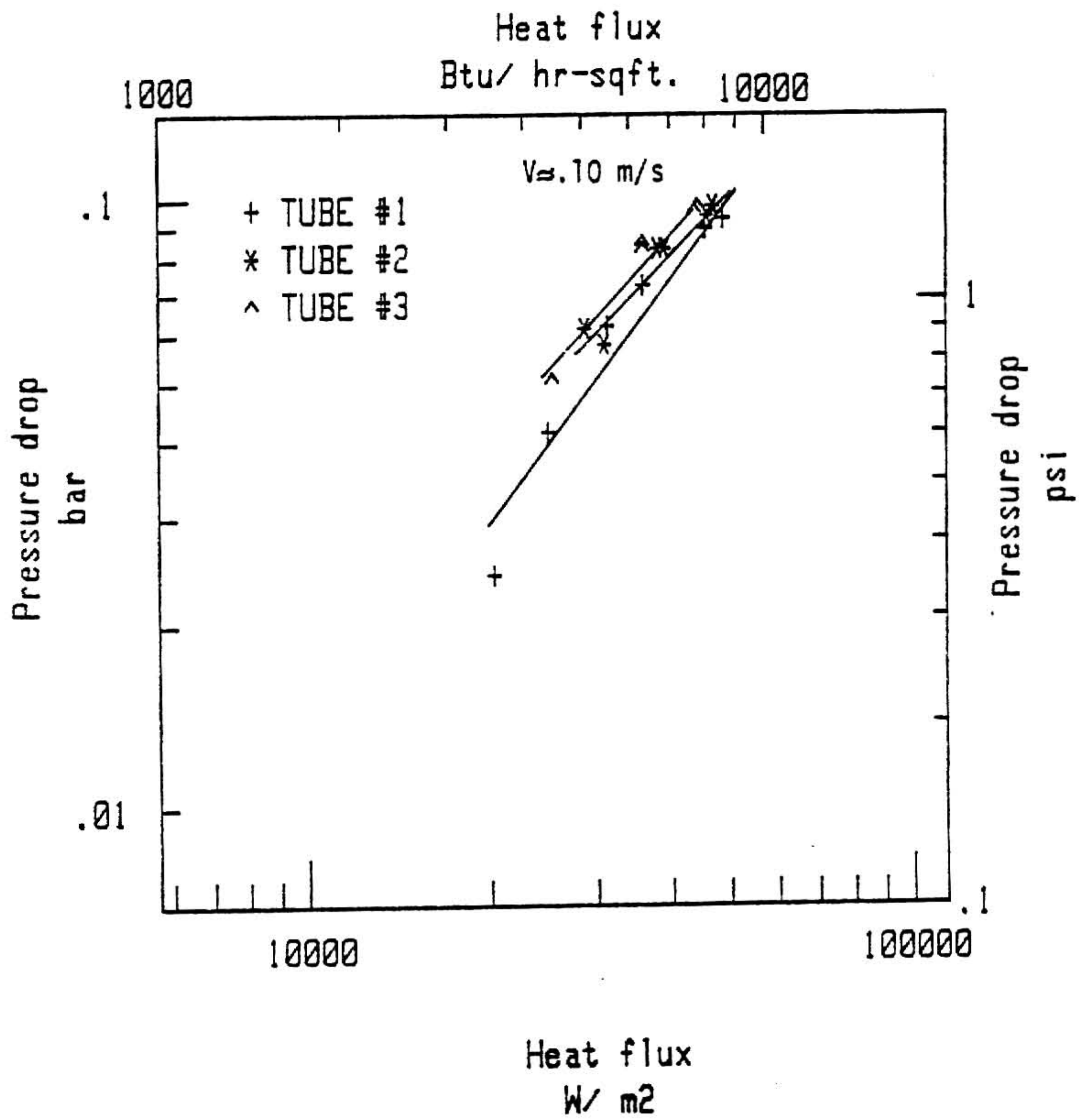

Fig. 4-19 Pressure Drop versus Heat Flux

for Subcooled Boiling

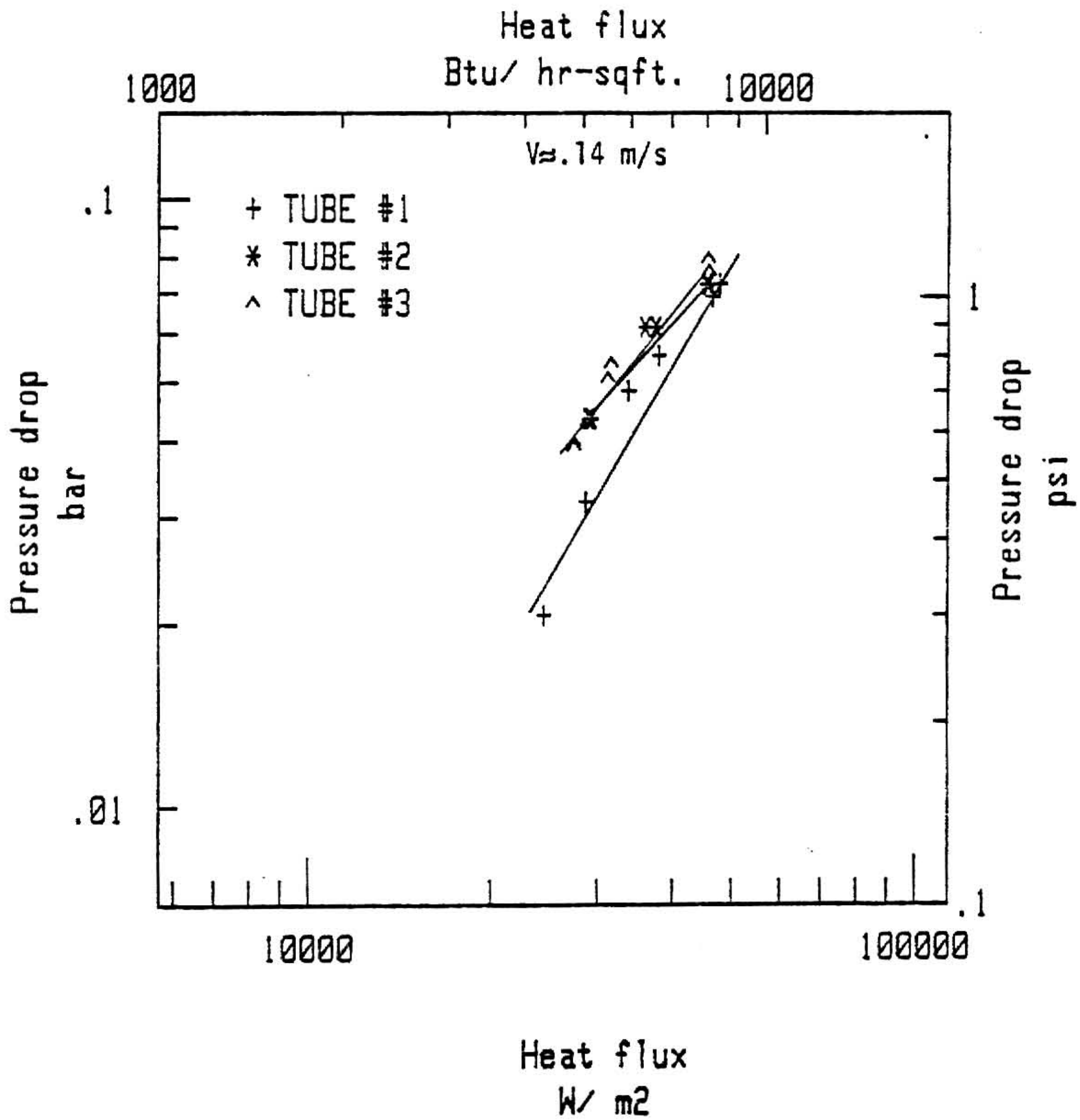

Fig. 4-20 Pressure Drop versus Heat Flux

for Subcooled Boiling

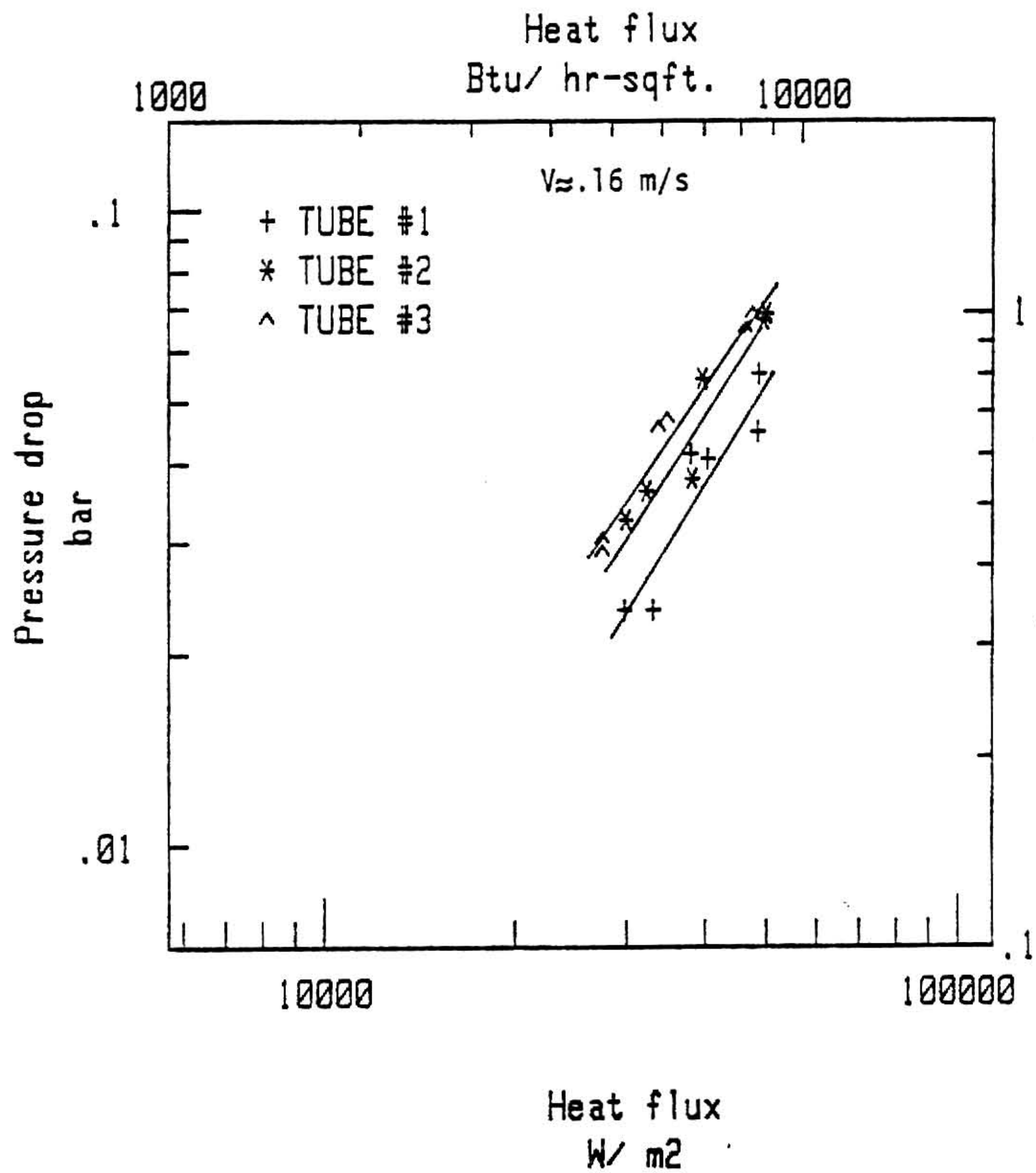

Fig. 4-21 Pressure Drop versus Heat Flux for Subcooled Boiling.

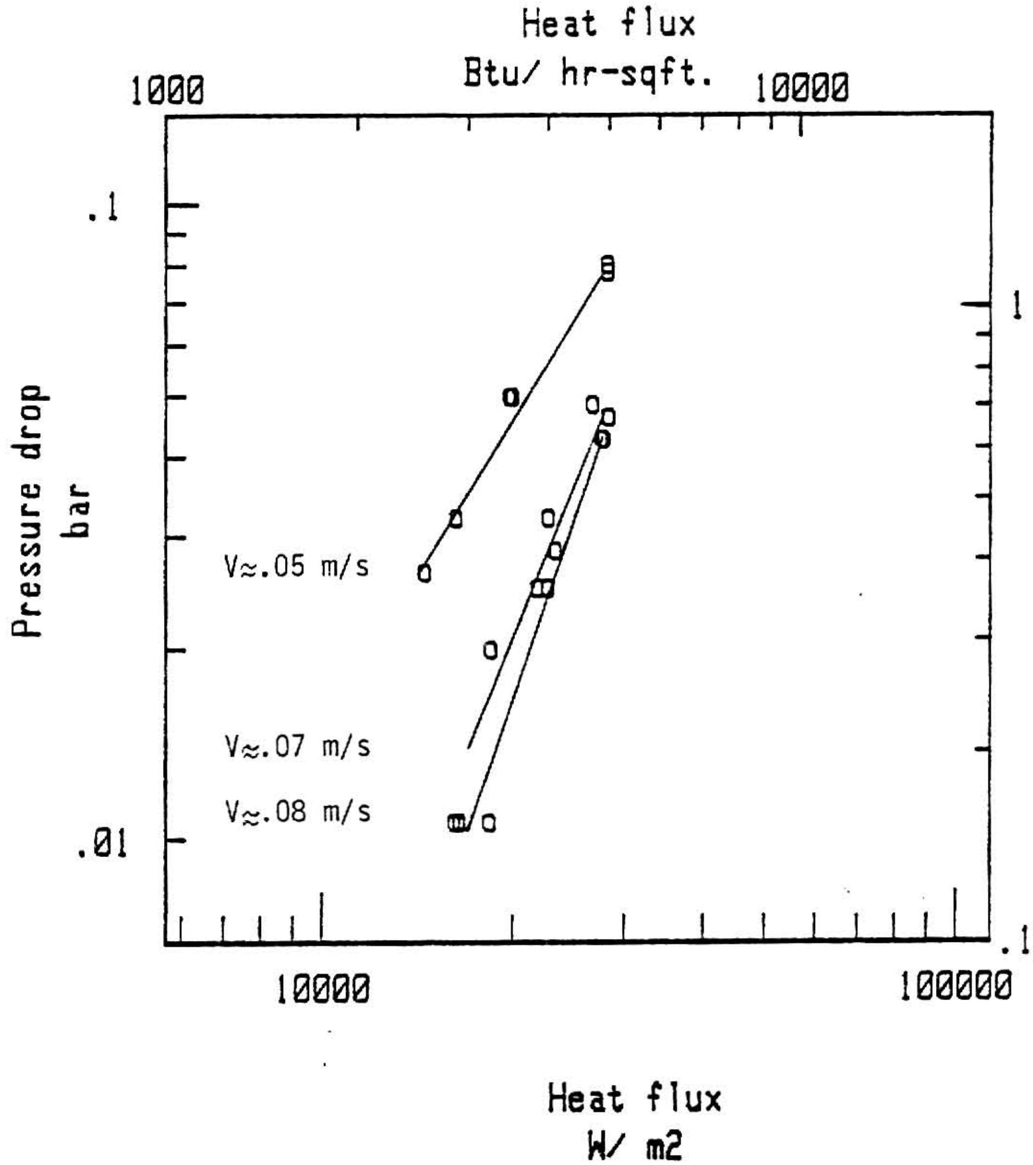

Fig. 4-22 Pressure Drop versus Heat Flux for Subcooled Boiling of Tube 4.

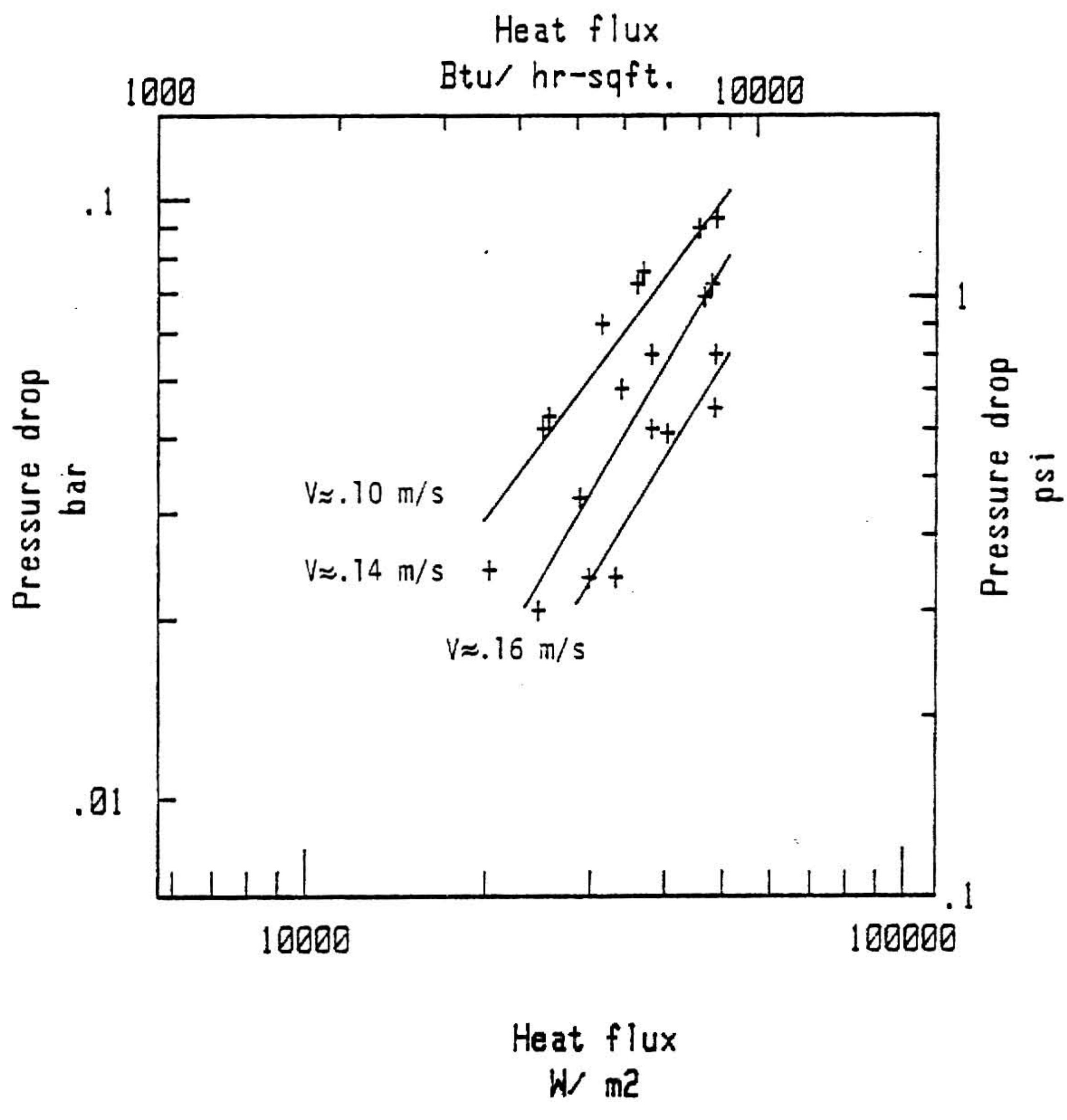

Fig. 4-23 Pressure Drop versus Heat Flux for Subcooled Boiling in Smooth Tube 1 at Different Velocities.

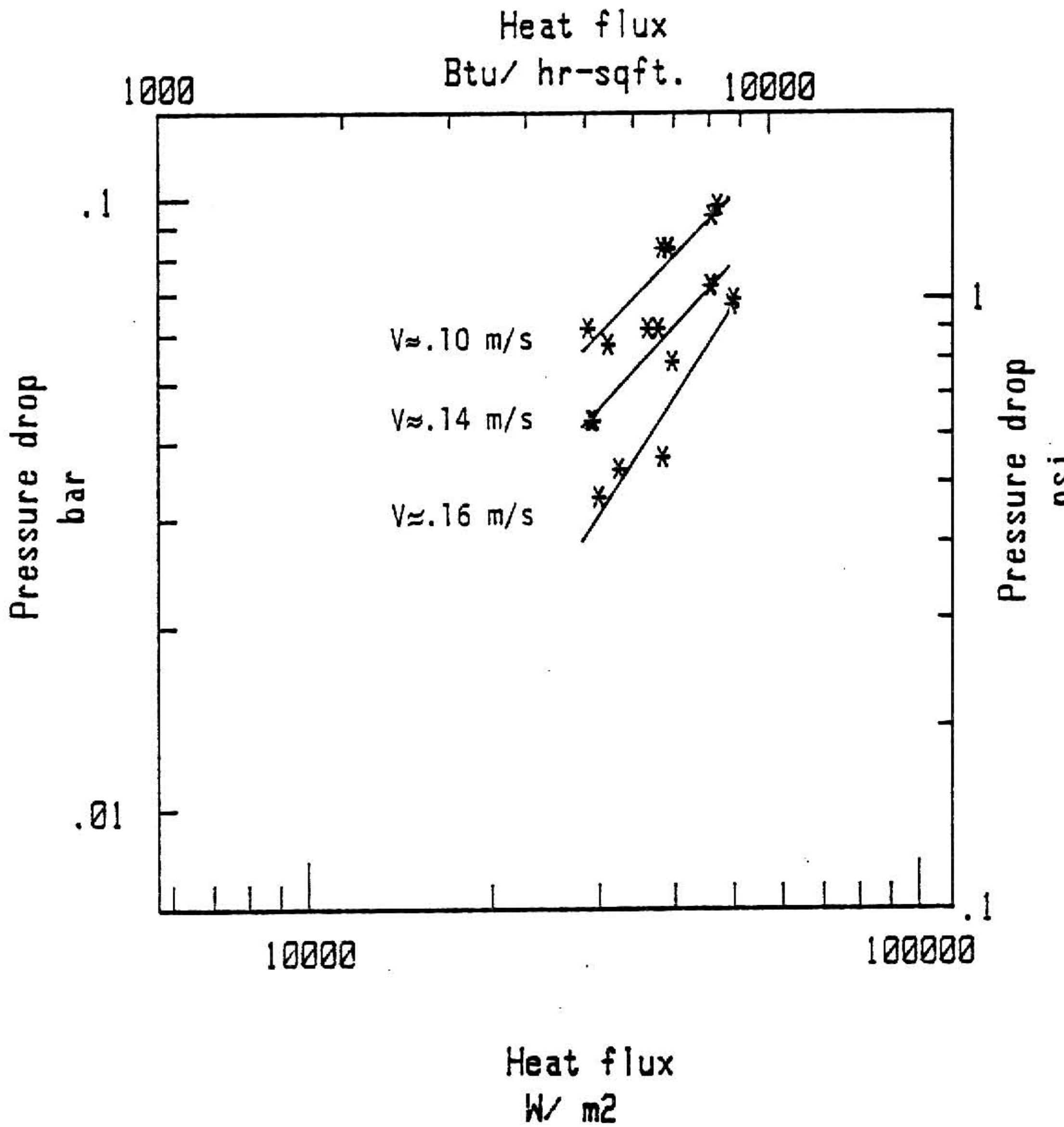

Fig. 4-24 Pressure Drop versus Heat Flux for Subcooled Boiling in Finned Tube 2 at Different Velocities.

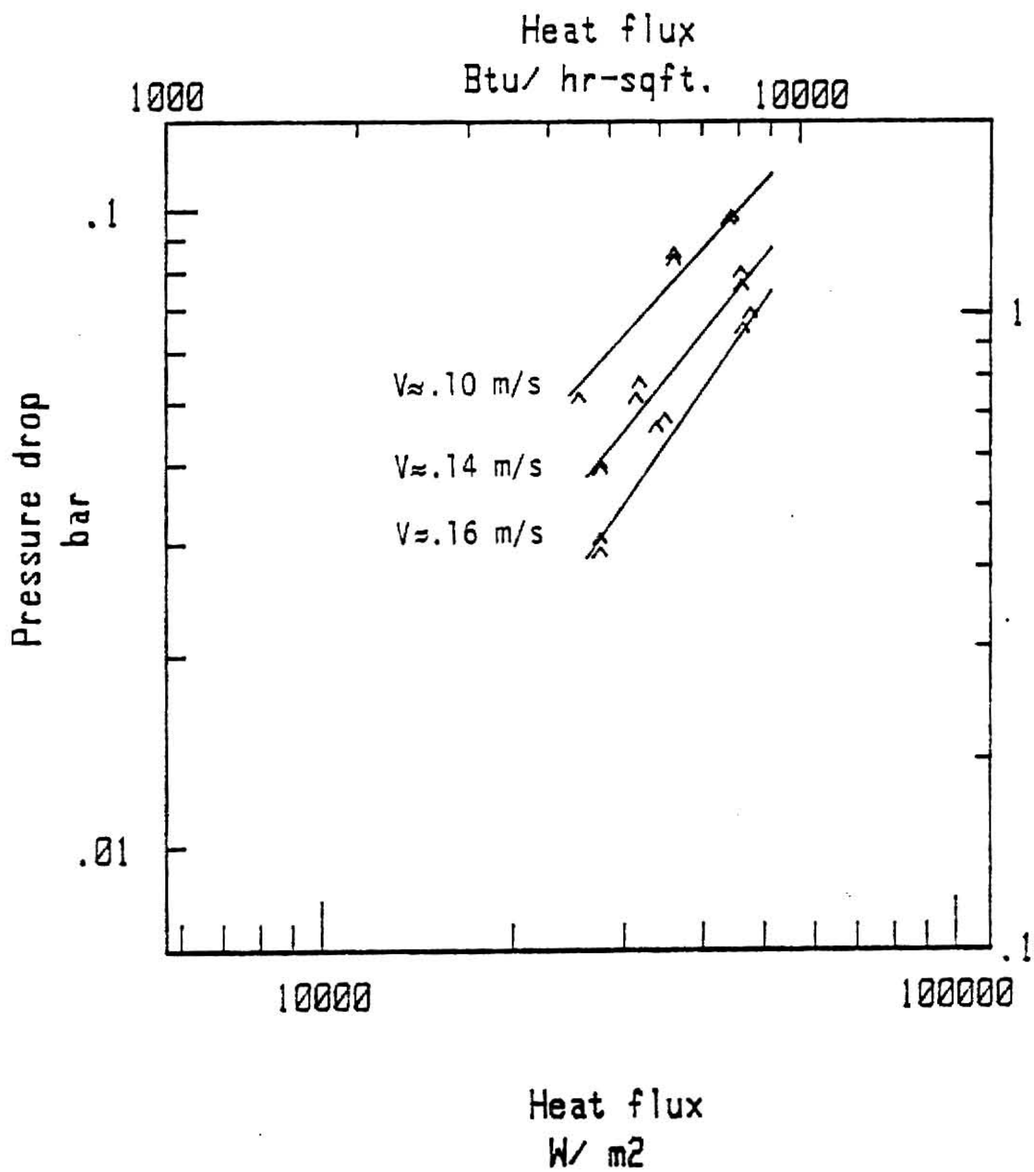

Fig. 4-25 Pressure Drop versus Heat Flux for Subcooled Boiling in Finned Tube 3 at Different Velocities.

CHAPTER V

PERFORMANCE EVALUATION

Any performance evaluation should be directed toward a certain objective. Depending on the objective, each evaluation index is determined by calculating the ratio of certain parameters of interest for the augmented and smooth surfaces, subject to certain constraints.

Bergles et al. [5] suggested eight performance evaluation indices for heat transfer augmentation in single phase flow. These indices were divided into two groups. One group was directed toward the use of heat transfer enhancement for existing heat-exchangers. Thus the basic geometry was fixed. The other group was directed toward evaluating the advantages of using the heat transfer augementation in the design of new heat exchangers. Therefore, the length and size and number of tubes were unrestricted, and the objective was the reduction of the size of heat exchangers.

The criteria suggested by Bergles et al. [5] are summarized below.

1) Basic Geometry Fixed, Flow Rate Fixed-Increase Heat Transfer

the ratio of the heat transfer coefficient h_{aug}, of the augmented surface to its value for the unaugmented surface h_s, is evaluated for constant flow rate m, constant temperature difference ΔT, the same length L and nominal diater D_i, and same inlet temperature T_i is given by

$$R_1 = \frac{q_{aug}}{q_s} = \left(\frac{h_{aug}}{h_s}\right)_{m,\ \Delta T,\ L,\ D_i,\ N,\ T_i} \tag{5-1}$$

q_{aug} and q_s are heat transfer rates for augmented and unaugmented surfaces, respectively.

2) Basic Geometry Fixed, Pressure Drop Fixed-Increase Heat Transfer.

Such criteria can be expressed as follows:

$$R_2 = \frac{q_{aug}}{q_s} = \left(\frac{h_{aug}}{h_s}\right)_{\Delta p,\ \Delta T,\ L,\ D_i,\ N,\ T_i} \qquad (5\text{-}2)$$

ΔP = Pressure drop.

3) Basic Geometry Fixed, Pumping Power Fixed-Increase Heat Transfer.

This criteria is appropriate when the cost of fluid pumping is a major consideration.

$$R_3 = \frac{q_{aug}}{q_s} = \left(\frac{h_{aug}}{h_s}\right)_{\mathbb{P},\ \Delta T,\ L,\ D_i,\ N,\ T_i} \qquad (5\text{-}3)$$

4) Basic Geometry Fixed, Heat Duty Fixed-Reduce Pumping Power

Such criterion can be expressed as follows:

$$R_4 = \left(\frac{\mathbb{P}_{aug}}{\mathbb{P}_s}\right)_{q,\ \Delta T,\ L,\ D_i,\ N,\ T_i} \qquad (5\text{-}4)$$

$\mathbb{P}$ = Pumping Power

q = rate of heat transfer

In this case, the heat transfer coefficient is constant.

5) Heat Duty Fixed, Pumping Power Fixed-Reduce Exchanger Size.

The performance index is given by

$$R_5 = \left(\frac{A_{aug}}{A_s}\right)_{q,\ \Delta T,\ \mathbb{P},\ D_i,\ T_i} = \frac{h_s}{h_{aug}} \qquad (5\text{-}5)$$

6) Heat Duty Fixed, Pressure Drop Fixed-Reduce Exchanger Size.

The desired ratio in this case is

$$R_6 = \left(\frac{A_{aug}}{A_s}\right)_{q,\ \Delta T,\ \Delta P,\ D_i,\ T_i} = \frac{h_s}{h_{aug}} \qquad (5\text{-}6)$$

7) Heat Duty Fixed, Flow Rate Fixed-Reduce Exchanger Size.

The following ratio is of interest for this case:

$$R_7 = \left(\frac{A_{aug}}{A_s}\right)_{q,\ \Delta T,\ m,\ D_i,\ T_i} = \frac{h_s}{h_{aug}} \qquad (5\text{-}7)$$

8) Heat Duty Fixed, Flow Rate Fixed, Pressure Drop Fixed-Reduce Exchanger Size.

The appropriate ratio is

$$R_8 = \left(\frac{A_{aug}}{A_s}\right)_{q,\ \Delta T,\ \Delta P,\ m,\ D_i,\ T_i} = \frac{h_s}{h_{aug}} \qquad (5\text{-}8)$$

several of the above eight criteria include certain constraints which cannot be realized. The first performance index R_1, can be used to evaluate the performance of the finned tubes 2 and 3 in relation to tube 1. The tree tubes had the same geometries.

Figure 5-1 shows a plot of the ratio $\frac{h_{aug}}{h_s}$ for tubes 2 and 3 subject to the constraints of fixed geometry, flow rate, inlet temperature. With an increase in Reynolds number the ratio for finned tube 2 slightly decreased while it increased for tube 3. The reason for such trend is similar to the reason given earlier while discussing the results in Fig. 4-4.

In evaluating the performance of static in-line mixers in augmenting the single phase flow and condensation heat transfer inside horizontal tubes Lin et al. [18, 19] used the ratio of pumping power to the rate of heat transfer $\mathbb{P}/q$ as an evaluation index. The pumping power was obtained from the product of the volumetric flow rate of the liquid at the circulating pump, and the pressure drop across the test section. The constraints under which $\mathbb{P}/q$ was evaluated were: the same mass flow rate, the same inlet temperature, and the same geometry. This index of evaluation was used in Figs. 5-2 and 5-3 with addition of the constraint of constant heat flux.

Figure 5-2 shows a plot of pumping power per unit heat transfer $\mathbb{P}/q$ versus Reynolds number of R-113 for tubes 1, 2, and 3 for single phase flow. The lower this index is, the lower is the power demand per unit heat transfer. The figure shows that the smooth tube is far better than the finned tubes as far as pumping power is concerned; this means that at a constant Reynolds number, $\mathbb{P}/q$ is lower for the smooth tube than for the finned tubes. Tubes 2 and 3 over the range of Reynolds number tested had the same pumping power per unit heat transfer $\mathbb{P}/q$ for the same Re. Also, it should be noticed that with an increase of the Reynolds number, $\mathbb{P}/q$ increases.

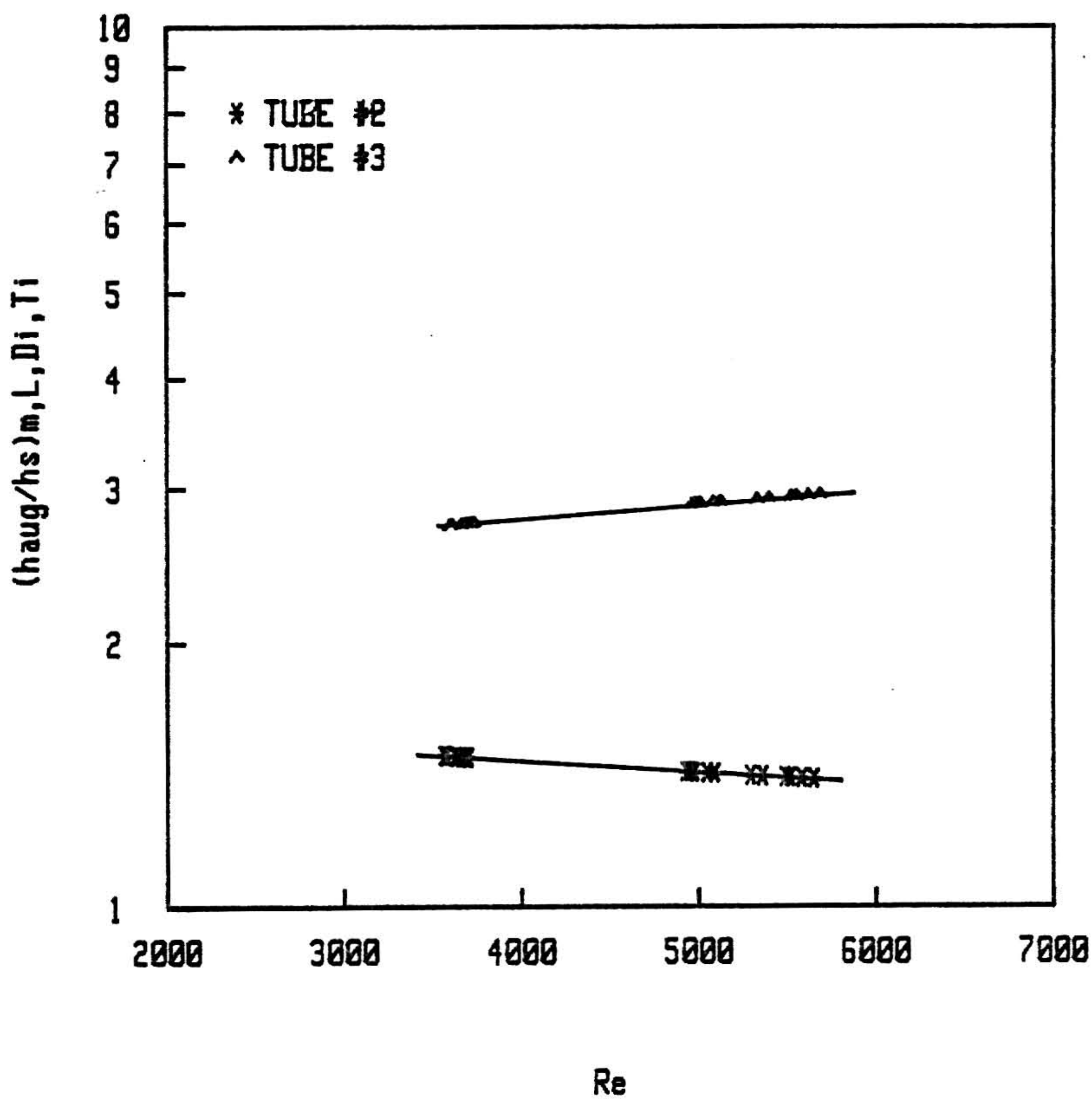

Fig. 5-1 Ratio of the Heat Transfer Coefficient of the Finned Tubes to the Smooth Tube versus Reynolds Number.

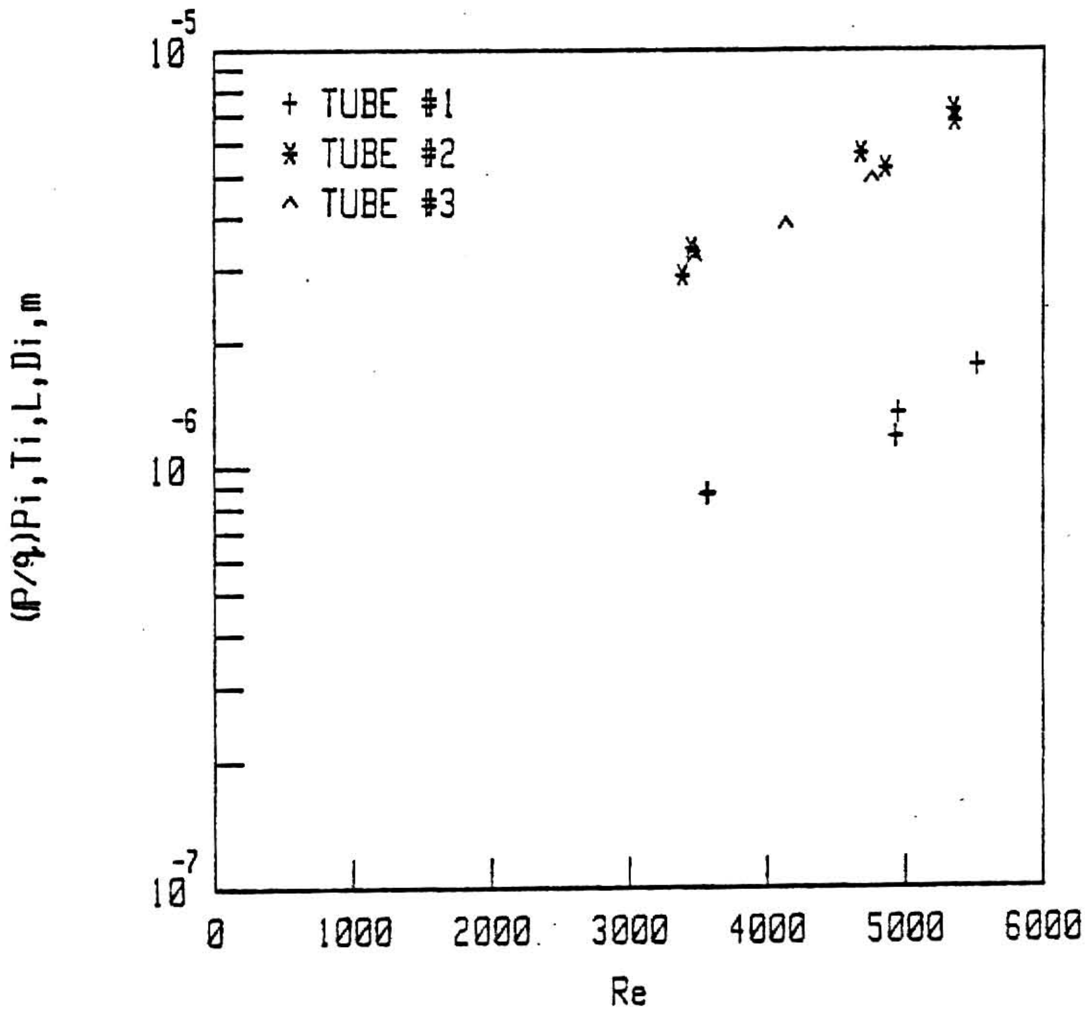

Fig. 5-2 Pumping power per Unit Heat-Transfer Rate versus the Reynolds Number for Single Phase Heating at Q ≈ 6500 Btu/hr-ft^2 (20501 w/m^2)

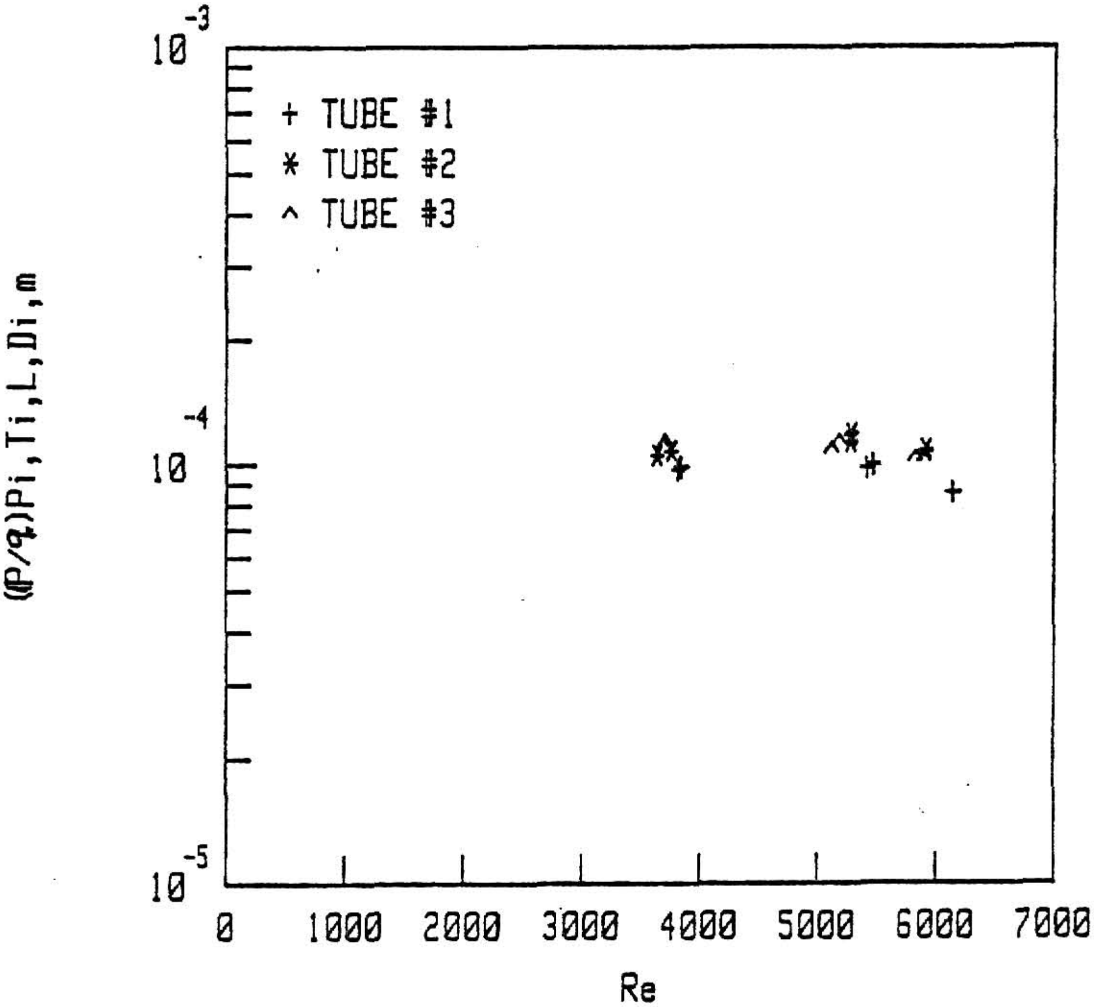

Fig. 5-3 Pumping power per Unit Heat-Transfer Rate versus Reynolds Number for Subcooled Boiling, at $Q \approx 6500$ Btu/hr-ft^2 (20501 w/m^2).

Figure 5-3 shows a comparison of $\mathbb{R}/q$ versus Re for subcooled boiling. The results show that the ratio $\mathbb{R}/q$ is nearly the same for all three tubes. The reason is due to the fact that this ratio is mainly controlled by the pressure drop of each tube. According to the pressure drop results in Figs. 4-19, 4-20, and (4-21) discussed earlier, tube 2 and 3 had nearly the same pressure drop which was slightly higher than tube 1. Additional data are needed to establish the effect of finned tubes on heat transfer enhancement in the subcooled boiling regime.

CHAPTER VI

SUMMARY, CONCLUSION AND RECOMMENDATIONS

In the present study, heat transfer and pressure drop data were taken during single phase heating and subcooled boiling of R-113 inside four vertical electrically heated tubes. One of these was a smooth tube and the remaining three were internally finned tubes. For single phase flow inside internally finned tubes, appropriate modifiers were identified and applied to the smooth tube heat transfer and pressure drop correlations to bring about the best agreement between the measurements and predictions for these tubes.

The results are summarized as follows:

1 - In single phase heating, over Re range tested, an enhancement in the heat transfer coefficient of 50%, 180%, and 140% was obtained by tubes 2, 3, and 4 respectively, over the smooth tube results on nominal area basis.

2 - A correlation equation, Eq. 4-15, was developed for predicting the heat transfer coefficient during single phase heating inside internally finned tubes. It is applicable to a range of $3300 < Re < 5500$.

3 - A pressure drop correlation, Eq. 4-21, for single phase heating inside finned tubes was developed.

4 - In the subcooled boiling regime, the finned tubes had a diminishing effect on the heat flux, at a specified wall to fluid temperature difference. No attempt was made to develop a heat transfer or a pressure drop correlation in this flow regime because of the limited data obtained.

5 - In the subcooled boiling regime, the pressure drop for finned tubes was slightly higher than the smooth tube at the same mass flow rate and heat flux input.

6 - The pumping power per unit heat transfer subject to the constraints of fixed geometry and the same flow rate was used to evaluate the performance of the tubes tested. The smooth tube required the least power in single phase flow.

Recommendations for Future Studies

1 - Additional data are needed for subcooled boiling inside finned tubes to be able to develop heat transfer and pressure drop correlations for these tubes.

2 - Additional data are needed for single phase heating inside finned tubes using other fluids and tubes geometries, over the range of Reynolds number of the present study, to check the generality of the proposed correlations.

3 - Several performance evaluation criteria for single phase flow inside augmented tubes have been suggested. Some of these criteria require certain constraints which cannot be realized experimentally. More attainable performance criteria need to be defined not only for single phase but also for two phase flow such as boiling and condensation.

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APPENDIX A

SAMPLE OF DATA REDUCTION AND PROCEDURE FOR CALCULATING HEAT TRANSFER COEFFICIENT

1. SINGLE PHASE HEATING

Experimental Run No. 1, Smooth Tube (Tube 1) Appendix D Table D-1

R-113 inlet temperature	=	83.88 ^oF
R-113 exit temperature	=	86.36 ^oF
Pump flow rate	=	0.25 GPM
Temperature at flow meter	=	83.84 ^oF
Pressure drop	=	0.0025 PSI

Surface Temperature

T(26,27)*	=	83.3 ^oF
T(28,29)	=	84.74 ^oF
T(30,31)	=	86.63 ^oF
T(32,33)	=	87.55 ^oF
T(34,35)	=	88.25 ^oF
T(36,37)	=	89.69 ^oF
T(38,39)	=	91.85 ^oF
T(40,41)	=	92.21 ^oF
T(42,43)	=	93.02 ^oF

*T(26,27) is the average of thermocouples readings at a given axial location. See Fig. 3-4.

R-113 Density = 97.136 lb/ft^3

Calculation Procedure

Mass flow rate, M_f:

$$M_f = 0.25\ \frac{\text{gallons}}{\text{minute}} \times 60\ \frac{\text{minute}}{\text{hour}} \times 0.13368\ \frac{\text{ft}^3}{\text{gallon}} \times 97.136\ \frac{\text{lbm}}{\text{ft}^3}$$

$$= 194.78\ \text{lbm/hr}$$

Enthalpy of R-113 at inlet = 25.47 Btu/lb

Enthalpy of R-113 at exit = 26.00 Btu/lb

Heat loss to the top by conduction:

$$= \frac{KA}{L} \times (T(40,41) - T(42,43))$$

$$= \frac{227 \times \pi[(\frac{0.625}{24})^2 - (\frac{0.545}{24})^2] \times [93.02 - 92.21]}{(2/12)}$$

$$= 0.563\ \text{Btu/hr}$$

Heat loss to the bottom by conduction:

$$= \frac{KA}{L}(T(26,27) - T(28,29))$$

$$= \frac{227 \times \pi[(\frac{0.625}{24})^2 - (\frac{0.545}{24})^2] \times [84.74 - 83.3]}{(2/12)}$$

$$= 1.00\ \text{Btu/hr}$$

Total Electrical Power Supplied:

$$= 10.68\ \text{Volt} \times 3.1\ \text{Amp.} \times 3.41\ \frac{\text{B/hr}}{\text{w}} \times 0.96 = 108.38\ \text{Btu/hr}$$

Heat gained by R-113:

$$= 194.78(26.0 - 25.47)$$

$$= 103.23$$

Percentage Heat Bal. Error:

$$= \frac{108.38 - 103.23 - 0.563 - 1.00}{108.38} \times 100 = 3.31\%$$

Calculation of Inside Heat Transfer Coefficient

Average R-113 temperature

$$= \frac{83.88 + 86.36}{2} = 85.12^{\circ}F$$

Average outside surface temperature:

$$= (83.3 + 84.7 + 86.6 + 87.4 + 88.3 + 89.7 + 91.9 +$$
$$92.2 + 93.0)/9$$

$$= \underline{88.56}\ ^{\circ}F$$

Temperature drop across the wall thickness:

$$\Delta T = \frac{Q x \ln(D_e/D_i)}{2\pi KL}$$

$$= \frac{106.82 \times \ln(\frac{0.625}{0.545})}{2\pi \times 227 \times 52.5/12}$$

$$= 0.0023\ ^{\circ}F$$

Average inside surface temperature:

$$= 88.56 - 0.0023 = 88.5533\ ^{o}F$$

$$h_i = \frac{Q}{A\ \ \Delta T} = \frac{106.82}{(\pi \times \frac{0.545}{12} \times \frac{52.5}{12})(88.5533 - 85.12)}$$

$$= \frac{106.82}{0.6242 \times 3.43} = 49.89\ \ \text{Btu/hr ft}^2\ ^{o}F$$

Calculation of Fanning Friction Coefficient

$$G = \frac{M_f}{A} = \frac{194.78}{\pi(\frac{0.545}{24})^2}$$

$$= 120233\ \frac{\text{lbm}}{\text{hr ft}^2}$$

$$f = \frac{g_c D \Delta P \rho}{2LG^2}$$

$$f = \frac{32.2\ \frac{\text{ft} \cdot \text{lbm}}{\text{Sec}^2 \cdot \text{lbf}} \times 0.545\ \text{in} \times 0.0025\ \frac{\text{lbf}}{\text{in}^2} \times 144\ \frac{\text{in}^2}{\text{ft}^2} \times 3600\ \frac{\text{Sec}^2}{\text{hr}^2} \times 97.14\ \frac{\text{lbm}}{\text{ft}^3}}{2 \times 50.5\ \text{in} \times (120233)^2\ \frac{\text{lbm}^2}{\text{hr}^2\text{ft}^4}}$$

$$f = 0.0054$$

2. SUBCOOLED BOILING

Experimental Run No. 1, Smooth Tube (Tube 1) Appendix D Table D-2

R-113 inlet temperature	= 92.43 ^{o}F
R-113 exit temperature	= 128.03 ^{o}F
Pump flow rate	= 0.25 GPM
Temperature at flow meter	= 93.2 ^{o}F
Pressure drop	= 0.6 PSI

Surface Temperature

T(26,27)*	= 153.68 ^{o}F
T(28,29)	= 137.57 ^{o}F
T(30,31)	= 122.63 ^{o}F
T(32,33)	= 125.60 ^{o}F
T(34,35)	= 127.85 ^{o}F
T(36,37)	= 132.26 ^{o}F
T(38,39)	= 134.42 ^{o}F
T(40,41)	= 134.15 ^{o}F
T(42,43)	= 137.30 ^{o}F
R-113 Density	= 92.57 lbm/ft^3

Calculation Procedure

Mass flow rate, M_f:

$$M_f = 0.25 \frac{gal.}{min.} \times 60 \frac{min}{hr} \times 0.13368 \frac{ft^3}{gal.} \times 92.57 \frac{lbm}{ft^3} = 185.62 \frac{lbm}{hr}$$

Enthalpy of R-113 at inlet = 27.33 Btu/lbm

Enthalpy of R-113 saturated liquid at exit, h_f = 35.29 Btu/lbm

Enthalpy of R-113 saturated vapor at exit, h_g = 97.59 Btu/lbm

Heat loss to the top by conduction:

$$= \frac{KA}{L} \times (T(40,41) - T(42,43))$$

$$= \frac{227 \times \pi[(\frac{0.625}{24})^2 - (\frac{0.545}{24})^2] \times [137.3 - 134.15]}{(2/12)}$$

$$= 2.19 \text{ Btu/hr}$$

Heat loss to the bottom by conduction:

$$= \frac{KA}{L} \times (T(26,27) - T(28,29)$$

$$= \frac{227 \times \pi[(\frac{0.625}{24})^2 - (\frac{0.545}{24})^2] \times [153.68 - 137.57]}{(2/12)}$$

$$= 11.20 \text{ Btu/hr}$$

Total Electrical Power Supplied:

$$= 50.1 \text{ vol.} \times 16.8 \text{ amp.} \times 3.41 \frac{\text{B/hr}}{\text{w}} \times 0.96$$

$$= 2755.32 \text{ Btu/hr}$$

Heat gained by R-113:

$$= 2755.32 - (11.2 + 2.19)$$

$$= 2741.93 \quad \text{Btu/hr}$$

Dryness fraction at Exit:

$$2741.93 \frac{Btu}{hr} = 185.62 \frac{lbm}{hr} \{[97.59 X + (1-x)35.29]-27.33\} \frac{Btu}{lbm}$$

$$X = 0.109$$

Average outside surface temperature:

$$= (153.68 + 137.57 + 122.63 + 125.60 + 127.85 + 132.26 +$$

$$134.42 + 137.3)/9$$

$$= 133.94\ ^{o}F$$

Temperature drop across the wall thickness:

$$\Delta T = \frac{Q \cdot \ln (\frac{D_e}{D_i})}{2\pi KL}$$

$$= \frac{2741.93 \times \ln (\frac{0.625}{0.545})}{2\pi \times 227 \times 52.5/12}$$

$$= 0.06$$

Average inside surface temperature:

$$= 133.94 - 0.06 = 133.88\ ^{o}F$$

The wall-fluid Temperature difference, $(T_w - T_f)$, can be calculated as:

$$(T_w - T_f) = 133.88 - (\frac{92.43 + 128.03}{2}) = 23.65\ ^{o}F$$

$$h_i = \frac{Q}{A\ \Delta T} = \frac{2741.93\ \text{B/hr}}{(\pi \times \frac{0.545}{12} \times \frac{52.5}{12})(23.65)}$$

$$= 185.73\ \frac{\text{Btu}}{\text{hr ft}^2\ {}^{o}\text{F}}$$

APPENDIX B

NOMENCLATURE

Symbol	
A	Heat transfer area or heat conduction area, m^2 (ft^2)
A_a	Actual heat transfer area, m^2/m (ft^2/ft)
A_{fa}	Actual free flow area, m^2 (ft^2)
A_{fc}	Open core free flow area, m^2 (ft^2)
A_{fn}	Nominal flow area based on inside tube diameter as if fins were not present, m^2 (ft^2)
A_n	Nominal heat transfer area based on inside tube diameter as if fins were not present, m^2/m (ft^2/ft)
b	Fin height
D_e	Diameter of internally finned tube if fins melted down, m (ft)
ΔT	Temperature difference ($T_w - T_f$), ^oC (^oF)
D_h	Hydraulic diameter ($D_h = 4\ A_{fa}/A_a$), m (ft)
D	Tube diameter, m, ft
f	Fanning friction factor
F	Modifying factor in Eq. (4-5)
F_1	Parameter defined in Eq. (4-7)
F_2	Parameter defined in Eq. (4-8)
F_3	Parameter defined in Eq. (4-9)
F_4	Parameter defined in Eq. (4-17)
G	Mass flux, kg/hr m^2 ($lbm/hr\text{-}ft^2$)

Symbol

$\bar{h}$	Heat transfer coefficient, w/m^2 ^{o}C (Btu/hr-ft^2 ^{o}F)
K	Thermal conductivity of test tube, w/m ^{o}C (Btu/hr-ft ^{o}F)
L	Length of test tube, m (ft)
M_f	Mass flow rate of R-113, kg/hr (lbm/hr)
Nu	Nusselt number
$\mathcal{P}$	Pitch of fin (length per turn), m (ft)
ΔP	Pressure drop, bar (PSI)
n	Number of fins
Pr	Prandtle number of saturated liquid
$\mathbb{P}$	Pumping Power
q	Heat transfer rate to the coolant, w (Btu/hr)
Q	Heat Flux w/m^2 (Btu/hr-ft^2)
Re	Reynolds number
t	Fin thickness, m (in)
T	Temperature ^{o}C(^{o}F)
T_w	Average inside Wall temperature
T_f	Average temperature of R-113 ^{o}C (^{o}F)
V	Inlet velocity of R-113, m/sec. (ft/sec.)
X	Dryness fraction (ratio of vapor mass to total mass)
W	Average distance between fins, m (ft)

Greek Letters

α	Spiral fin tube helix angle (angle between fin and tube axis), degrees
μ	Dynamic viscosity, kg/hr-m (lbm/ft-hr)
ρ	Density, kg/m^3 (lbm/ft^3)

Subscripts

aug	Augmented surface
Cal	Calculated
e	Equivalent
exp	Experimental
i	Inside
in	Inlet
Out	Outlet
s	Smooth surface
w	Wall
f	Fluid
fin	Finned surface

APPENDIX C

COMPUTER PROGRAM FOR DATA REDUCTION

This program was written for H.P. 9845B

```
10     PRINTER IS 0
20     DIM T(21),F(71),Re(71),Nop(71),Hfin(71),Q(71),Prdiff(71),F(71),Deltat(71)
30     N=17
40     PRINT LIN(7)
50     IMAGE 8X,AAAA,3X,AAA,2X,AAA,3X,AAAAAA,3X,A,5X,AA,5X,AA,2X,AAAAAAAA,3X,A,6X
,AAA
60     PRINT USING 50;"Tube","Run.","Flo","Prdiff","f","Re","Hi","Nu/Pr^.4","Q","
Q/A"
70     PRINT LIN(1)
80     DATA 28.8,28.82,30.2,33.9,33.9,33.0,33.9,33.1,33.4,32.0,32.1,31.3,31.2,30.
8,30.7,30.4,30.3,29.5,29.1,28.5,28.5,.25,10.68,3.1,.0025
90     DATA 29.0,30.3,32.53,39.0,39.0,37.6,37.6,37.9,37.9,36.5,36.6,35.6,35.6,34.
7,34.6,33.5,33.4,31.9,31.5,30.5,30.5,.25,13.48,4.2,.00260
100    DATA 29.2,30.7,33.85,41.8,41.8,40.3,41.7,41.1,41.5,39.4,39.5,38.4,38.3,37.
2,37.1,35.7,35.6,33.6,33.0,31.9,31.9,.25,15.3,5.2,.00272
110    DATA 28.60,28.50,29.85,33.7,33.7,32.9,33.6,32.7,33.1,31.9,31.9,31.3,31.3,3
0.8,30.7,30.2,30.3,29.6,29.3,28.7,28.7,.25,10.4,3.2,.0025
120    DATA 28.6,29.83,31.61,36.8,36.8,35.6,35.6,35.7,35.2,34.6,34.7,34.0,33.9,33
.1,33.0,32.2,32.3,31.4,31.0,30.3,30.3,.25,11.77,3.8,.00267
130    DATA 29.9,30.85,33.25,39.7,39.7,38.3,39.5,38.5,39.1,37.2,37.4,36.4,36.3,35
.5,35.4,34.2,34.1,32.8,32.4,31.5,31.5,.25,13.5,4.4,.00275
140    DATA 28.3,27.53,28.98,31.4,31.4,31.4,31.4,31.1,31.1,30.5,30.5,30.2,30.2,30
.2,30.1,30.2,30.1,29.9,29.6,29.2,29.2,.35,12.29,4.0,.00363
150    DATA 28.7,29.10,31.36,36.8,36.8,35.6,36.5,35.5,35.7,34.3,34.3,33.7,33.6,33
.4,33.3,33.0,32.8,32.2,31.8,31.0,31.0,.35,15.22,5.1,.00370
160    DATA 29.1,32.60,35.65,45.2,45.2,43.1,44.7,42.9,43.5,40.4,40.5,38.8,38.7,37
.5,37.4,36.1,36.0,34.7,34.2,33.1,33.1,.35,17.77,6.0,.00372
170    DATA 28.8,28.00,29.25,31.5,31.5,31.4,31.4,31.0,31.3,30.6,30.7,30.5,30.4,30
.5,30.4,30.4,30.5,30.2,29.9,29.4,29.4,.35,12.21,3.6,.00370
180    DATA 28.6,28.14,29.92,32.9,32.9,32.0,32.6,32.2,32.4,31.6,31.7,31.3,31.4,31
.3,31.3,31.2,31.3,30.8,30.6,29.9,29.9,.35,13.61,4.2,.00371
190    DATA 28.9,29.70,32.14,38.5,38.5,37.0,38.2,36.7,37.1,35.3,35.4,34.6,34.5,34
.2,34.1,33.6,33.5,32.7,32.3,31.4,31.4,.35,16.04,5.3,.00375
200     DATA 26.5,25.85,27.05,29.2,29.2,28.6,29.0,28.7,28.9,28.3,28.3,28.1,28.1,2
8.1,28.1,28.3,28.2,28.1,27.9,27.3,27.3,.4,12.43,3.7,.0044
```

```
210   DATA 27.1,26.43,28.44,31.9,31.9,31.0,31.7,31.3,31.4,30.6,30.7,30.4,30.4,30
.4,30.4,30.4,30.4,30.1,29.7,29.1,29.1,.4,15.02,5.2,.00450
220   DATA 27.4,27.20,29.60,33.9,33.9,32.8,33.7,33.1,33.3,32.3,32.4,31.9,32.0,32
.0,32.0,31.8,31.7,31.2,30.9,30.0,30.0,.4,16.56,5.6,.00460
230   DATA 23.4,23.03,24.77,27.9,27.9,26.9,27.6,27.1,27.4,26.6,26.7,26.4,26.3,26
.5,26.4,26.3,26.3,26.1,25.8,25.0,25.0,.4,14.39,4.7,.00432
240   DATA 24.2,24.62,26.88,31.1,31.1,29.9,30.7,30.2,30.4,29.4,29.5,29.2,29.1,29
.0,29.0,28.8,28.7,28.3,28.0,27.0,27.0,.4,16.58,5.3,.0045
250   DATA 32.8,32.38,55.54,63.3,63.3,60.7,60.7,59.8,58.6,56.2,56.7,53.6,53.7,53
.0,53.4,53.5,51.9,56.0,55.2,56.9,56.9,.4,57.9,19.2,.34
260   DATA 34.6,34.41,57.72,66.5,66.5,63.9,63.8,63.5,62.0,60.3,60.8,57.0,57.5,56
.4,56.8,56.3,56.4,58.0,56.9,60.1,60.1,.4,63.7,21.3,.59
270   DATA 36.6,37.24,60.8,70.7,70.7,65.7,67.2,67.2,64.8,64.6,65.6,61.4,61.9,60.
5,61.4,60.3,60.3,61.6,60.4,64.7,64.7,.4,70.3,23.2,.8
280   DATA 34.8,34.35,54.3,61.2,61.2,57.4,59.3,58.7,57.6,55.4,55.8,52.9,53.1,52.
7,53.2,53.0,51.6,59.3,60.5,68.6,68.6,.4,54.9,18.3,.34
290   DATA 36.2,35.75,56.34,65.0,65.0,61.1,62.7,62.6,61.2,59.9,60.5,57.1,57.5,56
.6,57.1,56.3,54.5,59.6,60.2,66.7,66.7,.4,62.0,20.6,.6
300   DATA 39.6,38.9,60.25,70.6,70.6,66.1,67.5,67.1,64.8,65.1,65.9,62.3,62.9,61.
9,62.7,61.7,61.7,63.3,62.4,66.4,66.4,.4,70.1,23.2,.65
310   FOR Z=1 TO 17
320   Tin=0
330   Tout=0
340   Tav=0
350   Tfav=0
360   Tfflo=0
370   Tflo=0
380   FOR J=1 TO 21
390   T(J)=0
400   NEXT J
410   READ Tfflo,Tin,Tout
```

```
420   CALL R(Tin,Tfin)
430   CALL R(Tout,Tfout)
440   CALL R(Tfflo,Tflo)
450   Tsurf=0
460   FOR I=4 TO 21 STEP 2.0
470   READ T(I),T(I+1)
480   T(I)=(T(I)+T(I+1))/2.0
490   Tf=0
500   CALL R(T(I),Tf)
510   Tsurf=Tsurf+Tf
520   NEXT I
530   Tavsurf=Tsurf/9
540   D=104.07-.0827*Tflo
550   Dl=104.07-.0827*Tfout
560   Df=104.07-.0827*Tavfl
570   Dv=.00005783491*Tfout^2.0-.0055223269*Tfout+.3108693
580   K=4.8457142857E-2-6.5535714286E-5*Tfout
590   Cp=.2122857+.0002595*Tavfl-5.9523809E-7*Tavfl^2.0
600   Cl=.2122857+.0002595*Tfout-5.9523809E-7*Tfout^2.0
610   Hfg=71.87867787-7.46960784E-2*Tfout
620   C=1.399E-3*(D-Dv)*(487.25-(Tout+273)-.9)^.925*6.85251E-5
630   READ Pfl
640   Mf=Pfl*60*.13368*D
650   Hin=7.308+.2165*Tfin
660   Hsl=7.308+.2165*Tfout
670   Hsv=78.8043+.14643*Tfout
680   T(4)=(T(4)+T(5))/2.0
690   CALL R(T(4),Tf1)
700   T(6)=(T(6)+T(7))/2.0
710   CALL R(T(6),Tf2)
720   T(20)=(T(20)+T(21))/2.0
730   CALL R(T(20),Tf3)
740   T(18)=(T(18)+T(19))/2.0
750   CALL R(T(18),Tf4)
760   Htlbot=227*(PI*((.625/24)^2.0-(.545/24)^2.0))*ABS(Tf1-Tf2)*6
770   Htltop=227*(PI*((.625/24)^2.0-(.545/24)^2.0))*ABS(Tf3-Tf4)*6
```

```
780   READ V,A
790   Electra=V*A*3.41*.96
800   Heatgain=Electra-Htltop-Htlbot
810   Dryness=(Heatgain/Mf+Hin-Hsl)/(Hsv-Hsl)
820   Hgainf=Mf*(Hsl-Hin)+Htlbot+Htltop
830   Hbe=(Electra-Hgainf)/Electra*100
840   READ Prtrans
850   Prdiff=Prtrans
860   P=Pfl*.13368*Prdiff*144*2546.4/(550*60)/Heatgain
870   Delt=Heatgain*LOG(.625/.545)/(2.0*PI*227*52.5/12)
880   Tavinsurf=Tavsurf-Delt
890   Tavfl=(Tfin+Tfout)/2.0
900   Deltat=Tavinsurf-Tavfl
910   Hi=Heatgain/(PI*.545/12*(52.5/12)*(Tavinsurf-Tavfl))
920   G=Mf/((.545/24)^2.0*PI)
930   Myu=5.57059-.90962*LOG(Tavfl)
940   Re(Z)=G*(.545/12)/Myu
950    F(Z)=Prdiff*32.2*.545*Df*144*3600^2.0/(2.0*50.50*G^2.0)
960  ! F(Z)=16/Re(Z)
970    Nop=Hi*.545/(12*K*(Cp*Myu/K)^.4)
980    Stpr=Hi*(Cp*Myu/K)^(2.0/3)/(Cp*G)
990   Area=PI*.545*52.5/144
1000  Q=Heatgain/Area
1010   PRINT
1020   IMAGE 8X,AAAAA,2X,DD,2X,.DDD,2X,.DDDD,2X,.DDDD,2X,DDDD.D,2X,DD.DD,2X,DD.D
D,2X,DDD.DD,2X,DDD.DD
1030   IF Z<17 THEN PRINT USING 1020;"SM.",Z,Pfl,Prdiff,F(Z),Re(Z),Hi,Npe,Heatga
in,Q
1040 ! IF Z>18 THEN PRINT USING 1340;"STFIN",Z,Pfl,Electra,Hgainf,Hbe,Deltat(Z),
Re,Dryness,Q(Z)
1050  NEXT Z
1060 ! CALL Plot(G(*),Hi(*),N)
1070    CALL Plot(Re(*),F(*),N)
1080  END
1090  SUB R(T,Tfarh)
1100  Tfarh=T*1.8+32
1110  SUBEND
```

APPENDIX D

REDUCED DATA

EXPLANATION OF TERMINOLOGY USED IN COMPUTER OUTPUT

TERM	
TUBE	Type of test tube
No.	Run number
SM	Smooth tube 1
STFIN	Straight fin tube 2
SP(3)	Spiral fin tube 3
SP(4)	Spiral fin tube 4
$T_{f\ in}$	R-113 temperature at the Inlet of test tube, ^oF
$T_{f\ out}$	R-113 temperature at the outlet of the test tube, ^oF
G	Mass flux, $lbm/hr\text{-}ft^2$
DELTAT	Temperature difference $(T_w - T_f)$, ^oF
T_{ow}	Average outside wall temperature, ^oF
Nu/Pr^0.4	$Nu/Pr^{0.4}$
Q	Heat gain by the R-113, Btu/hr
Prdiff	Pressure drop, PSI
f	Fanning friction coefficient
Re	Reynolds number
H_i	Heat transfer coefficient $Btu/hr\text{-}ft^2\ ^oF$
ERROR	Percent heat balance error
Q/A	Heat flux $Btu/hr\text{-}ft^2$
X_{out}	Dryness fraction at exit

TABLE (D-1): - REDUCED DATA

TUBE	NO.	Flo	Tfin	Tfout	G	DELTAT	Nu/Pr^.4	Tow	ERROR
SM.	1	.250	83.88	86.36	120231.8	3.44	23.57	88.56	2.28
SM.	2	.250	86.54	90.55	120195.0	6.84	19.77	95.39	7.01
SM.	3	.250	87.26	92.93	120158.1	9.88	19.51	99.98	7.44
SM.	4	.250	83.30	85.73	120268.7	3.92	20.04	88.44	4.82
SM.	5	.250	85.69	88.90	120268.7	5.35	19.86	92.65	5.98
SM.	6	.250	87.53	91.85	120029.1	7.26	19.77	96.95	5.51
SM.	7	.350	81.55	84.16	168453.5	3.91	29.76	86.77	3.81
SM.	8	.350	84.38	88.45	168350.4	6.71	27.70	93.13	4.77
SM.	9	.350	90.68	96.17	168247.2	9.08	28.83	102.51	6.42
SM.	10	.350	82.40	84.65	168324.6	3.57	29.08	87.10	7.10
SM.	11	.350	82.65	85.86	168376.1	4.47	30.32	88.73	-1.92
SM.	12	.350	85.46	89.85	168298.8	7.24	28.24	94.90	6.12
SM.	13	.400	78.53	80.69	193049.0	3.36	31.89	82.97	2.07
SM.	14	.400	79.57	83.19	192872.1	5.70	32.28	87.09	3.75
SM.	15	.400	80.96	85.28	192783.7	6.76	32.49	89.89	3.20
SM.	16	.400	73.45	76.59	193962.9	4.80	32.32	79.82	3.00
SM.	17	.400	76.32	80.38	193727.0	6.28	32.56	84.64	3.18

TABLE (D-1): - (CONTINUED)

TUBE	NO.	Flo	Prdiff	f	Re	Hi	Q	Q/A
SM.	1	.250	.0025	.0058	3573.2	49.94	107.22	171.77
SM.	2	.250	.0026	.0057	3658.1	42.68	182.21	291.90
SM.	3	.250	.0027	.0059	3696.0	41.89	258.32	413.82
SM.	4	.250	.0025	.0054	3559.2	44.01	107.76	172.62
SM.	5	.250	.0027	.0058	3628.8	43.09	143.91	230.54
SM.	6	.250	.0028	.0060	3681.8	42.52	192.57	308.50
SM.	7	.350	.0036	.0040	4927.4	65.73	160.34	256.85
SM.	8	.350	.0037	.0041	5048.6	60.24	252.32	404.21
SM.	9	.350	.0037	.0041	5293.5	61.13	346.37	554.88
SM.	10	.350	.0037	.0041	4946.8	64.16	143.05	229.16
SM.	11	.350	.0037	.0041	4973.8	66.50	185.62	297.37
SM.	12	.350	.0038	.0042	5090.7	61.15	276.29	442.62
SM.	13	.400	.0044	.0037	5517.5	71.28	149.37	239.28
SM.	14	.400	.0045	.0038	5582.9	71.46	254.34	407.44
SM.	15	.400	.0046	.0039	5649.5	71.49	301.83	483.52
SM.	16	.400	.0043	.0036	5361.4	73.40	219.71	351.98
SM.	17	.400	.0045	.0038	5486.8	72.80	285.57	457.47

TABLE (D-1); - (CONTINUED)

TUBE	NO.	Flo	Tfin	Tfout	Ġ	DELTAT	Tow	ERROR
STFIN	1	.250	76.73	80.42	114687.7	3.81	82.39	5.84
STFIN	2	.250	77.00	82.40	114635.2	6.37	86.07	4.93
STFIN	3	.250	77.36	84.47	114565.1	8.86	89.78	6.07
STFIN	4	.250	79.88	83.30	114390.0	3.42	85.01	5.73
STFIN	5	.250	80.24	85.21	114337.4	5.27	88.00	6.16
STFIN	6	.250	80.78	87.73	114284.9	8.25	92.51	6.62
STFIN	7	.350	81.50	84.56	159949.8	3.00	86.03	6.15
STFIN	8	.350	79.97	84.20	160146.0	4.59	86.68	6.15
STFIN	9	.350	78.44	84.02	160391.2	6.00	87.24	4.69
STFIN	10	.350	75.56	78.44	160857.0	3.02	80.02	3.58
STFIN	11	.350	76.28	80.60	160734.4	4.76	83.20	2.28
STFIN	12	.350	76.28	81.86	160709.9	6.23	85.31	6.07
STFIN	13	.400	75.92	78.53	183752.5	2.93	80.16	4.31
STFIN	14	.400	76.64	80.42	183780.6	4.61	83.14	5.91
STFIN	15	.400	76.10	81.14	183724.5	6.22	84.85	4.05
STFIN	16	.400	76.01	78.80	183752.5	3.13	80.54	4.25
STFIN	17	.400	75.74	79.52	183808.6	4.36	81.99	4.73
STFIN	18	.400	77.72	82.58	183556.4	5.33	85.49	4.01

TABLE (D-1): - (CONTINUED)

TUBE	NO.	Flo	Prdiff	f	Re	Hi	Nu/Pr^.4	Q	Q/A
STFIN	1	.250	.0120	.0310	3337.1	66.70	31.52	165.89	254.26
STFIN	2	.250	.0126	.0305	3362.7	57.86	26.66	240.34	368.35
STFIN	3	.250	.0128	.0310	3390.0	55.38	25.65	320.10	490.61
STFIN	4	.250	.0129	.0313	3401.2	68.71	31.84	153.20	234.80
STFIN	5	.250	.0131	.0318	3427.1	64.96	30.25	223.46	342.49
STFIN	6	.250	.0132	.0320	3462.7	58.31	27.32	313.92	481.13
STFIN	7	.350	.0180	.0223	4804.6	98.46	45.32	192.52	295.07
STFIN	8	.350	.0190	.0235	4778.4	88.95	41.29	266.41	408.31
STFIN	9	.350	.0195	.0240	4756.7	88.47	40.97	346.57	531.18
STFIN	10	.350	.0180	.0221	4627.2	90.08	41.01	177.32	271.77
STFIN	11	.350	.0185	.0228	4672.3	84.51	38.76	262.23	401.90
STFIN	12	.350	.0190	.0234	4692.9	86.63	39.83	352.36	540.05
STFIN	13	.400	.0208	.0196	5294.5	96.70	44.08	184.97	283.50
STFIN	14	.400	.0210	.0198	5345.7	90.70	41.59	272.52	417.68
STFIN	15	.400	.0230	.0217	5347.6	87.71	40.25	356.16	545.88
STFIN	16	.400	.0209	.0197	5301.5	96.72	44.13	197.61	302.87
STFIN	17	.400	.0210	.0198	5311.8	94.72	43.29	269.17	412.54
STFIN	18	.400	.0230	.0217	5401.9	98.55	45.49	342.96	525.65

TABLE (D-1): - (CONTINUED)

TUBE	NO.	Flo	Tfin	Tfout	G	DELTAT	Tow	ERROR
SP(3)	1	.300	88.43	93.29	127204.2	2.49	93.35	-4.81
SP(3)	2	.300	89.87	96.62	127263.0	3.36	96.61	-6.63
SP(3)	3	.300	92.55	100.49	127361.0	4.07	100.60	-8.59
SP(3)	4	.300	85.51	89.42	127400.1	2.02	89.49	-4.69
SP(3)	5	.300	87.55	93.43	127400.1	2.90	93.39	-8.66
SP(3)	6	.300	92.57	100.85	127223.8	4.29	101.00	-9.40
SP(3)	7	.350	82.62	87.08	149090.6	2.33	87.18	-5.66
SP(3)	8	.350	84.13	90.23	148976.3	3.18	90.36	-4.41
SP(3)	9	.350	86.02	94.10	148839.2	4.16	94.22	-4.26
SP(3)	10	.350	82.80	86.43	149113.4	1.94	86.56	-4.65
SP(3)	11	.350	84.69	90.68	148976.3	3.14	90.83	-1.68
SP(3)	12	.350	86.81	95.00	148770.6	4.17	95.08	-4.17
SP(3)	13	.400	82.00	89.10	170702.7	3.47	89.03	-7.02
SP(3)	14	.400	83.75	91.76	170598.2	3.91	91.67	-6.60
SP(3)	15	.400	83.61	92.52	170624.3	4.28	92.35	-6.76
SP(3)	16	.400	81.95	89.06	170702.7	3.44	88.95	-8.26
SP(3)	17	.400	82.58	90.77	170754.9	4.13	90.81	-4.44
SP(3)	18	.400	82.94	92.84	170807.2	4.99	92.89	-7.63

TABLE (D-1): - (CONTINUED)

TUBE	NO.	Flo	Prdiff	f	Re	Hi	Nu/Pr^.4	Q	Q/A
SP(3)	1	.300	.0150	.0294	4178.6	140.22	71.39	233.48	348.74
SP(3)	2	.300	.0152	.0298	4248.7	141.71	70.92	318.88	476.31
SP(3)	3	.300	.0155	.0304	4346.4	135.08	68.41	368.48	550.40
SP(3)	4	.300	.0150	.0294	4088.7	139.07	67.96	188.14	281.02
SP(3)	5	.300	.0154	.0302	4174.5	140.90	69.85	273.15	408.00
SP(3)	6	.300	.0157	.0308	4347.2	132.84	67.38	381.11	569.25
SP(3)	7	.350	.0180	.0258	4698.4	159.89	77.39	249.31	372.39
SP(3)	8	.350	.0190	.0273	4771.6	162.03	79.41	344.60	514.73
SP(3)	9	.350	.0200	.0287	4862.6	164.14	81.33	456.65	682.09
SP(3)	10	.350	.0180	.0258	4691.4	157.59	76.31	205.05	306.28
SP(3)	11	.350	.0190	.0273	4788.2	165.23	81.13	347.64	519.27
SP(3)	12	.350	.0200	.0287	4888.5	165.86	82.43	462.97	691.53
SP(3)	13	.400	.0195	.0214	5405.9	192.50	93.67	447.78	668.84
SP(3)	14	.400	.0220	.0241	5485.9	193.89	95.30	507.40	757.89
SP(3)	15	.400	.0220	.0241	5498.4	196.62	96.73	563.68	841.96
SP(3)	16	.400	.0195	.0214	5404.2	192.70	93.83	443.74	662.81
SP(3)	17	.400	.0220	.0241	5450.1	191.76	93.89	530.04	791.71
SP(3)	18	.400	.0230	.0252	5497.8	186.07	91.55	621.90	928.93

TABLE (D-1): - (CONTINUED)

TUBE	NO.	Flo	Tfin	Tfout	G	DELTAT	Tow	Nu/Pr^.4	ERROR
SP(4)	1	.40	81.54	86.41	88873.8	3.15	87.13	74.48	-5.85
SP(4)	2	.40	83.79	90.14	88792.2	4.37	91.34	68.91	-5.19
SP(4)	3	.40	84.56	91.58	88765.0	4.99	93.07	66.27	-6.13
SP(4)	4	.40	81.64	86.77	88887.4	3.54	87.75	70.25	-1.89
SP(4)	5	.40	82.40	89.67	88873.8	5.33	91.37	64.16	-5.86
SP(4)	6	.40	83.48	91.71	88860.2	5.85	93.45	65.68	-7.12
SP(4)	7	.25	86.68	89.92	55274.2	1.04	89.35	102.41	5.63
SP(4)	8	.25	85.77	89.83	55274.2	2.52	90.32	51.03	1.58
SP(4)	9	.25	85.82	89.24	55316.7	1.04	88.57	103.24	.86
SP(4)	10	.25	85.84	90.75	55265.7	2.81	91.11	51.39	-5.83
SP(4)	11	.25	86.38	93.65	55248.7	4.50	94.52	49.00	-3.36
SP(4)	12	.35	73.94	78.08	78311.9	2.84	78.85	60.47	-1.60
SP(4)	13	.35	75.56	81.14	78240.5	3.90	82.25	57.69	-5.81
SP(4)	14	.35	76.87	83.59	78192.9	4.75	84.99	58.03	-4.41
SP(4)	15	.35	77.05	82.17	78133.4	3.29	82.90	62.80	-5.77
SP(4)	16	.35	77.90	83.07	78109.6	4.01	84.50	58.60	5.80
SP(4)	17	.35	77.90	84.18	78097.7	5.01	86.06	55.77	3.47

TABLE (D-1): - (CONTINUED)

TUBE	NO.	Flo	Prdiff	f	Re	Hi	Q	Q/A
SP(4)	1	.40	.0064	.0409	3858.1	107.51	311.37	338.75
SP(4)	2	.40	.0065	.0416	3935.8	101.48	407.77	443.63
SP(4)	3	.40	.0069	.0441	3964.9	97.25	446.39	485.65
SP(4)	4	.40	.0061	.0390	3865.1	104.69	340.31	370.24
SP(4)	5	.40	.0060	.0383	3914.2	94.78	464.17	504.99
SP(4)	6	.40	.0062	.0396	3956.1	96.47	518.73	564.35
SP(4)	7	.25	.0038	.0624	2472.9	150.44	144.37	157.07
SP(4)	8	.25	.0040	.0657	2464.3	75.09	173.77	189.05
SP(4)	9	.25	.0041	.0673	2461.6	152.09	145.11	157.88
SP(4)	10	.25	.0043	.0707	2472.4	75.44	195.01	212.17
SP(4)	11	.25	.0042	.0691	2501.0	71.41	295.48	321.47
SP(4)	12	.35	.0052	.0431	3210.7	93.06	242.64	263.97
SP(4)	13	.35	.0054	.0448	3262.9	87.59	313.63	341.22
SP(4)	14	.35	.0057	.0473	3305.4	87.48	382.26	415.88
SP(4)	15	.35	.0054	.0448	3288.2	95.03	287.03	312.27
SP(4)	16	.35	.0057	.0473	3307.9	88.36	325.90	354.56
SP(4)	17	.35	.0053	.0440	3320.6	83.88	386.57	420.57

TABLE (D-2): - REDUCED DATA

TUBE	NO.	Flo	Tfin	Tfout	Prdiff	DELTAT	Tow	Hi
SM.	1	.25	92.43	128.03	.60	23.65	133.94	185.78
SM.	2	.25	95.23	131.27	1.05	25.78	139.12	244.91
SM.	3	.25	102.20	140.09	1.35	28.25	149.51	303.46
SM.	4	.25	90.39	118.81	.35	22.77	127.42	157.00
SM.	5	.25	94.21	130.46	.90	23.44	135.85	235.08
SM.	6	.25	98.69	139.53	1.30	28.08	147.30	285.27
SM.	7	.35	95.02	130.24	.46	23.81	136.51	212.77
SM.	8	.35	98.91	134.83	.80	25.41	142.37	262.60
SM.	9	.35	102.65	139.68	1.00	27.80	149.08	294.46
SM.	10	.35	95.00	129.06	.30	23.41	135.50	184.56
SM.	11	.35	97.12	132.10	.70	24.10	138.79	246.91
SM.	12	.35	102.72	139.53	1.05	28.67	149.91	293.57
SM.	13	.40	89.92	131.79	.34	23.40	134.34	248.77
SM.	14	.40	93.94	135.90	.59	25.27	140.28	281.18
SM.	15	.40	99.03	141.44	.80	27.02	147.37	316.00
SM.	16	.40	93.83	129.74	.34	24.15	136.01	217.24
SM.	17	.40	96.35	133.41	.60	26.06	141.03	256.33
SM.	18	.40	102.02	140.45	.65	27.59	148.94	308.62

TABLE (D-2): - (CONTINUED)

TUBE	NO.	Flo	G	Re	Xout	Q/A	Q
SM.	1	.25	119273.7	3747.00	.1037	4394.02	2742.87
SM.	2	.25	118831.5	3842.66	.2034	6314.03	3941.39
SM.	3	.25	118094.5	4003.53	.3204	8571.86	5350.80
SM.	4	.25	119384.3	3723.17	.0852	3574.58	2231.35
SM.	5	.25	118978.9	3810.71	.1603	5510.51	3439.81
SM.	6	.25	118241.9	3971.19	.2797	8010.16	5000.16
SM.	7	.35	166596.3	5329.41	.0656	5065.96	3162.31
SM.	8	.35	165951.4	5469.48	.1243	6672.19	4164.97
SM.	9	.35	165306.5	5610.60	.1794	8187.27	5110.72
SM.	10	.35	166854.2	5273.64	.0417	4321.00	2697.29
SM.	11	.35	166209.3	5413.33	.1000	5949.93	3714.11
SM.	12	.35	165280.7	5616.27	.1888	8415.99	5253.49
SM.	13	.40	191191.7	5918.99	.0429	5821.69	3634.06
SM.	14	.40	190661.1	6033.38	.0851	7104.21	4434.64
SM.	15	.40	190071.4	6161.05	.1319	8537.35	5329.25
SM.	16	.40	190602.1	6046.12	.0454	5247.13	3275.40
SM.	17	.40	190189.4	6135.47	.0886	6679.19	4169.34
SM.	18	.40	189187.0	6353.90	.1463	8514.32	5314.88

TABLE (D-2): - (CONTINUED)

TUBE	NO.	Flo	Tfin	Tfout	Prdiff	DELTAT	Tow	Hi
STFIN	1	.25	86.18	128.53	.80	23.41	130.83	227.51
STFIN	2	.25	89.74	131.18	1.15	25.01	135.55	267.69
STFIN	3	.25	92.35	137.52	1.30	26.49	141.51	297.64
STFIN	4	.25	91.76	129.69	.85	23.39	134.17	211.05
STFIN	5	.25	95.36	133.79	1.15	25.25	139.90	258.56
STFIN	6	.25	97.75	136.96	1.35	27.25	144.70	295.52
STFIN	7	.35	92.41	129.76	.60	22.33	133.47	225.73
STFIN	8	.35	95.49	133.70	.85	24.70	139.37	260.79
STFIN	9	.35	97.34	136.00	1.00	27.31	144.07	286.61
STFIN	10	.35	95.18	131.18	.60	22.09	135.33	225.82
STFIN	11	.35	95.99	132.55	.85	24.54	138.88	251.85
STFIN	12	.35	97.43	135.45	1.00	27.79	144.32	283.26
STFIN	13	.40	93.67	130.35	.45	22.65	134.72	227.18
STFIN	14	.40	95.04	132.87	.52	24.48	138.51	267.44
STFIN	15	.40	98.71	138.20	.95	26.13	144.68	327.96
STFIN	16	.40	92.03	129.78	.50	22.78	133.75	243.64
STFIN	17	.40	93.74	132.39	.75	24.70	137.84	275.01
STFIN	18	.40	97.11	137.08	.93	26.41	143.60	321.78

TABLE (D-2): - (CONTINUED)

TUBE	NO.	Flo	G	Re	Xout	Q/A	Q
STFIN	1	.25	114162.3	4037.8	.1391	5326.62	3475.40
STFIN	2	.25	113654.4	4100.6	.2180	6695.55	4368.57
STFIN	3	.25	113321.6	4206.2	.2724	7883.19	5143.46
STFIN	4	.25	113444.2	4099.8	.1354	4936.72	3221.01
STFIN	5	.25	113111.5	4188.9	.2218	6529.02	4259.91
STFIN	6	.25	112691.1	4247.0	.3049	8053.75	5254.74
STFIN	7	.35	158552.2	5743.2	.0654	5040.50	3288.72
STFIN	8	.35	158208.9	5859.7	.1178	6442.73	4203.61
STFIN	9	.35	157718.5	5918.5	.1715	7827.36	5107.03
STFIN	10	.35	158503.2	5818.4	.0683	4989.16	3255.22
STFIN	11	.35	158184.4	5846.8	.1132	6180.78	4032.70
STFIN	12	.35	157620.5	5906.1	.1756	7872.57	5136.52
STFIN	13	.40	181398.7	6609.6	.0468	5146.10	3357.62
STFIN	14	.40	181090.4	6680.1	.0909	6547.51	4271.98
STFIN	15	.40	180193.7	6837.8	.1562	8569.38	5591.17
STFIN	16	.40	181650.9	6572.3	.0565	5551.23	3621.95
STFIN	17	.40	181314.6	6650.8	.0960	6792.76	4432.00
STFIN	18	.40	180530.0	6792.5	.1513	8497.85	5544.50

TABLE (D-2): - (CONTINUED)

TUBE	NO.	Flo	Tfin	Tfout	Prdiff	DELTAT	Tow	Hi
SP(3)	1	.25	97.92	128.28	.70	23.34	136.47	186.84
SP(3)	2	.25	101.01	134.69	1.15	25.77	143.67	239.35
SP(3)	3	.25	101.01	134.19	1.18	26.22	143.87	234.85
SP(3)	4	.25	105.94	137.52	1.33	27.58	149.37	272.32
SP(3)	5	.25	107.20	137.73	1.35	28.04	150.57	270.91
SP(3)	6	.35	102.85	132.84	.74	23.86	141.75	227.69
SP(3)	7	.35	111.34	141.22	1.05	26.30	152.65	299.70
SP(3)	8	.35	102.54	132.55	.70	22.53	140.12	238.34
SP(3)	9	.35	110.93	140.92	1.10	25.85	151.84	303.98
SP(3)	10	.40	95.67	124.90	.25	20.11	130.42	174.02
SP(3)	11	.40	102.85	133.32	.63	24.28	142.41	238.40
SP(3)	12	.40	111.34	141.94	.95	27.19	153.90	299.50
SP(3)	13	.40	96.96	129.24	.26	19.00	132.13	225.09
SP(3)	14	.40	103.91	135.16	.65	22.14	141.72	269.90
SP(3)	15	.40	104.41	139.30	.90	26.21	148.13	302.42

TABLE (D-2): - (CONTINUED)

TUBE	NO.	Flo	G	Re	Xout	Q/A	Q
SP(3)	1	.25	105448.5	4007.3	.1370	4359.94	2918.92
SP(3)	2	.25	105203.6	4119.4	.2284	6167.70	4129.19
SP(3)	3	.25	105138.3	4110.4	.2297	6157.94	4122.66
SP(3)	4	.25	104942.4	4209.8	.3131	7510.33	5028.07
SP(3)	5	.25	104828.1	4224.6	.3221	7596.32	5085.63
SP(3)	6	.35	147125.1	5760.6	.1130	5433.61	3637.73
SP(3)	7	.35	146416.6	6041.4	.2150	7882.64	5277.32
SP(3)	8	.35	147399.3	5760.6	.1100	5370.12	3595.22
SP(3)	9	.35	146508.0	6031.8	.2134	7858.49	5261.15
SP(3)	10	.40	169083.3	6311.5	.0199	3499.52	2342.88
SP(3)	11	.40	168221.3	6596.6	.0965	5787.79	3874.85
SP(3)	12	.40	167124.3	6911.1	.1822	8143.24	5451.79
SP(3)	13	.40	169109.4	6426.5	.0364	4275.88	2862.65
SP(3)	14	.40	168534.7	6669.1	.1002	5974.94	4000.14
SP(3)	15	.40	168378.0	6759.9	.1569	7926.22	5306.50

TABLE (D- 2): - (CONTINUED)

TUBE	NO.	Flo	Tfin	Tfout	Prdiff	DELTAT	Tow	Hi
SP(4)	1	.40	95.05	129.29	2.74	16.59	128.80	168.63
SP(4)	2	.40	97.99	135.39	2.93	19.29	136.03	195.75
SP(4)	3	.40	104.00	142.56	3.23	20.44	143.78	239.75
SP(4)	4	.40	95.07	132.69	2.74	17.11	131.03	185.67
SP(4)	5	.40	97.41	136.36	2.94	19.54	136.48	201.24
SP(4)	6	.40	101.66	140.86	3.18	20.44	141.76	235.65
SP(4)	7	.25	92.30	126.00	2.97	17.77	126.95	140.82
SP(4)	8	.25	96.31	129.67	3.29	19.91	132.94	170.81
SP(4)	9	.25	102.07	135.75	3.71	20.23	139.20	240.95
SP(4)	10	.25	92.28	126.72	3.05	18.72	128.25	150.16
SP(4)	11	.25	96.40	129.78	3.29	19.20	132.33	179.58
SP(4)	12	.25	103.15	135.91	3.68	20.37	139.96	239.52
SP(4)	13	.35	85.87	120.61	2.69	23.55	126.82	99.92
SP(4)	14	.35	90.37	130.77	2.88	19.05	129.66	167.14
SP(4)	15	.35	92.52	134.11	3.05	20.64	134.00	190.07
SP(4)	16	.35	96.93	138.18	3.28	21.28	138.89	216.62
SP(4)	17	.35	92.50	136.74	3.00	21.01	135.68	191.50
SP(4)	18	.35	95.40	140.41	3.20	22.03	139.99	216.88

TABLE (D-2): - (CONTINUED)

TUBE	NO.	Flo	G	Re	Xout	Q/A	Q
SP(4)	1	.40	87949.2	4605.2	.0146	2798.20	2572.00
SP(4)	2	.40	87731.7	4726.8	.0511	3776.69	3471.39
SP(4)	3	.40	87514.1	4912.8	.1025	4900.93	4504.74
SP(4)	4	.40	88071.6	4661.9	.0208	3176.58	2919.79
SP(4)	5	.40	87881.2	4740.7	.0529	3932.88	3614.95
SP(4)	6	.40	87622.9	4857.7	.0957	4816.45	4427.10
SP(4)	7	.25	55061.7	2828.1	.0734	2502.58	2300.28
SP(4)	8	.25	54959.8	2892.8	.1437	3400.32	3125.44
SP(4)	9	.25	54781.3	2992.9	.2580	4873.66	4479.68
SP(4)	10	.25	55095.7	2836.2	.0941	2810.52	2583.32
SP(4)	11	.25	54951.3	2894.2	.1473	3447.43	3168.75
SP(4)	12	.25	54730.3	3001.7	.2619	4878.12	4483.78
SP(4)	13	.35	77621.8	3837.8	.0065	2352.76	2162.57
SP(4)	14	.35	77360.1	4009.7	.0325	3184.23	2926.82
SP(4)	15	.35	77288.7	4076.5	.0688	3922.44	3605.36
SP(4)	16	.35	77110.2	4177.1	.1086	4609.43	4236.81
SP(4)	17	.35	77300.6	4110.9	.0653	4023.52	3698.26
SP(4)	18	.35	77145.9	4188.2	.1047	4777.12	4390.95

APPENDIX E

ADDITIONAL INFORMATION ON THE INSTRUMENTATION AND COMPONENTS USED IN THIS STUDY

1. R-113 FLOW CIRCUIT

A. Components:

1. Refrigerant-113 Liquid Circulating Gear Pump:

Sherwood Alear Siegler Company
Bronze Rotary Gear Pump
Model: S and V Series
R.P.M.: 1725
Pipe Size: 1/4"
Shaft Diameter: 1/2"
H.P.: 1/3
Dripless Mechanical Shaft Seal, Self Lubricated.

2. Refrigerant-113 Liquid Circulating Pump Motor:

Dayton-Electric A.C. Motor
Model No.: 5K--1
R.P.M.: 1785
H.P.: 1/2
HZ: 60

3. Refrigerant-113 Filter:

Sparlan-Catchall Refrigerant Filter Type: C-304

4. Refrigerant-113 Liquid Receiver:

Midlan-Ross Refrigerant Type Circular Tank
Serial No.: 2193
Size: 3.5 gallons
Working pressure: Maximum Allowable Working Pressure
400 PSI at 650 ^{o}F

5. Refrigerant-113 Valves:

Diaphram Packless Line Valves
Superior Brand, Solder to Solder Type
A. Model No.: 214-45 (1/4")
B. Model No.: 216-105 (5/8")

6. Refrigerant-113 Tube Connectors:

Standard Copper Tube
Sweat Fitting Type

7. Thermocouples:

   Copper-Constantan Thermocouple of Type RIP - 24 gage.

8. Test Section, and Preheater, Locally constructed:

   Material: Copper Tubing
   Heating Element: Ribbon type chromel of $0.204\Omega/03.48$ cm
   Teflon tape: Saunder type S-17
   Epoxy: Armstrong A-68 and B-68 types

B. <u>Instrumentation</u>

1. Refrigerant-113 Liquid Level Gauge:

   Brooks Rotameter view meter
   Type: 6-1355-VB
   Serial No.: 6507-36340/4

2. Refrigerant-113 Flow Meters:

   a. Fischer-Porter Variable Area Type Flow Meter
      Range: 0~0.35 GPM liquid
      Model: 10A3565S
      Serial No.: 7207A4733A2
      Tube No.: FP-1/2-27-G-10/55

   b. Fischer-Porter Variable Area Type Flow Meter
      Range: 0~0.5 GPM liquid
      Model: 10A3565S
      Serial No.: 7207A4733A1
      Tube No.: FP-1/2-17-G-10/55

3. Refrigerant-113 Pressure Gauge:

   Heise Pressure Gauge of Type H28832
   Range: 0~200 psig

4. Pressure Transducer:

   Pace Wiancko Division of Whittaker Corporation
   Model: KP15 Pressure Transducer
   Serial No.: 150330

5. Transducer Indicator:

   Pace Wiancko Division of Whittaker Corporation
   Model: CD25
   Serial No.: 23449

6. Voltage Regulator

   Superior Electric Co.
   Powerstat Variable Autotransformer
   Input: 240 V, 60 HZ
   Output: 0-280 V, 28 A, 7.8 KW

7. A.C. Ampere Meter:

   Daystrom, Incorporated Weston Instruments Div.
   Westen Instruments, Inc.
   New York, New Jersey
   Model: 433 No. 164330

8. A.C. Volt Meter:

   Daystrom, Incorporated Weston Instruments Div.
   Weston Instruments, Inc.
   New York, New Jersey.
   Model: 433 No. 146652

9. Data Acquisition System:

   Esterline Angus an Esterline Company
   Model: PD-2064
   Type: Key Programmable

   The system can gather analog and digital data from up to 64 channels under the control of tiny microprocessor. The system outputs the measured values in engineering or scientific units through various output devices. The solid-state integrated circuit microprocessor is combined with RAMs (random access memory devices), ROMs (read-only memory devices), PROMs (Program-mable ROMs) to provide a keyboard-programmable system that permits the instrument to scan, measure, collect, identity, and record both analog and digital input signals.

   Accuracy:

   With Ambient Temperature at $77^{o}F \mp 9^{o}F$

   $\mp$ 0.01% of reading, $\mp$ 0.015% full scale, $\mp$ 1 Count on 4000 MV range;
   $\mp$ 0.01% of reading, $\mp$ 0.03% full scale, $\mp$ 1 Count on 400 MV range;
   $\mp$ 0.01% of reading, $\mp$ 0.04% full scale, $\mp$ 1 Count on 40 MV range.

   Over full Operation Ambient Temperature Range of $32^{o}F$ to $122^{o}F$

   $\mp$ 0.5 μV per ^{o}C, $\mp$ 0.01% of reading, $\mp$ .04% full scale, $\mp$ 1 Count on all ranges.

10. Vaccum Pump

Matheson Scientific
Division of Will Ross, Inc.
Serial No.: 1173
Power: 115 V, 60 HZ
Connections: 3 Conductor Power Cord with 2 prong adaptor. Inlet and outlet Connector to 3/8" I.D. hose.
Function: Portable A.C. Power Source of vaccum (to 686 mm/ 27"Hg) or Pressure (to 1.7 kg/cm^2, 25 psig)

2) WATER FLOW CIRCUIT

1. Cooling Water Flow Meters:

a. Brook Rotameter
Type: 110-09H3A1B
Serial No.: 7201-74650/1
Tube No.: R-9M-25-1 BR-3/4-14G10
Range: 0 ~ 3 GPM

b. Brooks Rotameter
Type: 1
Serial No.: R-9M-25-2
Range: 0 ~ 2 GPM

2. Cooling Water Pump

A.O.Smith Co. Pump.
Model No.: C48L2DA11A4
Serial No.: J69
H.P.: 1
R.P.M.: 3450
HZ: 60

APPENDIX F

UNCERTAINTY ANALYSIS IN EXPERIMENTAL MEASUREMENTS OF HEAT TRANSFER COEFFICIENTS

The uncertainty intervals for individual measurements are summarized as follows:

Measurements	Smooth Tube 1 Uncertainty Inverval
1. Tube radius	± 0.001 ft
2. R-113 Temperature	± 0.6 ^{o}F
3. Tube wall Temperature	± 0.6 ^{o}F
4. Tube length	± 0.002 ft
5. R-113 flow rate	± 1.0% of flow
6. Inlet enthalpy of R-113	± 0.15 Btu/lbm
7. Outlet enthalpy of R-113	± 0.15 Btu/lbm

Kline and McClintor [15] presented a method of analyzing the effect of uncertainty in each variable on the uncertainty of the result. This may be represented by:

$$W_R = \{\Sigma_i (\frac{\partial R}{\partial V_i} W_{V_i})^2\}^{\frac{1}{2}} \qquad \text{(F-1)}$$

where

W_R = Uncertainty in calculating of R

R = Result

V_i = Independent variable

W_{V_i} = Uncertainty in Measurement of V_i

For single phase flow and subcooled boiling, the heat transfer coefficient, $\overline{h}$, was calculated as follows:

$$\overline{h} = \frac{Q}{A(T_w - T_f)} \quad \text{(F-2)}$$

By applying Eq. (F-1), the uncertainty in calculating $\overline{h}$ can be written as

$$W_{\overline{h}} = \{ (\frac{\partial \overline{h}}{\partial q} W_Q)^2 + (\frac{\partial \overline{h}}{\partial A} W_A)^2 + (\frac{\partial \overline{h}}{\partial T_w} W_{T_w})^2 + (\frac{\partial \overline{h}}{\partial T_f} W_{T_f})^2 \}^{\frac{1}{2}} \quad \text{(F-3)}$$

From Eq. (F-2)

$$\frac{\partial \overline{h}}{\partial q} = \frac{Q}{A(T_w - T_f)} \quad \text{(F-4)}$$

$$\frac{\partial \overline{h}}{\partial A} = \frac{Q}{A^2 (T_w - T_f)} \quad \text{(F-5)}$$

$$\frac{\partial \overline{h}}{\partial T_w} = \frac{Q}{A (T_w - T_f)^2} \quad \text{(F-6)}$$

$$\frac{\partial \overline{h}}{\partial T_f} = \frac{Q}{A (T_w - T_f)^2} \quad \text{(F-7)}$$

For run number one of the smooth tube (Table D-2) Appendix (D)

$$A = \pi DL = \pi(\frac{0.545}{12}) \times 4.375 = 0.624 \text{ ft}^2$$

T_w = 133.94 ^oF

$\overline{T}_f$ = 110.23 ^oF

Q = 2742.87 Btu/hr

substituting in Eqs. (F-4) to (F-7) to get the numerical values

$$\frac{\partial \bar{h}}{\partial q} = \frac{1}{0.624\ (133.94 - 110.23)} = \frac{1}{14.8} \quad \text{(F-8)}$$

$$\frac{\partial \bar{h}}{\partial A} = \frac{12742.87}{(0.624)^2\ (133.94 - 110.23)} = -\ 296.91 \quad \text{(F-9)}$$

$$\frac{\partial \bar{h}}{\partial T_w} = \frac{2742.87}{(0.624)(133.94 - 110.23)^2} = -\ 7.82 \quad \text{(F-10)}$$

$$\frac{\partial \bar{h}}{\partial T_f} = \frac{2742.87}{(0.624)(133.94 - 110.23)} = 7.82 \quad \text{(F-11)}$$

It was estimated that the maximum uncertainty in the heat transfer Q to be ± 7%, therefore $W_Q = \pm .07$.

Uncertainty for energy transfer, W_Q:

$$= \frac{7}{100} \times 2742.87 = \pm\ 192.00 \text{ Btu/hr} \quad \text{(F-12)}$$

The estimation of the uncertainty for the heat transfer area, W_A:

$$A = 2\pi r L \quad \text{(F-13)}$$

Thus, the uncertainty

$$W_A = \{(\frac{\partial A}{\partial r} W_r)^2 + (\frac{\partial A}{\partial L} W_L)^2\}^{\frac{1}{2}} \quad \text{(F-14)}$$

From Eq. (F-13):

$$\frac{\partial A}{\partial r} = 2\pi L \quad \text{(F-15)}$$

$$\frac{\partial A}{\partial L} = 2\pi r \quad \text{(F-16)}$$

substituting Eqs. (F-14) and (F-15) into (F-13) yields

$$W_A = \{(2\pi L W_r)^2 + (2\pi r W_L)^2\}^{\frac{1}{2}} \qquad \text{(F-17)}$$

$$W_A = \{(2\pi \times 4.375 \times 0.001)^2 + (2\pi \times \frac{0.545}{24} \times 0.002)^2\}^{\frac{1}{2}}$$

$$W_A = 0.027 \text{ ft}^2 \qquad \text{(F-18)}$$

substituting Eqs. (F-8), (F-9), (F-10), (F-11), (F-12), and (F-18) into (F-3) yields

$$W_{\bar{h}} = \{(\frac{1}{14.8} \times 192.00)^2 + (-296.91 \times 0.027)^2 + (-7.82 \times 0.6)^2 + (7.82 \times 0.6)^2\}^{\frac{1}{2}}$$

$$W_{\bar{h}} = \pm 16.63 \text{ Btu/hr ft}^2 \text{ }^oF$$

Therefor, $\bar{h} = 185.8 \frac{\text{Btu}}{\text{hr ft}^2 \text{ }^oF} \pm 16.63$

Uncertainty for this run is about ± 8.95%

One can assume that this uncertainty is representative of the uncertaties of all other runs.

AUGMENTATION OF SINGLE PHASE HEATING AND SUBCOOLED BOILING BY INTERNALLY FINNED TUBES

by

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ABSTRACT

In the present study, heat transfer and pressure drop data were taken during single phase heating and subcooled boiling of R-113 inside four vertical electrically heated tubes. One of theses was a smooth tube and the remaining three were internally finned tubes. For single phase flow heating inside internally finned tubes, appropriate modifiers were identified and applied to the smooth tube heat transfer and pressure drop correlations to bring about the best agreement between the measurements and predictions for these tubes.

The results are summarized as follows:

1 - In single phase heating, over Reynolds number range tested, the highest enhancement in the heat transfer coefficient was 180%, over the smooth tube results on nominal area basis.

2 - A correlation equation was developed for predicting the heat transfer coefficient during single phase heating inside internally finned tubes. It is applicable to a Reynolds number in the range of 3300 to 5500.

3 - Over the same range of Reynolds number, for single phase heating inside finned tubes, a new pressure drop correlation was developed.

4 - In the subcooled boiling regime, the finned tubes had a diminishing effect on the heat flux, at a specified wall to fluid temperature difference. No attempt was made to develop a heat transfer or a pressure drop correlation in this flow regime because of the limited data obtained.

5 - In the subcooled boiling regime, the pressure drop for finned tubes was slightly higher than the smooth tube at the same mass flow rate and heat flux input.

6 - The pumping power per unit heat transfer subject to the constraints of fixed geometry and the same flow rate was used to evaluate the performance of the tubes tested. The smooth tube required the least power in single phase flow.