HEAT TRANSFER MEASUREMENTS IN THE TRANSITION REGION FOR A HORIZONTAL CIRCULAR TUBE WITH A SQUARE-EDGED ENTRANCE

By

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Throughout the course of this study much help was secured from many sources to help in the completion of the experimental calibrations and data reduction.

In the experimental calibrations area, software was developed by Kevin Howard, a graduate student of Mechanical and Aerospace Engineering at Oklahoma State University. He developed two programs used to manipulate the data collected during the isothermal calibration runs. The first program was called CALHT, and its function was to calculate and store the temperature deviations for each outside wall thermocouple from the isothermal average temperature of the test section. He then developed a second program GETAVE, which input the deviation data from CALHT and created an average deviation file called CORR.DAT.

In the data reduction area, the services of Kevin Howard were again employed in development of programs RED96, and REDUCE30. RED96 calculates and stores the corrected average outside wall temperatures for the data collected by the MAC-14 utilizing CORR.DAT. REDUCE30 is an averaging program used to handle the output from the MODEL 5100 data logger. I am appreciative to Kevin Howard for his endeavors in development of these programs, which eliminated the need for repetitive averaging calculations and provided systematic temperature data correction.

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NOMENCLATURE

A	area, having units of ft^2 or m^2
C_p	specific heat of the liquid evaluated at the bulk temperature, Btu/lbm·F or J/kg·K
d	inside diameter of the test section
h	heat transfer coefficient, Btu/hr·ft2·F or $W/m^2 \cdot K$
I	current carried by the test section in Amperes
Gr	local bulk Grashof number, $g\beta\rho^2 d^3(T_{wi}-T_b)/\mu$
Gz	Graetz number, RePr(x/d)
k_s	thermal conductivity of steel test section, Btu/hr·ft·F or W/m·K
k_f	thermal conductivity of test fluid, Btu/hr·ft·F or W/m·K
L	length of the test section in feet or meters
L_h	heated length of the test section, feet or meters
$\dot{m}$	mass flow rate of test fluid, lbm/s or kg/s
Nu	local radial average Nusselt number, hd/k
Pr	local bulk Prandtl number, $C_p\mu/k$
Pe	Peclet number, RePr
$\dot{q}$	rate of heat addition to the test section, Btu/hr or W
$\dot{q}''$	heat flux, Btu/hr·ft2 or W/m^2
$\dot{Q}$	volume flow rate of the test fluid, GPM or m^3/s
Re	local bulk Reynolds number, $\rho ud/\mu$
Ra	Rayleigh number, GrPr
t	time increment or period, seconds

T	temperature, F or C
T_{wi}	tube inside wall temperature, F or C
T_b	bulk test fluid temperature, F or C
T_1	temperature lost due to longitudinal heat conduction, F or C
T_2	linear decrease in temperature from the fluid bulk exit value, F or C
u	flow velocity in the test section, ft/hr or m/s
U	uncertainty associated with a variable, %
V	Voltage drop across the test section, volts
x	local distance along the test section from the inlet, ft or m

Greek Letters

β	coefficient of volume expansion, 1/F or 1/C
μ	fluid viscosity, lbm/hr·ft
μ_w	fluid viscosity evaluated at the tube wall, lbm/hr·ft
ρ	fluid density, lbm/ft^3

CHAPTER I

INTRODUCTION

This chapter will first present background as it relates to the study of transition flow heat transfer for horizontal square entrance tubes. The purposes of the investigation will be given and a summary of the performed experimental work will be included. Brief statements about the previous work of other investigations will justify the need for experimentation such as this.

Background

A vast amount of data and analytical modeling is available for fluid flow through tubes in the laminar and fully turbulent flow regimes. However, very little has been done to examine the problems in the transition flow regime. An important design problem in industrial heat exchangers arises when flow inside the tubes falls into the transition region including Reynolds number ranges as low as 400 to 10000. In practical engineering design, the usual recommendation is to avoid design and operation in this regime; however, this is not always feasible under design constraints. The usually cited transition Reynolds number of about 2100 applies to a very steady flow with a rounded entrance. If the flow has a disturbed entrance typical of heat exchangers, in which there is a sudden contraction and possibly even a reentrant entrance and if the Reynolds number is less than 2100, the flow will eventually become a fully-developed laminar flow in the tube. Langhaar (1942), estimated that a fully developed laminar flow would occur for $L/d > 0.0575$ Re for a square-edged entrance. For a Reynolds number of 1000, this amounts to about 57 diameters, or in a 1 inch OD tube about 5 feet. For many heat exchanger applications, this is comparable to the full length of the tubes inside the exchanger. Calming sections in the heat exchangers present an unrealistic solution to the transition problem as they include added expense and wasted space. Even in the turbulent flow regime above a Reynolds number of 2000 to 2500, where the entrance effect is usually estimated to be on the order of 50 diameters for the establishment of a fully developed flow, the heat transfer characteristics of the flow have been found to be poorly defined. It is only for Reynolds numbers less than a few hundred or greater than 10,000 that the ability for accurate heat transfer prediction is available.

Inside circular square-entrance heat exchanger tubes, the fluid is subjected to an abrupt contraction at the entrance which may cause turbulence in the fluid altering the velocity and temperature profiles at the entrance of the test section. Ideally, both profiles are flat at the entrance and simultaneously the hydrodynamic and thermal profiles begin to develop along the tube toward the fully developed profiles. Generally the velocity profile develops faster than the temperature profile for fluid Prandtl numbers greater than one.

If the Prandtl number is one, then heat and momentum are diffused through the fluid at the same rates. If the Prandtl number is greater than one, it must follow that the velocity profile developes faster than the thermal profile. Actually if the Prandtl number is greater than about five, the velocity profile leads the temperature profile such that a solution assuming fully developed velocity profile at the inlet will apply accurately even without a hydrodynamic starting length (Kays and Crawford 1980).

Once heating starts in the tube, the flow may change from the laminar regime to transition regime along the tube due to the change in physical properties caused by the temperature variation. As the fluid inside a horizontal tube is heated using a constant wall heat flux, it dilates, the difference between the fluid density at the wall and at the pipe center causes a circulation which displaces the wall fluid that spirals along the tube with a vertical plane of symmetry. This secondary flow (see Figure 1.1) is at right angles to the primary flow (forced flow) direction and is driven by radial temperature variation. At the same time the secondary flow circulation reduces the wall temperature, the imposed heat flux must maintain the wall temperature at some level above the fluid temperature. This provides a

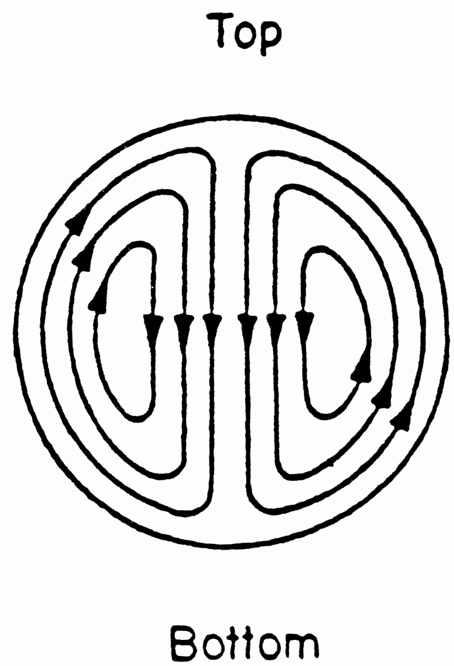


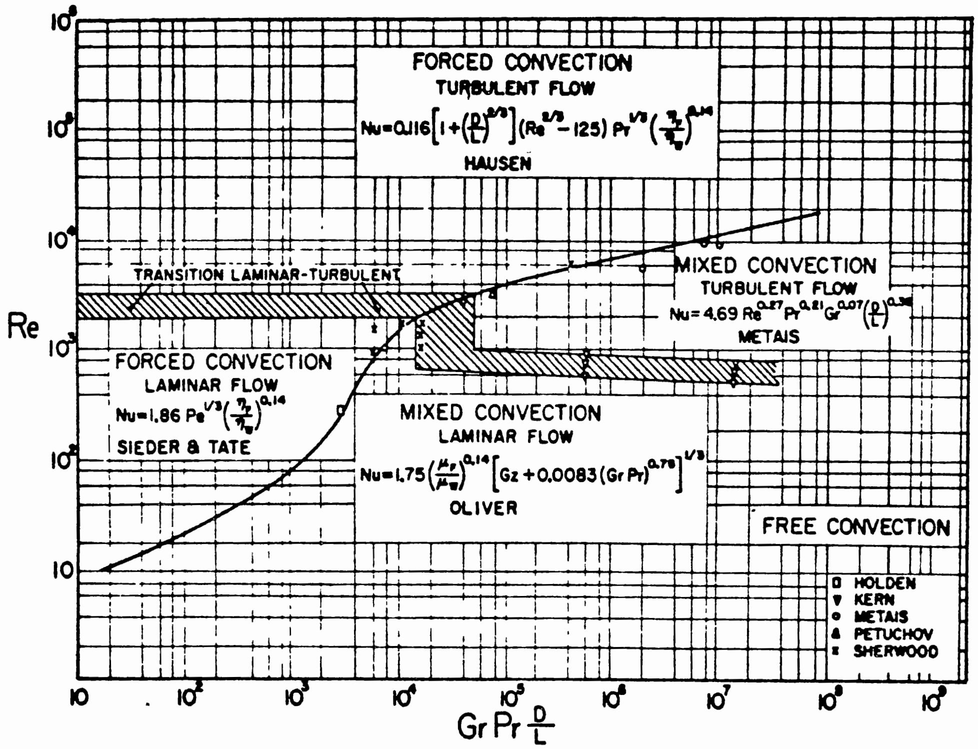
Figure 1.1 Secondary Flow Pattern

driving force for circulation resulting in mixed convective heat transfer. Consequently, the free convection contribution may be sustained and heat transfer is enhanced (Shannon & Depew 1968).

The boundary between natural, mixed (natural and forced), and forced convection is determined from the local heat transfer coefficient data. The ratio of the heat transfer coefficient at the top of the tube to that of the bottom (h_{top} / h_{bottom}) should be close to 1.0 for forced convection and is much less than 1.0 for a case in which natural convection dominates. Forced convection is primarily dependent upon the Reynolds and Prandtl numbers while free convection is dependant upon the Grashof and Prandtl numbers. Figure 1.2 illustrates the zones of free, mixed, and forced convection for a horizontal tube with constant wall temperature (Eckert & Metais 1964).

Objectives

The long range goal of this investigation is to study the effects of various entry configurations on the local heat transfer and pressure drop in tubes operating in the transition region under constant wall heat flux condition (for pressure drop see Augustine (1990)). In addition, it was desired to create an accurate heat transfer data base across all regimes of flow for a wide range of Reynolds, Prandtl, Grashof, and Nusselt numbers. From this data base, the development of accurate correlations for all flow



$$\left(10^{-2} < Pr\frac{d}{L} < 1\right)$$

Figure 1.2 Free, Forced, and Mixed Convection Flow Regimes

regimes and a clear definition of the transition region can be established for the square-edged inlet. The data base can also provide a comparison with established correlations to determine their validity under various flow conditions (see Chapter III).

Long Term Goals

This thesis details the preparations of a suitable experimental apparatus with inherent versatility for the successful fulfillment of all long term objectives. This required the ability for multiple inlet configurations to be employed without the necessity for re-construction of the apparatus. To deliver this ability, a unique calming and inlet section was fabricated which allowed the various inlets to be inserted between flanges connecting the two sections, requiring the removal of only eight bolts and the flexible end of the return tube (see Chapter II).

To operate in the transition region where even small disturbances of the flow can induce turbulent flow also required special test section abilities. Modifications included rubber damping material installed beneath the welder and pumps to isolate vibration transmission through structures. Special containment housings were constructed to suppress noise vibrations produced by both pumps and the welder. The welder was also located near an external wall so exhaust vibrations could be removed from the lab. Rubber hoses were installed at key locations to assist in damping pump vibrations through the return lines. Finally, damping material was located at all points where the test section and return loop came in contact with support structure. To enable velocity intermittency measurements via a hot film probe special test section provisions were required. An aluminum housing was constructed and located at the exit end of the test section and a slot was machined to allow insertion of the hot film probe. These provisions were implemented such that normal operation of the test section could occur with or without the hot film probe. Due to the precision and sensitivity of the probe, a filter was included in the return loop capable of removing particles down to five microns in size.

To facilitate massive and accurate data accumulation, digital acquisition systems were employed with fast sampling capability. When used in tandem with a personal computer, labor and data collection periods are reduced and accuracy is increased.

From the inception of the project, a flow visualization test loop was envisioned. The ability to adjoin a transparent test section to the pre-existing return loop has been retained by arranging adaptors and control valves at the inlet and exit destinations.

Data Base

This thesis discusses the entrance effects, the development of convective flow patterns (mixed and forced), and heat transfer rates over a wide range of Reynolds, Prandtl, Nusselt and Grashof numbers for a circular horizontal electrically-heated straight tube with a square-edged entrance. To provide the necessary data, 82 experimental runs were collected using distilled water, ethylene glycol, and a mixture of the two. Of these runs 15 used distilled water as the test fluid giving local Reynolds numbers of 3639 to 50529, local Prandtl numbers of 3.44 to 6.24, local Grashof numbers of 63011 to 2252505, and local Nusselt numbers of 31.3 to 232.0. The ethylene glycol runs numbered 14 and covered a local Reynolds number range of 281 to 3328, a local Prandtl number range of 95.89 to 157.80, a local Grashof number range of 1031 to 15911, and a local Nusselt number range of 15.0 to 80.2. The remaining 53 runs used a mixture of 60% mass fraction ethylene glycol covering a local Reynolds number range of 1180 to 12456, a local Prandtl number range of 16.81 to 55.29, a local Grashof number of 2604 to 197013, and a local Nusselt number of 12.9 to 146.3.

Correlation Comparisons

From this data base the existing heat transfer correlations were evaluated and those which best predicted the data in various flow regimes was determined. The laminar region data was difficult to evaluate and compare against existing equations due to the minimum heat flux available. The smallest attainable heat flux output from the welder at minimum current was observed to be 1437 Btu/hr·ft^2 (4.53 kW/m^2) and this forced the onset of mixed convection throughout the laminar flow region. However good verification of transition and turbulent region correlations was possible and recommendations for the ones which do the best job under forced and mixed convection are made. By modification of one pre-existing equation, good heat transfer predictions (forced and mixed convection) over the entire flow spectrum can be made (see Chapter III).

Literature Survey

The local heat transfer characteristics in pipes for both laminar and turbulent flow regimes have been treated extensively in the past including many analytical and numerical solutions for combined forced and free convection in horizontal tubes. In this section a brief review of many of the most important and pertinent works related to horizontal constant heat flux pipe flow is given. At the end of the section a table containing the most cited correlations is provided for all regimes of flow.

In the laminar regime Shah and London (1968) performed research for the case of uniform specified wall heat flux with fully developed flow. Kakac, Shah, and Bergles (1981) investigated the boundary condition for uniform wall heat flux with simultaneously developing velocity and temperature profiles. Siegal, Sparrow, and Hallman (1958) propose an analytical solution for laminar heat transfer without natural convection. They developed an equation for local Nusselt number with a fully developed velocity profile and uniform heat flux boundary condition. Their correlation has been used by many including Petukhov & Polyakov (1967), Hussain & McComas (1970), Bergles & Simonds (1971), and Hong & Bergles (1976) as a basic solution for laminar forced convection heat transfer. But their pure forced convection equation gives lower prediction for heat transfer coefficients than experimental data shows as found by Petukhov and Polyakov (1967). They observed that greater deviations from the forced convective prediction arise and that increased density variations (Rayleigh number) occur with increased heat flux. Research data is readily available using an array of fluids for combined natural and forced convection in the laminar flow region as investigated by Edé (1961), Metais & Eckert (1964), McComas & Eckert (1966), Shannon & Depew (1968-69), Bergles & Simonds (1971), and Morcos & Bergles (1975) to name a few. The results of these investigations commonly disagree.

To study the effects of free convection on fluid flow with constant heat flux, Ede (1956, 1961) used water and air at Reynolds numbers from 300 to 100,000 in electrically-heated aluminum-brass pipes with varying inside diameter and wall thickness for abruptly converging and diverging inlet geometry. He found that there was no consistent variation in Nusselt number with Grashof number less than 100,000 in the laminar region (Re <2300). He presented no correlations for transition or turbulent flows, and his laminar equation for Re < 2300 in fact had no Reynolds number dependency.

Using air as a test fluid in a uniformly heated horizontal tube, McComas and Eckert (1966) investigated free convection heat transfer for laminar flow Reynolds numbers from 100 to 900 and Grashof numbers from 0.33 to 1000. By comparing high Grashof number runs with low Grashof number runs at equal Reynolds numbers, they found that buoyancy created secondary flows increasing with the ratio of Grashof to Reynolds number.

Mori et al (1966) studied the effect of buoyancy on forced convection heat transfer in horizontal tubes with uniform heat flux. For air at Reynolds numbers from 100 to 13,000 in a brass tube with nichrome wires wound at constant pitch for nearly constant heat flux, they developed a correlation for laminar Nusselt number which is presented in Table II (end of this chapter).

Petukhov et al (1963, 1967a, 1967b, & 1969), using distilled water in a tube heated with internal electrical wall resistance (using AC current), studied local heat transfer coefficient. By measuring the temperature at both axial and radial locations, they plotted average Nusselt numbers versus $((x/d) / (RePr))$ and showed the combined free and forced convection effect on laminar heat transfer.

Shannon and Depew (1968, 1969) investigated natural convection effects for a wall resistance (DC current) heated stainless steel tube that also incorporated an unheated calming section. They used two fluids (water and ethylene glycol) to cover a Reynolds number range of 6 to 2300. Their results showed influence of free convection and was correlated using the parameter $((GrPr)^{1/4} / Nu_{GS})$ where Nu_{GS} is the theoretical local Nusselt number found from Siegal, Sparrow, and Hallman (1958).

Hussain and McComas (1970) researched the effect of free convection for Reynolds numbers between 670 and 3800 for air flowing through a uniformly heated horizontal circular tube. They discovered for a Reynolds number below 1200 and far downstream of the entrance, the heat transfer was below the pure forced convection prediction. For larger Reynolds numbers, the results were higher than predicted by forced convection theory. They experienced significant wall temperature variation in the radial direction also.

Siegwarth et al (1969) analyzed the effect of secondary flow on the temperature field and primary flow at the outlet of a long electrically-heated tube. They developed a model for the flow field by dimensional reasoning and found that the secondary flow controls the rate of heat transfer. Their model showed good agreement with the data measured by Readal (1969).

Using a flow visualization technique, Bergles and Simonds (1971) studied the effects of free convection on laminar water flow in horizontal circular tubes with constant heat flux. The tubes were Pyrex E-C Coated Tube with four thermocouples placed 90 degrees apart in the radial direction. Heat was generated in the coating to provide a constant heat flux with nearly zero radial conduction.

Morcos and Bergles (1975) investigated the effects of fluid property variations for laminar flow heat transfer with a fully developed velocity profile in horizontal tubes with constant heat flux. They used both glass and steel tubes in their study.

Ogawa and Kawamura (1986) experimented on air flow in the transition region for Reynolds number ranging from 1940 to 9120. They used a vertical steel tube subjected to a nearly constant heat flux provided by electric sheath heaters and studied the effect produced by four entrance configurations (providing different levels of disturbance) using 16 thermocouples and three pressure taps. They also gathered intermittency data in the transition region for Reynolds numbers of 566 to 16500 using a hot wire anemometer. Their results determined the intermittency factor to be very little influenced by the local bulk Reynolds number and primarily dependent upon the entrance condition. Additionally, their results were not influenced by free convection as the Rayleigh number experienced in their study was confined below 15. For the heat transfer results, they present both laminar and turbulent Stanton number equations and in the transition region, their correlation adds the turbulent and laminar Stanton numbers with an intermittentcy weight factor. Their data is claimed to agree with the correlation. However, by using air as the test fluid, they fail to encompass any range of Prandtl number and cover approximately 0.7-0.8 Prandtl number.

Chen (1988) investigated the entrance effects for square-entrance tubes and found correlations predicting heat transfer for high laminar, transition, and fully turbulent flows. He used an electrically-heated horizontal circular tube and accounted for the peripheral conduction of heat by using eight thermocouples in the radial direction. Using water and diethylene glycol, he covered a Reynolds number range from 121 to 12400 and a Prandtl number range from 3.5 to 282.4. He developed correlations for heat transfer (see Table V) and his data agrees well with the equations. The apparatus used in his study is similar to the setup we elected to use in the present study. However, differences are apparent in that he covers a higher Prandtl number range using diethylene glycol but at the expense of a lower Reynolds number range. He chose to make no provisions for intermittency data collection nor any accommodations for flow visualization experiments. Also missing from his apparatus was the ability to evaluate different entrance configuration and pressure drop data. His single test section was only 12 feet in length and includes only two pressure taps while ours is 20 feet and includes a separate test section using at least 20 pressure taps (see Augustine (1990)). Also in comparison with our digital data acquisition system (see Chapter II), his methods of data collection seem primitive and time consuming. But the biggest deficiency evident from his study was that he neglected to define the transition region based on his data and only refers to the theoretically defined zone of transition.

From this review of the important previous studies related to horizontal pipe flow with constant heat flux, it is important to note that only the work of Chen (1988) and Ogawa & Kawamura (1986) are performed with emphasis on the transition region. Most of the investigations were performed without multiple thermocouples at the various stations resulting in the inability to detect properly the occurrence of mixed convection and neglecting peripheral heat transfer. With the exception of Ogawa & Kawamura (1986), none of the results included the beginning and end of transition and most used short test sections where transition flow was likely to have existed for most of the tube length. The pertinent results of these studies and other similar studies are presented in Tables I through IV in the form of heat transfer correlations.

The correlations appearing in the tables are the most widely accepted and referred to equations for laminar, transition, and turbulent convective heat transfer. Most of the correlations are from the handbook of Kakac, Shah, and Aung (1987). The equations that can be found in chapter four of this handbook (authored by Bhatti and Shah) are all for forced convective turbulent flow and include those of

TABLE I

LAMINAR FLOW FORCED CONVE3CTION
HEAT TRANSFER CORRELATIONS

INVESTIGATOR	CORRELATION	CONDITIONS
Hausen (1943)	$\overline{Nu} = 3\ 66 + \dfrac{0\ 0668(D/L)RePr}{1+0.04[(D/L)RePr]^{2/3}} (\dfrac{\mu}{\mu_w})^{0\ 14}$	Uniform wall temperature
	$Nu = 4\ 36 + \dfrac{[(0.0445RePrD/L)+0\ 00356(RePrD/L)^{5/3})]}{[1+0\ 04(RePrD/L)^{2/3}]^2} (\dfrac{\mu}{\mu_w})^{0\ 14}$	Local value derived from average value and accounting for constant heat flux
Sieder & Tate (1936)	$\overline{Nu} = 1\ 86\ (RePr\ D/L)^{1/3} (\dfrac{\mu}{\mu_w})^{0\ 14}$	$0\ 48 < Pr < 16{,}700$ $0\ 0044 < (\mu/\mu_w) < 9.75$ Correlation recommended for values of $(RePr\ D/L)^{1/3}(\dfrac{\mu}{\mu_w})^{0\ 14} \geq 2$
	$Nu = 1\ 24\ (RePr\ D/L)^{1/3}(\dfrac{\mu}{\mu_w})^{0\ 14}$	Local value derived from average value
Kays (1955)	$\overline{Nu} = 4\ 36 + \dfrac{0.36RePr}{(L/D)} \ln [\dfrac{(L/D)}{0.0011RePr} + 1](\dfrac{\mu}{\mu_w})^{0\ 14}$	Application holds good only for small temperature differences between wall and bulk temperatures
	$Nu = 4\ 36 + \dfrac{0.36RePr}{(L/D + 0\ 0011RePr)} (\dfrac{\mu}{\mu_w})^{0\ 14}$	Local value derived from average value

TABLE II

LAMINAR FLOW MIXED CONVECTION
HEAT TRANSFER CORRELATIONS

INVESTIGATOR	CORRELATION	CONDITIONS
Mori et al (1966)	$\overline{Nu} = 0.6\,(GrPrRe)^{0.2}\{1 + \frac{1.8}{(GrPrRe)^{0.2}}\}$	Constant heat flux Re = 100 to 13,000
Eubank & Proctor (1951)	$\overline{Nu} = 1.75\,[PrRe\,D/L + 0.04\,(GrPr\,D/L)^{.75}]^{1/3}(\frac{\mu}{\mu_w})^{0.14}$	
	$Nu = 1.75\,[PrRe\,D/L + 0.04\,(GrPr\,D/L)^{0.75}]^{1/3}(\frac{\mu}{\mu_w})^{0.14}$ $- 0.583\,[PrRe\,D/L + 0.03\,(GrPr\,D/L)^{0.75}]/$ $[PrRe\,D/L + 0.04\,(GrPr\,D/L)^{0.75}]^{2/3}(\frac{\mu}{\mu_w})^{0.14}$	Local value derived from average value
Colburn (1933)	$\overline{Nu} = 1.5\,(RePr\,D/L)^{1/3}\,(\frac{\mu}{\mu_w})^{1/3}\,(1+0.015Gr^{1/3})$	
	$Nu = (RePr\,D/L)^{1/3}\,(\frac{\mu}{\mu_w})^{1/3}\,(1+0.015Gr^{1/3})$	Local value derived from average value

TABLE III

TURBULENT FLOW FORCED CONVECTION
HEAT TRANSFER CORRELATIONS

INVESTIGATOR	CORRELATION	CONDITIONS
Sieder & Tate (1936)	$\overline{Nu} = 0.023\ Re^{0.8} Pr^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.14}$	$0.7 \leq Pr \leq 16{,}700$ $Re \geq 10{,}000$ $L/D \geq 60$
Colburn (1933)	$\overline{Nu} = 0.023\ Re^{0.8} Pr^{1/3}$	$0.6 \leq Pr \leq 160$ $Re \geq 10{,}000$ This relation can be used only in cases of small to moderate temperature differences.
Dittus-Boelter (1930)	$\overline{Nu} = 0.023\ Re^{0.8} Pr^{0.4}$	$0.7 \leq Pr \leq 120$ $Re \geq 10{,}000$ $L/D \geq 60$ To be used for small to moderate temperature differences.
Sleicher & Rouse (1975)	$\overline{Nu} = 5 + 0.015\ Re^{a} Pr^{b}$ where $a = 0.88 - (0.24/(4+Pr))$ $b = 0.333 + 0.5\ exp(-0.6\ Pr)$	$0.1 < Pr < 10{,}000$ $10{,}000 < Re < 100{,}000$

TABLE III CONTINUED

INVESTIGATOR	CORRELATION	CONDITIONS
Petukhov & Popov (1963)	$\overline{Nu} = \frac{RePr}{a}\left(\frac{f}{8}\right)\left(\frac{\mu}{\mu_w}\right)^n$ where $a = 1.07 + 12.7\,(Pr^{2/3} - 1)\,(f/8)^{0.5}$ $n = 0.11$ for heating ($T_{wall} > T_{bulk}$) $f = (1.82 \log Re - 1.64)^{-2}$	$10^4 < Re < 5 \times 10^6$ $0.5 < Pr < 2{,}000$ $0.08 < \frac{\mu_w}{\mu} < 40$
McAdams (1954)	(a) $\overline{Nu} = 0.023\, Re^{0.8} Pr^{0.4}\left(1 + \frac{1.4}{L/D}\right)$	Fully developed velocity profile where the energy exchange is initiated Holds only for tube lengths less than twice the entrance length
	$Nu = 0.023\, Re^{0.8} Pr^{0.4}$	Local value derived from average value
McAdams (1954)	(b) $\overline{Nu} = 0.023\, Re^{0.8} Pr^{0.4}[1 + (L/D)^{-0.7}]$	For the case where there is a contraction at the point where heating or cooling starts (This includes the correction for L/D)
McAdams (1954)	(c) $\overline{Nu} = 0.0676\, Re^{0.852} Pr^{0.284}$	$2{,}600 < Re < 163{,}000$ $0.73 < \frac{RePr}{\overline{Nu}} < 95$ $11 < \overline{Nu} < 500$

TABLE III CONTINUED

INVESTIGATOR	CORRELATION	CONDITIONS
Nusselt (1931)	$Nu = 0.036\, Re^{0.8} Pr^{1/3} (D/L)^{0.055}$	$10 < L/D < 400$ For entrance region and the flow is not fully developed
von Karman (1934)	$\overline{Nu} = \dfrac{(f/8) Re Pr}{1 + 5\sqrt{f/8}\,\{(Pr-1) + \ln(1 + \frac{5}{6}(Pr-1))\}}$ where $f = 0.316 Re^{-0.25}$ $10^4 < Re < 5x10^4$; $f = 0.184 Re^{-0.2}$ $3x10^4 < Re < 10^6$	
Colburn (1933)	$\overline{Nu} = 1.86\, Re^{1/3} (L/D)^{-1/3} (Pr)^{0.42} (\frac{\mu}{\mu_w})^{0.14}$	$Re > 4x10^3$; $0.46 < Pr < 592$
	$Nu = 1.239\, Re^{1/3} (L/D)^{-1/3} (Pr)^{0.42} (\frac{\mu}{\mu_w})^{0.14}$	Local value derived from average value
Gnielinski (1976)	$\overline{Nu} = \dfrac{(f/2)(Re-1000) Pr}{1 + 12.7 (f/2)^{0.5} (Pr^{2/3} - 1)}$ where $f = (1.58 \ln Re - 3.28)^{-2}$	$2,300 < Re < 5x10^6$; $0.5 < Pr < 2000$
Friend & Metzner (1958)	$\overline{Nu} = 0.015\, Re^{0.83} Pr^{0.42} (\frac{\mu}{\mu_w})^{0.14}$	$Re > 4x10^3$; $0.46 < Pr < 592$

TABLE III CONTINUED

INVESTIGATOR	CORRELATION	CONDITIONS
Churchill (1977)	(a) $\overline{Nu} = 6.3 + \dfrac{0.079(f)^{0.5}RePr}{(1+Pr^{0.8})^{5/6}}$ where $(f)^{0.5} = \dfrac{1}{2.21 \ln (Re/7)}$	All Pr Re ≥ 10,000
Churchill (1977)	(b) $\overline{Nu}^{10} = \overline{Nu}_l^{10} + \{\dfrac{\exp[(2200 - Re)/365]}{\overline{Nu}_l^2} + \dfrac{1}{\overline{Nu}_t^2}\}^{-5}$ where $\overline{Nu}_l = 4.364$ $\overline{Nu}_t = 6.3 + \dfrac{0.079(f)^{0.5}RePr}{(1+Pr^{0.8})^{5/6}}$ $(f)^{0.5} = \dfrac{1}{2.21 \ln (Re/7)}$	All Pr $2,100 \leq Re \leq 10^6$ Recommended for transition flow regime

TABLE IV

MIXED CONVECTION HEAT TRANSFER CORRELATIONS
(SQUARE-EDGED ENTRANCE)

INVESTIGATOR	CORRELATION	CONDITIONS
Chen (1988)	(a) $Nu = \{4.364 + 0.00106Re^{0.81}Pr^{0.45}(1+14e^{-0.063x/D}) + 0.268(GrPr)^{1/4}(1-e^{-0.042x/D})\}(\mu/\mu_w)^{0.14}$	$121 < Re < 2{,}100$ $3.5 < Pr < 282.4$ $930 < Gr < 67{,}300$
	(b) $Nu = 0.00392RePr^{1/3}(1+1.19e^{-0.308x/D})(\mu/\mu_w)^{0.14}$	$4{,}600 < Re < 7{,}000$ $3.5 < Pr < 7.4$ $45{,}570 < Gr < 1.04 \times 10^6$
	(c) $Nu = 0.01426Re^{0.86}Pr^{1/3}[1+1.15e^{-x/(3D)}](\mu/\mu_w)^{0.14}$	$7{,}000 < Re < 12{,}400$ $3.5 < Pr < 7.4$ $45{,}570 < Gr < 1.04 \times 10^6$
	(d) $Nu = [(1-y)\ Nu_l + y\ Nu_{ut}]$ where $y = (Re-2100)/(4600-2100)$ Nu_l = correlation for $121 < Re < 2{,}100$ Nu_{ut} = correlation for $4{,}600 < Re < 7{,}000$	$2{,}100 < Re < 4{,}600$

Sleicher & Rouse, von Karman, Gnielinski, Petukhov & Popov, Dittus-Boelter, and Churchill. Additional turbulent forced convective equations (not in the handbook) include Sieder & Tate (1936), Friend & Metzner (1958), Nusselt (1931), two by Colburn (1933), and three by McAdams (1954). The equation by Churchill has limits including the commonly quoted start of transition to fully turbulent flow ($2100 < Re < 10^6$), and this is the equation that is modified to predict our forced and mixed convection data for all flow regions (see Chapter III). All of the above mentioned correlations can be viewed in Table III. The Table IV contains three correlations by Chen (1988) describing mixed convection for $121 < Re < 12400$. The laminar forced convection equations (Table I) included are from Hausen (1943), Sieder & Tate (1936), and Kays (1955) and for mixed convection laminar flow, Table II includes correlations by Eubank & Proctor (1951), Colburn (1933), and Mori et al (1966).

CHAPTER II

EXPERIMENTAL APPARATUS

Study of forced and mixed convective heat transfer under a constant wall heat flux condition in a horizontal circular tube in the transition region with different inlet configurations required a specialized and flexible setup. To achieve the required versatility, several unique features were incorporated into the system which enabled the use of any entrance and calming section configuration desired. A schematic diagram of the heat transfer experimental apparatus is shown in Figure 2.1. The figure also shows that a similar setup was used to study the pressure drop characteristics for transition flow, see Augustine (1990).

Presented in this chapter is a description of the experimental apparatus used including the necessary instrumentation details. Following the apparatus descriptions is an explanation of all necessary calibration procedures which ensured the accuracy of the apparatus and instrumentation elements. The final coverage of this chapter will detail the data reduction techniques and the software assisting in this task.

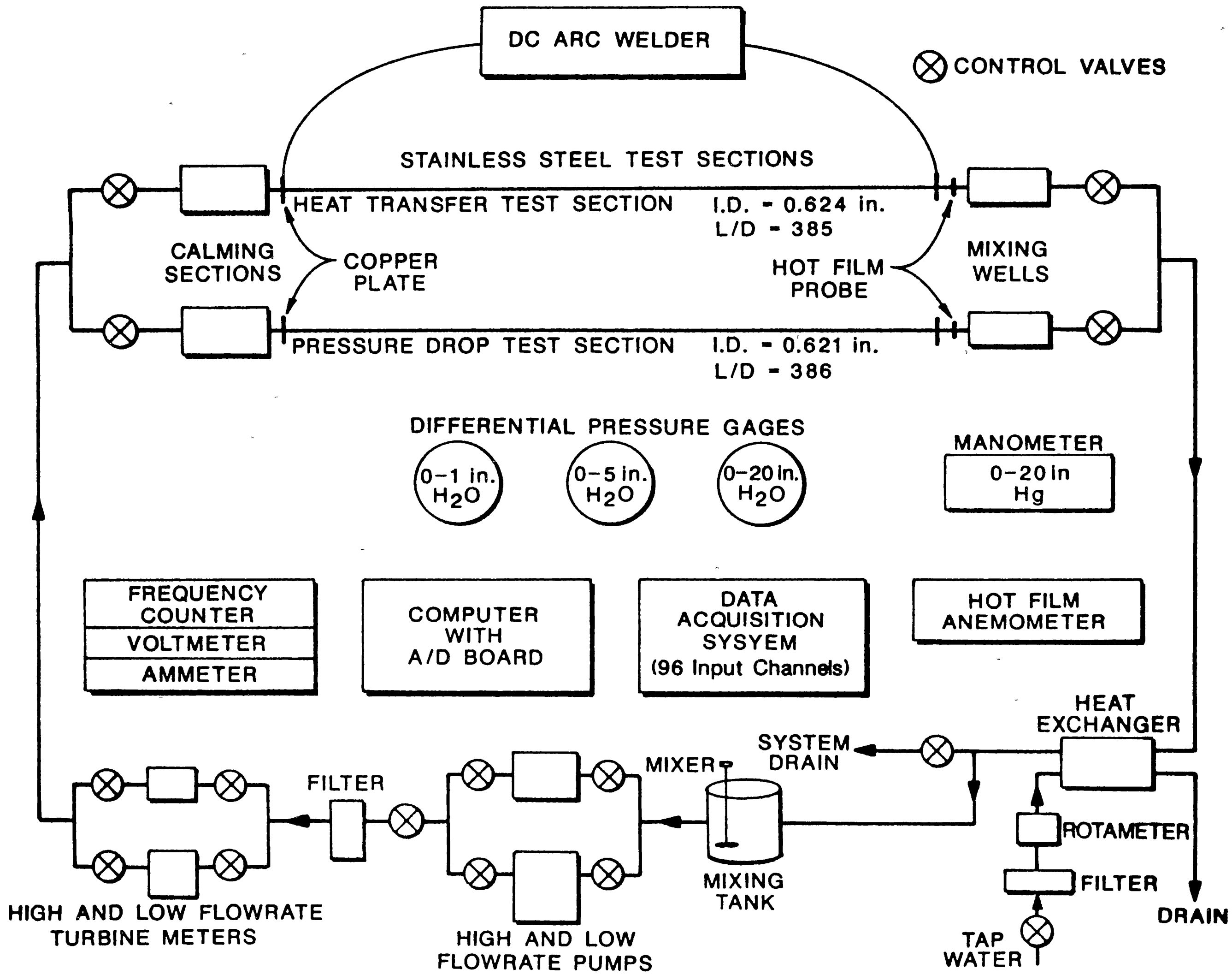


Figure 2.1 Schematic of Experimental Apparatus

Description of The Equipment

Test Section

The test section is a horizontal seamless 316 stainless steel circular tube with an average inside diameter 0.624 $\pm$ 0.0005 inches (15.850 $\pm$ 0.0127 mm) and outside diameter 0.748 $\pm$ 0.0005 inches (18.999 $\pm$ 0.0127 mm). The test section total length was 240.25 inches (6.10 m) providing a maximum length to diameter ratio (L/d) of 385. The tube was procured from Precision Fitting and Tubing Co. of Tulsa, Oklahoma.

Modifications made to the test section included two 5/64 inch holes drilled using a #60 R18C cobalt bit allowing the location of pressure taps near the inlet and exit, to give total test section pressure drop via a mercury manometer. To ensure that no flow disturbances were introduced a Reynold Tool Products (Anaheim CA) model T-A BA 5/64 inch ridge reamer was used to de-bur the pressur tap locations. A 1/4-inch-thick, 5-inch-wide, 9-inch-long copper plate was silver soldered to the exit end, and a 1/4-inch-thick, 10-inch by 10-inch square plate was silver soldered to the inlet of the test section. Supporting copper material was bolted to the end plates such that bus bars could be dropped into position to receive welding cables for heat addition. The plates and bus bars were sized as such only to conform to the space limitations of the apparatus structure.

At the end of the test section, a rectangular aluminum housing for the hot film probe was heat expanded, positioned, and cooled to water-tight fit over the test section. The approximate housing dimensions are 2.5 inches-tall, 2.25 inches-wide, and 1.5 inches-deep. A slot was machined through the tube wall to allow insertion of the probe. The probe housing and slot were machined by personnel at the Mechanical Engineering North Labs, at Oklahoma State University. The entire test section was surrounded with fiberglass pipe wrap insulation, followed by a thin polymer vapor seal to prevent moisture penetration. An approximate total thickness of the insulation materials is 1.25 inches.

When the test section was completed, it was leveled using a transit provided by the School of Civil Engineering, Oklahoma State University. This procedure required that a ruler with a 2 inch needle fastened to it be pushed through the insulation at several distances along the tube length until contact with the test section occurred. Then height measurements were recorded at each tube interval and adjustments made with the use of shims and damping material until all intervals indicated the same level.

Calibration runs were made to compare against the established correlations for fully turbulent flow and the details of those procedures are outlined in the experimental calibrations section of this chapter.

Calming and Inlet Sections

The calming section was constructed as a flow straightening and turbulence reduction device. Details of the construction and features incorporated into the calming and inlet sections can be seen in Figure 2.2. A uniform, low-turbulence flow was produced in the calming section which itself entered the inlet section. Inside the inlet section the test fluid flowed undisturbed through 9.25 inches of a 6.5 inch diameter plexiglass tube before it met the square-edged test section entrance. This was done to ensure a uniform velocity distribution upon entering the test section.

Inside the calming section are three screens which were machined from 1/2 inch acrylic plastic (plexiglass) sheet and contained uniformly distributed 5/16 inch diameter holes in an equilateral triangular pitch. These screens are the principal turbulence busters and are followed by a flow straightening component. This component is comprised of tightly packed 4 inch-long soda straws sandwiched between two 23 gauge galvanized steel mesh screens. Downstream of the flow straightening device an aluminum insect type mesh screen supported with 23 gauge steel mesh is located to eliminate any remaining low level turbulence before entering the inlet section. All the components were housed in a 7 inch O.D. plexiglass tube having 3/16 inch wall thickness. To provide lateral stability for all the internal calming section components, spacers were constructed from 6.5 inch

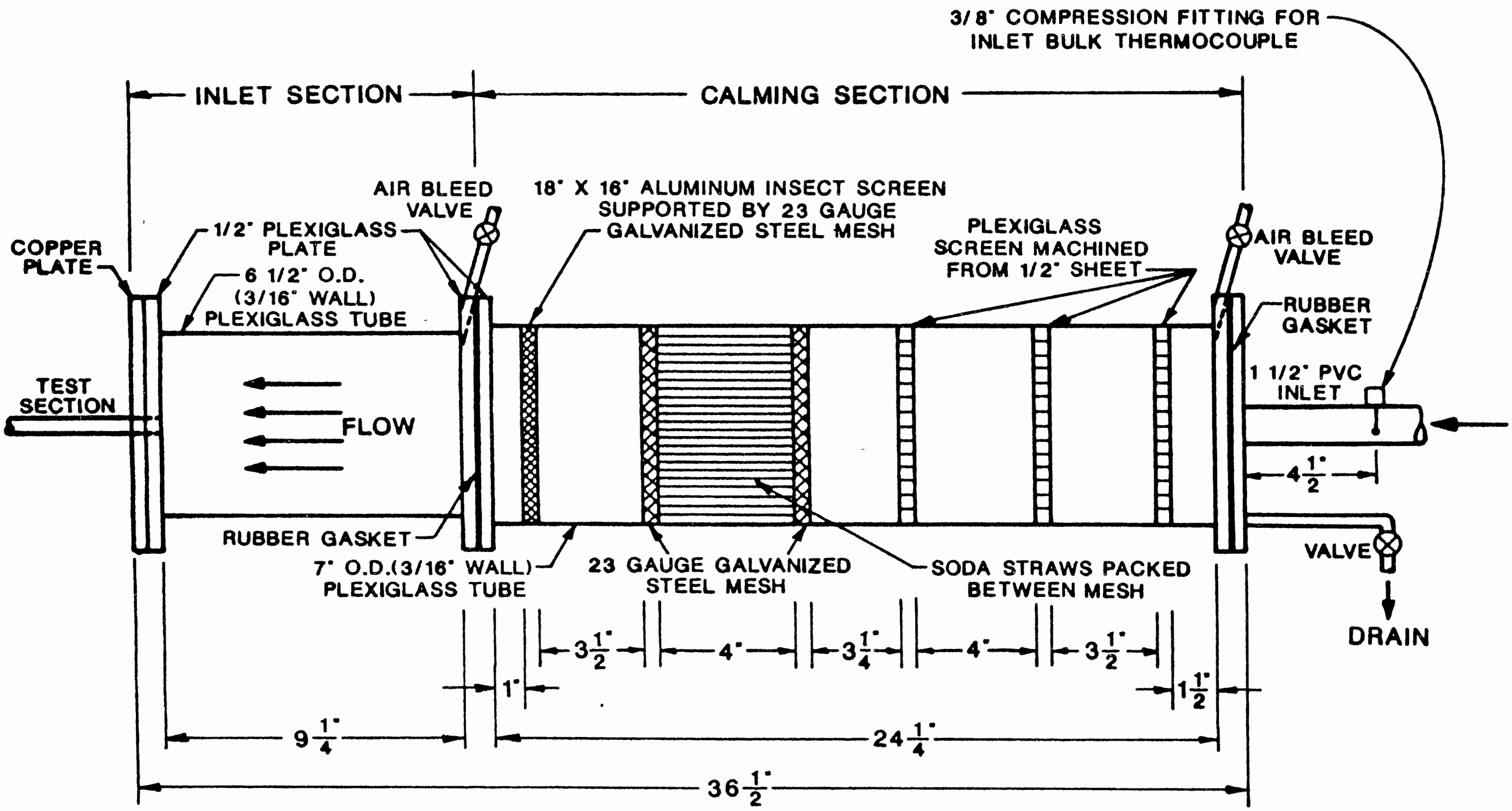


Figure 2.2 Detail Schematic of Calming and Inlet Sections.

O.D. plexiglass tube also having 3/16 inch wall thickness. The spacers were cut to the desired length and placed between the components such that no horizontal movement was possible.

The flow then entered the inlet section which was constructed of 6.5 inch plexiglass tube and 3/16 inch wall thickness. The inlet section inserted into the calming section with only a slight annular gap which was sealed using rubber O-rings. Acrylic flanges were machined and cemented to both ends of the inlet and calming sections. When the sections are joined, the flanges meet and are bolted together with eight bolts and a rubber gasket between to further protect against leakage. With this arrangement, either the square-edged or reentrant entrances could be used and with some inlet section modification, a smooth bell shaped inlet could be employed.

Air-escape valves were located on both the calming and inlet sections such that any accumulation of air could be eliminated.

Thermocouples

Thermocouples were placed on the outer surface of the tube wall at close intervals near the entrance region and at greater intervals further downstream. Thirty-one stations were designated with four thermocouples per station, placed at 90 degree intervals around the periphery. Omega TT-T-30 Copper-Constantan insulated T-type thermocouple wire was used with Omega EXPP-T-20 extension wire for relay to the data acquisition system. The thermocouple beads were fabricated using a Tigtech Inc. model 116 SRL thermocouple welder. Figure 2.3 shows the positioning of the thermocouples. Each thermocouple was labeled with two identification numbers, the first number specifying the station (1-31), the second number specifying the peripheral location (1-4). Due to the limitations of the data acquisition system only 96 of the 124 thermocouples were used, requiring some stations and peripheral locations be ignored during data collection. The extra unused thermocouples provide additional flexibility should insight at their location be desired. These excluded stations were numbers 2, and all even stations 24 to 30. All stations up to and including station twenty-two were labeled looking at the tail of the fluid flow with peripheral location number one being at the top of the tube, number two being 90 degrees in the clockwise sense, number three being at the tube bottom, and number four 90 degrees from the bottom (see Figure 2.3). Starting with station number twenty-three and including just the odd stations, terminating at station thirty-one, only two peripheral locations were used. For these stations, location number one was at the top of the tube and number two was at the tube bottom (see Figure 2.3). The first two channels of the data acquisition system were allocated to thermocouples monitoring the room temperature. These are the same type and material as all others and were

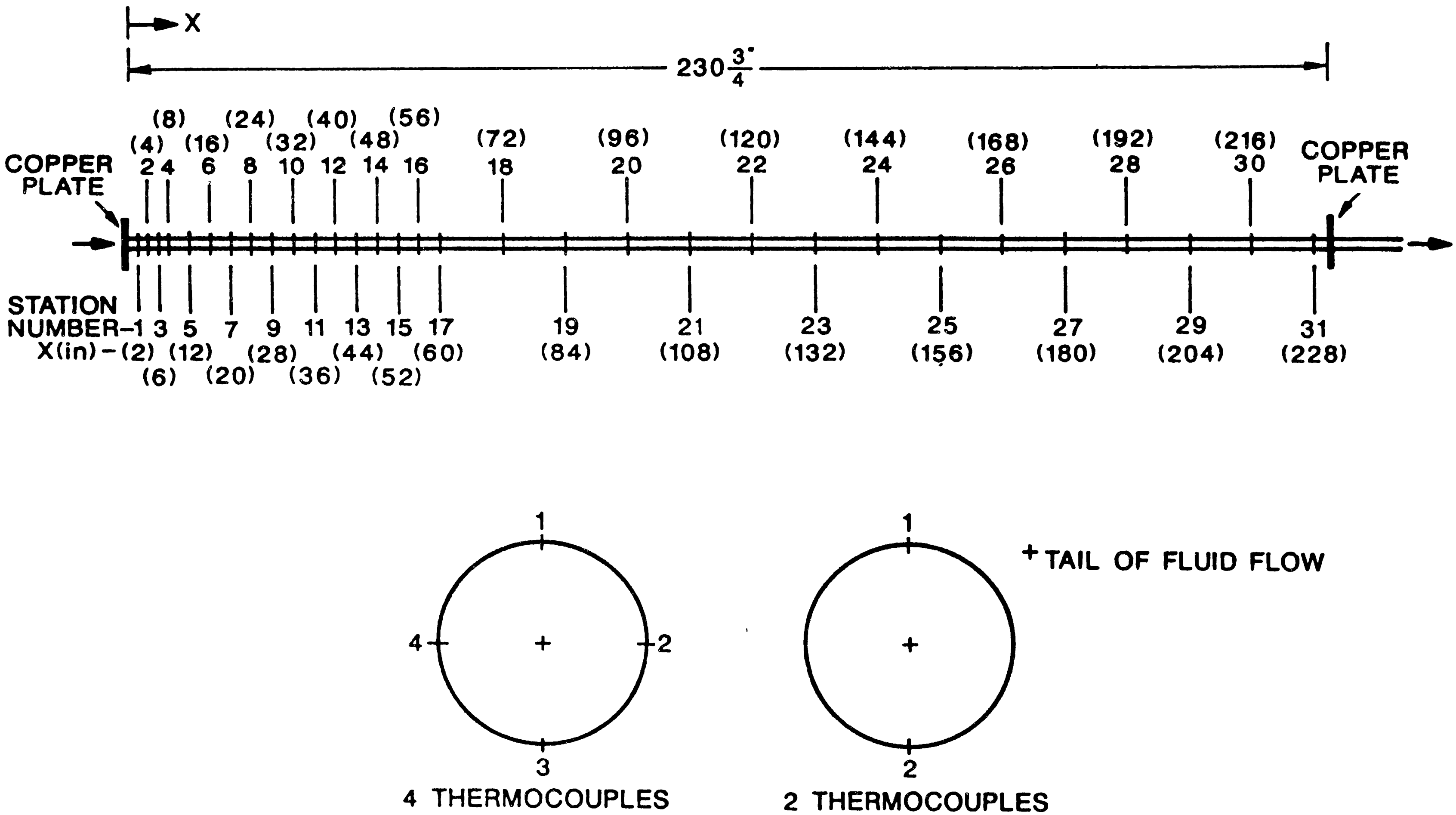


Figure 2.3 Heat Transfer Test Section Thermocouple Distribution

located between the two test sections near the entrance and exit of the tubes. The readings from both room temperature thermocouples were averaged to provide a representative value.

Thermocouple beads were attached to the outside of the tube wall using Omegabond 101, a two-part epoxy adhesive providing a high thermal conductivity (0.6 Btu/hr-ft-F), and very high electrical resistivity of (1E15 ohm-m). An initial drop (1mm radius), was placed at each thermocouple location and allowed to cure for twenty four hours. Each drop was then filed to a height above the tube of roughly 0.5mm, with an ordinary file. Each thermocouple was placed upon the hardened Omegabond surface (preventing direct contact with the tube surface), held in place with a strip of electrical tape such that the bead and hardened surface were exposed, and coated with another drop of Omegabond to ensure permanent positioning. When completely hardened, the electrical tape supporting the thermocouples was removed. The thermocouple extension wires were then bundled in groups of four, fastened to the wood channels supporting the test section using wiring bundle rings, and connected to the data acquisition system.

The thermocouples on the tube required calibration and these procedures are detailed in the experimental calibrations section of this chapter.

Data Acquisition System

A Cole-Parmer ninety-six input MAC-14 data logger was interfaced with an AT personal computer to provide digital data acquisition for the temperature measurements. It accepts input voltages from 0.3 micro-volts to 10 volts, has an accuracy of $\pm 0.02\%$ of range, and has 16 bit resolution. For further inquiries regarding specific parameters associated with the MAC-14, consult the operation manual.

Connection to the computer is through shielded cable to an RS232 port, and to the printer via the printer port. Menu-driven software (MS), is used in conjunction with signal conditioning (SC), real time graphics (RTG), and printer driver (PD) software to handle data input. The MS software allows each channel to be marked with the thermocouple identification labels and lets the user specify the logging interval, disk storage, or screen-only monitoring. The SC software provides additional columns on each channel for scale factors and units, also it performs the thermocouple conversions for all channels (volts to degrees). The RTG is a much expanded version of MS which gives real time graphic display in scientific graph or industrial stripchart format. RTG requires both MS and SC options for its use. The MAC-14 data logger is entirely self-contained requiring no special mounting. Additional software was required for post-experimental data reduction and was not available with the data logger (see experimental procedure section of this chapter for software discussion).

The IBM compatible AT personal computer has a 40 MB hard drive, dual floppy disk drives, an EGA monitor, a 80827-8 coprocessor, and a Panasonic 1091i printer. The computer was used for data logger and printer interfacing, data storage and reduction, graphics, and text production.

Supplemental Data Acquisition

An Electronic Controls Design (ECD) model 5100 digital data logger with forty channel capacity was used to support the temperature recording capabilities of the MAC-14 by storing the fluid bulk temperatures, and the four stream temperatures of the heat exchanger. In addition to supplemental heat transfer data logging duty, the model 5100 was the main acquisition system for the pressure drop tube thermocouples. The model 5100 has a resolution of 0.1 degree F, over a temperature range of -158 to 752 degrees F and a $\pm$0.1 degree F conformity error over a range of -105 to 400 degrees F. For additional information regarding the model 5100, consult the operation manual. Using PC-TALK, the model 5100 can interface with the personal computer through a shielded cable to a second RS232 port. The data cache memory is transferable to the computer hard disk or floppy diskette via PC-TALK, and it incorporates a strip chart recorder for instant data access.

The model 5100 required calibration both actively and passively. The details of the calibrations are described in the experimental calibrations section of this chapter.

Mixing Well

To ensure a uniform fluid bulk temperature at the exit of the test section, a mixing well was utilized. The cylindrical frame was constructed from high temperature tolerant poly-vinyl-chloride (CPVC). Alternating acrylic baffles were placed first from the top, blocking nearly 60 percent of the flow area, and then from the bottom, also blocking approximately 60 percent of the flow area. This provided an overlapping baffled passage forcing the fluid to encounter flow reversal and swirling regions. Near the end of the well a T-type thermocouple (TT-T-30) permanently fixed inside a 1/4 inch aluminum tube approximately 2.5 inches long was inserted into the flow area through a brass compression fitting. Using EXPP-T-20 extension wire the bulk exit thermocouple was connected to the model 5100 data logger. The mixing well is connected to the test section by an acrylic reducing adaptor cemented inside an extension junction. By using cement and silicone sealant, sealing is ensured and construction details are shown in Figure 2.4.

Constant Heat Flux Source

A Lincoln Idealarc DC-600 three phase rectified type electric welder was used with variable voltage to produce DC current. The current is delivered to the test section through Radaflex AWG 4/0 welding cable attached to the copper bus bars located at the inlet and exit flanges. Heat is generated internally to the steel tube wall due to its

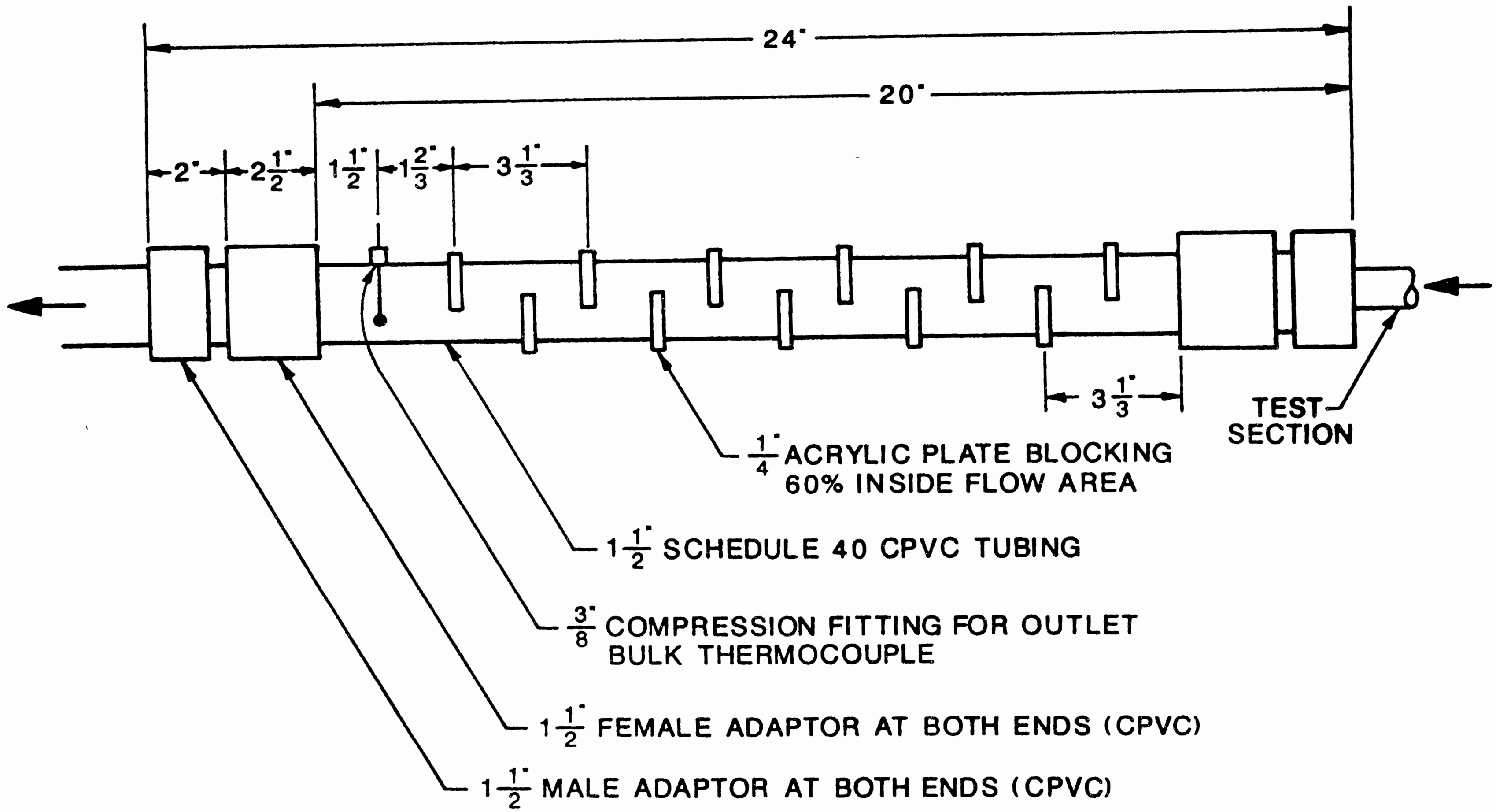


Figure 2.4 Detail Schematic of Mixing Well

electrical resistivity. The DC-600 is rated for 100% duty cycle at 600 amps and 44 volts for either 60 or 50 Hertz, giving a maximum power output of 26.4 KW.

To ensure minimal room heating and vibrational effects from the welder, it was located at an external wall where it exhausted hot air (outside) through a custom made plenum to a 12.25 inch square duct, and brought in cooler air through a 7.25 inch-wide, 18.25 inch-tall duct protected by a steel grate. Much of the exhaust noise is alleviated in this manner. A large 37 inch-tall, 41 inch-wide, and 57 inch-deep plywood box fitting flush with the external wall and layered on all internal sides with approximately 2 inches of duct insulation isolates the vibration effects due to the welder's operation. To control transmission of vibration through the floor, the welder was placed on approximately 10 refrigerator type rubber damping pads.

Voltmeter

A Hewlett-Packard model 3468B digital multimeter was used to calculate the actual voltage drop across the test section. The range available for DC volt measurement is 1 microvolt to 300 volts. An accuracy of 1% of the reading, and a resolution of 10 microvolts are possible with this model.

DC Ammeter

The current passing through the test section wall was measured with a Weston Instruments Division model 931 ammeter placed in parallel with a 50 millivolt shunt. It was calibrated by Mr. Gerald Stotts, manager of Electronics Laboratory, Electrical and Computer Engineering at Oklahoma State University. The accuracy is less than one percent of its 750 amp full scale reading. In addition the ammeter attached to the welder was calibrated against the model 931 as a back-up method to calculating the corrected amperage through the test section should we ever be without the ammeter.

Heat Exchanger

An ITT Standard model BCF 4036 one shell and two tube pass heat exhanger purchased from Thermal Engineering Company of Tulsa was used to cool the test fluid to an allowable and steady state inlet bulk temperature. The shell has an effective surface area of 21.2 ft^2 (1.97 m^2) and a maximum duty of 67190 Btu/hr (19.7 KW). The cooling water was provided from tap through an Omega FL-9028 rotameter with a maximum flow rate range of 4 to 28 GPM, accuracy of $\pm 5\%$ of full scale, and a repeatability of $\pm 1\%$ of full scale. A Teek Water Systems model 2P277 double cartridge filter removed impurities, and 1.25 inch schedule 40 poly-vinyl-chloride (PVC) tubing carried the coolant to the heat exchanger, exiting to the waste water trough. Four

TT-T-30 T-type thermocouples mounted inside 1/4 inch aluminum tubes approximately 2 inches long with silicone sealant were inserted through brass compression fittings allowing the four stream temperatures to be monitored with the model 5100 data logger. The heat exchanger was 39.625 inches in length (1.0 m), and was mounted on a wooden saw horse for stability.

Fluid Reservoir

A 16 gallon cylindrical polyethelene tank, cover and mixer was purchased from Industrial Plastics Corp. The approximate tank dimensions are 16 inches in diameter, 21 inches in height. The mixer having a 20 gallon capacity with 1/40 HP 1550 rpm motor was used to ensure uniform temperature distribution in the bath.

Pumps

For low flow rates, a pump manufactured by Little Giant Pump Co. model 4-MD, was used. It produces a flow rate of 4.5 GPM at 3100 rpm using a 1/12 HP motor.

At high flow rates, another pump manufactured by Oberdorfer Pumps, model SKH35FN193T, was used. It produces a flow rate of 11 GPM at 3450 rpm using a General Electric 1/3 HP motor.

To minimize the noise and vibrational effects of the pumps during operation, they were mounted inside a plywood box using rubber damping material approximately 3/8 inchesthick. The box has dimensions 57 inches-wide, 30 inches-high, and 16 inches-deep. On all interior box walls duct insulation material was placed nearly 1 inch-thick, providing acoustic vibration suppression. The whole pump containment box was isolated from the floor with 12 refrigerator type rubber dampers to prevent vibration transmission. In addition flexible hoses connect the pump box at both upstream and downstream ends, attempting to prevent vibration transmission to the fluid return tubing.

<u>Turbine Meters</u>

For small flow rates a Cox Flowmeters model AN 8-6, 1/2 inch turbine meter was used over a frequency range of 140 to 1140 Hertz, giving flow rates from 0.5 to 6.0 GPM. For larger flow rates a Halliburton 1 inch turbine meter was used over a frequency range of 50 to 150 Hertz, for flow rates to approximately 10.5 GPM. Both turbine meters had a linear accuracy of ±0.5% of reading, and repeatability of less than ±0.10% of reading.

The turbine meters were calibrated on several occasions and the procedure details are outlined in the experimental calibrations section of this chapter.

The turbine meters were incorporated into an assembly using 1.5 inch PVC ball valves at both ends of each to ensure no fluid entrainment from the one which is inactive at any given time.

Frequency Meter

A Hewlett-Packard model 5314A universal counter was used to indicate the frequency of the turbine meters during data collection. Input frequency range for the 5314A is 10Hz to 100 MHz, with a sensitivity of 25 millivolts rms to 100 MHz, and 0.075 volts peak to peak at a minimum pulse of 5 nanoseconds. The resolution and accuracy are to the least significant digit (LSD).

During operation of the turbine meters at low frequency, frequency instability problems were observed. To counter this problem, a variable gain amplifier was constructed by an Instruments Technician in the Mechanical & Aerospace Engineering Instruments Shop. It has a possible gain from 1 to 5, and produced a stable signal to the counter under the previously mentioned conditions.

Test Fluids

The fluids used were distilled water, ethylene glycol, and a 60% ethylene glycol mixture (40% distilled water by mass fraction). The ethylene glycol was obtained from Delta Distributors in Tulsa, Oklahoma, and the distilled water from Mr. Charles Baker, lab manager Chemical Engineering, at Oklahoma State University.

Fluid Return Tubing

At the end of the mixing well, the return tubing begins, high temperature 1.5 inch schedule 40 CPVC tubing takes the hot fluid to the heat exchanger. Exiting the heat exchanger, standard 1 inch schedule 40 PVC tubing allows for system drain or return to reservoir via 1 inch PVC ball valves. Leaving the reservoir, 1/2 inch schedule 40 PVC tubing carries the fluid to the pump box assembly described earlier. Upon exiting the pump assembly, a 1/2 inch stainless steel ball valve is used for flow control and a Teel Water Systems model 2P277 double cartridge filter system using two model 1P753A cartridges removes all impurities as small as 5 microns from the flow. This is essential to protect the fragile hot film probe. Next the fluid traveled via 1.5 inch PVC schedule 40 tubing to the turbine meter assembly previously described. Control valves are used to provide flow to only the test section operating at any given time, with 1.5 inch PVC ball valves. The last element in the loop is the inlet bulk temperature sensing thermocouple. It is T-type with extension wire the same as all other thermocouples and is inserted into 1.5 inch schedule 40 PVC pipe with a 3/8 inch compression fitting. The same PVC tube attaches to the calming section approximately 4 inches downstream of the bulk thermocouple.

The return loop of the test apparatus has inherent to its design and construction, flexibility for the addition of a flow visualization test tube, anticipated from inception of the project.

Experimental Calibrations

Upon completing the experimental setup construction and acquiring the monitoring equipment, calibrations of all the equipment and most of the apparatus was required. The accompanying manuals gave instructions for calibration of the equipment while existing standard procedures were used for calibrating apparatus components. The following sections outline the details of these calibrations.

Thermocouples

For the MAC-14 data acquisition system no calibrations were required. However the thermocouples connected to the system were calibrated by means of a constant temperature bath. A steady state condition was accomplished with no heat addition when all thermocouples indicated nearly the same temperatures (within 0.5 F). Eleven sets of data were collected using the MAC-14. Using CALHT (see Appendix C) an average test section temperature and deviation from this temperature for each thermocouple location was calculated for all eleven data sets. The thermocouple deviations were then stored in files corresponding to each of the eleven isothermal runs. Using GETAVE (Appendix C), an average deviation for each test section thermocouple was calculated (over all eleven files) and stored in a file named CORR.DAT (Appendix C). This file was then used to correct the thermocouple data collected throughout the experimental data collection period. In Figure 2.5 the average variation in deviation along the tube for a typical constant temperature run is shown. When the deviations are subtracted from the uncorrected temperatures the corrected temperatures are found for each run. However, as described above we found an average correction value for each thermocouple stored in CORR.DAT. Figure 2.6 shows four comparison plots of both uncorrected and corrected thermocouple temperatures for peripheral locations 1 to 4 along the tube length.

The thermocouples measuring the temperatures of the heat exchanger streams, and the inlet and exit bulk temperature thermocouples were calibrated against a platinum resistance thermometer in the Chemical Engineering Lab. A Yellow Springs Instruments Co. platinum thermometer model 8163Q was calibrated and certified by the National Bureau of Standards (NBS), and connected to a Leeds and Northrup model 8069-B resistance bridge. Also a Leeds and Northrup galvanometer model 9834-2 was used as a null detector. A Rosemount model 910AB, adjustable constant temperature bath provided the environment for the thermocouples during calibration, using a silicon oil grade SF1154. The thermocouples of interest with extension wires were placed in the oil and allowed to reach an initial temperature of 78 degrees F. The null detector was zeroed and the reading from the resistance bridge was compared to the NSB conversion chart for the platinum thermometer giving temperature. Calibration data was collected to a maximum temperature of 150 F, by 1/2 F intervals. After conversion

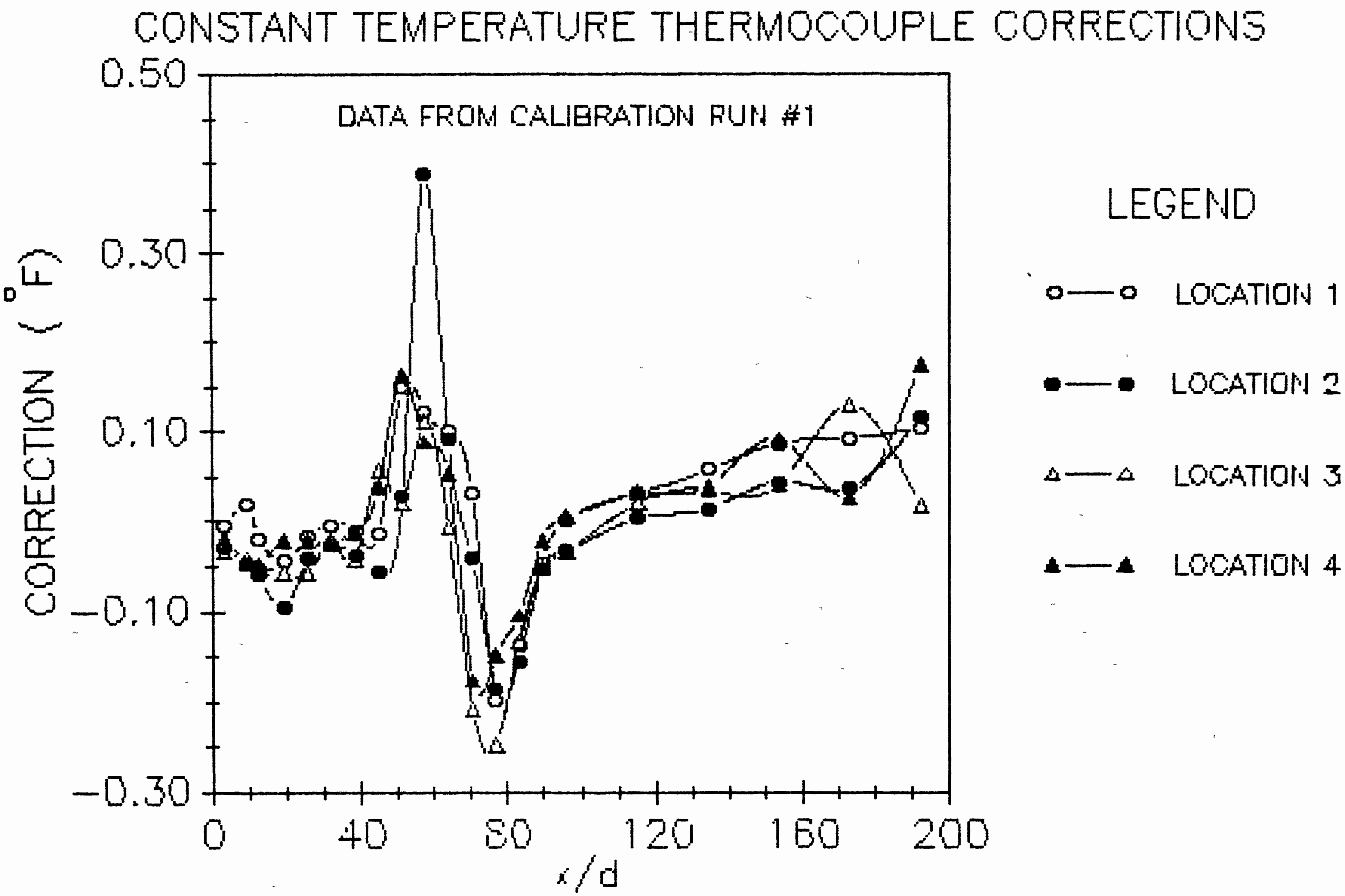


Figure 2.5 Thermocouple Location Correction Data

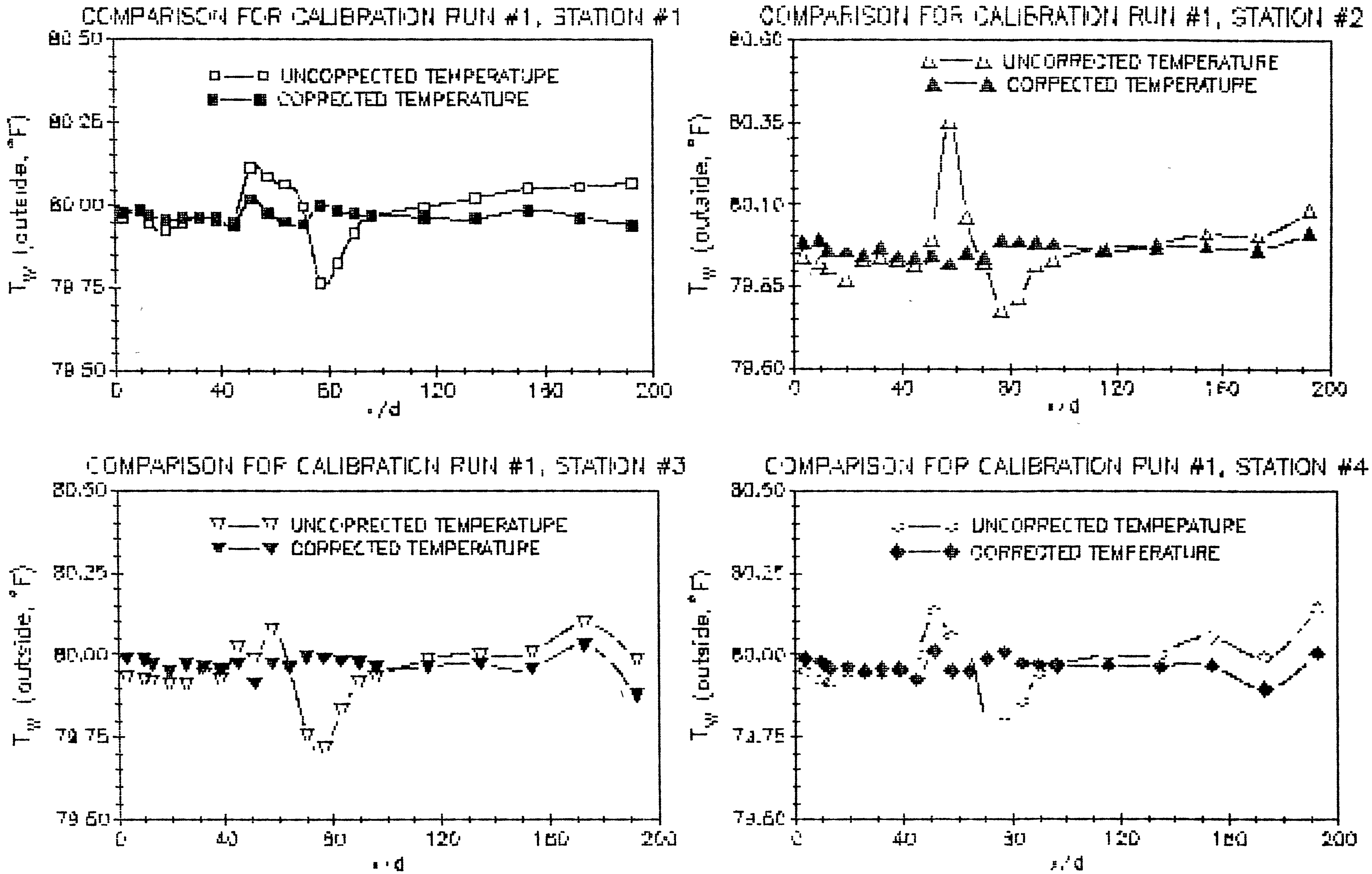


Figure 2.6 Comparisons of Location Uncorrected and Corrected Temperatures

a comparison with the model 5100 data logger readings allowed a deviation table to be assembled. The conversion data did not prove to have linear deviations with increasing temperature so a program INTERP (see Appendix C) was written such that a scaled deviation could be calculated for any of the calibrated thermocouples. After calibration the thermocouples and their associated extension wires remained in their assigned model 5100 data logger channel locations during the data collection process. This ensured that all deviations arising from material inconsistencies among thermocouples would be eliminated.

Model 5100 Logger

The model 5100 data logger required a calibration procedure outlined in the operation manual. To perform the calibration a DC voltage standard model MV116, made by Electronic Development Corporation was required. It has an accuracy of ± 1 microvolt. To begin the calibration it was required to perform the setup procedure as described in the manual. With the logger held on channel number one a 2.00000 volt ± 10 microvolts standard voltage was applied to the channel. On the accessory card the R32 unit was adjusted until the mainframe display indicated exactly 2.0000 volts. Once this calibration had been performed the thermocouples connected to the logger were calibrated as described previously.

Turbine Meters

The flow rate through the two turbine meters was calibrated against the frequency of impeller rotation. The calibration required a stopwatch, a frequency meter, and at least two people to perform correctly. A twenty-five inch ruler graduated to 1/16 inches was taped to the inside of the reservoir. The pump was switched on and allowed to run for a few seconds until normal operation occurred and to allow the test fluid to reach uniform flow. Noting the initial height and frequency the reservoir was allowed to drop in height at the instant the stopwatch was activated. The fluid passed through the entire system and exited to a catch container. At certain time increments during the fluid collection the frequency indicated by the meter was recorded. When roughly three to five gallons of fluid drained from the reservoir, or a reasonable period had elapsed for the considered flow rate, the stopwatch and fluid flow were simultaneously stopped. The final height of the fluid in the reservoir was recorded, and an average frequency over the time interval calculated. Using the total time interval, the tank diameter, and the change in reservoir height, the volume flow rate was approximated as a cylindrical volume over a time interval. The procedure was repeated at representative values over the available frequency range of each turbine meter (approximately 15 points). This provided data of flow rate versus average frequency. After completing three 15-point calibration runs of each turbine meter the respective data was correlated using a linear least squares curve fit. The minimum correlation coefficient attained was 99.90%, to a maximum of 99.99%. A program called GPMCALC (see Appendix C) was written to enable quick identification of flow rate given frequency, or given flow rate the program will return frequency, and this is available for both turbine meters over all the fluid concentrations studied. Figures 2.7 and 2.8 illustrate the data collected for both meters and the curve fits (including the correlation coefficients) for all fluid concentrations used. A complete uncertainty analysis of the data is given in Appendix A.

<u>Calibration Runs</u>

To compare the accuracy of the test section to the established correlations for turbulent flow the test fluid (distilled water), was put in steady-state fully turbulent flow with Reynolds numbers above 10000. Heat addition was accomplished in the form of constant heat flux with the DC welder. When the first and last two active test section thermocouple stations, all four heat exchanger thermocouples, and the two bulk fluid temperatures no longer indicated temperature deviation greater than 1/2 degree F, the steady state condition was assumed to be present. Data collection and reduction were performed (see the experimental procedure section of this chapter) and the resulting Nusselt numbers were compared with those predicted

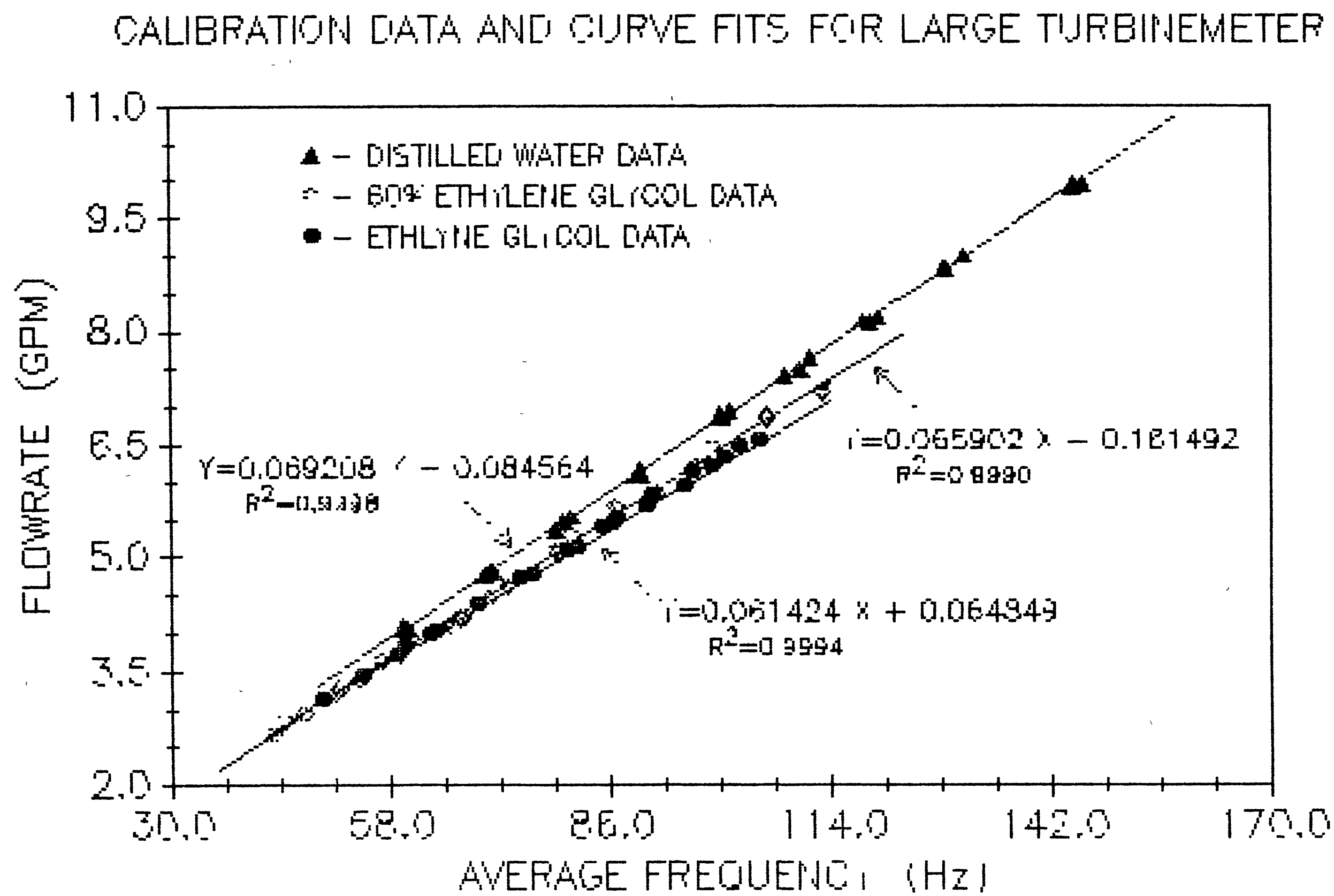


Figure 2.7 Large Turbinemeter Flowrate Calibration Curve

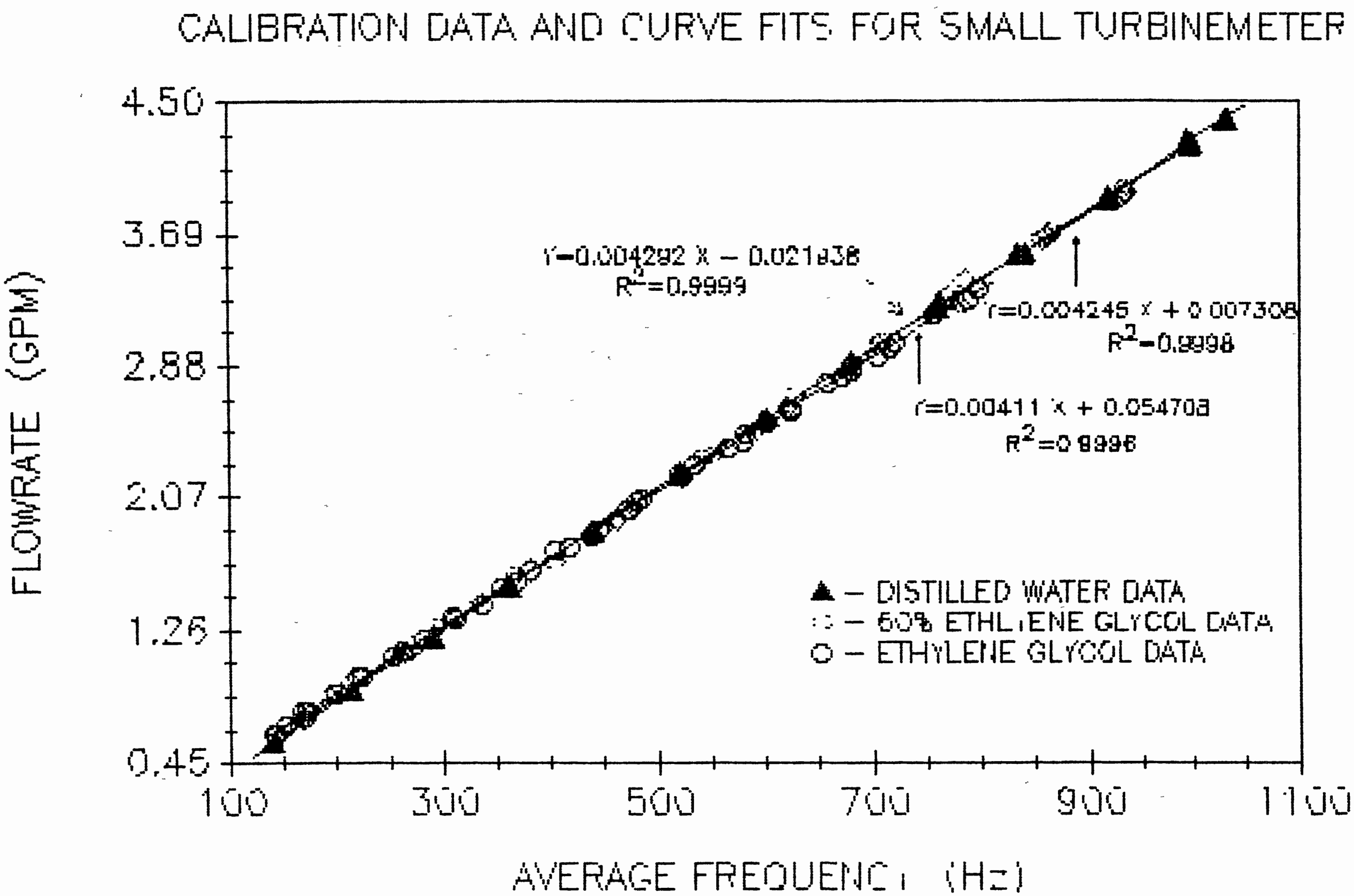


Figure 2.8 Small Turbinemeter Flowrate Calibration Curve

by the accepted correlations outlined in the literature survey section of Chapter I. Recalling from Table III the fully turbulent forced convection constant heat flux correlations of Sieder & Tate (1936), Dittus-Boelter (1930), and Gneilinski (1976) while observing all dimensionless parameter restrictions, acceptable errors were observed. An exhaustive comparison for all equations from Chapter I is presented in Chapter III; however, for the For Sieder & Tate equation data errors from 3.1 to 9.1% were observed similar errors for Dittus-Boelter correlation of 2.9 to 9.2% occurred, and for the Gneilinski equation higher errors of 4.3 to 21.7% were evident. This reflected the desired accuracy and experimentations continued into the transition region. Figure 2.9 shows the comparison of collected data to the Sieder-Tate correlation for fully developed turbulent flow.

Experimental Procedures

The system warm up, data collection, and shut off procedures were conceived with consideration for accuracy, repeatability, safety, and ease of performance. Although several concentrations and types of fluids were used the procedures were consistent throughout data collection.

Warm Up

Before each data collection experiment occurred a quick check of all apparatus and equipment was performed to ensure

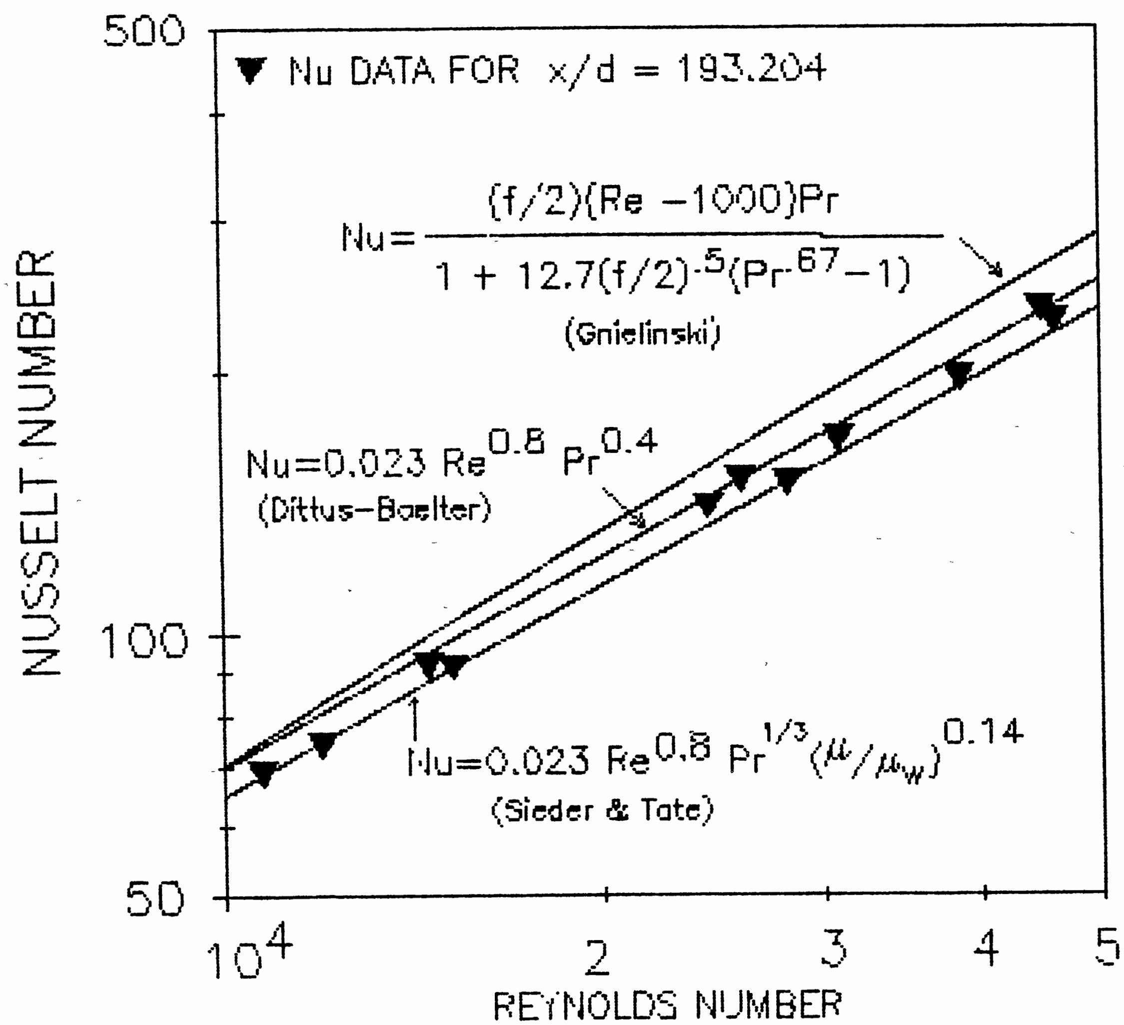


Figure 2.9 Fully Turbulent Calibration Run Comparisons

no leaks nor failed components were present in the system. When a decision about desired flow rate and heat input had been made the warm up procedure was instituted. The steps are as follows:

1. Set the inlet and exit control valves such that flow is provided to the heat transfer test section.
2. Set the turbine meter assembly control valves in the correct position to allow flow through the correct meter based on the desired flow rate.
3. Connect the shielded coaxial cable from the correct turbine meter to either the amplifier or the frequency meter depending upon the frequency of operation. Turn on the amplifier and/or the frequency meter.
4. Set the control valves of the pump box assembly in correct position such that the correct pump will provide flow to the system.
5. Open the flow control valve to the full open position and provided all valves are in the correct position, switch on the pump.
6. Using program ETH (see Appendix C) determine the bulk inlet temperature and GPM necessary for the desired Reynolds number.
7. Consult program GPMCALC to determine the necessary frequency to attain the desired flow rate.
8. Adjust the flow rate with the flow control valve until the frequency indicated by the frequency

meter is near the desired value.

9. Check the welder cable and all connections to ensure their integrity and proper fitting. Check the fuse box to see that the breaker is in the on position. Turn on the voltmeter, the model 5100 data logger, and the MAC-14 data logger.
10. Switch the welder on and adjust the current output to the near desired value on the welder ammeter. Check the DC ammeter in the test section circuit and re-adjust the welder current until this meter is reading the correct value.
11. After approximately 10 minutes of operation switch on the reservoir mixer and the heat exchanger coolant. The coolant flow through the heat exchanger is set such that the inlet bulk temperature of steady state operation is in the desired vicinity of operation.

Data Collection and Shut Down

When the test section reaches the steady state condition the initiation of data collection begins. By monitoring the first two and last two test section thermocouple stations, as well as the inlet and exit bulk temperature thermocouples, a decision as to when this condition is reached can be made. The procedure follows:

1. On the AT computer bring up the ACQCOMP software monitoring the MAC-14 data logger output. Set

the model 5100 data logger to print data on strip chart every 2 minutes.

2. Re-adjust the heat exchanger coolant flow such that the maximum inlet bulk temperature does not exceed 90 F if necessary.
3. When the six thermocouple readings on the model 5100 data logger in addition to test section thermocouple stations 1, 3, 29, and 31 all indicate less than 1/2 F deviations, assume the steady state condition to be present.
4. Record the frequency of the flow meter, the voltage at the meter, and amperage at the meter.
5. Prepare the model 5100 data logger for a maximum channel recording of only 6, turn off the strip printer. Set a 10 second dwell and 2 minute log interval.
6. Set the MAC-14 logging parameters through ACQCOMP such that disk storage of data occurs every minute for all 96 channels. Monitor all equipment during operation and discontinue data collection on the MAC-14 at 100 samples stored. This is the maximum samples input to data reduction software (explained in Data Reduction section of this chapter).
7. When the data collection period is complete, repeat step 4 for all final values. Disable all data recording devices.

8. Turn off the DC welder, voltmeter, amplifier, and/or frequency meter. When the inlet and exit bulk temperatures approach room temperature shut off the coolant water to the heat exchanger and the reservoir mixer.
9. Turn off the pump, close the flow control valve and the inlet and exit test section valves. Switch off the MAC-14, model 5100 data logger, and the AT computer.
10. Inspect the test section apparatus and insure that no leaks have become evident.

Problems

It was discovered that failure to close all test section control valves and the flow control valve could result in siphoning of the test fluid to the reservoir, exceeding its capacity and causing leakage. This was easily solved by checking the valve positions after terminating data collection. Another problem was observed during higher heating experiments, the test section would expand such that the tube entrance would no longer remain flush with the inlet section. This had twofold consequences, first the silicone seal at the inlet would occasionally leak during these incidents and the flow would not enter through the desired square entrance. The remedy for this problem was already present. Due to the inherent flexibility of the inlet/calming section design the bolts fastening it to the system were loosened and horizontal adjustment could be made in either direction to correct the inlet alignment. Incorrect control valve positioning during start-up caused pressure fractures in the calming section bonds. This was corrected by developing strict warm up procedures and adhering to them. Many weeks of troubleshooting and diagnosis led to solution of the severest problem encountered. Originally Omegatherm (a non-curing variant of Omegabond) was used to attach the thermocouples to the test section, leading to the thermocouple beads being in direct contact with the electrified tube wall and transmitting a contaminated signal to data logging equipment. Removal of insulation, all thermocouples, the Omegatherm, and re-application with the use of Omegabond, prevented direct contact and solved the problem.

Data Reduction Procedures

A computer program called HEAT was the major data reduction tool. A listing of the computer program HEAT is given in Appendix B. The program inputs include the type of fluid used, the voltage drop across the tube, the current carried by the tube, the volume flow rate, the bulk fluid temperatures at the inlet and exit, and the outside wall temperature data for all stations (a sample input file is included in Table X of Appendix B). The program then uses a finite-difference technique to calculate the inside wall temperatures taking into account heat conduction in both the longitudinal and peripheral directions. Fluid bulk temperatures from inlet to exit were assumed to vary linearly along the axial direction and were used to calculate the local dimensionless group values. The program automatically generates two output files, the first has the extension OUT and gives a complete listing of all output calculations. The second has extension SUM and gives summary output in a headingless table for easy input to graphics software or curve fit programs. Examples of both the OUT and SUM files can be seen in Tables XI and XII of Appendix B. The following steps show the data input and reduction procedure:

1. Obtain the data concerning the bulk temperatures stored in the cache memory from the model 5100 data logger using PC-TALK. Calculate the uncorrected average bulk temperatures using REDUCE30 (see Appendix C).
2. Use INTERP to calculate the corrected bulk temperatures from step 1 and obtain the flow rate from GPMCALC by inputing average frequency.
3. Calculate the corrected average outside wall temperatures by inputing the file collected from ACQCOMP and the correction file CORR.DAT into program RED96 (see Appendix C). Also input to the RED96 program for relay to HEAT are the corrected inlet and exit bulk temperatures, voltage drop across the test section, current carried through

the tube, and the volume flow rate. The RED96 program generates a formatted input file for use in HEAT.

4. Use the heat transfer default values in HEAT and input the file generated by step 3 to calculate inside local temperatures and heat transfer coefficients, mass flow rate of test fluid, local heat fluxes, heat balance error, local and average Reynolds, Prandtl, Nusselt, and Grashof numbers, local ratios of absolute viscosity (bulk to wall), ratios of top to bottom heat transfer coefficient, and finally local bulk fluid temperatures. Other miscellaneous calculations are given by HEAT, but those listed above are of primary interest.
5. Use the output data files with SUM extensions in various programs to generate further reduced data which is input to the SIGMAPLOT graphics software to produce the figures presented in the this thesis.

CHAPTER III

RESULTS AND DISCUSSION

From eighty-two complete sets of experimental data a transition region for forced convection heat transfer of a horizontal circular tube utilizing a square entrance inlet configuration will be clearly defined. The effects of both forced and mixed convection heat transfer will be discussed as a preliminary to the transition region definition which will be followed by the influence of the tube entrance. Finally some comparisons of heat transfer data with existing correlations will be presented and one existing equation will be modified to predict the Nusselt number data over all flow regimes.

Forced and Mixed Convection

To develop the compliment of experimental data two test fluids were used in three concentrations. Full strength concentrations of distilled water and ethylene glycol were followed by a mixture of 60% ethylene glycol (mass fraction) that gave local bulk Reynolds number ranging from 281 to 50529. Similarly the local bulk Prandtl number ranged from 3.44 to 157.80, the local bulk Grashof number ranged from 1031 to 2252505, and the local bulk Nusselt number ranged from 12.9 to 232.0. The experimental numbers were achieved using a constant wall average heat flux ranging from 1437 to 19904 Btu/hr·ft^2 (4.53 to 62.7 kW/m^2). Heat balance errors were calculated for all experimental runs by taking a percent difference between two methods of calculating the heat addition. The product of the voltage drop across the test section and the current carried by the tube is the primary method while the fluid enthalpy rise from inlet to exit is the second method. The primary method is the one used in the HEAT program for all heat flux and heat transfer coefficient calculations. The heat balance errors ranged from -1.56% to +8.99% (only nine data runs exceeded +3.5%) giving an average absolute heat balance error of 1.88%. A complete listing of all experimental runs is given in Table IX of Appendix B.

The data given in Appendix B is for all fluids used regardless of flow regime or mode of convection heat transfer. Concentrating on only the 100% distilled water runs, the Reynolds number ranged from 3639 to 50529, the Prandtl number ranged from 3.44 to 6.24, the Grashof number ranged from 63011 to 2252505, and the Nusselt number ranged from 31.3 to 232.0. The next test fluid was 100% ethylene glycol and for this fluid the Reynolds number ranged from 281 to 3328, the Prandtl number ranged from 95.89 to 157.80, the Grashof number ranged from 1031 to 15911, and the Nusselt number ranged from 15.0 to 80.2. When mixing the two test fluids to achieve a 60% mass fraction of ethylene glycol the Reynolds number ranged from 1180 to 12456, the

Prandtl number ranged from 16.81 to 55.29, the Grashof number ranged from 2604 to 197013, and the Nusselt number ranged from 12.9 to 146.3. Again note that this data is for both forced and mixed convection heat transfer and can be found in the contents of Table IX (Appendix B).

Using pure ethylene glycol fluid the lower laminar flow could be reached, however the laminar forced convective flow was influenced by strong free convection (modes of convection heat transfer will be discussed in more detail later in this chapter). The fully turbulent flow was established using distilled water and was dominated by forced convection. For the Reynolds number ranges between the laminar and fully turbulent flows both mixed and forced convection modes of heat transfer were encountered. By using the minimum heat addition available from the welder, forced convection would dominate only for flows slightly above the laminar region (Re > 2700). With increased flow rates the amount of heat addition could also be increased without the onset of mixed convection.

The heat transfer coefficient (h) is defined as:

$$h = q_{wm} / (T_{wi} - T_b) \tag{3.1}$$

The average wall heat flux (q_{wm}) is obtained from the electric power input to the tube from the welder (see previous page). The wall temperature was measured outside the tube and using a finite difference technique the inside wall temperature (T_{wi}) was calculated. The bulk test fluid temperature was measured at the tube inlet and exit, and was assumed to vary in a linear fashion between such that bulk fluid temperature (T_b) calculations at all locations could be made. Kays and Crawford 1980 state, for a constant heat flux problem, the mixed mean temperature of the fluid varies linearly with dimensionless axial distance. All the bulk fluid temperature and heat transfer coefficient calculations are performed by the HEAT program. By isolating all experimental runs where the mean local heat transfer coefficient ratio (top to bottom) remained above 0.90 a table of forced convection dominated experimental data was compiled and is presented as Table V. In Table V the heading subscripts 1 and 31 indicate data for stations 1 and 31 with x/d = 3.205 and x/d = 365.38 respectively. Of the 82 experimental runs 47 were determined to be dominated by forced convection. For these runs the Reynolds number ranged from 2285 to 50529, the Prandtl number ranged from 3.44 to 125.65, the Grashof number ranged from 2560 to 2252505, and the Nusselt number ranged from 31.3 to 232.0. In a similar fashion Table VI for mixed convection dominated runs was compiled where the heat transfer coefficient ratio was less than 0.90 and 35 of the experimental runs fit this category. For the mixed convection runs the Reynolds number ranged from 281 to 4070, the Prandtl number ranged from 16.92 to 157.80, the Grashof number ranged from 1031 to 205639, and the Nusselt number ranged from 12.9 to 65.1.

TABLE V

SUMMARY OF FORCED CONVECTION RESULTS

RUN	%WATER	Re_1	Re_{31}	Pr_1	Pr_{31}	Nu_1	Nu_{31}	Gr_1	Gr_{31}
1005	100	41239	50529	5.61	4.47	226.5	217.5	112841	2252505
1006	100	41338	48203	5.53	4.66	220.3	232.0	85994	132591
1004	100	33768	43204	5.57	4.23	190.6	191.6	137196	292799
1007	100	30907	35290	5.72	4.93	151.5	145.5	76857	122908
1014	100	28667	33036	5.99	5.11	160.2	163.3	63011	98855
1008	100	23669	28002	5.63	4.67	142.9	144.5	81194	136278
1015	100	22116	26472	5.98	4.89	134.8	138.8	75415	131855
1016	100	11076	18057	5.92	3.44	85.4	97.7	181837	697699
1009	100	10243	13656	5.76	4.18	71.4	75.4	119154	276169
2654	40	10202	12456	32.61	26.68	146.3	151.2	19231	28906
1010	100	8982	12424	5.70	3.97	66.6	71.1	131712	334573
1012	100	7346	10196	5.86	4.06	59.1	59.9	113844	311718
2653	40	6829	7069	40.19	38.82	66.6	64.8	3588	3961
2663	40	6483	10610	34.28	20.93	117.4	130.2	36698	93587
1013	100	6263	9163	5.81	3.80	51.5	53.1	133958	417667
2659	40	5918	7358	40.15	32.27	101.5	102.4	12862	20151
2601	40	5858	7918	31.62	23.38	90.0	98.1	28230	48895
2664	40	5688	9899	33.92	19.48	104.1	118.1	42426	120422
2602	40	5570	7685	31.55	22.85	84.9	94.3	30548	54243
2660	40	5533	6984	40.30	31.90	93.7	95.9	13833	22071
2610	40	5370	7347	30.42	22.22	85.4	91.8	30625	55256
2687	40	5031	7351	42.07	28.76	99.0	90.8	18837	45749
2665	40	4818	9232	33.99	17.73	90.5	106.4	48819	163979

TABLE V CONTINUED

RUN	%WATER	Re_i	Re_{3i}	Pr_i	Pr_{3i}	Nu_i	Nu_{3i}	Gr_i	Gr_{3i}
1017	100	4802	7482	6.17	3.77	41.1	42.0	130775	508312
2661	40	4762	6237	40.67	31.02	83.3	85.6	15289	26256
2655	40	4352	5827	41.28	30.80	78.3	80.8	15785	28273
2652	40	4277	4532	39.67	37.44	51.3	54.5	4975	5290
2656	40	4237	5720	41.1	30.42	76.4	78.7	16332	29804
1018	100	4170	6724	6.22	3.65	36.6	38.1	136155	575094
2662	40	4106	5620	40.75	29.74	74.0	76.4	17172	32200
2686	40	4053	5843	44.24	30.66	81.8	73.1	16181	39230
2657	40	3993	5495	40.91	29.70	73.1	75.2	17252	32817
2658	40	3794	5283	40.77	29.25	70.4	71.3	18050	35772
1019	100	3639	5622	6.24	3.85	32.7	31.3	122102	499159
2614	40	3507	5690	28.62	17.64	63.7	67.1	47479	125702
2685	40	3396	4376	47.68	36.97	68.4	53.6	9875	21556
2683	40	3083	4091	45.37	34.15	60.7	48.6	11808	26891
2691	40	3012	3604	53.88	44.99	58.5	42.7	5797	11599
2609	40	2927	5150	29.58	16.81	54.9	57.2	51551	164026
2613	40	2872	3713	34.89	26.97	46.8	47.3	19849	33801
2692	40	2683	3277	53.68	43.90	53.8	39.0	6355	13406
2684	40	2679	3702	44.94	32.48	53.4	43.5	13736	33424
2688	40	2635	3658	47.81	34.39	58.0	46.2	11857	29818
2608	40	2571	4518	30.65	17.43	49.5	50.4	48528	157419
2690	40	2509	3113	53.56	43.13	51.3	36.9	6698	14719
2011	0	2364	3328	120.91	96.55	80.2	66.5	3120	6097
2010	0	2285	2878	125.54	100.56	77.3	56.1	2560	5693

TABLE VI

SUMMARY OF MIXED CONVECTION RESULTS

RUN	%WATER	Re_i	Re_{3i}	Pr_i	Pr_{3i}	Nu_i	Nu_{3i}	Gr_i	Gr_{3i}
2651	40	2985	3253	38.49	35.31	46.2	33.4	5675	9419
2680	40	2605	2859	48.78	44.43	43.7	26.8	4146	8234
2682	40	2398	3050	47.21	37.09	47.8	18.7	9373	40683
2677	40	2397	3042	46.99	36.99	46.2	17.7	9804	43083
2679	40	2386	2634	48.8	44.19	39.0	12.9	4361	16507
2681	40	2385	2784	48.24	41.29	41.7	15.7	6549	24458
2678	40	2379	2737	48.27	41.93	42.8	14.7	5931	23589
2689	40	2368	2672	55.29	48.98	42.7	14.4	4020	15623
2676	40	2360	3383	45.28	31.55	47.4	21.1	15278	75058
2675	40	2335	3045	45.8	35.09	46.7	17.6	10728	51119
2612	40	2280	4070	30.22	16.92	43.6	43.8	50018	170625
2674	40	2250	2620	47.25	40.55	41.5	14.2	6395	26292
2673	40	2174	2430	47.84	42.78	39.4	13.2	4726	18153
2603	40	2133	3362	35.14	22.28	40.3	21.0	30011	153757
2671	40	2056	2389	46.43	39.95	39.4	13.8	6282	24900
2009	0	2021	2544	128.99	103.34	65.1	27.5	2518	9806
2607	40	2005	3489	32.56	18.70	39.7	31.5	40866	167282
2670	40	1919	2351	45.52	37.14	39.6	15.0	8180	33740
2672	40	1880	2139	47.57	41.78	35.7	13.3	5289	18919
2611	40	1859	3254	32.35	18.47	38.7	28.6	39830	177907
2008	0	1827	2349	129.81	101.91	65.1	21.3	2484	13242
2669	40	1716	2065	46.11	38.29	36.3	13.9	7080	27737
2604	40	1684	2972	33.89	19.19	37.6	22.5	34760	197013

TABLE VI CONTINUED

RUN	%WATER	Re_1	Re_{31}	Pr_1	Pr_{31}	Nu_1	Nu_{31}	Gr_1	Gr_{31}
2666	40	1639	2109	43.31	33.63	36.7	15.6	10295	42007
2668	40	1576	1855	45.32	38.48	34.3	13.4	6394	23405
2007	0	1483	2035	130.02	95.89	62.1	21.6	2782	15911
2667	40	1398	1676	45.03	37.54	31.0	12.9	6925	24647
2606	40	1377	2575	34.11	18.23	35.2	22.4	34245	205639
2006	0	1331	1831	133.04	97.92	58.2	20.9	2642	14681
2605	40	1180	2156	34.68	18.97	32.1	20.9	29744	165470
2004	0	1150	1465	150.10	118.86	54.4	18.8	1564	7680
2002	0	949	1148	153.77	127.88	48.2	16.1	1069	4839
2005	0	730	930	155.66	123.38	41.6	16.1	1210	5226
2003	0	559	721	157.80	123.48	38.3	15.3	1031	4422
2001	0	281	421	149.90	101.45	27.1	15.0	1287	5377

To illustrate the different trends in heat transfer coefficient ratio for flows dominated by forced convection and mixed convection heat transfer Figure 3.1 is presented. The figure includes representative Reynolds number ranges from laminar to fully turbulent flow (local bulk Reynolds number range of 281 to 14567). As the figure demonstrates, the higher Reynolds number flows are dominated by forced convection heat transfer and the heat transfer ratio does not fall below 0.90 and exceeds 1.0 due to round-off errors in the property evaluation subroutine of the HEAT program. The flows dominated by mixed convection heat transfer have heat transfer coefficient ratios beginning near 1.0 but dropping off rapidly as the length to diameter ratio increases. Beyond 125 diameters of flow the ratio tends to stabilize to an approximate value of 0.30 indicating a much less dominant heat transfer role for forced convection and increased natural convection activity.

Since convection heat transfer depends upon temperature differences, the changes in the heat transfer coefficient can be related to temperature. For a fluid heated by ideal forced convection the peripheral inside wall temperatures are constant resulting in uniform local heat transfer and near constant local heat transfer coefficient with respect to the periphery. As the flow rate is decreased for the heated case the fluid encounters a longer contact residence with the surface causing the fluid nearer the wall to become warmer and less dense than that near the center. Due to

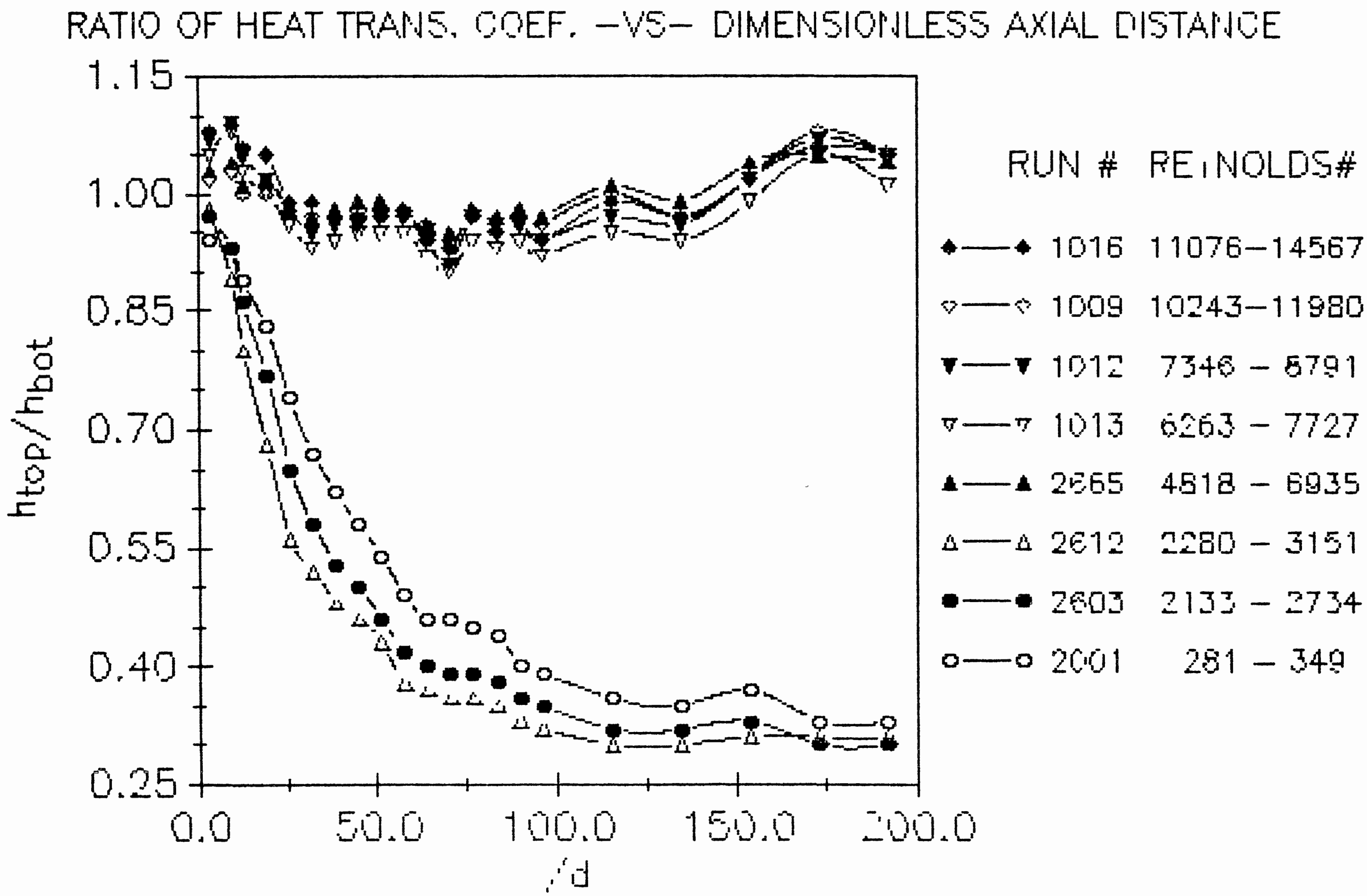


Figure 3.1 Forced and Mixed Convection Heat Transfer Coefficient Ratio Comparisons

buoyant forces the less dense fluid rises along the fluid wall and the denser fluid flows down near the center. This produces a temperature difference along the periphery of the tube wall with the maximum temperature at the tube top and the minimum at the tube bottom. The temperature difference along the periphery drives a secondary flow pattern at right angles to the primary (forced flow) direction consisting of two vertically symmetric vortices. These temperature differences also lead to free convective heat transfer occurring simultaneously with forced convection, resulting in a non-uniform heat transfer coefficient with respect to the peripheral location. Figure 3.2 shows the variation in inside wall temperature with peripheral location for run 2007 (pure ethylene glycol). For this run the local bulk Reynolds number ranged from 1483 at station 1 (x\d=3.2) to 1757 at station 22 (x\d=192.3). The station 1 temperature difference (between top and bottom) is -2.86 degrees F and at station 22 the temperature difference is 33.69 degrees F. From station 11 to station 22 (x\d=57.69 to 192.30) the bottom surface temperature change (160.04 to 170.36 F) was not as extreme as the top surface temperature change (172.23 to 204.05 F) exhibiting the extent to which natural convection removed the lower surface heat. This also results in the top to bottom heat transfer coefficient ratios at these stations changing from 0.77 to 0.50.

In contrast Figure 3.3 shows data from run 1014 (pure distilled water) where the local bulk Reynolds number

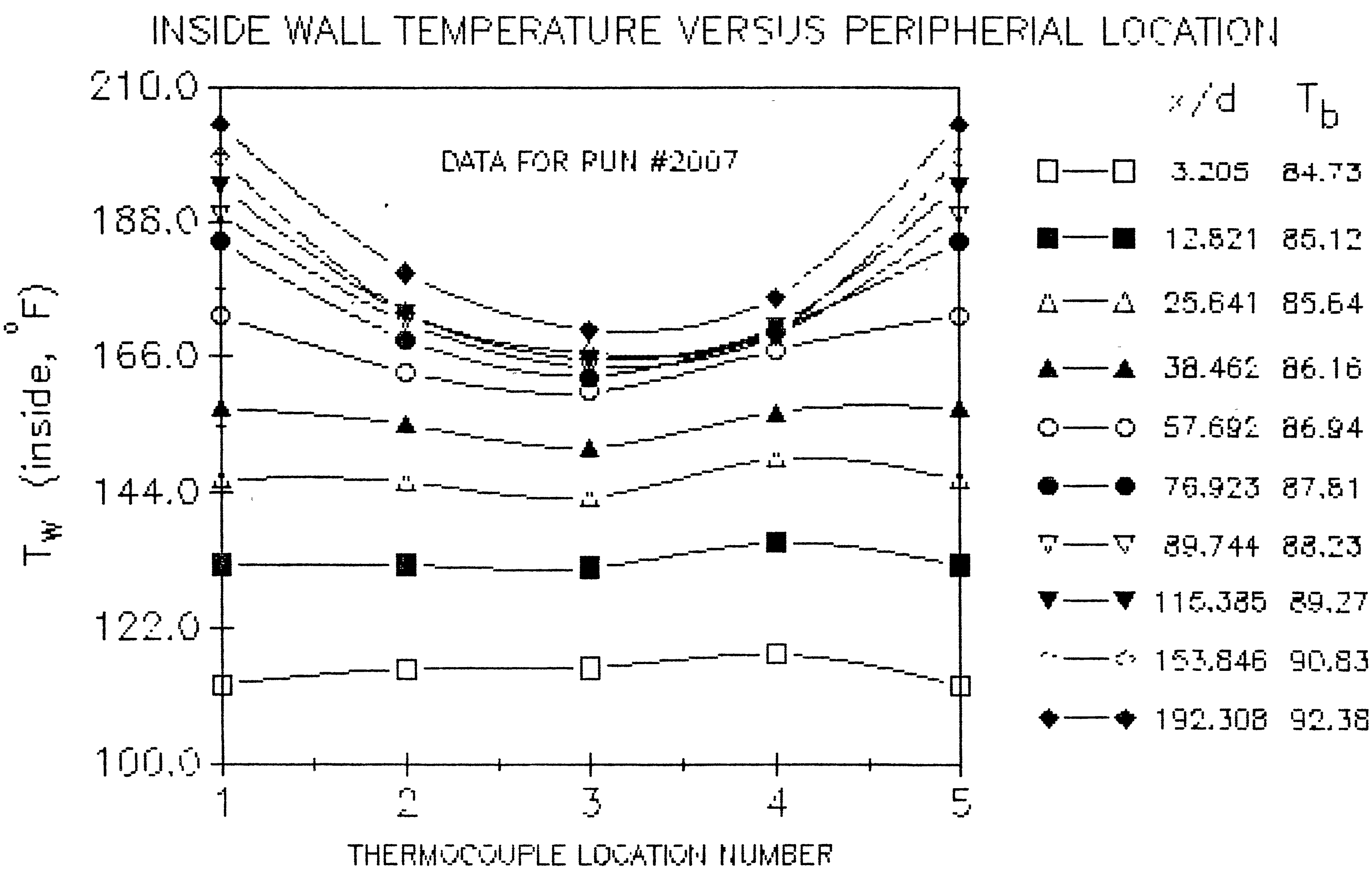


Figure 3.2 Peripheral Inside Wall Temperature Distribution for Mixed Convection

changes from 28667 to 30918 (stations 1 to 22). The top to bottom temperature difference at station 1 is -0.53 and at station 22 is -0.55 showing a nearly uniform radial temperature difference along the tube. Furthermore from stations 11 to 22 the top surface temperature difference is 3.52 while the lower surface difference is only 4.02. This indicates a nonexistent role for natural convection and is further supported by the nearly uniform top to bottom heat transfer coefficient ratios of 0.95 and 1.08 at the aforementioned stations.

Defining The Transition Region

Due to the limitation on the minimum heat flux that can be furnished from the welder to our test section (1437 Btu/hr·ft^2 (4.53 kW/m^2)) the transition region definition relies on data of both forced and mixed convection heat transfer. Of the total experimental runs collected (82) over eighty-five percent (70) were used to establish the transition flow region for forced and mixed convection. Table VII presents a list of the run numbers, the bulk local Reynolds number, the bulk local Prandtl number, the bulk local Nusselt number, the bulk local Grashof number, the average heat flux, the heat transfer coefficient ratio (top to bottom), and the heat transfer mode (forced or mixed), for all 70 runs used to define the transition region. All dimensionless group data in Table VII is for station 22 (x/d=192.308) and is designated with a subscript 22.

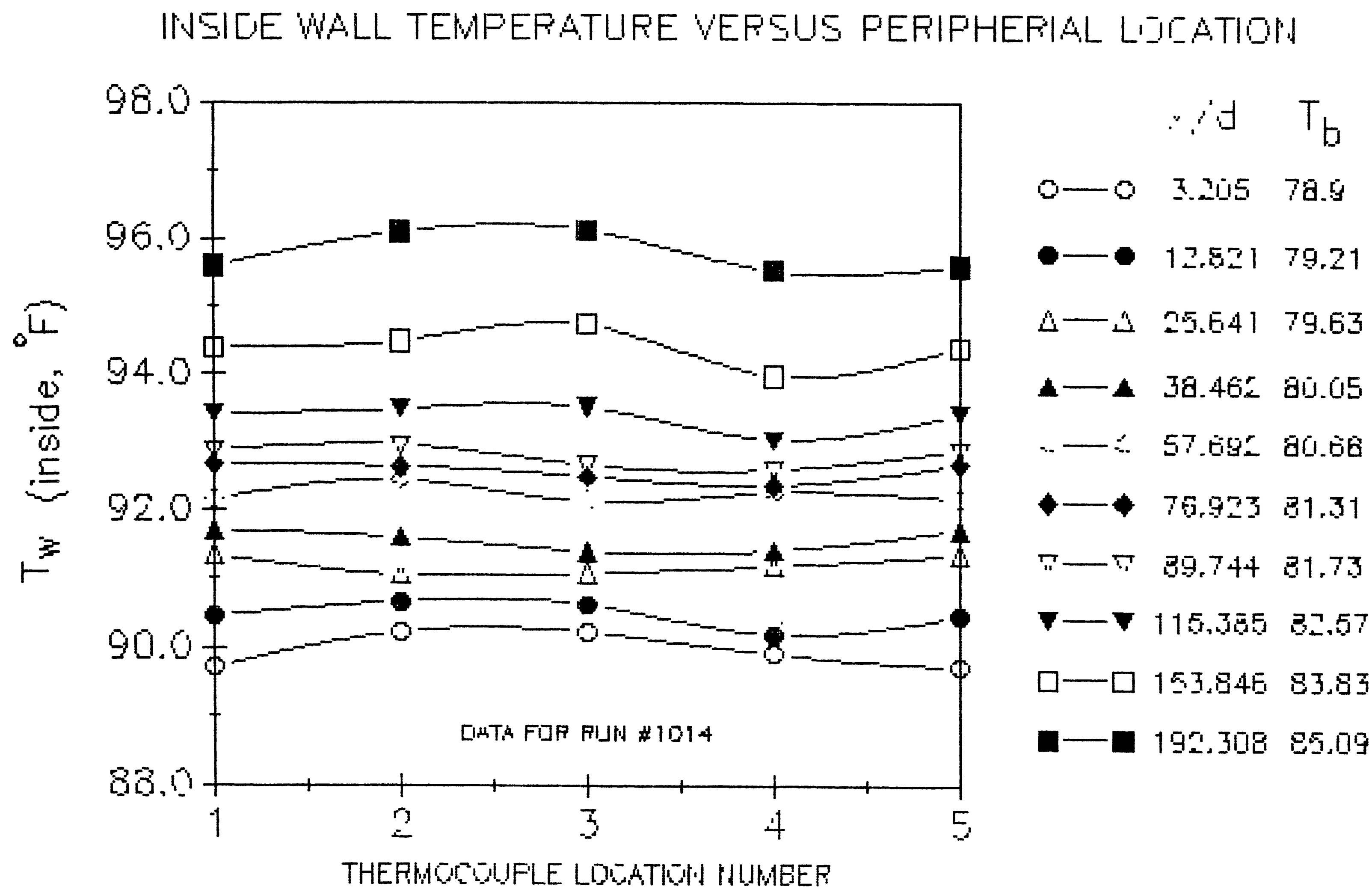


Figure 3.3 Peripheral Inside Wall Temperature Distribution
for Forced Convection

TABLE VII CONTINUED

RUN#	Re_{22}	Pr_{22}	Nu_{22}	Gr_{22}	Q/A	$h_{t/b}$	MODE
2603	2734	27.40	21.6	96339	5464	0.30	MIXED
2682	2725	42.06	18.2	31971	3687	0.47	MIXED
2677	2720	41.38	17.4	34687	3706	0.46	MIXED
2675	2690	39.74	17.6	39181	3901	0.43	MIXED
2607	2725	23.94	23.1	137076	6256	0.27	MIXED
2681	2588	44.44	15.3	21536	2350	0.49	MIXED
2010	2583	111.58	46.6	5517	6049	1.05	FORCED
2678	2562	44.81	14.5	20810	2146	0.52	MIXED
2611	2536	23.70	22.8	132786	5879	0.26	MIXED
2689	2524	51.87	13.9	14538	1963	0.57	MIXED
2674	2438	43.58	14.2	22556	2169	0.47	MIXED
2671	2225	42.89	13.8	21486	1924	0.46	MIXED
2670	2137	40.87	15.1	27485	2431	0.42	MIXED
2008	2088	114.13	21.9	10119	5402	0.57	MIXED
2672	2012	44.42	13.0	16924	4818	0.47	MIXED
2669	1893	41.79	13.8	23200	1965	0.43	MIXED
2668	1718	41.57	13.2	20119	1585	0.44	MIXED
2667	1539	40.89	12.8	20840	1559	0.40	MIXED
2004	1308	132.62	19.3	5925	3755	0.54	MIXED
2002	1049	139.49	16.3	3971	2409	0.55	MIXED
2005	830	137.60	16.2	4129	2389	0.49	MIXED
2003	640	138.54	15.1	3503	1933	0.46	MIXED
2001	349	121.54	14.3	3837	9120	0.33	MIXED

* Units in $Btu/hr \cdot ft^2$

To accurately define the transition region it was important to avoid any entrance effects (described in detail later in this chapter) so only data of a fully established thermal distribution was used. Data with developing thermal profiles provides a non-uniform and useless transition region definition. All three fluid concentrations were required to encompass the onset of transition flow from the laminar region until the decay from transition to the turbulent region. To illustrate the transition region Figure 3.4 plots a dimensionless heat transfer parameter (similar to the Colburn j factor) versus Reynolds number at a length to diameter ratio of 192.308 (station 22). This ratio (L/d) is far enough downstream of the entrance to guarantee a fully developed thermal profile and includes four peripheral thermocouple locations to ensure accuracy. The heat transfer parameter plotted is the product of Stanton number and $\text{Prandtl}^{0.6}$ retaining a non-dimensional parameter valid for any fluid. Due to the large increments in mass fraction of ethylene glycol used as test fluids (0%, 60%, and 100%) large gaps resulted over the range of Prandtl number (refer to the second paragraph of this chapter) leading to discontinuities on the standard Stanton versus Reynolds plot. Including the Prandtl number term in the heat transfer parameter ($StPr^{0.6}$) eliminated the influence of this dimensionless group (Pr) and provided a smooth curve describing the region (again similar to the function of the Colburn j factor described by Shah (1983)). The figure

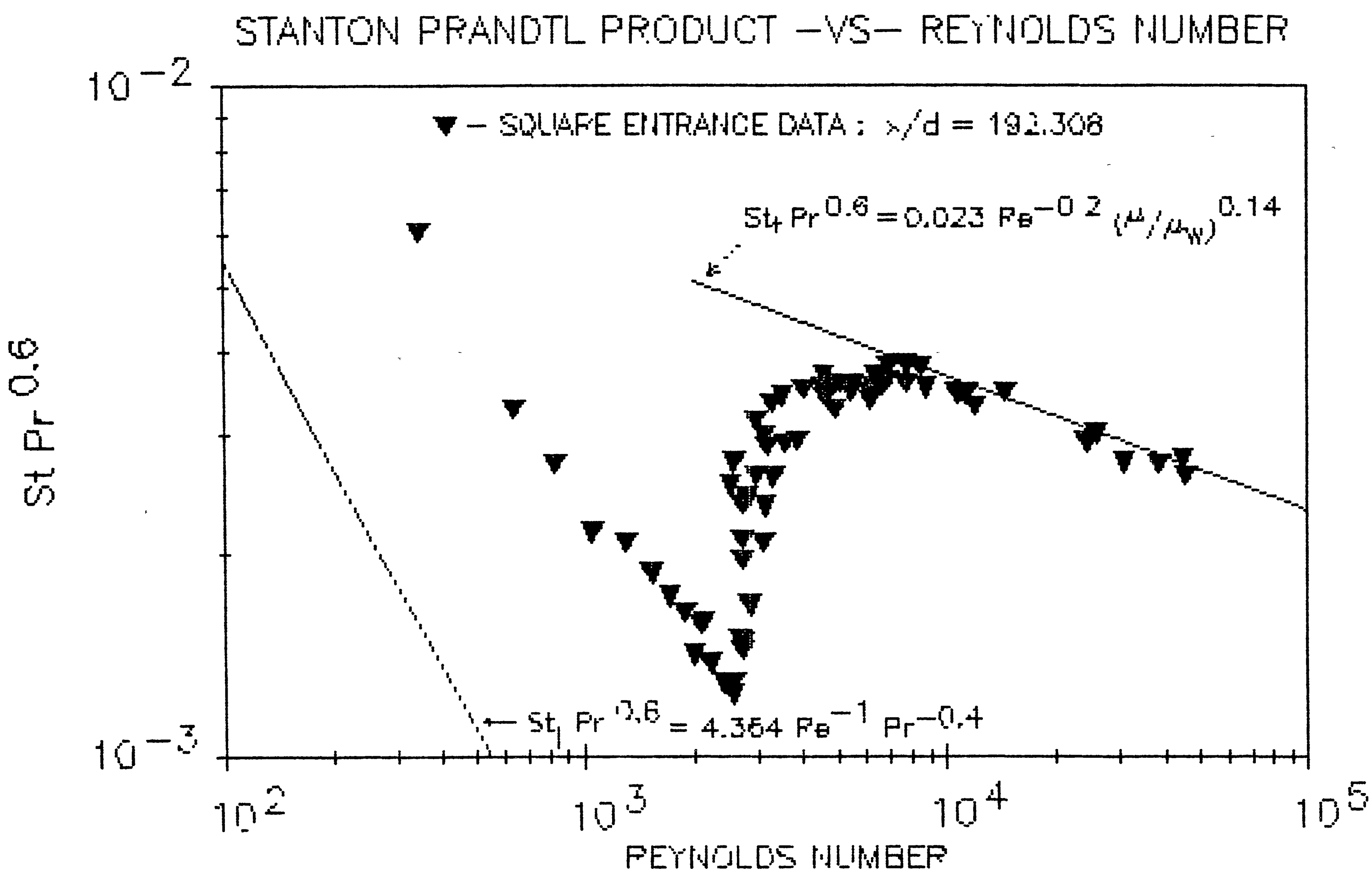


Figure 3.4 Dimensionless Heat Transfer Data

includes local bulk fluid properties of Reynolds number from 349 to 46000, Prandtl number from 4.36 to 139.49, Nusselt number from 12.9 to 239.9, and Grashof number from 3503 to 403636 for average constant heat flux range from 1434 to 19904 $Btu/hr \cdot ft^2$ (4.83 to 62.7 kW/m^2). This results in 44 of the 70 experimental runs used to define the transition region being dominated by forced convection with 26 runs containing mixed convection heat transfer.

While defining the transition region as described in Figure 3.4 we attempted to employ data demonstrating similar trends. It was critical to extract data deviating from the natural pattern displayed by the plot. This required the deletion of less than fifteen percent (12) of the 82 runs from the plot. Table VIII illustrates the runs not appearing on Figure 3.4 and provides the dimensionless data in the same format as Table VII. Three-fourths of these runs were in the laminar flow region and were eliminated due to exceedingly high wall heat fluxes forcing the data away from the laminar trend. Low bulk temperature variation from inlet to exit was determined as the cause of deviation from the established pattern for the remaining data appearing in Table VIII.

All the experimental data points on Figure 3.4 were calculated by the following equation:

$$StPr^{0.6} = Nu\ Re^{-1}\ Pr^{-0.4} \qquad (3.2)$$

resulting in three distinct regions which can be quantified.

TABLE VIII

RESULTS NOT APPEARING IN STANTON PRANDTL PLOT

RUN	Re_{22}	Pr_{22}	Nu_{22}	Gr_{22}	Q/A*	$H_{t/b}$	MODE
1007	31700	5.28	153.8	95489	9395	1.07	FORCED
2653	6953	39.47	66.4	3737	1437	1.02	FORCED
2652	4409	38.48	54.3	5014	1469	1.03	FORCED
2679	2513	46.32	12.4	15482	1498	0.56	MIXED
2604	2308	24.72	22.4	115567	5454	0.26	MIXED
2673	2306	45.10	12.9	16512	1533	0.52	MIXED
2009	2284	114.66	25.2	8637	5373	0.73	MIXED
2606	1953	24.04	22.8	112054	5033	0.23	MIXED
2666	1875	37.85	15.6	32682	2522	0.39	MIXED
2007	1757	110.48	22.1	11523	5764	0.50	MIXED
2605	1650	24.79	21.3	92017	4162	0.22	MIXED
2006	1579	112.93	21.6	10468	5388	0.49	MIXED

* Units in $Btu/hr \cdot ft^2$

These three regions include the laminar, transition, and the turbulent regions.

The Laminar Region

The data of Figure 3.4 includes a local laminar flow Reynolds number range of 349 to an approximate upper bound of 2438, with bulk temperature differences (from inlet to exit) up to 11.6 F. This provides for the local bulk dimensionless parameter ranges of Prandtl number from 40.87 to 139.49, Nusselt number from 12.9 to 21.9, Grashof number from 3503 to 23200, and a dimensionless heat transfer parameter ($StPr^{0.6}$) from a maximum of 0.00601 decreasing to 0.00129. The Reynolds number at the departure from laminar to transition flow is affected significantly by the intensity of entrance disturbance and can vary from 2000 to 50000 depending on experimental conditions as stated by Ogawa & Kawamura (1986).

As a reference line for the laminar region the accepted Stanton correlation for internal fully developed forced convective laminar pipe flow under a constant wall heat flux was modified to fit the heat transfer parameter of Figure 3.4. In the equation 4.364 is the conventional fully developed constant wall heat flux forced convective Nusselt number for internal laminar pipe flow:

$$St_1Pr^{0.6} = 4.364\ Re^{-1}\ Pr^{-0.4} \qquad (3.3)$$

The data has a pronounced and parallel shift above the accepted line. This is directly due to the presence of strong natural convection mixed with forced convection. This in turn results in a higher fully developed laminar constant heat flux Nusselt number than the accepted 4.364 value (the slope estimates a value of 14.5 for the data). For these laminar flow data points no forced convection dominated data could be obtained due to the inability of the welder to supply less than its minimum heat output. At the minimum welder current setting (approximately 150-155 amps) the heat generation at the tube wall was above that required to bring about peripheral temperature variations extensive enough to cause free convection and secondary flow (refer to the previous sections of this chapter). Due to the automatic onset of mixed convection in the laminar region poor agreement with the conventional laminar heat transfer correlations was expected and observed.

The Transition Region

The data of Figure 3.4 suggests that the transition region is bounded by local bulk Reynolds number of 2438 to 8791. Using bulk temperature differences from roughly 5 to 40 F gives dimensionless local bulk transition Prandtl number ranges of 4.60 to 111.58, Nusselt number ranges from 13.9 to 120.0, Grashof number ranges from 5301 to 385358, and heat transfer parameter values from 0.00130 increasing to 0.00384.

For the transition region as described above, it can be seen in Table VII that some data in this territory is affected by mixed convection while other data is dominated by forced convection. This is similar to the behavior encountered in the laminar region where low flow rates in tandem with high heat fluxes that could not be attenuated to the degree necessary to prevent secondary flow occurrences resulted in data affected by free convection. Even so it is evident that the lower bound of the transition data joins the laminar data line in the neighborhood of 2400 local bulk Reynolds number and appears at lower Reynolds number as if it is attempting to progress away from laminar flow as low as 2000 Reynolds number. Wall heat flux could account for the early departure behavior.

As the transition line makes its departure vertically from the laminar line it is clearly shown that the heat transfer parameter value increases disproportionately with small increments in Reynolds number. This is probably due to random intermittent bursts of turbulent heat transfer activity. Ogawa & Kawamura (1986) indicates that it has long been known that in the transition from laminar to turbulent flow in a circular tube, laminar and turbulent flows pass alternatively through the tube. Near the middle area of transition the curve encounters a pronounced re-direction and the behavior tends increasingly towards turbulent heat transfer with larger increments of turbulent flow present. Finally, near the upper bound of transition flow the response approaches that of complete turbulence as the dominant intermittent flow is turbulent and only periodic episodes of laminar behavior occur.

The Turbulent Region

The turbulent region from Figure 3.4 is determined as local bulk Reynolds number above 9000, (the maximum point on the figure being 46001) using bulk fluid temperature differences from 12 to 43.7 F. The data in the turbulent region follows a correlation modified to give values of the heat transfer parameter ($St_t Pr^{0.6}$) from the Sieder and Tate equation predicting Nusselt number for fully turbulent flow and appears on Figure 3.4 as follows:

$$St_t Pr^{0.6} = 0.023\ Re^{-0.2}\ (\mu/\mu_w)^{0.14} \qquad (3.4)$$

the data proves to have good agreement with the correlation. Other important local bulk dimensionless group ranges for the turbulent region include Prandtl number from 4.36 to 29.3, Nusselt number from 69.0 to 239.9, Grashof number from 22724 to 403636, and heat transfer parameter of 0.00349 decreasing to 0.00262. Due to high flow rates, high heat transfer, and rapid test fluid mixing in the turbulent flow region no significant temperature differences occur in the peripheral direction resulting in insignificant if not nonexistent levels of secondary flow and free convection.

Entrance Effects

Using the specialized configuration of the experimental setup for horizontal square entrance geometry provided a uniform velocity and temperature distribution at the tube inlet. After the flow traveled through the tube to a point far from the inlet the velocity and thermal profiles became fully developed. The distance from the inlet to this fully developed point is known as the entrance length. Sparrow, Hallman & Siegel (1958) define the thermal entrance region as the length required for the local heat transfer coefficient to approach to within a few percent of the fully developed value of the coefficient. Others such as Shah (1983) define thermal entry length as it relates to Nusselt number and because the Nusselt number is a function of the heat transfer coefficient:

$$Nu = hd/k \tag{3.5}$$

the criteria for thermal entrance effect can be related to heat transfer coefficient or the Nusselt number deviation equally well.

Figure 3.5 shows Nusselt number variation with dimensionless axial distance for all test fluids used over a local bulk Reynolds number range from laminar to fully turbulent (281 to 14567). The local bulk Nusselt number at station 1 ($x\backslash d$ = 3.205) vary from 27.1 to 85.4 and at station 22 (x/d = 192.308) vary from 14.3 to 91.6. The entrance effect is identified as the change in local Nusselt

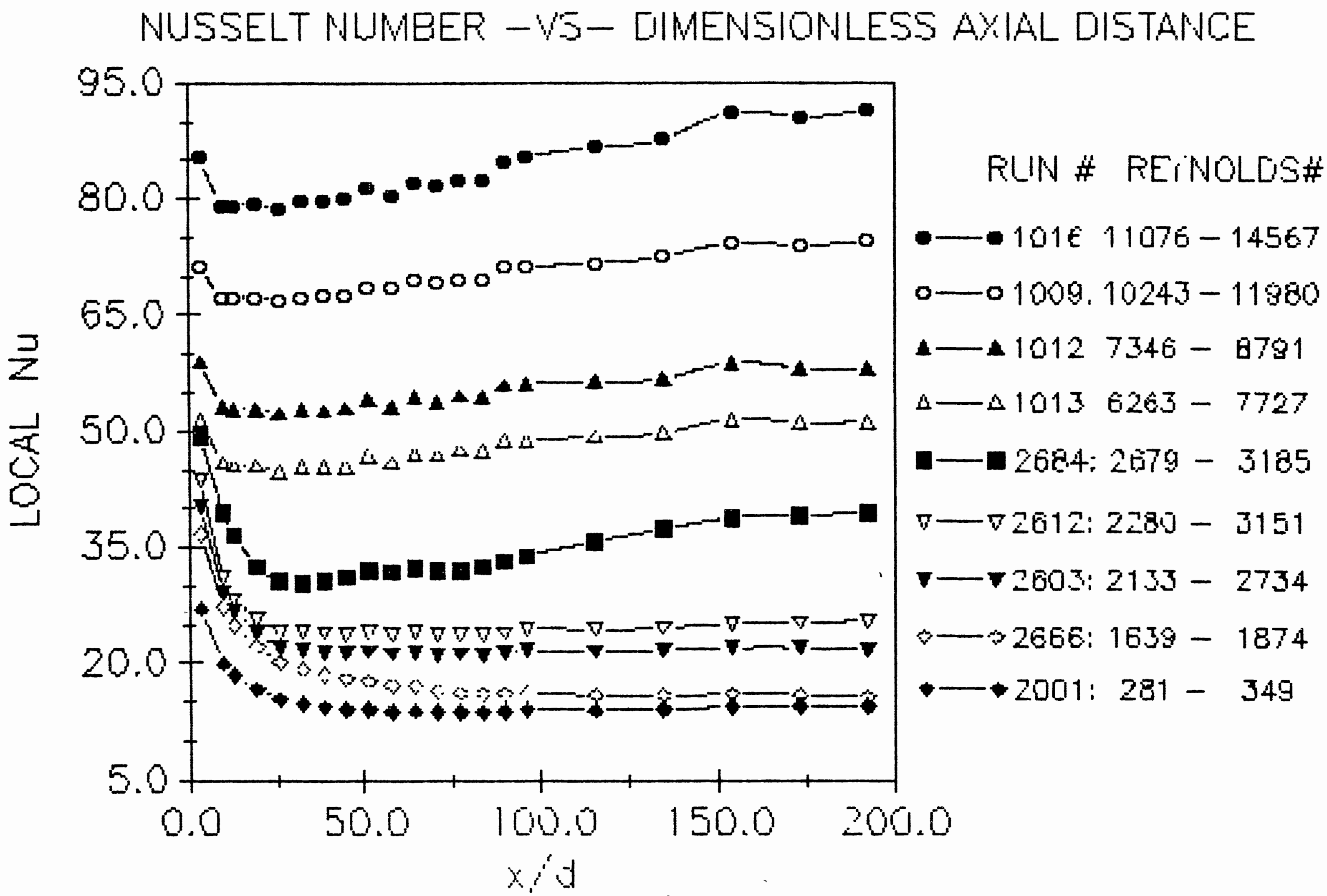


Figure 3.5 Nusselt Number Variation Over all Flow Regimes

number with axial direction. Determining the entrance length and how it affects heat transfer of all flow regimes will be studied in this section. Because Figure 3.5 presents only an overall view of the Nusselt number trends no entrance length calculations are made for this figure.

The appearance of the runs shown in Figure 3.5 shows the laminar runs to have nearly constant Nusselt numbers downstream of the inlet, while the transition and turbulent runs gradually drift upward with tube length. Investigation of the data associated with these runs (and others) reveals the temperature difference to be responsible for the drift. From equation (3.5) we expect a rising Nusselt number to result from an increasing average heat transfer coefficient. Equation (3.1) shows that for a constant average heat flux the heat transfer coefficient will increase with decreasing temperature difference (T_{wi} - T_b). For laminar run 2001 of Figure 3.5 the data reveals this temperature difference decreases only 0.65 F from stations 9 to 22 producing a nearly constant Nusselt number across this length. However, for transition run 1013 a temperature difference decrease of 2.19 F is experienced over the same axial distance resulting in a slightly drifting Nusselt number. In the turbulent region run 1009 shows a temperature difference decline of 2.02 F while run 1016 yields a reduced value of 3.39 F, again over the same length. Comparing these associated temperature differences with the degree of drift evident in Figure 3.5, it can be seen that as larger decreases in temperature difference along the tube length occur, an increasing drift in Nusselt number results.

Further investigation into this trend indicates that free convection appears to play a less significant role. To illustrate this point we examined the minimum heat transfer coefficient ratios experienced by these runs. The runs exhibiting the drifting Nusselt number trend show minimum ratio values of 0.90, 0.94, and 0.92 for runs 1013, 1009, and 1006 respectively. This suggests that forced convection is dominating these runs and free convection is not driving the trend. This is not to say that free convection is not slightly affecting the fluid behavior, for even at these high heat transfer coefficient values, small free convection effects are expected to be present.

Laminar Region Thermal Entrance Length

For the experimental setup used the fluid experiences a combined hydrodynamic and thermal-entry-length scenario. As Kays and Crawford suggest (1980) it has been shown for laminar flows with Prandtl number greater than five the velocity profile develops much faster than the thermal profile such that a fully developed velocity profile can be assumed at the tube inlet. Since the minimum Prandtl number experienced in the laminar region of this study was found to be 129.81 at station 1 (3.205 diameters) any hydrodynamic entry length considerations are ignored.

Figure 3.6 presents eight laminar flow runs with local bulk Reynolds number from 281 to 2088. It is evident from the figure that the accepted fully developed constant heat flux forced convective laminar Nusselt number value of 4.364 is not achieved. Due to the low flow rates and relatively high heat fluxes used in the laminar region mixed convection dominated all the runs in various degrees of strength. Combined forced and free convection results in Nusselt number (and heat transfer coefficients) higher than predicted by pure forced convection correlations. Bergles and Simonds (1971) experienced similar occurrences for high heat fluxes on laminar flows.

To establish the fully developed thermal entry length arbitrarily deviations less than or equal to 5% in local Nusselt number were chosen as the criteria. This criteria remained the standard for determining the entry lengths for all regions of flow. The fully developed laminar thermal entry length ranged from 32.051 test section diameters for run 2001 to 83.332 diameters for run 2008. Because data was collected at specific locations and not in a continuous fashion it did not lend itself to curve fitting analysis. However, the thermal entrance length for the laminar region generally appears to increase with Reynolds number. Evidence in Figure 3.6 shows for all laminar data no thermal entry length effects are present for distances greater than 84 diameters downstream of the inlet.

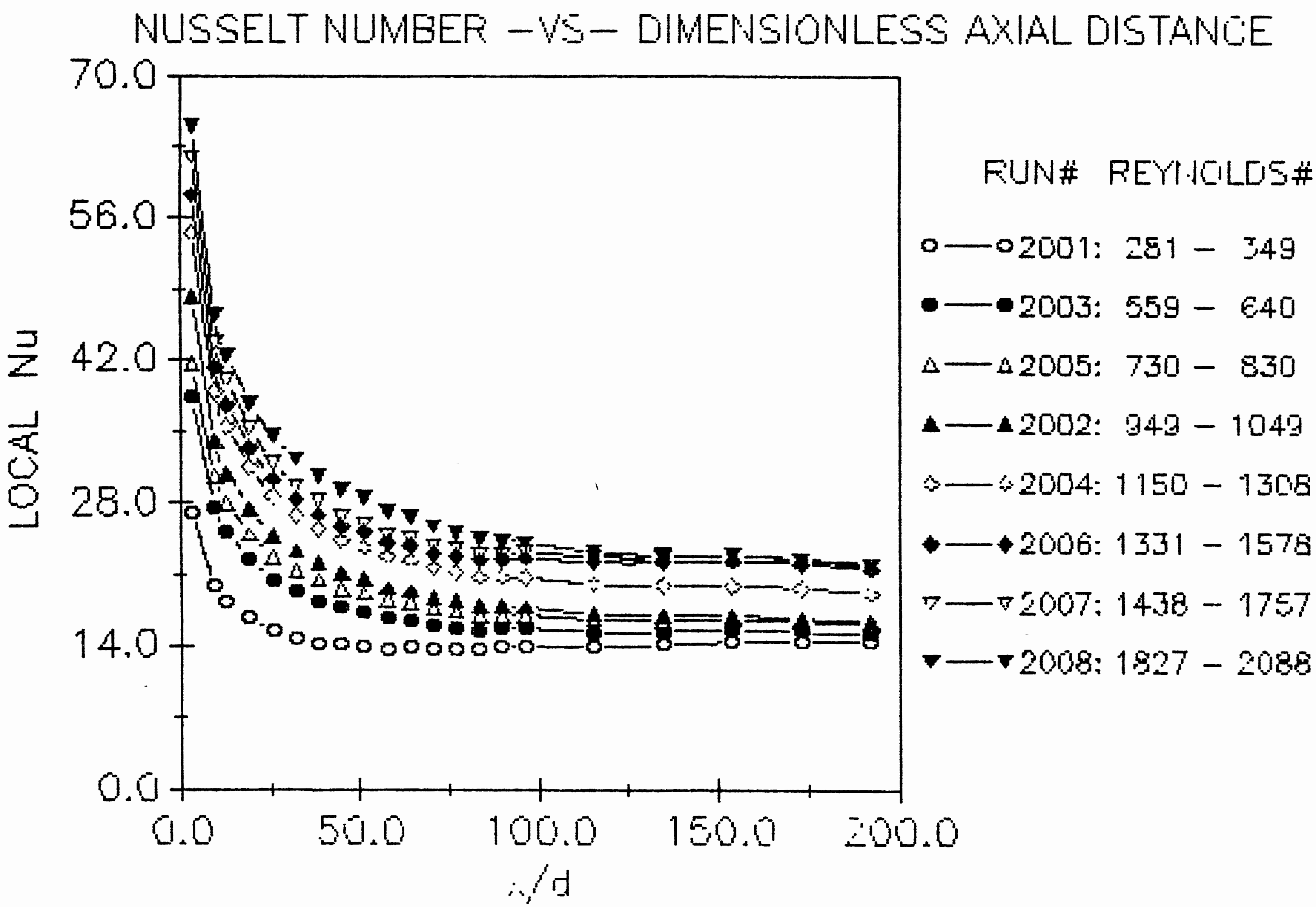


Figure 3.6 Nusselt Number Trends for Laminar Flow

Transition Region Thermal Entry Length

Transition from laminar to turbulent flows occurs when the tube cross section temporarily fills with alternating laminar and turbulent bursts and is affected by many factors such as tube roughness, inlet geometry, and natural convection in the flow as suggested by Ogawa & Kawamura (1986) and White (1974). As described in this chapter the transition flow for this particular inlet geometry occurs at local Reynolds number of approximately 2400 to 9000. Within this region the entrance length behavior can be categorized as having laminar to transition tendencies when laminar flow bursts are predominant and transition to turbulence trends as turbulent bursts prevail.

Figure 3.7 illustrates the transition region with nine runs encompassing local Reynolds number of 2133 at station 1 to 8494 at station 22. To define the transition region thermal entrance length, data of both forced and mixed convective heat transfer is used. For inlet bulk Reynolds number less than roughly 3000, mixed convection effects were present, with forced convection dominated data beyond. The data has Nusselt number ranging from 40.3 to 117.4 for station 1 (x/d = 3.205) and 21.6 to 120.0 for station 22 (x/d = 192.305). It should be noted that although run 2603 appears to be out of the pre-defined transition region it actually contributes to the transition data as from stations 18 (115.385 diameters) and on the Reynolds number lies in the transition region.

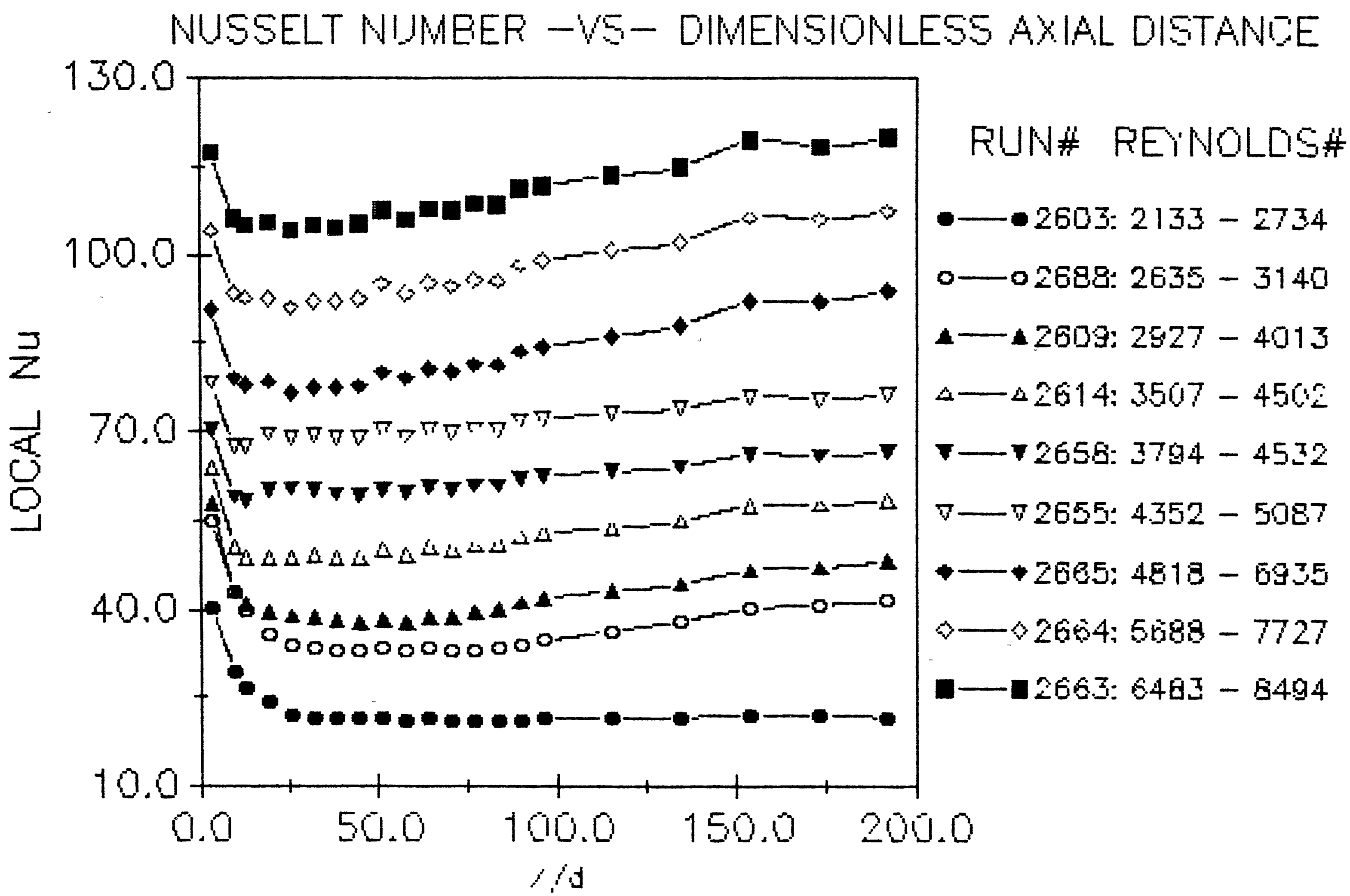


Figure 3.7 Nusselt Number Trends for Transition Flow

Again employing the 5% deviation criteria for judging the end of thermal entrance effects and assuming no hydrodynamic entry effects (minimum transition Prandtl number was 29.58) Figure 3.7 indicates a thermal entrance length ranging from 12.821 tube diameters to 19.231 diameters with all but two of the runs less than 13.0. For these determinations it was necessary to disregard the entrance length data from run 2603 as up to and beyond its fully developed region the flow was still laminar. Accordingly, from Figure 3.7 it can be determined that no thermal entrance effects are evident beyond 20 diameters downstream of the entrance for transition flow. From the data of Chen (1988) it is evident that he experienced similar trends such as nearly constant transition region thermal entry length however at a slightly higher value of 25.3 diameters downstream. He suggests the magnitude of the thermal entry length seems to be inversely proportional to the Reynolds number.

Investigation into the upward drifting Nusselt number trends evident in Figure 3.7, for axial distances greater than 50 diameters from the inlet, reveals only a slight influence of mixed convection. It has been determined that the rising Nusselt number in these runs are attributable to slightly diminishing temperature differences, between the tube wall and the bulk fluid, as the tube length increases. This is further supported by examining one of the data runs. Run 2658 has mid-transition data and shows a decrease in temperature difference between the wall and the fluid of 2.94 F from stations 7 to 22. This allows the Nusselt number to slightly increase for distances greater than the thermal entrance length (as described previously). Similarly, for a minimum heat transfer coefficient ratio of 0.96 (for run 2658) we expect only slight free convection influence affecting the data behavior.

Concentration on the right side of Figure 3.7 indicates the data can be divided into three distinct areas of drift. Examination of the average heat flux in these areas suggests a possible cause of the decreasing temperature differences. The first region includes only the first data run 2603 (not exhibiting this trend), and has an average heat flux of 5464 Btu/hr·ft^2 (17.2 kW/m^2). The next 5 runs (showing moderate upward drift) have mean average heat fluxes of 7745 Btu/hr·ft^2 (24.4 kW/m^2) representing a 29.5% increase over the initial transition run. Runs 2665, 2664, and 2663 (showing strong upward tendency) display mean average heat fluxes of 18360 Btu/hr·ft^2 (57.8 kW/m^2) representing a 57.8% increase over the moderate drift group and a 70.2% increase over the first run. This gives evidence that there is some relationship between the average test section wall heat flux and the decrease in temperature difference along the tube length, resulting in an upward drifting Nusselt number.

Turbulent Thermal Entry Length

For turbulent heat transfer it is seen that at a fixed Prandtl number the entry length is little affected by the Reynolds number but there is a marked decrease as the Prandtl number increases (Sparrow, Hallman & Siegel (1958)). This has been observed to be true in all experimental cases for both laminar and transition also. Ross (1956) suggests that the ratio of turbulent hydrodynamic quantities in the entrance region to the fully developed values are also independent of Reynolds number indicating that the velocity profile can be assumed developed at the inlet for turbulent flows. Data shows that the smallest station one Prandtl number (from Figure 3.8) in the turbulent region is 5.61 and this further supports the assumption of developed velocity profile.

Figure 3.8 introduces the local Nusselt number trends for fully turbulent flow with Reynolds Number ranging from 10243 at station 1 (3.205 diameters) to 44873 at station 22 (192.305 diameters). The local Nusselt number varied from 71.4 to 226.5 at station 1 and from 150.1 to 230.8 at station 22. Calling the standard used to judge the thermal entry length resulted in a nearly uniform entrance length of 9.615 diameters for the turbulent region as the Nusselt number deviation was less than 5% between stations 1 and 2 in all cases but one. From the statements made in the last paragraph this was to be expected. Therefore it was found that for fully turbulent forced convective flow the thermal

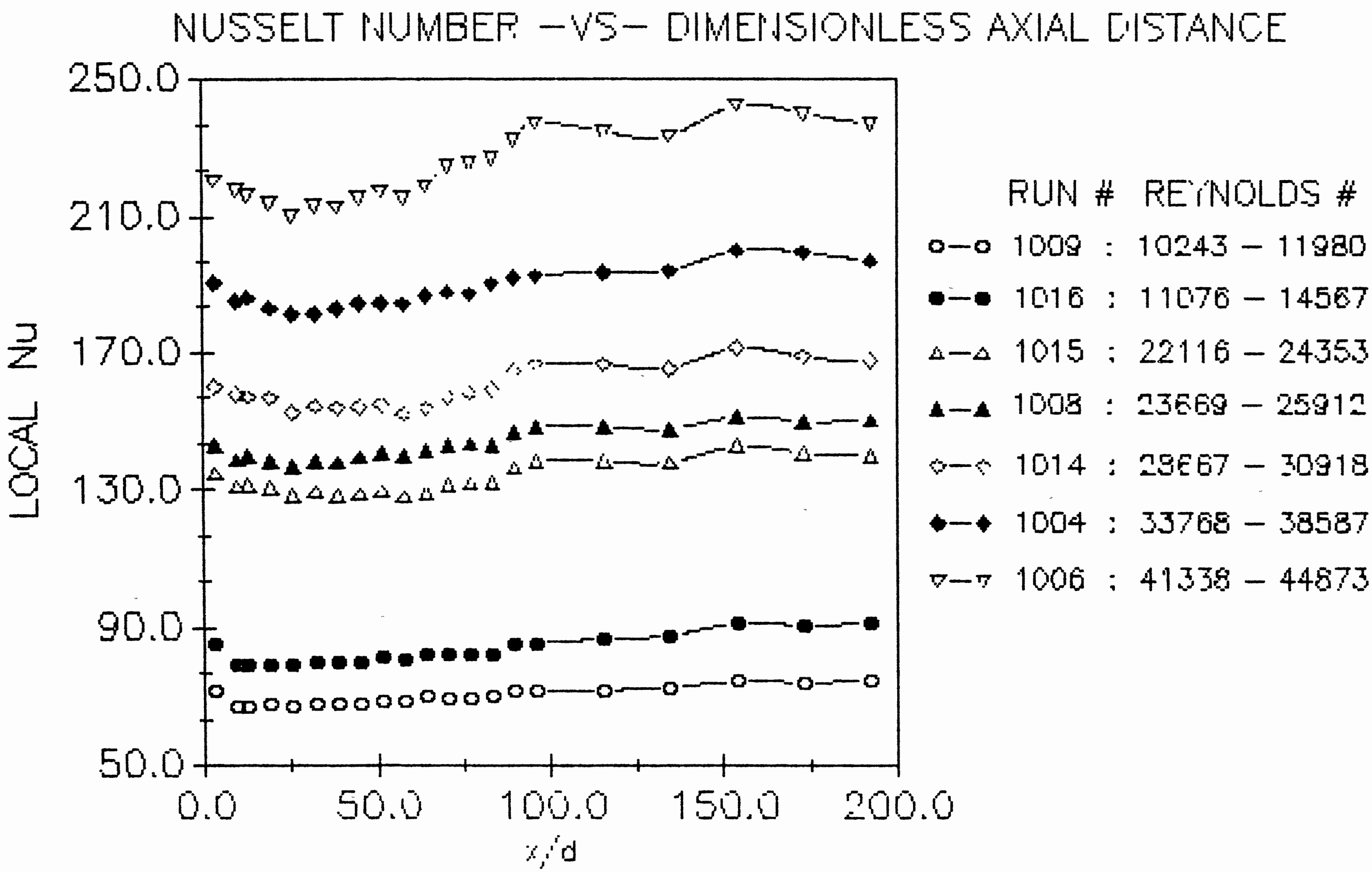


Figure 3.8 Nusselt Number Trends for Turbulent Flow

entrance effects are less than 10 diameters downstream.

Due to the high flow rates associated with turbulent flow natural convection did not influence the data. As seen in the transition region, the drifting trends in the Nusselt number with respect to axial direction is due to high heat fluxes resulting in diminishing temperature differences in the axial direction.

Comparison With Established Correlations

Using the data accumulated throughout the study it was desirable to consider how accurate it could be predicted by the conventional correlations. To accomplish this goal the correlations provided in Tables I through IV (see chapter I) were compared with the data conforming to the respective correlation limitations. The data sets were divided in two groups; the laminar and lower transition and the upper transition and turbulent flow regions. An arbitrary assessment of where the laminar and lower transition region would end and where the upper transition and turbulent region began was a prerequisite to making the comparisons. We determined the Reynolds number for the laminar and lower transition region to range from 4352 down to the lowest laminar data collected (281). To give the most complete comparisons possible we elected to overlap the mid-transition region by assigning the upper transition point at a Reynolds number (3639) below that defined for the lower transition. This resulted in a total of 57 runs compared in the laminar and lower transition flow region while 33 runs were compared in the upper transition and turbulent flow region (eight runs being compared in both defined areas).

The comparisons took the form of percent deviations between the equation prediction and the actual data for all stations along the entire length of the tube. Taking sum of the 26 station deviations and dividing by 26 resulted in the average absolute percent deviations for each run versus all pertinent correlations. Also the maximum percent deviation of each comparison was identified. The comparison data appears as described in Tables XIII and XIV of Appendix D. As a limitation for this table correlations providing absolute average deviation greater than 30% were not included. Generally the reported maximum deviations did not represent the data deviation trend and was therefore not considered as the major correlation criteria.

Laminar and Upper Transition Flow Comparisons

Mentioned previously in the thesis many problems were discovered with the laminar forced convection constant heat flux case. Because no data was available in this area no comparisons were applicable with the exception of the Sieder & Tate equation (1936). For this equation comparisons with only seven data runs occurred and the average absolute error ranged from 6.5 to 23.2%.

In the laminar mixed convective constant heat flux area Mori et al (1966) had absolute average deviations from 10.2 to 27.8% for only seven of the 57 region data runs. Colburn's equation (1933) shows better results with absolute average deviations ranging from 6.5 to 19.0% for 19 data sets.

Lower transition flow comparisons show Colburn's forced convective turbulent (1933) equation has absolute average deviations ranging from 11.9 to 23.2% over nine data sets. The Gnielinski equation (1976) shows absolute average deviations ranging from 2.4 to 16.9% for 15 data runs. The Friend & Metzner equation compares for only nine data runs and includes absolute average deviations ranging from 9.5 to 18.5% and appears to always over-predict the data. The Churchill correlation did not show good comparison and only four data sets were in the reasonable range and all near the 30% absolute average deviation limit. Chen's (1988a), (1988b), and (1988d) equations predicted our data well for the laminar and lower transition region. They displayed absolute average deviations ranging from 4.0 to 20.5% over 21 data runs, 3.7 to 22.4% for 17 runs, and 5.2 to 27.8% for 23 runs respectively.

The Colburn (1933) laminar flow mixed convection correlation displayed the best and most consistent absolute average deviations (none above 20% and only two above 14%) in the laminar and lower transition flow region and is highlighted in Figures 3.9 and 3.10. In Figure 3.9 the deviations in our experimental data is compared with the Colburn equation versus Reynolds number. The correlation shows good agreement with our data even beyond the range of validity for the equation. Figure 3.10 plots the same deviations for the dimensionless axial distance along the tube. In this figure it is evident that the Colburn equation overpredicts the data for the initial 100 diameters downstream of the inlet and underprediction occurs beyond that point. Also evident from this plot is that the deviations are bounded by 20% magnitudes above and below zero for virtually the entire tube length (x/d < 300).

Upper Transition and Turbulent Flow Comparisons

The data in these flow regions is typically for forced and mixed convective heat transfer while the correlations are for forced convective heat transfer only. However in these regions the equations generally correlated better and for many more experimental data runs than the previous regions. Sieder & Tate equation (1936) had absolute average deviations ranging from 2.5 to 13.9% over 28 of the 33 region data runs. The Dittus-Boelter (1930) equation shows absolute average deviations ranging from 2.6 to 21.7% for 28 runs and appears to overpredict the forced convective heat transfer data. Several other equations were in the overpredicting category and the Sleicher & Rouse (1975) equation yielded an absolute average deviation range from

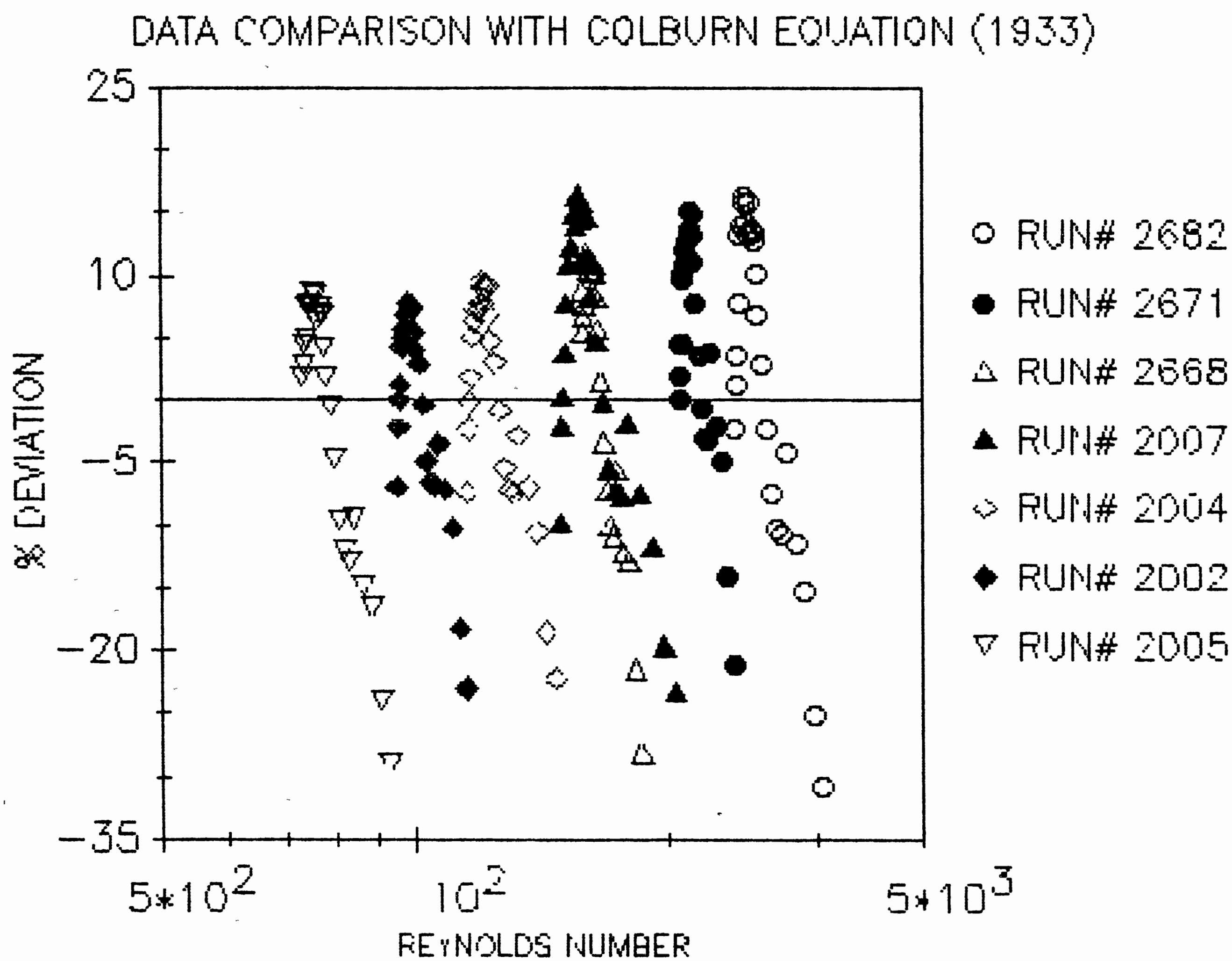


Figure 3.9 Laminar and Lower Transition Flow Data Reynolds Number Deviation Trend

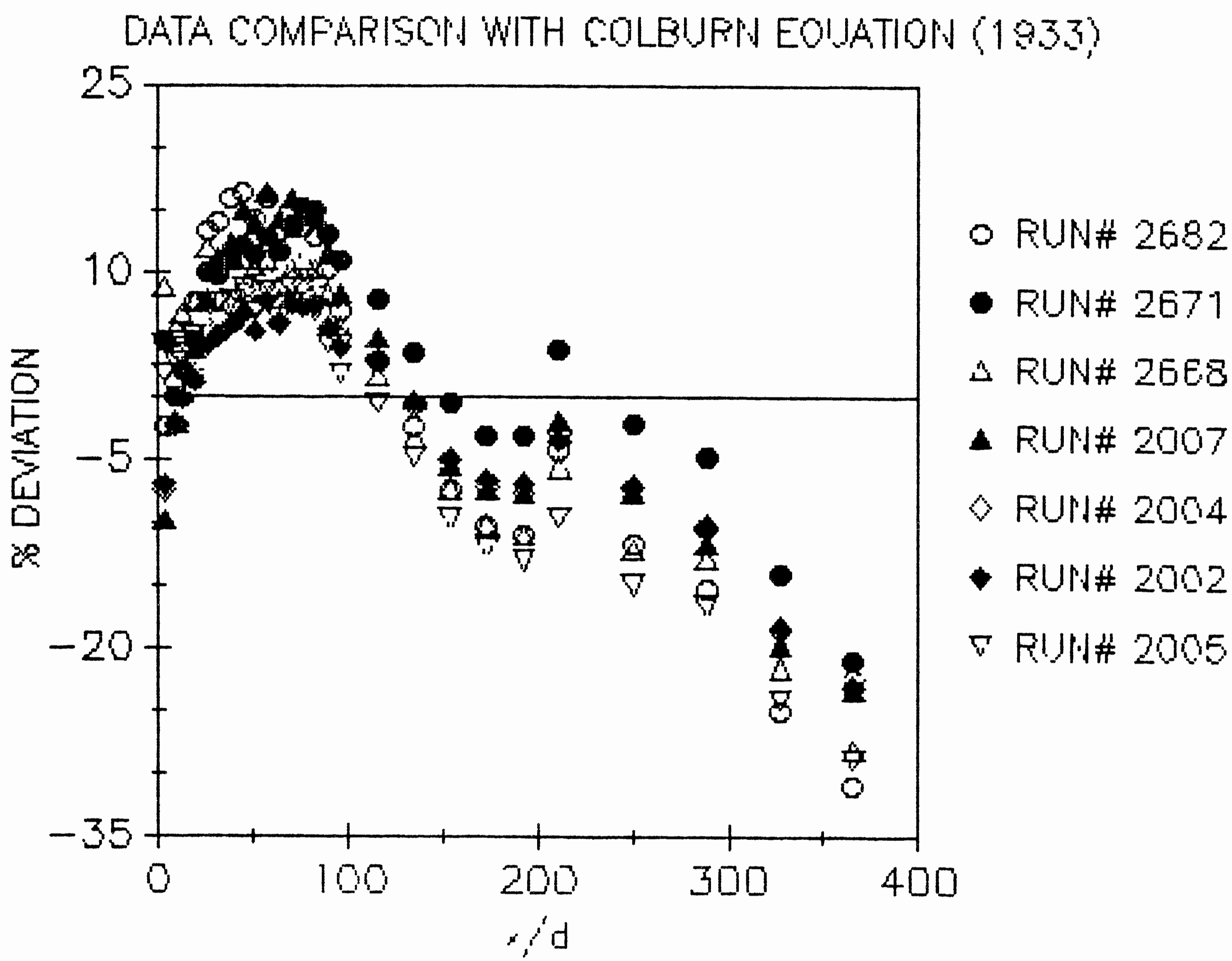


Figure 3.10 Laminar and Lower Transtion Flow Data
Tube Length Deviation Trend

3.4 to 21.8% and correlated for 25 data sets. Others included McAdams (1954a) and (1954b) equations that produced absolute average deviations ranging from 3.0 to 21.7% and 1.9 to 21.7% respectively. Both of the equations gave overpredictions of the data and correlated within reason for 28 data runs.

The Nusselt equation (1931) agreed with 23 experimental data runs and included absolute average deviations that ranged from 5.5 to 20.4%. Though the von Karman equation agreed with more data runs (30) it included generally higher absolute average deviations from 3.4 to 28.7% with only four being less than 10%. As with the laminar and lower transition region the Gnielinski equation (1976) had good absolute average deviations that ranged from 1.0 to 21.7% but for more data runs (27) in the upper transition and turbulent region with only seven having magnitudes greater than 10%.

Other equations that yielded generally poor results in the lower transition region improved in the upper transition and turbulent flow region. Some equations included the Friend and Metzner (1958), the Churchill (1977b), and the Chen (1988b), (1988c), & (1988d) equations. These equations conveyed absolute average deviations ranging from 2.1 to 19.5%, 6.4 to 27.0%, 1.8 to 11.6%, 1.8 to 26.8%, and 5.2 to 30.0% respectively. The first three had extensive correlation with the data (1958) and (1977b) showing agreement with 28 and 25 data sets respectively while

(1988b), (1988c), and (1988d) agreed with 23, 27, and 11 data sets each.

From the data of Table XIV it is evident that the Sieder & Tate equation does the best data prediction in the fully turbulent force convective regime (this is also apparent in Figure 2.9). This equation consistently yields the lowest maximum and absolute average deviations. In Figures 3.11 and 3.12 our experimental data deviations are plotted against the Sieder & Tate correlation. Figure 3.11 suggests that this equation is generally an underpredictor for high Reynolds number and gives good prediction for all runs shown. The equation underpredicts because it does not account for any natural convection effects. In Figure 3.12 the trends of deviation for dimensionless tube length are apparent. Again good comparison is achieved and the deviations are bounded by +5 and -10% for all but five data points. The underpredicting nature of the equation is also evident in this plot.

Another correlation showing good agreement with our data is the Gnielinski equation. It is advertised for use at Reynolds number above 2300 and up to 5 million. Figure 3.13 shows our experimental data deviation from this equation plotted against Reynolds number and indicates excellent agreement at low Reynolds number and increasing deviation with higher Reynolds number (also evident in Figure 2.9). Of the 27 runs we made comparison to this correlation a maximum deviation of 28% was observed and only

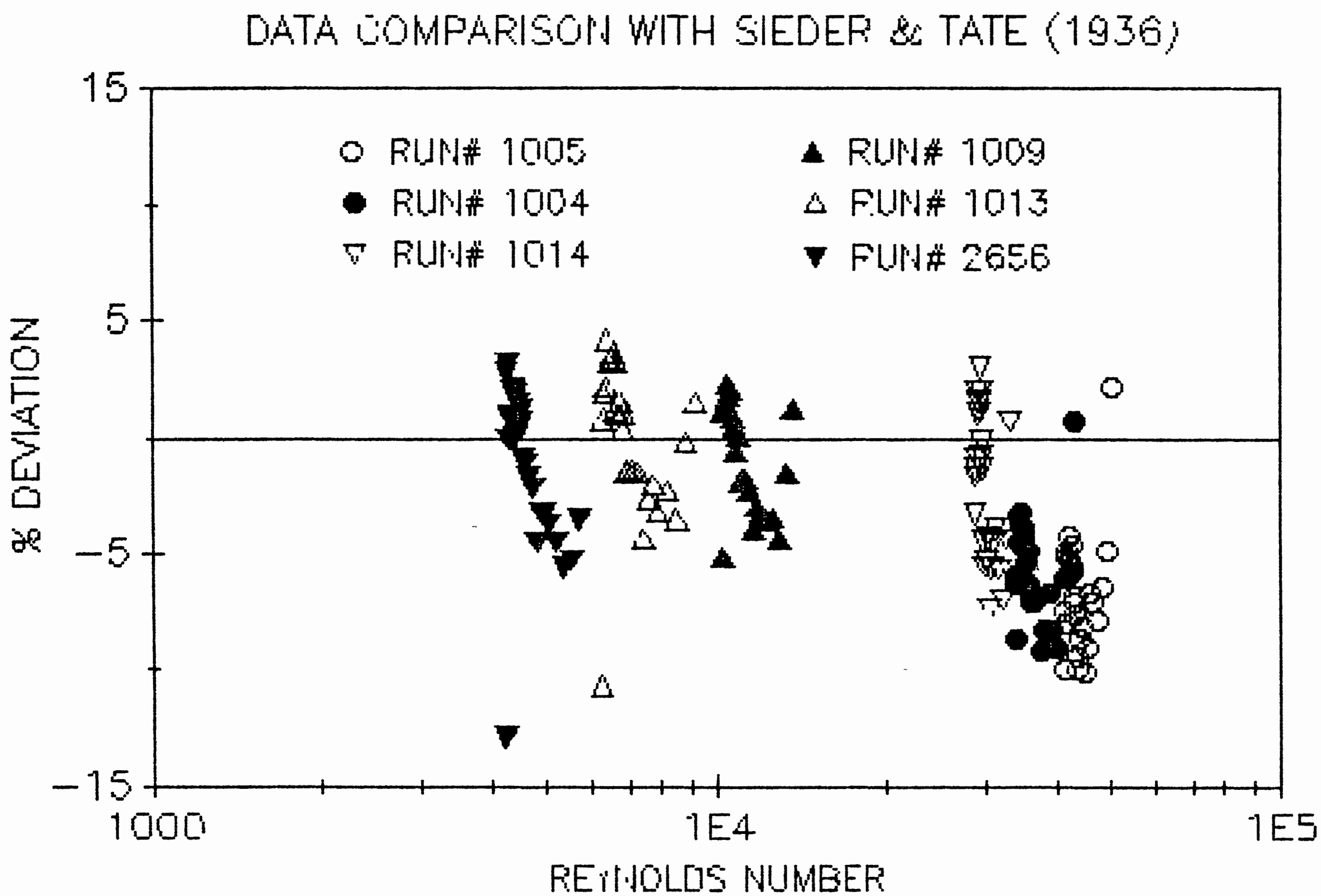


Figure 3.11 Upper Transition and Turbulent Flow Data Reynolds Number Deviation Trend I

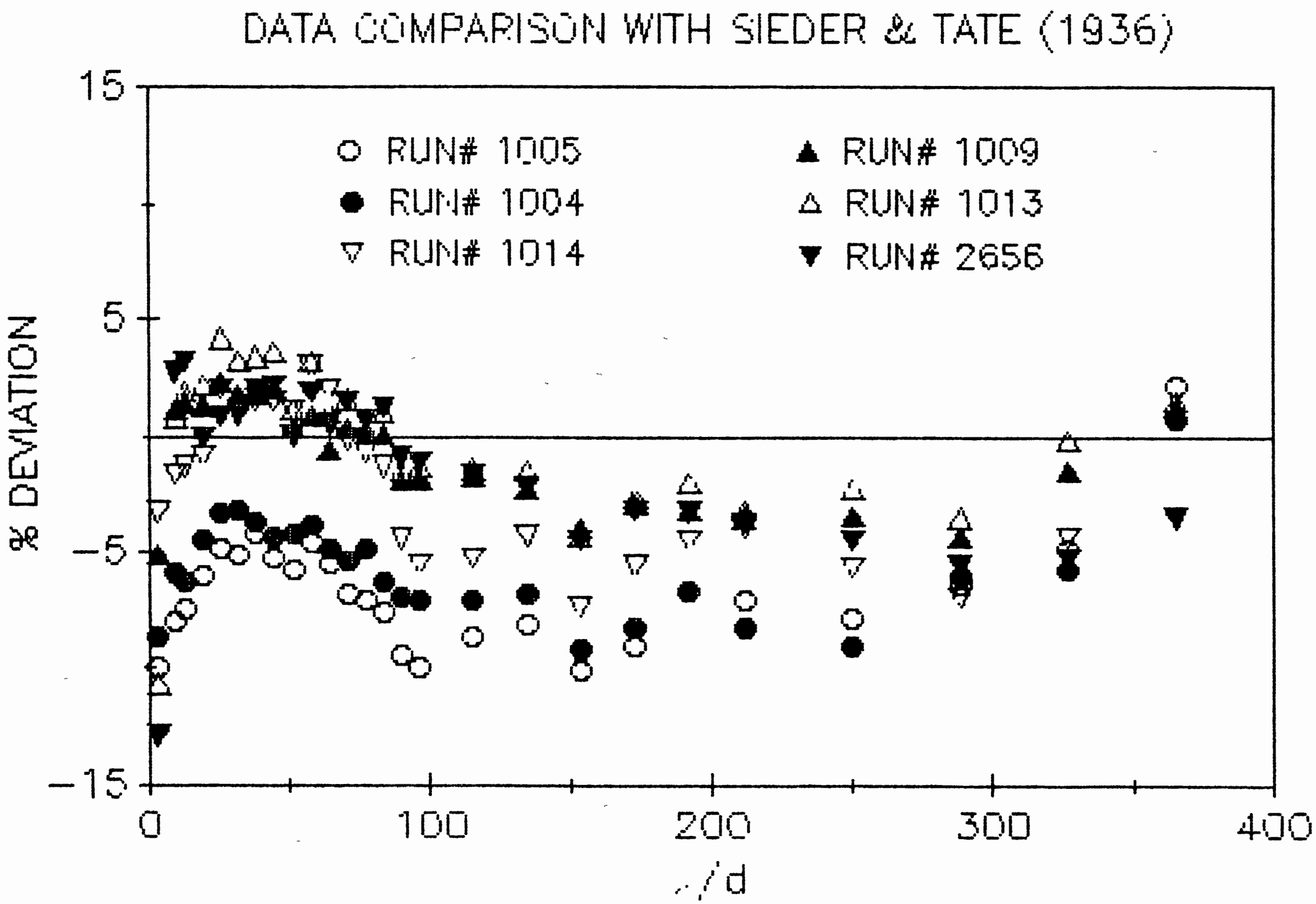


Figure 3.12 Upper Transition and Turbulent Flow Data Tube Length Deviation Trend I

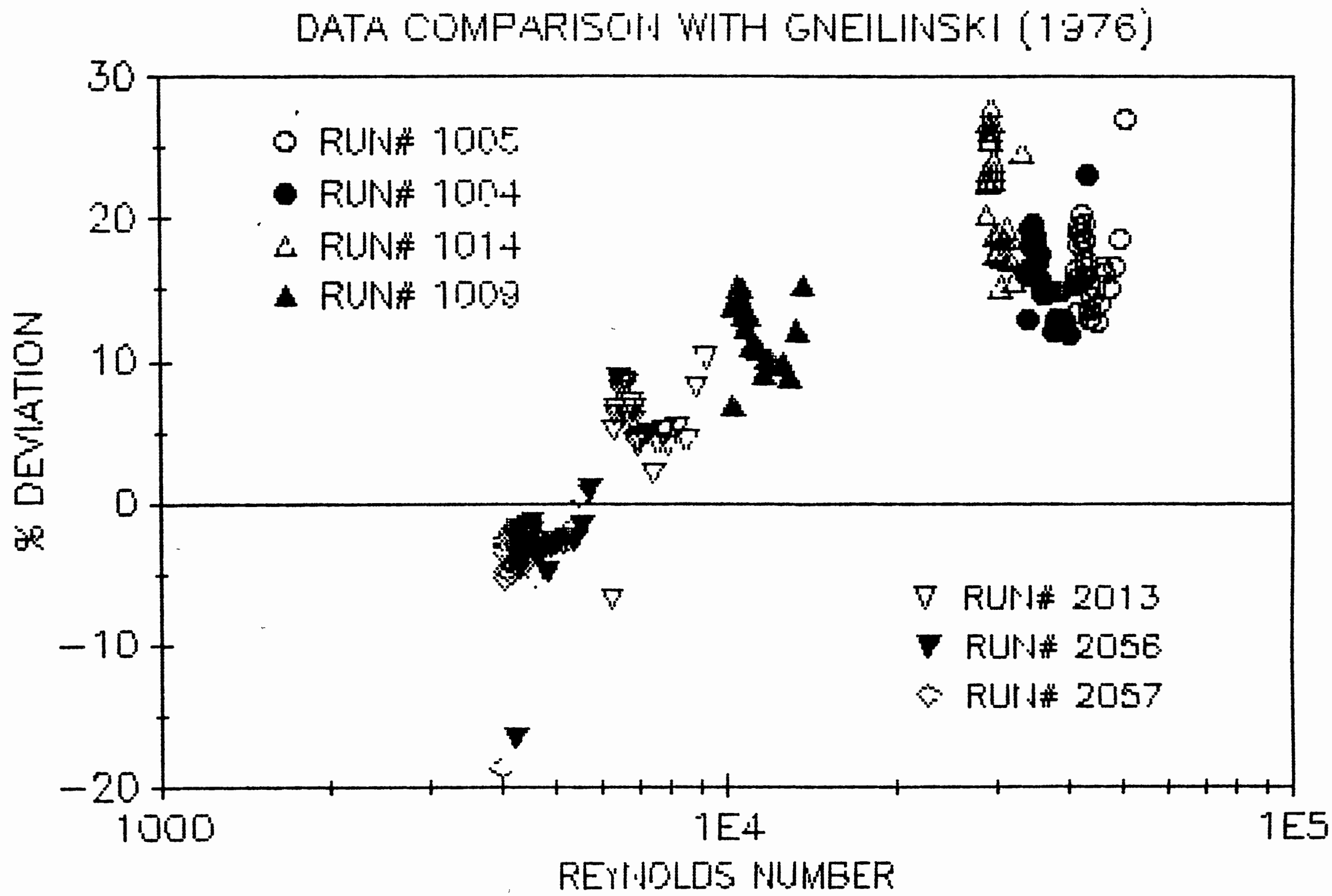


Figure 3.13 Upper Transition and Turbulent Flow Data Reynolds Number Deviation Trend II

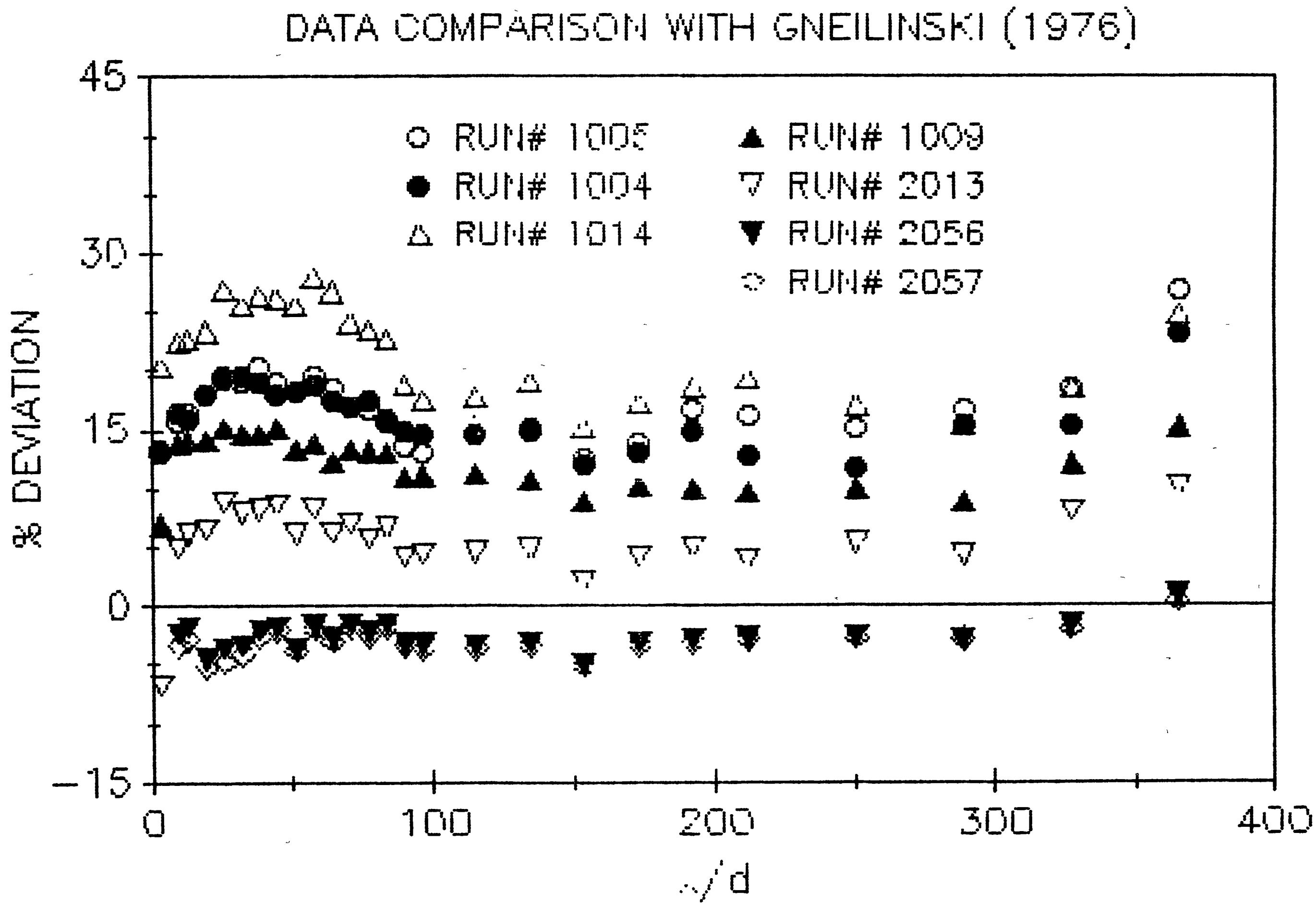


Figure 3.14 Upper Transition and Turbulent Flow Data Tube Length Deviation Trend II

seven runs produced absolute average deviations greater than 10%. Figure 3.14 shows the Gnielinski correlation as mostly an overpredictor of our data with the deviations remaining fairly constant over the entire tube length. Good agreement is supported by the figure as the deviations fall between -5 and 20% for all but one of the data runs shown.

To compare Chen's equations it was necessary to encompass the entire flow spectrum from laminar through lower and upper transition and into the fully turbulent regime. Good agreement is evident across all flow regimes and Figure 3.15 illustrates that his equations underpredict our data for low Reynolds number and slightly overpredicts at high Reynolds number. All of his equations compare well with our data and from Figure 3.15 it can be seen that for all equations over the entire flow range the data is predicted within -20% to 15% excepting only two points. Figure 3.16 presents the same deviations versus the dimensionless tube length. Chen's four equations appear to give the most uniform predictions of our data as evidenced by the nearly constant deviations with tube length. On this figure it can be clearly seen that our nine plotted data sets when compared with Chen's equations deviated within a 15% tolerance band excluding only 13 data points.

As one would expect Chen's correlations fit his data better than ours in general. However, equation (1988a) has an absolute average deviation of 12.9% for his data and a mean absolute average deviation of only 10.2% over 21 of our

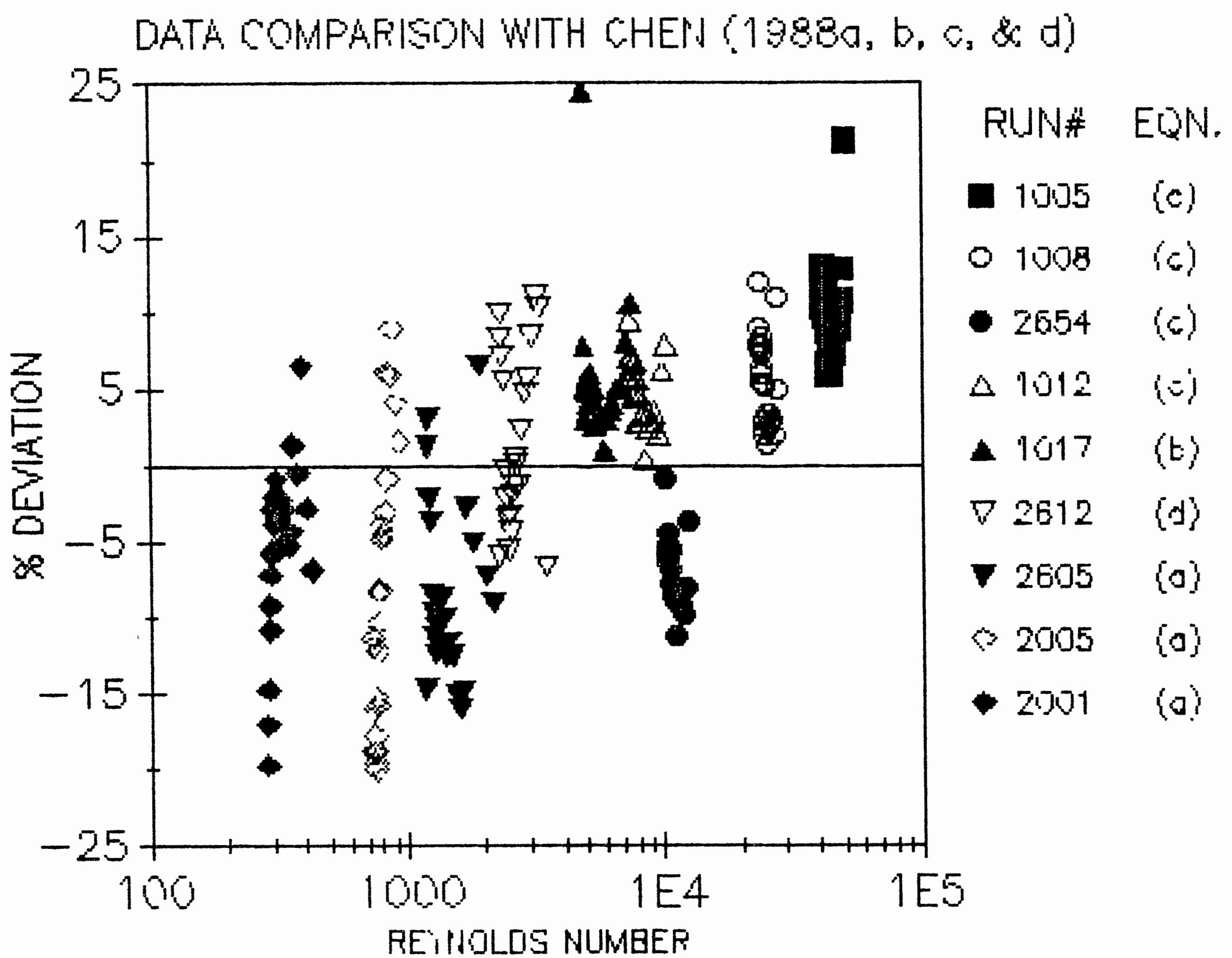


Figure 3.15 Reynolds Number Data Deviations
for all Flow Regions

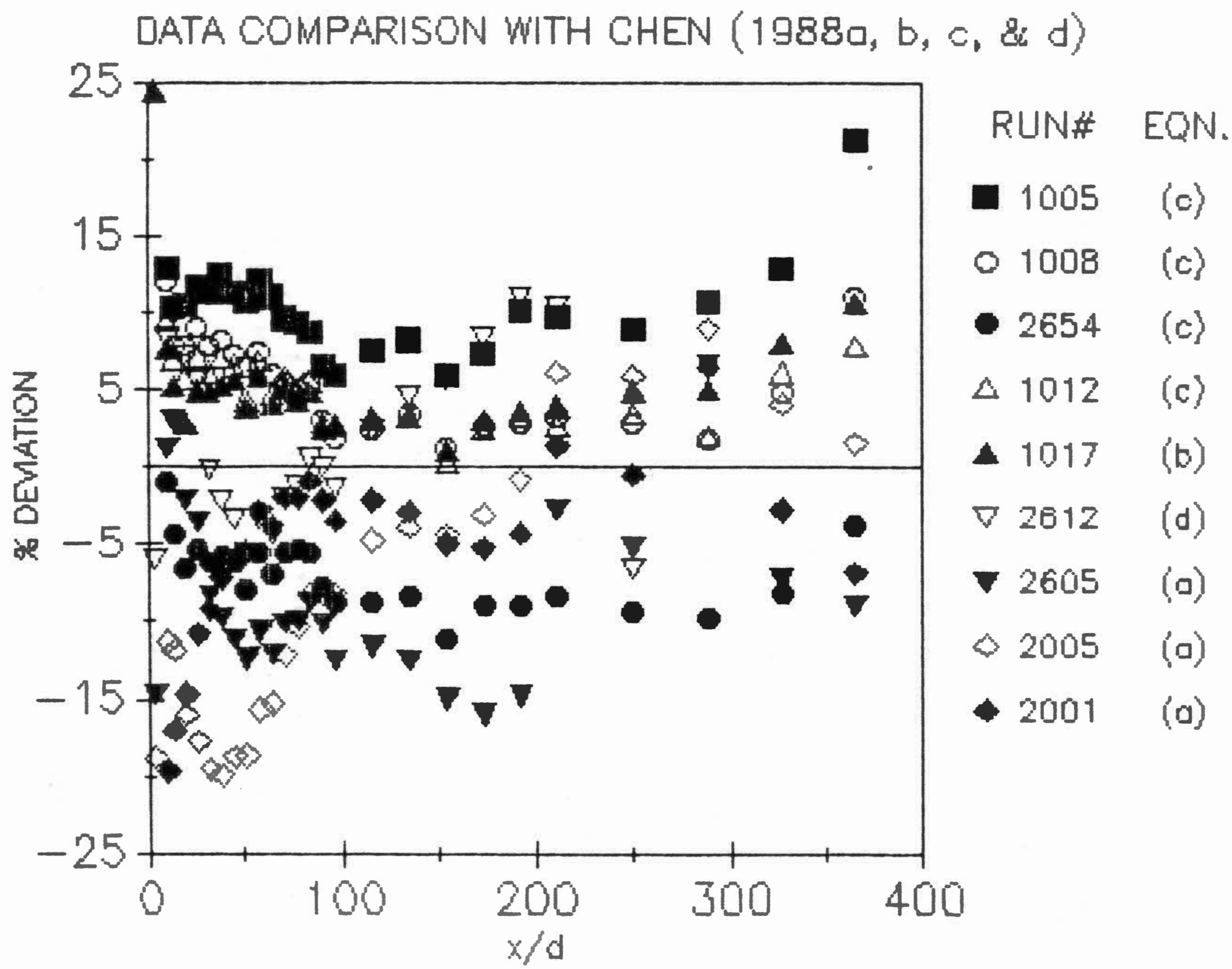


Figure 3.16 Tube Length Data Deviation Trends for all Flow Regions

laminar and lower transition flow data sets. Equation (1988c) shows a 5% tolerance band for his data while ours fell within roughly a 15% deviation limit. Using equation (1988b) Chen reports nearly exact agreement with his data while this correlation provided a mean absolute average deviation of 8.7% for 33 laminar, transition, and turbulent runs. His final equation (1988d) is reported to have an absolute average deviation of 15.9% against his data while we find a mean absolute average deviation of 15% for 22 of our data runs.

Data Prediction Correlation

For the experimental data runs collected it was desirable to develop a new correlation or modify an existing correlation to predict the heat transfer data values with reasonable accuracy. Greatest precision is probably attained when separate new correlations are developed corresponding to the data for each region of flow. However this requires much labor to develop the equations as well as make use of them. Difficulty arises from the necessity of determining the mode of flow before utilizing an individual equation to predict heat transfer for that region of flow. Additional problems occur when the potential for mixed convective heat transfer is present in one or more of the flow regions. Therefore we found that acceptable accuracy could be attained over the entire region of flow with forced convective dominated or mixed convective heat transfer using a modified form of the Churchill correlation (1977b). Use of this equation requires no determination of the region of flow, uses only the Reynolds and Prandtl numbers as inputs, and was easily developed.

We began with and retained the general structure of the equation as seen in Table III, but by using our data and a curve fitting program call MARQ (see Appendix C) we evaluated new constants appearing throughout the equation. The modified form of the equation is presented below:

$$Nu = Nu_l + \left\{ \frac{(\exp\,(2311 - Re)/533.7)}{Nu_l^{2}} + \frac{1}{Nu_t^{2}} \right\}^{-1/2} \qquad (3.6)$$

where:

$$Nu_t = 6.3 \quad + \left[\frac{0.079\ (f)^{0.5}\ Re\ Pr}{(1 + Pr^{0.8})^{5/6}}\right] \qquad (3.7)$$

$$(f)^{0.5} = \left[\frac{1}{2.21\ \ln\ (Re/7)}\right] \qquad (3.8)$$

and Nu_l is a constant equal to 14.5 representing our fully developed mixed convective constant heat flux Nusselt number. Comparing this value to the forced convection counterpart (4.364) we find a 330% increase for fully developed laminar flow.

This form of the equation provided excellent agreement with our data in the transition flow region and good comparison with our laminar and turbulent flow data. Figure 3.17 conveys the deviations between our data and equation (3.7). Only fully developed data from station 22 (x/d = 192.308) is used for the comparison and for all 70 runs used in Figure 3.4, the absolute average deviation was 14.16%. Nearly all the deviations are confined to a 25% deviation band with only nine having greater magnitude. This equation represents the most useful and applicable correlation for our forced and mixed convective constant heat flux data over all regions of flow.

After converting the Nusselt number predictions provided by equation (3.7) to the Stanton Prandtl parameter ($StPr^{0.6}$) and plotting them together with the data from Figure 3.4, good point by point comparison is evident from Figure 3.18. The correlation appears to best predict data in transition flow from about 2000 to 9000 Reynolds number.

In both the laminar and turbulent flow regions the prediction data appears to deviate slightly from the experimental data but the general trends are maintained. It is suggested that a separate equation (Sieder & Tate or Gnielinski) for constant heat flux forced convective heat transfer in the turbulent regime would probably predict our data better for Reynolds number greater than 9000. In the the laminar flow region the Colburn (1933) equation for mixed convection will yield greater accuracy and if laminar forced convection is present it is expected that our modified Churchill equation will fail.

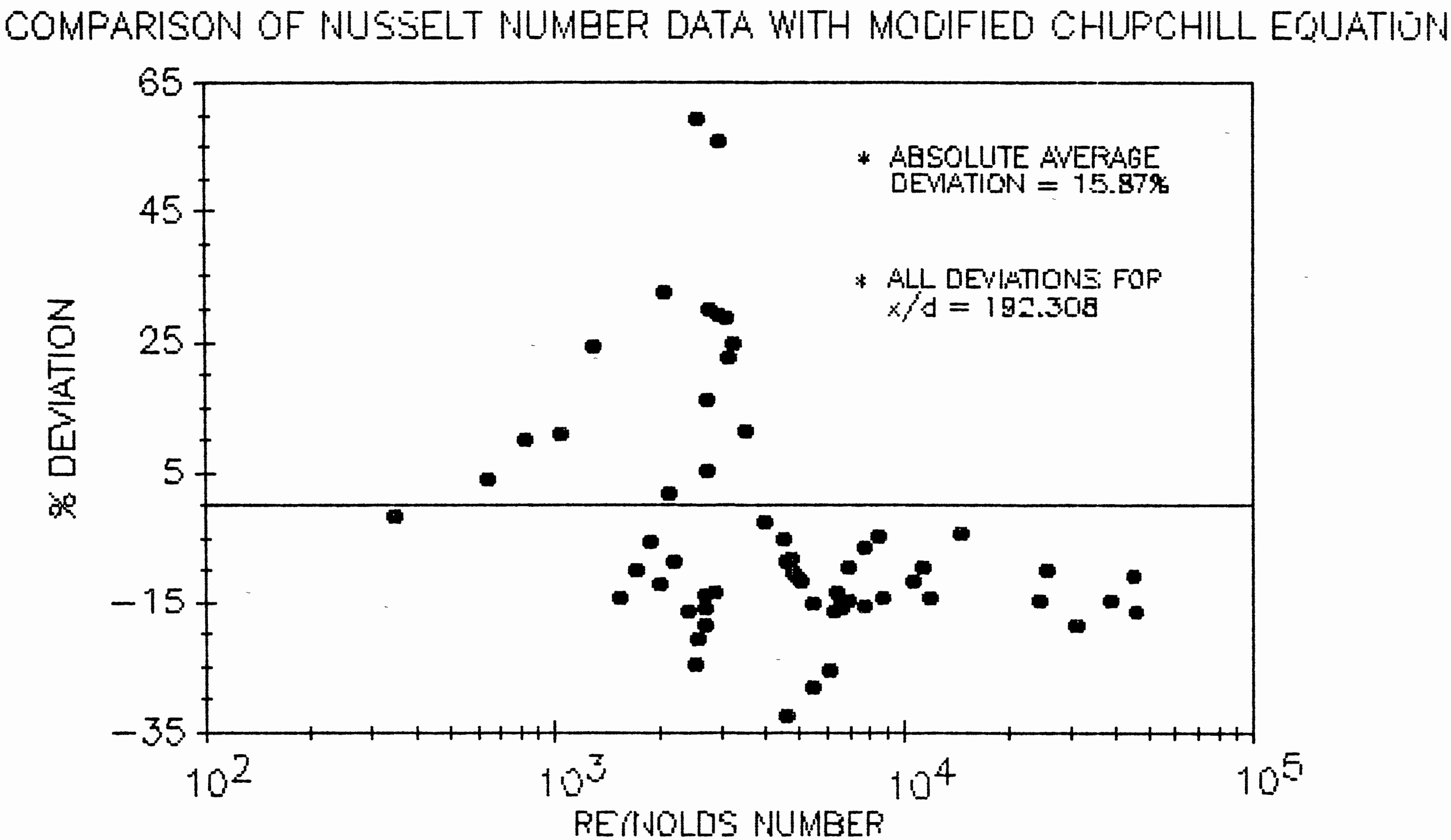


Figure 3.17 Reynolds Number Deviation Trends for Data of Fully Developed Thermal Profile

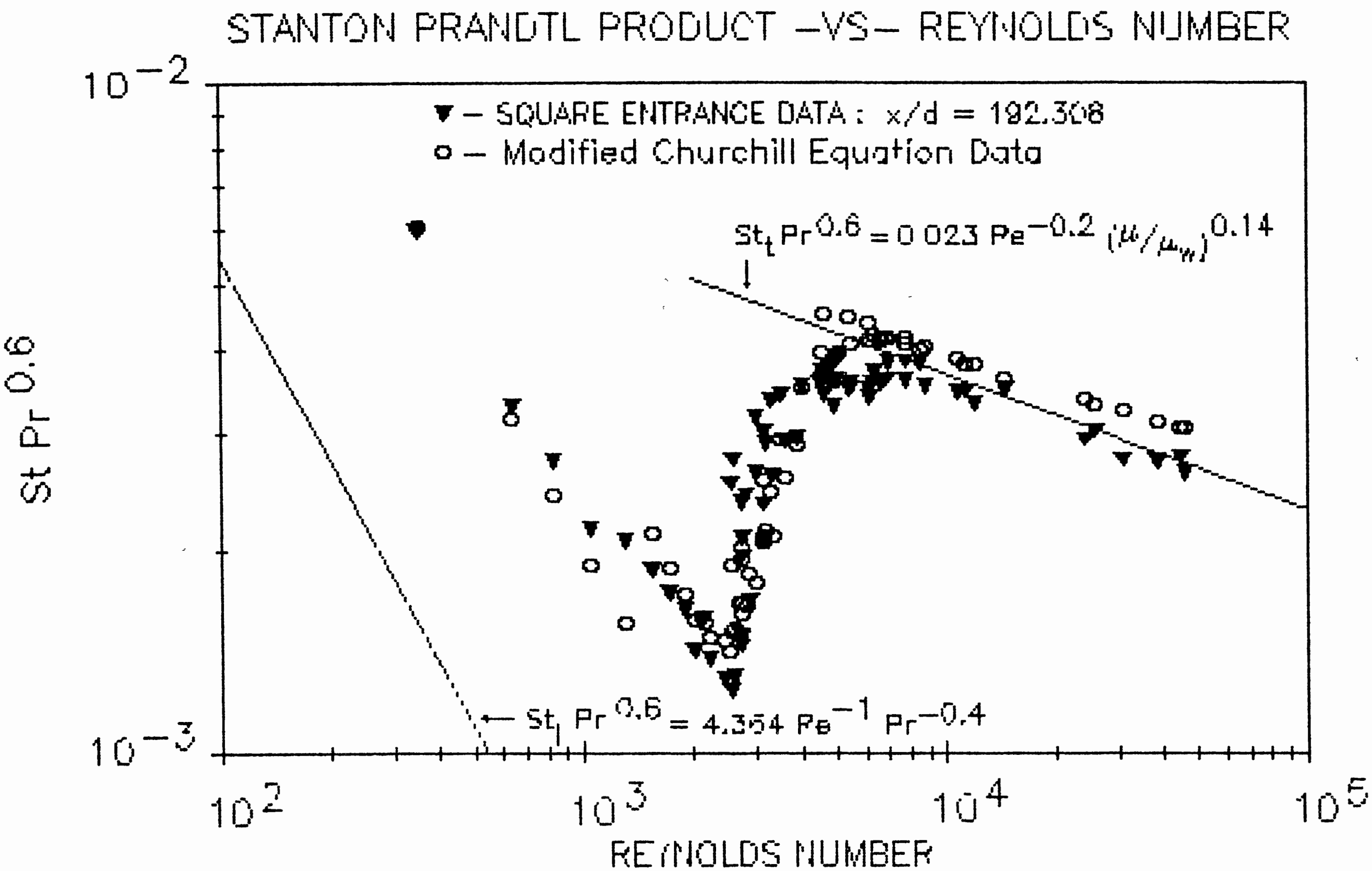


Figure 3.18 Square Entrance Heat Transfer Data Comparison
With Modified Churchill Equation

CHAPTER IV

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

The goals of this thesis as set forth in Chapter I were successfully achieved. Construction of a versatile setup was completed, the entrance effects for a constant heat flux heated tube with square-edged inlet were studied, a complete heat transfer data base encompassing all flow regions was assimilated, transition flow region boundaries were defined, and the correlations from Tables I through IV were compared against our data.

Experimental Apparatus

With the completion of the setup future transition flow experiments will continue. These studies will include transition flow velocity intermittency, the effects of various inlet geometry, and the utilization of flow visualization techniques. Although the flow visualization test section is not yet in place, the existing test loop design has incorporated features for easy acceptance of such an addition. Most of the apparatus problems have been solved and trouble-free experimentation should occur with its use.

The Laminar Region

From our data it is evident that laminar flow has persisted in the tube from as low as 281 Reynolds number up to 2400. Using the Square-edged inlet configuration with a uniform wall heat flux we determined the laminar flow thermal profile to be fully established for distances greater than 84 diameters downstream of the inlet. It was also determined that the thermal entrance effects generally lasted longer as the Reynolds number increased. However, for laminar constant heat flux tube flow no thermal entrance effects were evident beyond 52.4 inches (4.37 ft) from the inlet.

Throughout the laminar flow regime the heat transfer data was affected by the presence of free convection and no laminar forced convection data was obtained. It was found that high heat flux was responsible for the mixed convection and could not be reduced such that forced convection heat transfer would dominate.

Resulting from the comparisons of all the laminar mixed convection equations of Table II (Chapter I) it was discovered that most did not predict our data well. Only the Colburn (1933) equation gave good comparison over a reasonable amount of data runs. The deviations between our data and the equation as illustrated in Figures 3.9 and 3.10 are restricted to a 20% tolerance band for distances less than 300 diameters downstream of the inlet. Also evident was the correlation tendency to underpredict our data for tube lengths less than 100 diameters and overpredict beyond.

The Transition Region

The data collected defines a transition flow region from Reynolds number of 2400 up to a value of 9000. This was determined by plotting the heat transfer data as a Stanton and Prandtl parameter ($StPr^{0.6}$) versus Reynolds number (see Figure 3.4). The data makes a clearly evident shift from the laminar mixed convection trend and blends smoothly with the turbulent forced convection pattern. The mixed convective heat transfer persisted through the lower transition region to a Reynolds number of approximately 3000 causing the transition region definition to be based on both forced and mixed convection data.

With the square-edged entrance design thermal entrance effects were determined to be nonexistent for transition tube flow greater than 20 diameters downstream of the entrance. This corresponds to 12.5 inches (1.04 ft) and was considered to be nearly constant with Reynolds number.

By comparison of the data with the correlations of Chen, we determined that good agreement for both of his transition region equations occurs for our data. The problem using his equations is that they are bulky and require separate equations for lower and upper transition flow. In general we determined that our data was predicted nearly as well by these equations as his own. From curve fitting our data we evaluated new constants for the general form of the Churchill equation which compared well with our data and this was determined as the best prediction equation for the transition region.

The Turbulent Region

We determined the turbulent region to be bounded by Reynolds number greater than 9000 for uniform heat flux and square-edged inlet. It was also clear that no free convection effects were present throughout the turbulent regime.

From the forced convective heat transfer data it is found that no thermal entrance effects were present beyond 10 diameters of the inlet. This translates to a distance of 6.24 inches (0.52 ft) and remained essentially independent of Reynolds number.

Virtually all of the correlations for forced convective heat transfer given in Table III of Chapter I provided reasonable agreement with our data. However, it was determined that the Seider & Tate (1936) equation gave the best correlation over most of the data sets.

Recommendations

As alluded to in Chapter III it is believed that the heat flux could be responsible for the lower boundary of the transition region. It appeared that for high heat flux at low Reynolds number there could have been an early departure tendency from the laminar mixed convection line.

It is therefore recommended that additional attention be given to the effect of heat flux variation at Reynolds number below 2400 and the resulting changes in transition flow boundary.

Also pertaining to heat flux, future investigations are recommended to include research into the causes behind the upward drift in the fully developed Nusselt number for transition and turbulent flows. It was suggested that this might be caused by high wall heat flux and slight free convection effects. However, other factors not yet discovered may be influencing the diminishing temperature differences in these flow regions.

From the literature comparisons no recommendations of the forced laminar convective heat transfer equations can be made. However, in the laminar mixed convection regime it is recommended that the Colburn (1933) equation be used to predict experimental data. Similarly for the forced convective turbulent flow region the Seider & Tate (1936) equation is recommended. Furthermore, in the transition region the modified Churchill equation should be used to give heat transfer predictions.

Some slight apparatus modifications are recommended such that data in the laminar forced convective region can be obtained for comparison with the respective correlations. This might include the addition of a smaller capacity welder for low heat flux application.

REFERENCES

Augustine, J. R. (1990), Pressure Drop Measurements in the Transition Region for a Square-Entrance Tube, M.S. Thesis, Oklahoma State University, Stillwater, Oklahoma.

Bergles, A. E., and Simonds, R. R. (1971), Combined Forced and Free Convection for Laminar Flow in Horizontal Tubes with Uniform Heat Flux, Int. J. Heat Mass Transfer, v. 14, pp. 1989-2000.

Bhatti, M. S. and Shah, R. K. (1987), Turbulent and Transition Flow Convective Heat Transfer in Ducts, from Handbook of Single-Phase Convective Heat Transfer (edited by Kakac, S., Shah, R. K. and Aung, W.), John Wiley & Sons (New York).

Chen, J. (1988), Heat Transfer in High Laminar, Transition and Lower Turbulent Flow Regimes for Square-Edged Contraction Entrance in a Circular Tube, Ph.D. dissertation, Oklahoma State University, Stillwater, Oklahoma.

Ede, A. J. (1961), The Heat Transfer for Flow in a Pipe, Int. J. Heat Mass Transfer, v. 4, pp. 105-110.

Ede, A. J., Hislop, C. I. and Morris, R. (1968), Effect on the Local Heat-Transfer Coefficients in a Pipe of an Abrupt Disturbance of the Fluid: Abrupt Convergence and Divergence of Diameter Ratio 2/1, Proc. Instn. Mech. Engrs, v. 170, pp. 1113-1130.

Hong, S. W. and Bergles, A. E. (1976), Laminar Flow Heat Transfer in the Entrance Region of Semi-Circular Tubes with Uniform Heat Flux, Int. J. Heat Mass Transfer, v. 19, pp. 123-124.

Hussain, N. A. and McComas, S. T. (1970), Experimental Investigation of Combined Convection in a Horizontal Circular Tube with Uniform Heat Flux, Int. Heat Transfer Conf., Paris, v. 4, paper no. NC3.4.

Kakac, S., Shah, R. K. and Bergles, A. E. (1981), Low Reynolds Number Flow Heat Exchangers, Hemisphere Pub. Corp. (New York).

Kays, W. M. (1955), Numerical Solution for Laminar Flow Heat Transfer in Circular Tubes, ASME Trans., v. 77, p. 1265.

Kays, W. M. and Crawford, B. S. (1980), Convective Heat and Mass Transfer, McGraw-Hill (New York).

Kline, S. J. and McClintock, F. A. (1953), Describing Uncertainties in Single-Sample Experiments, Mech. Engr., v. 1, pp. 3-8.

Langhaar, H. L. (1942), Steady Flow in the Transition Length of a Straight Tube, J. Appl. Mech., v. 9.

Moffat, R. J. (1988), Describing the Uncertainties in Experimental Results, Exp. Thermal and Fluid Sci., v. 1, pp. 3-17.

McComas, S. T. and Eckert, E. R. G. (1966), Combined Free and Forced Convection in a Horizontal Circular Tube, Int. J. Heat Mass Transfer Trans. ASME, v. 88, pp. 147-153.

Metais, B. and Eckert, E. R. G. (1964), Forced, Mixed, and Free Convection Regimes, J. Heat Transfer Trans. ASME, pp. 295-296.

Morcos, S. M. and Bergles, A. E. (1975), Experimental Investigation of Combined Forced and Free Laminar Convection in Horizontal Tubes, Int. J. Heat Mass Transfer Trans. ASME, v. 91, pp. 212-219.

Mori, Y. and Futagami, K. (1967), Forced Convection Heat Transfer in Uniformly Heated Tubes (2nd Report), Int. J. Heat Mass Transfer, v. 10, pp. 1801.

Ogawa, M. and Kawamura, H. (1986), Effects of Entrance Configuration on Pressure Loss and Heat Transfer of Transitional Gas Flow in a Circular Tube, JSME Trans., 52 (477), pp. 2164-2169.

Petukhov, B. S. and Polyakov, A. F. (1967), Effect of Free Convection on Heat Transfer During Forced Flow in Horizontal Pipe, High Temperature, v. 5, pp. 75-81.

Petukhov, B. S. and Polyakov, A. F. (1967), Effect of Free Convection on Heat Transfer During Forced Flow in Horizontal Pipe, High Temperature, v. 5, pp. 348-351.

Petukhov, B. S., Polyakov, A. F. and Strigin, B. K. (1969), Heat Transfer in Tubes with Viscous-gravity Flow, Heat Transfer-Soviet Research, v. 1, pp. 24-31.

Petukhov, B. S. and Popov, V. N. (1963), Theoretical Calculation of Heat Exchange and Frictional Resistance in Turbulent Flow in Tubes of an Incompressible Fluid with Variable Physical Properties, Teplifiz. Vys. Temp, v. 1, pp. 85-101.

Readal, T. C. (1969), Effect of Density Gradients on the Rate of Heat Transfer for Flow Through a Horizontal Pipe, M.S.Thesis, University of Illinois, Urbana.

Shah, R. K. and London, A. L. (1978), Laminar Flow Forced Convection in Ducts, A Supplement to Advances in Heat Transfer, Academic Press (New York).

Shah, R. K. (1983), Fully Developed Laminar Flow Forced Convection in Channels, from Low Reynolds Number Heat Exchangers (edited by Kakak, S., Shah, R. K. and Bergles, A. E., Hemisphere/McGraw-Hill (Washington D. C.).

Shannon, R. L. and Depew, C. A. (1968), Combined Free and Forced Laminar Convection in a Horizontal Tube With Uniform Flux', J. Heat Transfer Trans. ASME, v. 90, pp. 353-357.

Shannon, R. L. and Depew, C. A. (1969), Combined Free and Forced Laminar Convection in a Horizontal Tube With Variable Viscosity and Free Convection Effect, J. Heat Transfer Trans. ASME, v. 91, pp. 251-258.

Sieder, E. N. and Tate, G. E. (1936), Heat Transfer and Pressure Drop of Liquids in Tubes, Ind. and Eng. Chem., v. 28, pp. 1429-1435.

Siegal, R., Sparrow, E. M. and Hallman, T. M. (1958), Steady Laminar Heat Transfer in a Circular Tube with Prescribed Wall Heat Flux, Appl. Sci. Res., Sec. A, v. 7, pp. 386-392.

Siegwarth, D. P., Mikesell, R. D. Readal, T. C. and Hanratty, T. J. (1969), Effect of Secondary Flow on the Temperature Field and Primary Flow in a Heated Horizontal Tube, Int. Heat Mass Transfer, v. 12, pp. 1535-1552.

Sparrow, E. M., Hallman, T. M. and Siegal, R., (1958), Turbulent Heat Transfer in the Thermal Entrance Region of a Pipe With Uniform Heat Flux, App. Sci. Res., Sec. A, v. 7, pp. 37-52.

White, F. M. (1974), Viscous Fluid Flow, McGraw-Hill (New York).

APPENDIXES

APPENDIX A

UNCERTAINTY ANALYSIS

Experimental Error

The probable error associated with the measurement of flow rate and heat transfer coefficient are presented in this appendix. The flow rate curve fit equations for all mixtures used in this thesis are presented in Figures 2.7 and 2.8 while the Nusselt number trends appear in Chapter III.

Volume Flow Rate

The equation for volume flow rate is defined as:

$$\dot{Q} = \frac{dV}{dt} \tag{A.1}$$

Using a cylindrical reservoir and measuring the change in height over a specific time interval the volume change (dV) can be calculated from the volume of a cylinder given as:

$$V = (pi)r^2h \tag{A.2}$$

and

$$dV = 2(pi)rhdr + (pi)r^2dh \tag{A.3}$$

The reservoir manufacture advertises the tank diameter to be 15.75 inches and supplies no tolerance. Therefore it is assumed that the change in radius is neglect-able (dr = 0). From this it is clear that the first dV term contributes nothing and the uncertainty associated with the volume change is a function only of the measured quantities for initial tank height and final tank height and the flow rate uncertainty is therefore given as:

$$U_Q = \left[\left(\frac{dh}{h_{init}}\right)^2 + \left(\frac{dh}{h_{final}}\right)^2 + \left(\frac{dt}{t_f - t_i}\right)^2\right]^{1/2} \qquad (A.4)$$

The average initial tank height was 15 inches and for a sample calculation the final tank height was 10.5 inches for an experimental time period of 71.22 seconds. Other necessary quantities are shown below:

dh The probable error in height measurements is 0.0625 inches.

dt The estimatable error in reaction time is given to be 0.1 seconds.

Then the uncertainty for flow rate is given as:

$$U_Q = [(0.0625/15.0)^2 + (0.0625/10.5)^2 + (0.1/71.22)^2]^{1/2}100$$
$$= 0.74\%$$

This is approximately the normal uncertainty for all flow rates independent of fluid type however, the above calculation was for 100% ethylene glycol.

<u>Heat Transfer Coefficient</u>

The heat transfer coefficient is defined as:

$$h = \dot{q}''/(T_{wi} - T_b) \tag{A.5}$$

The percent probable error for h is given by:

$$U_h = [(\frac{d\dot{q}''}{\dot{q}''})^2 + (-\frac{d\ T}{T})^2]^{1/2} \tag{A.6}$$

The heat flux rate can be calculated by either of two methods. The first method is the enthalpy rise attributed to the fluid and the second method is the product of the voltage drop across the tube and the current carried in the test section. Because we used the second method to calculate the heat flux it will also be used here and is written as:

$$q'' = (VI)/((pi)d_iL_h) \tag{A.7}$$

where

V is the voltage drop across the tube.

I is the current carried by the tube.

d_i is the inside diameter of the tube.

L_h is the heated length of the tube.

Then the uncertainty in heat flux is calculated using the equation below:

$$U_{q''} = \left[\left(\frac{dV}{V}\right)^2 + \left(\frac{dI}{I}\right)^2 + \left(-\frac{dd_i}{d_i}\right)^2 + \left(-\frac{dL_h}{L_h}\right)^2\right]^{1/2} \qquad (A.8)$$

The uncertainty in each variable is accounted for in the equation and described below.

dV The voltage meter has a manufacturers advertised accuracy of 1% of reading. Our readings ranged from 9 to 31 volts giving an average error of 0.2 volts.

dI The ammeter was calibrated and had an error of less than 1% of full scale and was used from 150 to 600 amps giving an average error of 3.25 amps.

dd_i The inside diameter was measured accurately to 0.0005 inches using a micrometer and the inside diameter is 0.624 inches.

dL_h The heated length of the test section is 230.75 inches and was measured to within 0.0625 inches.

Then by using equations (A.?) and (A.?) below:

$$T_{wi} = T_{wo} - \left(\frac{\dot{q}}{2\ AK_s L_h}\right)\left[d_o^2 \ln\left(\frac{d_o}{d_i}\right) - \left(\frac{A}{2}\right)\right] \qquad (A.9)$$

$$T_b = T_{bo} - \frac{(T_{bo} - T_{bi})(L_h - x)}{L_h} \qquad (A.10)$$

the quantity $(T_{wi} - T_b)$ can be arranged into terms of measured quantities:

$$(T_{wi} - T_b) = T_{wo} - T_{bo} + T_1 + T_2 \qquad (A.11)$$

where:

$$T_1 = -\left(\frac{q}{2\,AK_w L_h}\right)\left[d_o^2 \ln\left(\frac{d_o}{d_i}\right) - \left(\frac{A}{2}\right)\right] \tag{A.12}$$

$$T_2 = \frac{(T_{bo} - T_{bi})(L_h - x)}{L_h} \tag{A.13}$$

and

$$A = (d_o^2 - d_i^2) \tag{A.14}$$

For this analysis the following uncertainties of each term are as follows:

dT_{wo} The assumed error in outside wall temperature was estimated to be 0.3 from the calibration runs.

dT_{bo} The bulk average temperature deviation was assumed to be 0.3 from the calibration of the Model 5100 data logger.

dT_2 The deviation ratio dT_2/T_2 was assumed to be 0.05.

dT_1 The deviation ratio dT_1/T_1 was assumed to be 0.05 also.

Therefore the uncertainty associated with the quantity $(T_{wi} - T_b)$ can be estimated from the equation:

$$U_t = \left[\left(\frac{|dT_{wo}| + |dT_{bo}| + |dT_2| + |dT_1|}{T_{wi} - T_b}\right)^2\right]^{1/2} \tag{A.15}$$

Applying typical values from run# 1008 at station 31 (x\d = 365.378):

q = 28677 Btu/hr | q'' = 9129 Btu/hr·ft^2

V = 21/55 volts | I = 390 amps

T_{bi} = 81.51 F | T_{bo} = 98.21 F

d_o = 0.748 inches | d_i = 0.624 inches

T_{wo} = 109.34 F | K_s = 7.686 Btu/hr·ft·F

Using the necessary wall temperature value in the equation below the K_s value is shown above.

$$K_s = [7.27 + 0.0038(T_{wo})] \tag{A.16}$$

Substitution of all values into the respective equations (respecting proper units) yields the following values.

$T_1 = -2.967$ F $\qquad$ $T_2 = 0.199$ F

$(T_{wi} - T_b) = 8.332$ F

These values result in the expected experimental uncertainties of:

$$U_T = \{ [(0.3 + 0.3 + 0.148 + 0.010)/8.332]^2\}^{1/2}$$
$$= 0.0909$$

$$U_{q''} = [(0.2/21.55)^2 + (7.5/390)^2 + (0.0005/0.625)^2 + \ldots$$
$$\ldots + (0.0625/230.75)^2]^{1/2}$$
$$= 0.0214$$

$$U_h = [(0.0214)^2 + (0.0909)^2]^{1/2} * 100$$
$$= 9.34\%$$

This experimental heat transfer uncertainty is within the range of acceptability and is believed to be a good representative value.

APPENDIX B

EXPERIMENTAL DATA AND RESULTS

In this appendix the briefest results from all 82 experimental runs are presented. Table IX gives all pertinent findings by descending inlet bulk Reynolds number (column 3). The runs are also identified by number (column 1). The results consist mostly of dimensionless groups data (Prandtl, Grashof, and Nusselt numbers) with subscripts denoting station number. Other information includes the percent water used in the particular run, the average heat flux, and the heat balance error (explained in Chapter III).

Since the HEAT program is the major tool used to calculate the results, a sample input file (including description) and examples of the output files are included in this appendix in addition to the program listing. The sample is for run# 1008 only and the entire data set and results are available from:

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TABLE IX

SUMMARY OF EXPERIMENTAL RESULTS

RUN	%WATER	Re_i	Re_{si}	Pr_i	Pr_{si}	Nu_i	Nu_{si}	Gr_i	Gr_{si}	Q/A*	HBAL
1005	100	41239	50529	5.61	4.47	226.5	217.5	112841	2252505	19898	1.86
1006	100	41338	48203	5.53	4.66	220.3	232.0	85994	132591	14592	0.65
1004	100	33768	43204	5.57	4.23	190.6	191.6	137196	292799	19904	1.44
1007	100	30907	35290	5.72	4.93	151.5	145.5	76857	122908	9593	1.99
1014	100	28667	33036	5.99	5.11	160.2	163.3	63011	98855	9515	0.09
1008	100	23669	28002	5.63	4.67	142.9	144.5	81194	136278	9129	0.10
1015	100	22116	26472	5.98	4.89	134.8	138.8	75415	131855	9567	1.76
1016	100	11076	18057	5.92	3.44	85.4	97.7	181837	697699	14223	0.78
1009	100	10243	13656	5.76	4.18	71.4	75.4	119154	276169	7123	0.12
2654	40	10202	12456	32.61	26.68	146.3	151.2	19231	28906	10733	3.31
1010	100	8982	12424	5.70	3.97	66.6	71.1	131712	334573	7168	0.94
1012	100	7346	10196	5.86	4.06	59.1	59.9	113844	311718	5960	0.50
2653	40	6829	7069	40.19	38.82	66.6	64.8	3588	3961	1437	8.99
2663	40	6483	10610	34.28	20.93	117.4	130.2	36698	93587	18268	1.99
1013	100	6263	9163	5.81	3.80	51.5	53.1	133958	417667	5974	0.33
2659	40	5918	7358	40.15	32.27	101.5	102.4	12862	20151	7660	3.24
2601	40	5858	7918	31.62	23.38	90.0	98.1	28230	48895	9120	1.04
2664	40	5688	9899	33.92	19.48	104.1	118.1	42426	120422	18358	2.66
2602	40	5570	7685	31.55	22.85	84.9	94.3	30548	54243	9263	0.59
2660	40	5533	6984	40.30	31.90	93.7	95.9	13833	22071	7648	2.49
2610	40	5370	7347	30.42	22.22	85.4	91.8	30625	55256	8680	2.49
2687	40	5031	7351	42.07	28.76	99.0	90.8	18837	45749	11950	2.32
2665	40	4818	9232	33.99	17.73	90.5	106.4	48819	48819	18454	0.87

TABLE IX CONTINUED

RUN	%WATER	Re_1	Re_{31}	Pr_1	Pr_{31}	Nu_1	Nu_{31}	Gr_1	Gr_{31}	Q/A	HBAL
1017	100	4802	7482	6.17	3.77	41.1	42.0	130775	508312	5618	1.34
2661	40	4762	6237	40.67	31.02	83.3	85.6	15289	26256	7648	1.50
2655	40	4352	5827	41.28	30.80	78.3	80.8	15785	28273	7664	1.55
2652	40	4277	4532	39.67	37.44	51.3	54.5	4975	5290	1469	6.66
2656	40	4237	5720	41.1	30.42	76.4	78.7	16332	29804	7664	1.55
1018	100	4170	6724	6.22	3.65	36.6	38.1	136155	575094	5313	1.04
2662	40	4106	5620	40.75	29.74	74.0	76.4	17172	32200	7672	0.53
2686	40	4053	5843	44.24	30.66	81.8	73.1	16181	39230	9428	1.10
2657	40	3993	5495	40.91	29.70	73.1	75.2	17252	32817	7667	1.34
2658	40	3794	5283	40.77	29.25	70.4	71.3	18050	35772	7699	3.04
1019	100	3639	5622	6.24	3.85	32.7	31.3	122102	499159	4290	3.89
2614	40	3507	5690	28.62	17.64	63.7	67.1	47479	125702	8767	0.71
2685	40	3396	4376	47.68	36.97	68.4	53.6	9875	21556	5586	0.58
2683	40	3083	4091	45.37	34.15	60.7	48.6	11808	26891	5463	-0.37
2691	40	3012	3604	53.88	44.99	58.5	42.8	5797	11599	3679	-1.40
2651	40	2985	3253	38.49	35.31	46.2	33.4	5675	9419	4506	2.15
2609	40	2927	5150	29.58	16.81	54.9	57.2	51551	164026	8904	0.78
2613	40	2872	3713	34.89	26.97	46.8	47.3	19849	33801	4029	2.19
2692	40	2683	3277	53.68	43.90	53.8	39.0	6355	13406	3684	-0.73
2684	40	2679	3702	44.94	32.48	53.4	43.5	13736	33424	5483	0.44
2688	40	2635	3658	47.81	34.39	58.0	46.2	11857	29818	5691	0.41
2680	40	2605	2859	48.78	44.43	43.7	26.8	4146	8234	1534	-0.43
2608	40	2571	4518	30.65	17.43	49.5	50.4	48528	157419	8123	3.24
2690	40	2509	3113	53.56	43.13	51.3	36.9	6698	14719	3684	-1.56

TABLE IX CONTINUED

RUN	%WATER	Re_i	Re_{si}	Pr_i	Pr_{si}	Nu_i	Nu_{si}	Gr_i	Gr_{si}	Q/A	HBAL
2682	40	2398	3050	47.21	37.09	47.8	18.7	9379	40683	3687	-0.03
2677	40	2397	3042	46.99	36.99	46.2	17.7	9804	43083	3706	1.72
2679	40	2386	2634	48.8	44.19	39.0	12.9	4361	16507	1498	-0.21
2681	40	2385	2784	48.24	41.29	41.7	15.7	6549	24458	2350	-0.40
2678	40	2379	2737	48.27	41.93	42.8	14.7	5931	23589	2146	0.91
2689	40	2368	2672	55.29	48.98	42.7	14.4	4020	15623	1963	1.23
2011	0	2364	3328	120.91	96.55	80.2	66.5	3132	6097	7116	5.38
2676	40	2360	3383	45.28	31.55	47.4	21.1	15278	75058	5581	2.95
2675	40	2335	3045	45.8	35.09	46.7	17.6	10728	51119	3901	-0.15
2010	0	2285	2878	125.54	100.56	77.3	56.1	2560	5693	6049	2.47
2612	40	2280	4070	30.22	16.92	43.6	43.8	50018	170625	7160	0.09
2674	40	2250	2620	47.25	40.55	41.5	14.2	6395	26292	2169	0.44
2673	40	2174	2430	47.84	42.78	39.4	13.2	4726	18153	1533	0.45
2603	40	2133	3362	35.14	22.28	40.3	21.0	30011	153757	5464	0.18
2671	40	2056	2389	46.43	39.95	39.4	13.8	6282	24900	1924	0.27
2009	0	2021	2544	128.99	103.34	65.1	27.5	2518	9806	5373	1.20
2607	40	2005	3489	32.56	18.70	39.7	31.5	40866	167282	6256	1.33
2670	40	1919	2351	45.52	37.14	39.6	15.0	8180	33740	2431	0.68
2672	40	1880	2139	47.57	41.78	35.7	13.3	5289	18919	4818	0.15
2611	40	1859	3254	32.35	18.47	38.7	28.6	39830	177907	5879	1.74
2008	0	1827	2349	129.81	101.91	65.1	21.3	2484	13242	5402	2.26
2669	40	1716	2065	46.11	38.29	36.3	13.9	7080	27737	1965	-0.81
2604	40	1684	2972	33.89	19.19	37.6	22.5	34760	197013	5454	0.20
2666	40	1639	2109	43.31	33.63	36.7	15.6	10295	42007	2522	0.55

TABLE IX CONTINUED

RUN	%WATER	Re_i	Re_{3i}	Pr_i	Pr_{3i}	Nu_i	Nu_{3i}	Gr_i	Gr_{3i}	Q/A	HBAL
2668	40	1576	1855	45.32	38.48	34.3	13.4	6394	23405	1585	0.58
2007	0	1483	2035	130.02	95.89	62.1	21.6	2782	15911	5764	5.14
2667	40	1398	1676	45.03	37.54	31.0	12.9	6925	24647	1559	0.17
2606	40	1377	2575	34.11	18.23	35.2	22.4	34245	205639	5033	0.61
2006	0	1331	1831	133.04	97.92	58.2	20.9	2642	14681	5338	6.01
2605	40	1180	2156	34.68	18.97	32.1	20.9	29744	165470	4162	0.66
2004	0	1150	1465	150.10	118.86	54.4	18.8	1564	7680	3755	5.99
2002	0	949	1148	153.77	127.88	48.2	16.1	1069	4839	2409	3.71
2005	0	730	930	155.66	123.38	41.6	16.1	1210	5226	2389	4.16
2003	0	559	721	157.80	123.48	38.3	15.3	1031	4422	1933	3.19
2001	0	281	421	149.90	101.45	27.1	15.0	1287	5377	1541	3.64

* Units in Btu/hr·ft²

Input Data For HEAT

The files output by the RED96 program are the essential input for the HEAT program. Table X illustrates the input in its raw form and lines one and two appear as follows:

```
1008 26
 1   0.00 3.8610 390.00  21.55  83.30  98.21  79.43
```

Where:

Run#	1008
Total number of thermocouples stations used	26
Fluid index (1=water, 2=ethylene glycol)	1
Mass concentration of ethylene glycol	0.00
Flow rate in (GPM)	3.8610
Current carried by test section (amps)	390.00
Voltage drop across tube (volts)	21.55
Corrected inlet bulk temperature (F)	83.30
Corrected exit bulk temperature (F)	98.21
Room (ambient) temperature (avg. of 2 TC's)	79.43

The origin of the above numbers is described in the Data Reduction section of Chapter II. All subsequent lines appear as below:

```
 1  4    2.00  94.61  95.12  95.09  94.90
 3  4    6.00  95.33  95.56  95.83  95.13
 4  4    8.00  95.50  95.66  95.67  95.24
 5  4   12.00  95.86  95.82  96.00  95.81
 6  "      "      "      "      "      "
22  4  120.00 101.80 102.44 102.35 101.74
23  2  132.00 102.74 102.97
25  2  156.00 104.35 104.24
27  2  180.00 105.93 105.46
29  2  204.00 107.60 107.20
31  2  228.00 109.07 109.61
```

and the first column designates the station number origin of the data. The missing numbers correspond to the stations not used during data collection (Table X). Column two indicates the number of thermocouple locations used at the designated station. Column three gives the length at each station and columns four through seven are the outside wall thermocouple readings for run# 1008. They correspond to the thermocouple locations as illustrated on Figure 2.3. When only two columns are present (stations 23-31) the thermocouple readings are for the top and bottom tube locations (also evident on Figure 2.3). The data of Table X is the only input necessary to execute the HEAT program.

TABLE X

SAMPLE INPUT DATA FOR HEAT PROGRAM

1008	26						
1	0.00	3.8610	390.00	21.55	83.30	98.21	79.43
1	4	2.00	94.61	95.12	95.09	94.90	
3	4	6.00	95.33	95.56	95.83	95.13	
4	4	8.00	95.50	95.66	95.67	95.24	
5	4	12.00	95.86	95.82	96.00	95.81	
6	4	16.00	96.44	96.16	96.14	96.18	
7	4	20.00	96.55	96.43	96.19	96.35	
8	4	24.00	96.93	96.69	96.50	96.53	
9	4	28.00	97.04	96.71	96.88	96.67	
10	4	32.00	97.03	97.20	96.77	96.98	
11	4	36.00	97.44	97.45	97.08	97.32	
12	4	40.00	97.71	97.62	97.17	97.31	
13	4	44.00	98.10	97.54	97.39	97.54	
14	4	48.00	98.04	97.89	97.80	97.71	
15	4	52.00	98.16	98.09	97.94	98.35	
16	4	56.00	98.41	98.42	98.23	97.66	
17	4	60.00	98.79	98.35	98.38	97.83	
18	4	72.00	99.19	99.07	99.24	98.92	
19	4	84.00	99.95	100.10	99.90	99.73	
20	4	96.00	100.47	100.52	100.77	100.19	
21	4	108.00	100.87	101.41	101.67	101.37	
22	4	120.00	101.80	102.44	102.35	101.74	
23	2	132.00	102.74	102.97			
25	2	156.00	104.35	104.24			
27	2	180.00	105.93	105.46			
29	2	204.00	107.60	107.20			
31	2	228.00	109.07	109.61			

TABLE XI

SAMPLE OUTPUT FILE FROM HEAT WITH EXTENSION .SUM*

Nu_{ave}	Re_{ave}	Pr_{ave}	Gr_{ave}	x/d	Mew_b/Mew_w	h_{ave}	T_b	T_w
142.924	23699.750	5.627	81194.160	3.205	1.107	969.941	83.43	92.16
138.574	23773.520	5.608	84633.040	9.615	1.111	940.746	83.69	92.69
139.702	23810.440	5.598	84379.980	12.821	1.110	948.568	83.82	92.74
138.191	23884.340	5.579	86193.820	19.231	1.111	938.632	84.07	93.10
136.673	23958.340	5.560	88056.980	25.641	1.112	928.643	84.33	93.46
138.277	24032.420	5.540	87923.910	32.051	1.110	939.869	84.59	93.61
137.883	24106.590	5.521	89079.630	38.462	1.110	937.512	84.85	93.89
139.324	24180.850	5.502	89051.930	44.872	1.109	947.643	85.11	94.05
140.678	24255.190	5.484	89085.760	51.282	1.107	957.180	85.36	94.22
139.568	24329.630	5.465	90707.170	57.692	1.108	949.951	85.62	94.55
141.563	24404.150	5.446	90322.450	64.103	1.106	963.861	85.88	94.68
142.636	24478.760	5.428	90538.730	70.513	1.105	971.501	86.14	94.87
143.275	24553.450	5.409	91034.160	76.923	1.105	976.187	86.39	95.09
142.976	24628.240	5.391	92134.880	83.333	1.105	974.484	86.65	95.36
146.515	24703.110	5.373	90788.140	89.744	1.102	998.948	86.91	95.41
148.234	24778.060	5.354	90615.760	96.154	1.100	1011.008	87.17	95.56
148.283	25003.460	5.301	93278.840	115.385	1.100	1012.376	87.94	96.33
147.500	25229.620	5.248	96532.860	134.615	1.100	1008.046	88.72	97.14
151.114	25456.560	5.195	96949.050	153.846	1.097	1033.790	89.49	97.71
149.814	25684.270	5.144	100609.200	173.077	1.097	1025.919	90.26	98.55
150.137	25912.750	5.093	103246.800	192.308	1.096	1029.159	91.04	99.31
150.097	26141.980	5.044	106178.500	211.538	1.095	1029.909	91.81	100.08
151.945	26602.730	4.946	110771.300	250.000	1.093	1044.642	93.36	101.52
154.612	27066.480	4.852	114840.200	288.462	1.090	1065.052	94.91	102.92
151.538	27533.210	4.760	123503.000	326.923	1.091	1045.897	96.45	104.62
144.532	28002.880	4.671	136378.900	365.385	1.095	999.448	98.00	106.56
.000	.000	.000	.000	.000	.000	.000	.00	.00

* Data file is created in a headingless format.

TABLE XII

SAMPLE OUTPUT FROM HEAT PROGRAM WITH EXTENSION .OUT

RUN NUMBER 1008
TEST FLUID IS DISTILLED WATER

VOLUMETRIC FLOW RATE	=	3.86	GPM
MASS FLOW RATE	=	1925.4	LBM/HR
MASS FLUX	=	906637	LBM/(SQ.FT-HR)
FLUID VELOCITY	=	4.03	FT/S
ROOM TEMPERATURE	=	79.43	F
INLET TEMPERATURE	=	83.30	F
OUTLET TEMPERATURE	=	98.21	F
AVERAGE RE NUMBER	=	25829	
AVERAGE PR NUMBER	=	5.11	
CURRENT TO TUBE	=	390.0	AMPS
VOLTAGE DROP IN TUBE	=	21.55	VOLTS
AVERAGE HEAT FLUX	=	9129	BTU/(SQ.FT-HR)
Q=AMP*VOLT	=	28677	BTU/HR
Q=M*C*(T2-T1)	=	28648	BTU/HR
HEAT BALANCE ERROR	=	.10	%

OUTSIDE SURFACE TEMPERATURES - DEGREES F

	1	3	4	5	6	7	8	9	10
1	94.61	95.33	95.50	95.86	96.44	96.55	96.93	97.04	97.03
2	95.12	95.56	95.66	95.82	96.16	96.43	96.69	96.71	97.20
3	95.09	95.83	95.67	96.00	96.14	96.19	96.50	96.88	96.77
4	94.90	95.13	95.24	95.81	96.18	96.35	96.53	96.67	96.98

	11	12	13	14	15	16	17	18	19
1	97.44	97.71	98.10	98.04	98.16	98.41	98.79	99.19	99.95
2	97.45	97.62	97.54	97.89	98.09	98.42	98.35	99.07	100.10
3	97.08	97.17	97.39	97.80	97.94	98.23	98.38	99.24	99.90
4	97.32	97.31	97.54	97.71	98.35	97.66	97.83	98.92	99.73

	20	21	22	23	25	27	29	31
1	100.47	100.87	101.80	102.74	104.35	105.93	107.60	109.07
2	100.52	101.41	102.44					
3	100.77	101.67	102.35	102.97	104.24	105.46	107.20	109.61
4	100.19	101.37	101.74					

TABLE XII CONTINUED

INSIDE SURFACE TEMPERATURES - DEGREES F

	1	3	4	5	6	7	8	9	10
1	91.83	92.56	92.73	93.09	93.67	93.78	94.16	94.27	94.26
2	92.35	92.79	92.89	93.05	93.38	93.66	93.92	93.93	94.43
3	92.32	93.06	92.90	93.23	93.37	93.41	93.72	94.11	93.99
4	92.13	92.35	92.46	93.04	93.41	93.58	93.75	93.89	94.21

	11	12	13	14	15	16	17	18	19
1	94.67	94.94	95.34	95.27	95.38	95.64	96.03	96.42	97.18
2	94.68	94.85	94.76	95.12	95.32	95.65	95.57	96.29	97.33
3	94.30	94.39	94.61	95.03	95.16	95.46	95.61	96.47	97.12
4	94.55	94.53	94.76	94.93	95.58	94.87	95.04	96.14	96.95

	20	21	22	23	25	27	29	31
1	97.70	98.09	99.02	99.96	101.57	103.15	104.82	106.29
2	97.74	98.64	99.67					
3	98.00	98.90	99.58	100.19	101.46	102.68	104.42	106.83
4	97.41	98.60	98.96					

REYNOLDS NUMBER AT THE INSIDE TUBE WALL

	1	3	4	5	6	7	8	9	10
1	26148	26363	26414	26521	26696	26728	26843	26876	26871
2	26302	26432	26462	26509	26610	26692	26769	26774	26923
3	26293	26515	26466	26564	26605	26619	26712	26827	26792
4	26236	26302	26335	26506	26616	26667	26720	26762	26856

	11	12	13	14	15	16	17	18	19
1	26994	27076	27195	27175	27210	27288	27404	27522	27752
2	26998	27049	27023	27129	27189	27289	27267	27484	27798
3	26884	26911	26978	27102	27143	27232	27278	27537	27736
4	26958	26954	27023	27074	27269	27057	27107	27438	27684

	20	21	22	23	25	27	29	31
1	27910	28028	28313	28602	29098	29587	30107	30566
2	27924	28196	28512					
3	28002	28276	28484	28673	29063	29440	29981	30738
4	27822	28184	28294					

TABLE XII CONTINUED

INSIDE SURGACE HEAT FLUXES BTU/HR/FT2

	1	3	4	5	6	7	8	9	10
1	8485	8470	8467	8469	8460	8466	8461	8460	8481
2	8453	8469	8465	8477	8480	8471	8477	8489	8464
3	8461	8446	8459	8463	8476	8484	8482	8469	8493
4	8464	8490	8486	8477	8478	8476	8485	8491	8475

	11	12	13	14	15	16	17	18	19
1	8477	8469	8455	8472	8488	8468	8453	8482	8494
2	8471	8472	8492	8485	8484	8480	8496	8498	8487
3	8495	8495	8490	8484	8499	8477	8473	8479	8497
4	8477	8488	8492	8493	8471	8517	8522	8505	8506

	20	21	22	23	25	27	29	31
1	8494	8530	8524	8517	8521	8524	8536	8561
2	8504	8498	8491					
3	8479	8490	8496	8511	8524	8538	8547	8546
4	8520	8500	8526					

PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/(SQ.FT-HR-F)

	1	3	4	5	6	7	8	9	10
1	1009	955	950	939	905	921	908	923	953
2	947	930	933	944	936	934	934	961	933
3	951	900	931	924	938	961	955	940	984
4	973	980	981	945	934	943	952	966	958

	11	12	13	14	15	16	17	18	19
1	937	934	919	954	972	969	954	1000	1004
2	935	944	984	972	979	970	1011	1017	985
3	978	998	1001	983	998	991	1003	994	1010
4	949	980	984	994	948	1069	1082	1037	1032

	20	21	22	23	25	27	29	31
1	1035	1090	1068	1044	1037	1033	1019	1033
2	1030	1015	983					
3	996	983	994	1015	1052	1098	1073	967
4	1076	1020	1076					

TABLE XII CONTINUED

RUN NUMBER 1008
SUMMARY

	X/D	TBULK	RE NUMBER	PR NUMBER	NU NUMBER	GR NUMBER	MUB/ MUW	HTOP/ HBOT	H AVG
1	3.21	83.43	23699	5.63	142.9	81194	1.107	1.06	969
3	9.62	83.69	23773	5.61	138.6	84633	1.111	1.06	940
4	12.82	83.82	23810	5.60	139.7	84379	1.110	1.02	948
5	19.23	84.07	23884	5.58	138.2	86193	1.111	1.02	938
6	25.64	84.33	23958	5.56	136.7	88056	1.112	.97	928
7	32.05	84.59	24032	5.54	138.3	87923	1.110	.96	939
8	38.46	84.85	24106	5.52	137.9	89079	1.110	.95	937
9	44.87	85.11	24180	5.50	139.3	89051	1.109	.98	947
10	51.28	85.36	24255	5.48	140.7	89085	1.107	.97	957
11	57.69	85.62	24329	5.46	139.6	90707	1.108	.96	949
12	64.10	85.88	24404	5.45	141.6	90322	1.106	.94	963
13	70.51	86.14	24478	5.43	142.6	90538	1.105	.92	971
14	76.92	86.39	24553	5.41	143.3	91034	1.105	.97	976
15	83.33	86.65	24628	5.39	143.0	92134	1.105	.97	974
16	89.74	86.91	24703	5.37	146.5	90788	1.102	.98	998
17	96.15	87.17	24778	5.35	148.2	90615	1.100	.95	1011
18	115.38	87.94	25003	5.30	148.3	93278	1.100	1.01	1012
19	134.62	88.72	25229	5.25	147.5	96532	1.100	.99	1008
20	153.85	89.49	25456	5.20	151.1	96949	1.097	1.04	1033
21	173.08	90.26	25684	5.14	149.8	100609	1.097	1.11	1025
22	192.31	91.04	25912	5.09	150.1	103246	1.096	1.07	1029
23	211.54	91.81	26141	5.04	150.1	106178	1.095	1.03	1029
25	250.00	93.36	26602	4.95	151.9	110771	1.093	.99	1044
27	288.46	94.91	27066	4.85	154.6	114840	1.090	.94	1065
29	326.92	96.45	27533	4.76	151.5	123502	1.091	.95	1045
31	365.38	98.00	28002	4.67	144.5	136378	1.095	1.07	999

NOTE: TBULK IS GIVEN IN DEGREES "F"
H(AVG) IS GIVEN IN BTU/(FT^2*HR*F)

```
C **********************************************************************
C *                             "HEAT"                                 *
C *                                                                    *
C *   A PROGRAM TO CALCULATE THE INSIDE WALL TEMPERATURES AND          *
C *  LOCAL HEAT TRANSFER COEFFICIENTS FOR GIVEN OUTSIDE WALL           *
C *  TEMPERATURES FOR SINGLE PHASE HEAT TRANSFER STUDIES IN            *
C *  HORIZONTAL TUBES.  THE PROGRAM ALSO CALCULATES THE PERTINENT      *
C *  FLUID FLOW & HEAT TRANSFER DIMENSIONLESS NUMBERS.                 *
C *                                                                    *
C *  THE MATHEMATICAL ALGORITHM OF THIS PROGRAM HAS BEEN DEVELOPED     *
C *  BY THE STUDENTS OF DR. J.D. PARKER & DR. K.J. BELL OF             *
C *  OKLAHOMA STATE UNIVERSITY.                                        *
C *                                                                    *
C *  THE PROGRAM WAS MODIFIED BY:                                      *
C *                                                                    *
C *               Y. H. ZURIGAT  (APRIL 1989)                          *
C *                                                                    *
C *  AND REMODIFIED FOR INTERACTIVE USE ON PC's BY:                    *
C *                                                                    *
C *               D. R. MAIELLO  (DECEMBER 1989)                       *
C *                                                                    *
C *  UNDER THE SUPERVISION OF: DR. A.J. GHAJAR                         *
C *                            SCHOOL OF MECHANICAL &                  *
C *                               AEROSPACE ENGINEERING                *
C *                            OKLAHOMA STATE UNIVERSITY               *
C *                            STILLWATER, OK 74078                    *
C *                                                                    *
C **********************************************************************
C
C **********************************************************************
C *                                                                    *
C *                       SUBROUTINE LISTING                           *
C *                                                                    *
C *     NAME         FUNCTION                                          *
C *    --------      ------------------------------------              *
C *    GEOM          Prompts for pipe dimensions and                   *
C *                  calculates geometry for finite                    *
C *                  differencing                                      *
C *                                                                    *
C *    BET           Calculates fluid Thermal Expansion Coefficient    *
C *                                                                    *
C *    CONDFL        Calculates fluid Thermal Conductivity             *
C *                                                                    *
C *    DENS          Calculates fluid Density                          *
C *                                                                    *
C *    MEW           Calculates fluid Viscosity                        *
C *                                                                    *
C *    PRNUM         Calculates fluid Prandtl Number                   *
C *                                                                    *
C *    SPHEAT        Calculates fluid Specific Heat                    *
C *                                                                    *
C *    PRNT          Prints calculated data to output files            *
C *                                                                    *
C **********************************************************************
C
```

```
C
C ********************************************************************
C *                                                                  *
C *                          MAIN PROGRAM                            *
C *                                                                  *
C ********************************************************************
C
C
      CHARACTER INFILE*36,OUTFILE*11,SUMFILE*11,SUMDOC*11,FNAME*4
      DIMENSION TCHCK1(8),TCHCK2(8),QAVG(31),PWP(31),
     +          PRNU(31),CONDK(31,8),RSVTY(31,8)
C
      COMMON /PRINT/ IPICK,REN(31,8),TBULK(31),VEL,REYNO,PRNO,GW,
     +               HTCOFF(31,8),H(31),RENO(31),GRNO(31),PR(31),
     +               SNUS(31),VISBW(31),SHTHB(32),QFLXID(31,8),QFLXAV,
     +               QGEXPT,QBALC,QPCT,IGO,IPMAX,TAVG(31)
     +       /INPUT/ TROOM,VOLTS,TAMPS,RMFL,MFLUID,X2,FLOWRT,NRUN,VFLOW,
     +               TIN,TOUT,TOSURF(31,8),TISURF(31,8),IP(32),KST(32)
     +       /TEMP1/ TWALL(31,8),AMPS(31,8),RESIS(31,8),POWERS(32),
     +               TPOWER
     +       /MAIN1/ IST,KOUNT,NSTN
     +       /GEOM1/ XAREA(31),R(31),LTP(32),LTH(32),DELZ(31),LHEAT,
     +               LTEST,LOD(31),DOUT,DIN,DELR,NODES,NSLICE,PI
C
      REAL*4 LTH,LTP,LTEST,LHEAT,H,HAV,HTCOFF,HEF,HMIX,LOD
C
C -------------------------
C ----- PRINT HEADING -----
C -------------------------
C
      WRITE(*,1009)
C
      READ(*,1003)FNAME
C
C -------------------------------------------------
C ----- INITIALIZE OUTPUT DATA ARRAYS TO ZERO -----
C -------------------------------------------------
C
    1 DO 101 I=1,8
         DO 101 J=1,31
           TOSURF(J,I)=0.
           TISURF(J,I)=0.
           REN(J,I)=0.
           QFLXID(J,I)=0.
  101      HTCOFF(J,I)=0.
C
      G=32.174
C
C -------------------------------------------------------------------
C ----- PRINT EXPLANATIONS TO SCREEN & PROMPT FOR OUTPUT OPTION -----
C -------------------------------------------------------------------
C
      WRITE(*,1001)
C
      READ(*,*)IGO
```

```
C
C -------------------------------------------
C ----- PROMPT FOR INPUT DATA FILE NAME -----
C -------------------------------------------
C
      DO 2 J=1,18
    2 WRITE(*,*)'  '
      WRITE(*,*)'   ENTER INPUT DATA FILE NAME:'
      WRITE(*,*)'    (COMPLETE PATH & EXTENSION; 36 CHARACTER MAX)'
      WRITE(*,*)'************************************'
      WRITE(*,*)'  '
      READ(*,1002)INFILE
      OPEN(5,FILE=INFILE,STATUS='OLD')
C
C ------------------------------------------------
C ----- READ RUN NUMBER FROM INPUT DATA FILE -----
C ------------------------------------------------
C
      READ(5,1003)FNAME
      REWIND 5
C
C ----------------------------------------------------------------
C ----- ASSIGN FILE NAMES TO VARIABLES AND OPEN OUTPUT FILES -----
C ----------------------------------------------------------------
C
      OUTFILE='RN'//FNAME//'.OUT'
      SUMFILE='RN'//FNAME//'.SUM'
C
      OPEN(6,FILE=OUTFILE,STATUS='NEW') OPEN(9,FILE=SUMFILE,STATUS='NEW')
C
C ----------------------------------
C ----- PROMPT FOR UNITS INPUT -----
C ----------------------------------
C
      WRITE(*,*)'  '
      WRITE(*,*)'  '
    3 WRITE(*,*)'   SELECT ENGLISH OR S.I. UNITS FOR OUTPUT FILE:'
      WRITE(*,*)'       Enter "1" for English'
      WRITE(*,*)'       Enter "2" for S.I.'
      WRITE(*,*)'  '
      READ(*,*)IPICK
      IF(IPICK.NE.1.AND.IPICK.NE.2)GO TO 3
C
C ----------------------------------------------------------
C ----- READ RUN NUMBER AND # STATIONS FROM INPUT FILE -----
C ----------------------------------------------------------
C
    4 READ(5,1004) NRUN,NSTN
C
C ---------------------------------
C ----- CHECK FOR END OF FILE -----
C ---------------------------------
C
      IF (NRUN .EQ. 0) GO TO 99
C
```

```
C ------------------------------------
C ----- READ DATA FROM INPUT FILE -----
C ------------------------------------
C
      X2=0.0
      IPMAX=0
      READ(5,1005)MFLUID,X2,FLOWRT,TAMPS,VOLTS,TIN,TOUT,TROOM
C
      IF(X2.LT.0.0.OR.X2.GT.1.0)THEN
        WRITE(*,*)' WARNING : MASS CONCENTRATION IS OUT OF RANGE'
        STOP
      END IF
C
      DO 5 IST=1,NSTN
       READ(5,1006)KST(IST),IP(IST),LTH(IST),
     +     (TOSURF(IST,IPR),IPR=1,IP(IST))
       IF(IST.NE.1)THEN
            IF(IP(IST).GE.IPMAX)IPMAX=IP(IST)
       ELSE
    5  ENDIF
C
      VFLOW=FLOWRT
C
C --------------------------------------------------
C -----CALCULATION OF MASS FLOW RATE IN LBM/HR -----
C --------------------------------------------------
C
      CALL DENS(TIN,MFLUID,X2,ROW)
      RMFL=VFLOW*0.133666*60.0*ROW
C
C -----          -----
        CALL GEOM
C -----          -----
C
      NNODE=NODES-1
C
C ----------------------------------------
C ----- START SOLUTION WITH STATION 1 -----
C ----------------------------------------
C
      DO 30 IST=1,NSTN
          IPP= IP(IST)
          DO 10 IPR=1,IPP
   10       TCHCK1(IPR)=0.0
C
C ----------------------------------------------
C -----  SET ALL RADIAL TEMPERATURES EQUAL  -----
C ----- TO THE OUTSIDE SURFACE TEMPERATURES -----
C ----------------------------------------------
C
          DO 11 ISL=1,NODES
             DO 11 IPR=1,IPP
   11           TWALL(ISL,IPR)=TOSURF(IST,IPR)
          KOUNT=1
C
```

```
C -----------------------------------------------------------
C ----- CALCULATE THERMAL CONDUCTIVITY OF STAINLESS STEEL -----
C -----      FOR EACH NODE IN BTU/(HR-FT-DEGF)           -----
C -----------------------------------------------------------
C
   12 DO 13 ISL=1,NODES
        DO 13 IPR=1,IPP
          CONDK(ISL,IPR)=7.27+0.0038*TWALL(ISL,IPR)
   13 CONTINUE
C
C -------------------------------------------------------------
C ----- CALCULATE ELECTRICAL RESISTIVITY OF STAINLESS STEEL -----
C -----        FOR EACH NODE IN OHMS-SQIN/IN               -----
C -------------------------------------------------------------
C
      DO 14 ISL=1,NODES
        IPP= IP(IST)
        DO 14 IPR=1,IPP
         RSVTY(ISL,IPR)=(27.67+0.0213*TWALL(ISL,IPR))/1.E6
   14 CONTINUE
C
C -----------------------------------------------------------------
C -----      CALCULATE RESISTANCE FOR EACH SEGMENT, ALSO      -----
C ----- CALCULATE EQUIVALENT RESISTANCE FOR PARALLEL CIRCUITS -----
C -----------------------------------------------------------------
C
      DELR = (DOUT-DIN)/2.0/NSLICE
      R(1) = DOUT/2.0
      DO 15  I=1,NSLICE
   15   R(I+1)=R(I)-DELR
      IPP= IP(IST)
      XAREA(1)=(R(1)-DELR/4.0)*PI*DELR/IPP
      XAREA(NODES)=(R(NODES)+DELR/4.0) *PI*DELR/IPP
      DO 16 I=2,NSLICE
   16   XAREA (I)= 2.0*R(I)*PI*DELR/IPP
C
      RINV = 0.0
      DO 17 ISL=1,NODES
        DO 17 IPR=1,IPP
          RESIS(ISL,IPR) = RSVTY(ISL,IPR)*DELZ(IST)/XAREA(ISL)
          RINV = RINV +1.0/RESIS(ISL,IPR)
   17 CONTINUE
C
C ----------------------------------------------
C ----- CALCULATE CURRENT FOR EACH SEGMENT -----
C ----------------------------------------------
C
      OHMS = 1.0/RINV
      AMP=0.0
      DO 18 ISL=1,NODES
        DO 18 IPR=1,IPP
          AMPS(ISL,IPR) = TAMPS*OHMS/RESIS(ISL,IPR)
          AMP=AMP+AMPS(ISL,IPR)
   18 CONTINUE
C
```

```
C ------------------------------------------------------------
C -----           CALCULATE TEMPERATURES AT NODE 2        -----
C ----- TEMPERATURES AT NODE 1 ARE OUTSIDE WALL TEMPERATURES -----
C ------------------------------------------------------------
C
      ISL=1
      DO 20 IPR=1,IPP
         ITHCTL=IPP
         IMINS=IPR-1
         IPLUS=IPR+1
         NMINS = ISL - 1
         NPLUS = ISL + 1
            IF(IMINS.EQ.0 .AND. IPP.EQ. ITHCTL) IMINS=ITHCTL
            IF(IPLUS.EQ.(ITHCTL+1) .AND.  IPP.EQ. ITHCTL) IPLUS=1
         A= 3.41214*12.0*AMPS(ISL,IPR)*AMPS(ISL,IPR)
     +          *RSVTY(ISL,IPR)/XAREA(ISL)
         B = IPP*DELR*(CONDK(ISL,IPR)+CONDK(ISL,IPLUS))
     +         *(TWALL(ISL,IPR)-TWALL(ISL,IPLUS))/(8.0*PI*R(ISL))
         C = IPP*DELR*(CONDK(ISL,IPR)+CONDK(ISL,IMINS))
     +         *(TWALL(ISL,IPR)-TWALL(ISL,IMINS))/(8.0*PI*R(ISL))
         X = PI*(R(ISL)-DELR/2.0)*(CONDK(ISL,IPR)+CONDK(NPLUS,IPR))
     +         /(IPP*DELR)
   20    TWALL(NPLUS,IPR) = TWALL(ISL,IPR)-(A-B-C)/X
C
C ------------------------------------------------
C ----- CALCULATE REMAINING NODAL TEMPERATURES -----
C ------------------------------------------------
C
      DO 21 ISL=2,NNODE
         DO 21 IPR=1,IPP
            ITHCTL=IPP
            IMINS=IPR-1
            IPLUS=IPR+1
            NMINS=ISL-1
            NPLUS=ISL+1
            IF(IMINS.EQ.0 .AND. IPP .EQ. ITHCTL) IMINS=ITHCTL
            IF(IPLUS.EQ.(ITHCTL+1) .AND. IPP .EQ. ITHCTL) IPLUS=1
            A= 3.41214*12.0*AMPS(ISL,IPR)*AMPS(ISL,IPR)
     +          *RSVTY(ISL,IPR)/XAREA(ISL)
            B =PI*(R(ISL)+DELR/2.)*(CONDK(ISL,IPR)+CONDK(NMINS,IPR))
     +           *(TWALL(ISL,IPR)-TWALL(NMINS,IPR))/(IPP*DELR)
            C = IPP*DELR*(CONDK(ISL,IPR)+CONDK(ISL,IPLUS))
     +            *(TWALL(ISL,IPR)-TWALL(ISL,IPLUS))/(4.0*PI*R(ISL))
            D = IPP*DELR*(CONDK(ISL,IPR)+CONDK(ISL,IMINS))
     +            *(TWALL(ISL,IPR)-TWALL(ISL,IMINS))/(4.0*PI*R(ISL))
            X =PI*(R(ISL)-DELR/2.)*(CONDK(ISL,IPR)+CONDK(NPLUS,IPR))
     +            /(IPP*DELR)
   21    TWALL(NPLUS,IPR) = TWALL(ISL,IPR)- (A-B-C-D)/X
C
C ------------------------------------------------------------
C ----- CHECK FOR THE CONVERGENCE OF THE WALL TEMPERATURES -----
C ------------------------------------------------------------
C
      TCHCK = 0.0
      DO 22 IPR=1,IPP
```

```
          TCHCK2(IPR)=TWALL(NODES,IPR)
   22     TCHCK = TCHCK + ABS(TCHCK2(IPR)-TCHCK1(IPR))
        IF (TCHCK .GT. 0.001) GO TO 23
        GO TO 26
   23   DO 24 IPR=1,IPP
   24     TCHCK1(IPR) = TCHCK2(IPR)
        IF (KOUNT .GT. 20) GO TO 25
        KOUNT = KOUNT+1
        GO TO 12
        WRITE(6,1007) IST,KOUNT
   25   WRITE(*,1007) IST,KOUNT
   26   DO 27 IPR=1,IPP
   27     TISURF( IST ,IPR)=TWALL(NODES,IPR)
C
C -------------------------------------------------------------
C ----- CALCULATE POWER GENERATED IN EACH SEGMENT IN BTU/HOUR -----
C -------------------------------------------------------------
C
        POWER =0.0
        DO 28 ISL=1,NODES
          DO 28 IPR=1,IPP
          POWER=POWER+AMPS(ISL,IPR)*AMPS(ISL,IPR)*RESIS(ISL,IPR)
   28   CONTINUE
C
        POWERS(IST)=POWER*3.41214
C
C ----------------------------------------------
C ----- CALCULATE HEAT FLUX AT INSIDE SURFACE -----
C ----------------------------------------------
C
      ISL=NODES
      IPP= IP(IST)
        ITHCTL=IPP
      DO 29 IPR=1,IPP
        IPLUS=IPR+1
        IMINS=IPR-1
        IF(IMINS.EQ.0 .AND. IPP .EQ. ITHCTL) IMINS=ITHCTL
        IF(IPLUS.EQ.(ITHCTL+1).AND. IPP.EQ. ITHCTL) IPLUS=1
        Q1 = PI*(CONDK(ISL-1,IPR)+CONDK(ISL,IPR))*(R(ISL-1)-DELR/2.0)*
     +       (TWALL(ISL,IPR)-TWALL(ISL-1,IPR))/(IPP*DELR)
        Q2 = IPP*(CONDK(ISL,IPLUS)+CONDK(ISL,IPR))*DELR
     +       *(TWALL(ISL,IPR)-TWALL(ISL,IPLUS))/(PI*R(ISL)*8.0)
        Q4 = IPP*(CONDK(ISL,IPR)+CONDK(ISL,IMINS))*DELR
     +       *(TWALL(ISL,IPR)-TWALL(ISL,IMINS))/(PI*R(ISL)*8.0)
        QGEN=3.41214*12.0*AMPS(ISL,IPR)*AMPS(ISL,IPR)
     +       *RSVTY(ISL,IPR)/XAREA(ISL)
   29   QFLXID(IST,IPR) =(QGEN-Q1-Q2-Q4)*IPP*12.0/(2.0*PI*R(ISL))
C
   30 CONTINUE
C
C ----------------------------------------------------------
C ----- CALCULATE REYNOLDS NUMBERS AT INSIDE TUBE SURFACE -----
C ----------------------------------------------------------
C
      DO 40 IST=1,NSTN
```

```
      IPP= IP(IST)
      DO 40 IPR=1,IPP
         TR=TISURF(IST,IPR)
         CALL MEW(TR,MFLUID,X2,VISS)
         REN(IST,IPR)=RMFL*48.0/(PI*DIN*VISS)
   40 CONTINUE
C
C -------------------------------------------------------
C ----- CALCULATE TOTAL POWER GENERATED IN BTU/HOUR -----
C -------------------------------------------------------
C
      TPOWER=0.0
      DO 45 IST=1,NSTN
   45 TPOWER=TPOWER+POWERS(IST)
C
C -----------------------------------------------------------------
C ----- CALCULATE BULK FLUID TEMPERATURE AT EACH STATION,DEG.F -----
C -----------------------------------------------------------------
C
      TBULK(1)=TIN+(TOUT-TIN)*LTP(1)/LTEST
      DO 50 IST =2,NSTN
   50    TBULK(IST) = TBULK(IST-1) + (TOUT-TIN)*LTP(IST)/LTEST
C
C --------------------------------------------------------------------
C ----- CALCULATION OF INPUT AND OUTPUT HEAT TRANSFER RATE,BTU/HR -----
C -----    AND OVERALL AVERAGE REYNOLDS AND PRANDTL NUMBERS       -----
C --------------------------------------------------------------------
C
      QGCALC=TPOWER
      QGEXPT =TAMPS*VOLTS*3.41214
      QIN=QGEXPT
      QFLXAV=QIN/(3.1416*DIN/12.0*(LHEAT/12.0))
C
C ----- CALCULATE FLUID PROPERTIES AT TAVE -----
C
      T=(TOUT+TIN)/2.0
      CALL SPHEAT(T,MFLUID,X2,SPHT)
      CALL MEW(T,MFLUID,X2,VISC)
      CALL CONDFL(T,MFLUID,X2,COND)
C
      QBALC=RMFL*SPHT*(TOUT-TIN)
      QPCT=(QIN-QBALC)*100.0/QIN
      AID=PI*DIN*DIN/4.0/144.0
      GW=RMFL/AID
      REYNO=GW*DIN/12.0/VISC
      PRNO=VISC*SPHT/COND
C
C --------------------------------------------------------------
C ----- CALCULATION OF PERIPHERAL HEAT TRANSFER COEFFICIENT -----
C -----    FROM EXPERIMENTAL DATA,BTU/(HR-SQ.FT-DEG.F)      -----
C --------------------------------------------------------------
C
      DO 55 IST=1,NSTN
         IPP= IP(IST)
         DO 55 IPR=1,IPP
```

```
         HTCOFF(IST,IPR) =QFLXID(IST,IPR)/(TISURF(IST,IPR)-TBULK(IST)) 55
      CONTINUE
C
C ------------------------------------------------------------------
C ----- CALCULATE RATIO OF TOP/BOTTOM HEAT TRANSFER COEFFICIENTS -----
C ------------------------------------------------------------------
C
      DO 65 IST=1,NSTN
         IPP= IP(IST)
         IF (IPP.EQ. 4) GO TO 60
            SHTHB(IST)=HTCOFF(IST,1)/HTCOFF(IST,2)
            GO TO 65
   60    SHTHB(IST)=HTCOFF(IST,1)/HTCOFF(IST,3)
   65 CONTINUE
C
C ------------------------------------------------------------
C ----- CALCULATION OF OVERALL HEAT TRANSFER COEFFICIENT -----
C ------------------------------------------------------------
C
      DO 75 IST=1,NSTN
         QQ=0.0
         TT=0.0
         IPP= IP(IST)
         DO 70 J=1,IPP
            TT=TT+TISURF(IST,J)
            QQ=QQ+QFLXID(IST,J)
   70    CONTINUE
         TAVG(IST)=TT/IPP
         QAVG(IST)=QQ/IPP
         H(IST)=QAVG(IST)/(TAVG(IST)-TBULK(IST))
   75 CONTINUE
C
C --------------------------------------
C ----- CALCULATE FLUID PROPERTIES -----
C --------------------------------------
C
      DO 85 IST=1,NSTN
         T=TBULK(IST)
         CALL MEW(T,MFLUID,X2,VISC)
         CALL SPHEAT(T,MFLUID,X2,SPHT)
         CALL CONDFL(T,MFLUID,X2,COND)
         CALL DENS(T,MFLUID,X2,ROW)
         CALL BET(T,MFLUID,X2,BETA)
         PR(IST) = VISC*SPHT/COND
         RENO(IST) = GW*DIN/12.0/VISC
         GRNO(IST)=G*BETA*ROW**2*(DIN/12)**3*(TAVG(IST)-TBULK(IST))
     +         /VISC**2 *3600.0**2
         TIS=0.0
         IPP= IP(IST)
         DO 80 IPR=1,IPP
   80       TIS=TIS+TISURF(IST,IPR)
         T=TIS/IPP
         CALL MEW(T,MFLUID,X2,VISWL)
         VISBW(IST) = VISC/VISWL
         SNUS(IST)=H(IST)*DIN/(12.0*COND)
```

```
        TWALL(IST,1)=TAVG(IST)
   85 CONTINUE
C
C ---------------------------------------------
C ----- CALCULATE FLUID VELOCITY IN FT/SEC -----
C ---------------------------------------------
C
        VEL = VFLOW/(2.462557*DIN*DIN)
C
C -------------------------
C ----- PRODUCE OUTPUT -----
C -------------------------
C
      CALL PRNT
C
C -------------------------------------------------------------
C ----- PROMPT USER FOR PROGRAM TERMINATION OR CONTINUATION -----
C -------------------------------------------------------------
C
      WRITE(*,1008)NRUN
      READ(*,*)KEEP
      IF(KEEP.EQ.1)GO TO 1
      GO TO 4
C
   99 STOP
C
 1001 FORMAT(///////,6X,'HEAT will automatically create output files'
     +       ' with',/,6X,'the following destinations: ',//,6X,
     +       '"RN(run #).OUT"     - Formatted Output Data File ',/,6X,
     +       '"RN(run #).SUM"     - Output Plot/Reduction File ',///,
     +    6X,'The Output Data File may be produced with a format',/,
     +    6X,'specifically created for Dr. Ghajar's research project',/,
     +    6X,'using 26 stations with four T.C.'s.',/,
     +    6X,'You may select a more general format that will accept',/,
     +    6X,'up to 31 stations with up to eight T.C.'s per station.',//
     +   ,6X,'Enter "1" to select the restricted format (for Ghajar)',/,
     +    6x,'Enter "2" to select the generalized format',///)
 1002 FORMAT(A36)
 1003 FORMAT(A4)
 1004 FORMAT(I4,I3)
 1005 FORMAT(I2,F7.2,F7.4,5F7.2)
 1006 FORMAT(I3,I3,F9.2,8F7.2)
 1007 FORMAT(//5X,'TEMPERATURES AT STATION',I3,' DO NOT CONVERGE AFTER',
     +       I3,' ITERATIONS. JUMP TO NEXT STATION')
 1008 FORMAT(//////////////,6X,'DATA REDUCTION COMPLETED FOR RUN # ',I4,
     +     //6X,'To reduce a data set from another file, ENTER "1"',/,
     +       6x,'To continue program with current file, ENTER "2"',
     +       ///////)
 1009 FORMAT(////////,
     + 7X,'**********************************************************',
     +/7X,'*                         "HEAT"                         *',
     +/7X,'*                                                        *',
     +/7X,'*  A PROGRAM TO CALCULATE THE INSIDE WALL TEMPERATURES &  *',
     +/7X,'*  LOCAL HEAT TRANSFER COEFF.'s FOR GIVEN OUTSIDE WALL    *',
     +/7X,'*  TEMPERATURES FOR SINGLE PHASE HEAT TRANSFER STUDIES IN *',
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     +/7X,'*  HORIZONTAL TUBES.  THE PROGRAM ALSO CALCULATES THE     *',
     +/7X,'*  FLUID FLOW & HEAT TRANSFER DIMENSIONLESS NUMBERS.      *',
     +/7X,'*                                                         *',
     +/7X,'*  THE MATHEMATICAL ALGORITHM OF THIS PROGRAM HAS BEEN    *',
     +/7X,'*  DEVELOPED BY THE STUDENTS OF DR. J.D. PARKER &         *',
     +/7X,'*  DR. K.J. BELL OF OKLAHOMA STATE UNIVERSITY.            *',
     +/7X,'*                                                         *',
     +/7X,'*  THE PROGRAM WAS MODIFIED BY: Y.H. ZURIGAT (APR 1989)   *',
     +/7X,'*                               D.R. MAIELLO (DEC 1989)   *',
     +/7X,'*                                                         *',
     +/7X,'*  UNDER THE SUPERVISION OF:    DR. A.J. GHAJAR           *',
     +/7X,'*                               COLLEGE OF MECHANICAL     *',
     +/7X,'*                                & AREOSPACE ENGINEERING  *',
     +/7X,'*                               OKLAHOMA STATE UNIVERSITY *',
     +/7X,'*                                                         *',
     +/7X,'***********************************************************',
     + //,3X,'Press RETURN to Continue.')
C
      END
C
C
C *********************************************************************
C *                                                                   *
C *                       SUBROUTINE GEOM                             *
C *                                                                   *
C *                     ALL LENGTH IN INCHES                          *
C *                                                                   *
C *********************************************************************
C
      SUBROUTINE GEOM
C
      COMMON /MAIN1/ IST,KOUNT,NSTN
     +       /GEOM1/ XAREA(31),R(31),LTP(32),LTH(32),DELZ(31),LHEAT,
     +               LTEST,LOD(31),DOUT,DIN,DELR,NODES,NSLICE,PI
C
      REAL*4 LTH,LTP,LTEST,LHEAT,LOD
C
      NSLICE=10
      NODES= NSLICE + 1
C
C -------------------------------
C ----- PROMPT FOR PIPE SIZE -----
C -------------------------------
C
    1 WRITE(*,*)'   '
      WRITE(*,*)'   '
      WRITE(*,*)'   '
      WRITE(*,*)'   '
      WRITE(*,*)'    SELECT DESIRED PIPE SIZE OPTION:'
      WRITE(*,*)'   '
      WRITE(*,*)'       (1)  USE "HEAT TRANSFER" DEFAULT VALUES'
      WRITE(*,*)'       (2)  USE "PRESSURE DROP" DEFAULT VALUES'
      WRITE(*,*)'       (3)  VIEW DEFAULT SETTINGS'
      WRITE(*,*)'       (4)  INPUT YOUR OWN PIPE SIZE VALUES'
      READ(*,*)IPSO
```

```
C
      IF(IPSO.EQ.1)THEN
         DOUT=.748
         DIN=.624
         LHEAT=230.75
       ELSE
         IF(IPSO.EQ.2)THEN
              DOUT=.749
              DIN=.621
              LHEAT=230.75
           ELSE
              IF(IPSO.EQ.3)THEN
C
      WRITE(*,*)'  '
      WRITE(*,*)'  '
      WRITE(*,*)'      DEFAULT VALUES ARE (in inches):'
      WRITE(*,*)'  '
      WRITE(*,*)'        PIPE         O.D.    I.D.   LENGTH '
      WRITE(*,*)'    -------------  ------  ------  ------'
      WRITE(*,*)'    HEAT TRANSFER   0.748   0.624  230.75 '
      WRITE(*,*)'    PRESSURE DROP   0.749   0.621  230.75 '
      WRITE(*,*)' '
C
              GO TO 1
               ELSE
                  IF(IPSO.EQ.4)THEN
C
      WRITE(*,*)'  '
      WRITE(*,*)'  '
      WRITE(*,*)'   ENTER PIPE OUTSIDE DIAMETER (in inches):'
      READ(*,*)DOUT
      WRITE(*,*)'  '
      WRITE(*,*)'   ENTER PIPE INSIDE DIAMETER (in inches):' READ(*,*)DIN
      WRITE(*,*)'  '
      WRITE(*,*)'   ENTER PIPE LENGTH (in inches)'
      WRITE(*,*)'  (0.5 inches will be assumed & added for end plates):'
      READ(*,*)LHEAT
C
                    ELSE
                       GO TO 1
                   ENDIF
                ENDIF
            ENDIF
       ENDIF
C
C
      IF(DOUT.GT.DIN)GO TO 2
      WRITE(*,*)'  '
      WRITE(*,*)'  '
      WRITE(*,*)'  ************* !!!!!! WARNING !!!!!! ***************'
      WRITE(*,*)'   YOU HAVE ENTERED PIPE DIAMETERS WHICH RESULT IN'
      WRITE(*,*)'  A WALL THICKNESS OF ZERO OR LESS. PLEASE TRY AGAIN!'
      WRITE(*,*)'  ***************************************************'
      GO TO 1
C
```

```
C ----------------------------------------------------
C ----- CALCULATE GEOMETRY FOR FINITE DIFFERENCING -----
C ----------------------------------------------------
C
    2 PI = 3.141593
      LTEST = LHEAT+0.5
C
      DO 3 I=1,NSTN
    3    LOD(I)=LTH(I)/DIN
      LTH(NSTN+1)=LHEAT
      LTP(1)=LTH(1)
         SUM=LTP(1)
       DO 4 I=2,NSTN
         LTP(I) =  LTH(I)-LTH(I-1)
    4    SUM=SUM+LTP(I)
         LTP(NSTN+1)=LHEAT-SUM
      DELZ(1) = LTH(1)+( LTH(2)-LTH(1))/2.0
      DO 5 I=2,NSTN
    5    DELZ(I) = ( LTH(I+1)-LTH(I-1))/2.0
      RETURN
      END
C
C
C ********************************************************************
C *                                                                  *
C *                       SUBROUTINE BET                             *
C *                                                                  *
C *   CALCULATES THE THERMAL EXPANSION COEFFICIENT (BETA) FOR PURE   *
C *  WATER AND ANY CONCENTRATION OF ETHYLENE GLYCOL/WATER SOLUTION.  *
C *   THE INPUT IS TEMPERATURE IN DEGREES F AND THE OUTPUT IS 1/F.   *
C *                                                                  * C
******************************************************************** C
      SUBROUTINE BET(TF,MFLUID,X,BETA)
      T = (TF-32.0)/1.8
C
C ----- PURE WATER -----
C
      IF(MFLUID.GT.1)GO TO 1 PDRT=0.0615-0.01693*T+2.06E-4*T**2-1.77E-
       6*T**3+6.3E-9*T**4
        GO TO 2
C
C ----- ETHYLENE GLYCOL -----
C
    1  PDRTA = -1.2379*1.E-4 - 9.9189*1.E-4*X +4.1024*1.E-4*X*X
       PDRTB = 2.*((-2.9837E-06*T+2.4614E-06*X*T -9.5278E-8*X*X*T))
       PDRT=(PDRTA+PDRTB)*1000.
    2 CALL DENS(TF,MFLUID,X,ROW)
      ROW=ROW/.062427
      BETAC= -(1.0/ROW)*(PDRT)
      BETAF =(1.0/BETAC)*1.8
      BETA  = 1.0/BETAF
C
      RETURN
      END
C
```

```
C
C ********************************************************************
C *                                                                  *
C *                      SUBROUTINE CONDFL                           *
C *                                                                  *
C *     CALCULATES THE THERMAL CONDUCTIVITY (COND) FOR PURE WATER    *
C *      AND ANY CONCENTRATION OF ETHYLENE GLYCOL/WATER SOLUTION.    *
C *              THE INPUT IS TEMPERATURE IN DEGREES F               *
C *               AND THE OUTPUT IS IN BTU/HR-FT-'F                  *
C *                                                                  *
C *          TEMPERATURE RANGE:                                      *
C *               PURE WATER           0 - 100 C                     *
C *               E.G. MIXTURES        0 - 150 C                     *
C *                                                                  * C
******************************************************************** C
       SUBROUTINE CONDFL(TF,MFLUID,X,COND)
C
      T=(TF-32.0)/1.8
      CONW=0.56276+1.874E-3*T-6.8E-6*T**2
C
      IF(MFLUID.GT.1) GO TO 1
C
C ----- PURE WATER -----
C
      IF(T.LT.0.0.OR.T.GT.100.0)THEN
       WRITE(*,*)' TEMPERATURE IS OUT OF RANGE IN SUBROUTINE CONDFL' STOP
      END IF
C
       COND=CONW*0.5778
        GO TO 2
C
C ----- ETHYLENE GLYCOL -----
C
    1 IF(T.LT.0.0.OR.T.GT.150.0)THEN
       WRITE(*,*)' TEMPERATURE IS OUT OF RANGE IN SUBROUTINE CONDFL' STOP
      END IF
C
       CETH=0.24511+0.0001755*T-8.52E-7*T*T CF=0.6635-0.3698*X-0.000885*T
       COND=(1.0-X)*CONW+X*CETH-CF*(CONW-CETH)*(1.0-X)*X COND=COND*0.5778
    2 RETURN
      END
C
C
C ********************************************************************
C *                                                                  *
C *                      SUBROUTINE DENS                             *
C *                                                                  *
C *        CALCULATES THE FLUID DENSITY (ROW) FOR PURE WATER         *
C *      AND ANY CONCENTRATION OF ETHYLENE GLYCOL/WATER SOLUTION.    * C
* THE INPUT IS TEMPERATURE IN DEGREES F AND THE OUTPUT IS LB/FT**3. *
C *                                                                  *
C *          TEMPERATURE RANGE:                                      *
C *               PURE WATER           0 - 100 C                     *
C *               E.G. MIXTURES        0 - 150 C                     *
C *                                                                  * C
```

```
************************************************************************ C
      SUBROUTINE DENS(TF,MFLUID,X,ROW)
      DIMENSION D(3,3),AD(3,3)
C
      T=(TF-32.0)/1.8
      IF(MFLUID.GT.1) GO TO 1
C
C ----- PURE WATER -----
C
      IF(T.LT.0..OR.T.GT.100.0)THEN
        WRITE(*,*)' TEMPERATURE IS OUT OF RANGE IN SUBROUTINE DENS'
        STOP
      END IF
C
C
      ROWSI=999.86+.061464*T-.0084648*T**2+6.8794E-5*T**3-4.4214E-7
     +      *T**4+1.2505E-9*T**5
      ROW=ROWSI*0.062427
      GO TO 4
C
C ----- ETHYLENE GLYCOL -----
C
    1 IF(T.LT.0.0.OR.T.GT.150.0)THEN
        WRITE(*,*)' TEMPERATURE IS OUT OF RANGE IN SUBROUTINE DENS'
        STOP
      END IF
C
      AD(1,1)=1.0004
      AD(1,2)=0.17659
      AD(1,3)=-0.049214
      AD(2,1)=-1.2379E-04
      AD(2,2)=-9.9189E-04
      AD(2,3)= 4.1024E-04
      AD(3,1)=-2.9837E-06
      AD(3,2)= 2.4614E-06
      AD(3,3)= -9.5278E-08
C
      DO 2 I=1,3
       DO 2 J=1,3
    2    D(I,J)=AD(I,J)*X**(J-1)*T**(I-1)
      SUM=0.0
      DO 3 I=1,3
       DO 3 J=1,3
    3    SUM=SUM+D(I,J)
      SUM=SUM*1.E6/1000.0
      ROW=SUM*0.062427
C
    4 RETURN
      END
C
C
C **********************************************************************
C *                                                                    *
C *                      SUBROUTINE MEW                                *
C *                                                                    *
```

```
C *       CALCULATES THE DYNAMIC VISCOSITY (VISC) FOR PURE WATER       *
C *      AND ANY CONCENTRATION OF ETHYLENE GLYCOL/WATER SOLUTION.      *
C * THE INPUT IS TEMPERATURE IN DEGREES F AND THE OUTPUT IS LB/HR.FT.  *
C *                                                                    *
C *          TEMPERATURE RANGE:                                        *
C *               PURE WATER          10 - 100 C                       *
C *               E.G. MIXTURES        0 - 150 C                       *
C *                                                                    *
C **********************************************************************
C
      SUBROUTINE MEW(TF,MFLUID,X,VISC)
      DIMENSION V(3,3),AV(3,3),V2(3)
C
      T=(TF-32.0)/1.8
      IF(MFLUID.GT.1) GO TO 1
C
C ----- PURE WATER -----
C
      IF(T.LT.10..OR.T.GT.100.0)THEN
       WRITE(*,*)' TEMPERATURE IS OUT OF RANGE IN SUBROUTINE MEW'
       STOP
      END IF
C
      VISC=2.4189*1.0019*10.0**((1.3272*(20.0-T)-0.001053*(20-T)
     +     **2)/(T+105.0))
        GO TO 4
C
C ----- ETHYLENE GLYCOL -----
C
    1 IF(T.LT.0..OR.T.GT.150.0)THEN
       WRITE(*,*)' TEMPERATURE IS OUT OF RANGE IN SUBROUTINE MEW'
       STOP
      END IF
C
      AV(1,1)=0.55164
      AV(1,2)=2.6492
      AV(1,3)=0.82935
      AV(2,1)=-0.027633
      AV(2,2)=-0.031496
      AV(2,3)= 0.0048136
      AV(3,1)= 6.0629E-17
      AV(3,2)= 2.2389E-15
      AV(3,3)= 5.879E-16
C
      DO 2 I=1,2
        DO 2 J=1,3
          V(I,J)=AV(I,J)*X**(J-1)*T**(I-1)
    2     V2(J)=AV(3,J)*X**(J-1)
C
          SUM=0.0
      DO 3 I=1,3
    3   SUM=SUM+V2(I)
       V3=SUM**0.25*T*T
       VISC=V3 + V(1,1)+V(1,2)+V(1,3)+V(2,1)+V(2,2)+V(2,3)
      VISC=EXP(VISC)*2.4189
```

```
C
    4 RETURN
      END
C
C
C  ******************************************************************
C  *                                                                *
C  *                    SUBROUTINE PRNUM                            *
C  *                                                                *
C  *        CALCULATES THE PRANDTL NO.(PRN) FOR PURE WATER          *
C  *    AND ANY CONCENTRATION OF ETHYLENE GLYCOL/WATER SOLUTION.    *
C  *           THE INPUT IS TEMPERATURE IN DEGREES F.               *
C  *                                                                *
C  *           TEMPERATURE RANGE:                                   *
C  *                PURE WATER         10 - 100 C                   *
C  *                E.G. MIXTURES       0 - 150 C                   *
C  *                                                                *
C  ******************************************************************
C
      SUBROUTINE PRNUM(TF,MFLUID,X,PRN)
      DIMENSION P(3,3),AP(3,3),P2(3)
C
      T=(TF-32.0)/1.8
      IF(MFLUID.GT.1) GO TO 1
C
C ----- PURE WATER -----
C
      IF(T.LT.10.OR.T.GT.100.0)THEN
       WRITE(*,*)' TEMPERATURE IS OUT OF RANGE IN SUBROUTINE PRNUM'
       STOP
      END IF
C
        CALL SPHEAT(TF,MFLUID,X,SPHT)
        CALL NEW(TF,MFLUID,X,VISC)
        CALL CONDFL(TF,MFLUID,X,COND)
        PRN=SPHT*VISC/COND
        RETURN
C
C ----- ETHYLENE GLYCOL -----
C
    1 IF(T.LT.0.0.OR.T.GT.150.0)THEN
       WRITE(*,*)' TEMPERATURE IS OUT OF RANGE IN SUBROUTINE PRNUM'
       STOP
      END IF
C
      AP(1,1)=2.5735
      AP(1,2)=3.0411
      AP(1,3)=0.60237
      AP(2,1)=-0.031169
      AP(2,2)=-0.025424
      AP(2,3)= 0.0037454
      AP(3,1)= 1.1605E-16
      AP(3,2)= 2.5283E-15
      AP(3,3)= 2.3777E-16
C
```

```
      DO 2 I=1,2
        DO 2 J=1,3
          P(I,J)=AP(I,J)*X**(J-1)*T**(I-1)
    2     P2(J)=AP(3,J)*X**(J-1)
C
         SUM=0.0
C
         DO 3 I=1,3
    3     SUM=SUM+P2(I)
       P3=SUM**0.25*T*T
       PRN=P3+P(1,1)+P(1,2)+P(1,3)+P(2,1)+P(2,2)+P(2,3)
      PRN=EXP(PRN)
C
      RETURN
      END
C
C
C *******************************************************************
C *                                                                 *
C *                    SUBROUTINE SPHEAT                            *
C *                                                                 *
C *       CALCULATES THE SPECIFIC HEAT (SPHT) FOR PURE WATER        *
C *     AND ANY CONCENTRATION OF ETHYLENE GLYCOL/WATER SOLUTION.    *
C *            THE INPUT IS TEMPERATURE IN DEGREES F                *
C *              AND THE OUTPUT IS IN BTU/(LBM-DEGF).               *
C *                                                                 *
C *          TEMPERATURE RANGE:                                      *
C *               PURE WATER          0 - 100 C                      *
C *               E.G. MIXTURES       0 - 150 C                      *
C *                                                                  * C
*******************************************************************
C
      SUBROUTINE SPHEAT(TF,MFLUID,X,SPHT)
C
      T=(TF-32.0)/1.8
      IF(MFLUID .GT. 1.0)GO TO 1
C
C ----- PURE WATER -----
C
      IF(T.LT.0.0.OR.T.GT.100.0)THEN
       WRITE(*,*)' TEMPERATURE IS OUT OF RANGE IN SUBROUTINE SPHT'
       STOP
      END IF
C
      SPHT=-1.475E-7*T**3+3.66E-5*T*T-.0022*T+4.216
      SPHT=SPHT/4.1868
      RETURN
C
C ----- ETHYLENE GLYCOL -----
C
    1 IF(T.LT.0.0.OR.T.GT.150.0)THEN
       WRITE(*,*)' TEMPERATURE IS OUT OF RANGE IN SUBROUTINE SPHT'
       STOP
      END IF
C
```

```
       CALL MEW(TF,MFLUID,X,VISC)
       CALL CONDFL(TF,MFLUID,X,COND)
       CALL PRNUM(TF,MFLUID,X,PRN)
       SPHT = PRN*COND/VISC
      RETURN
      END
C
C
C  *********************************************************************
C  *                                                                   *
C  *                    SUBROUTINE PRINT-OUT                           *
C  *                                                                   *
C  *          PRINTS DATA TO OUTPUT FILES:                             *
C  *                                                                   *
C  *                  "RN(run #).OUT"     - Device #6                  *
C  *                  "RN(run #).SUM"     - Device #9                  *
C  *                                                                   *
C  *********************************************************************
C
      SUBROUTINE PRNT
C
      DIMENSION SITOC(31,8),SITB(31),HAVG(31),PWP(31),
     +          SIH(31),SIHP(31,8),SIQIN(31,8),PRNU(31),SITIS(31,8)
C
      INTEGER IREN(31,8),IDFLX(31,8),IHCOF(31,8)
      COMMON /PRINT/ IPICK,REN(31,8),TBULK(31),VEL,REYNO,PRNO,GW,
     +               HTCOFF(31,8),H(31),RENO(31),GRNO(31),PR(31),
     +               SNUS(31),VISBW(31),SHTHB(32),QFLXID(31,8),QFLXAV,
     +               QGEXPT,QBALC,QPCT,IGO,IPMAX,TAVG(31)
     +        /INPUT/ TROOM,VOLTS,TAMPS,RMFL,MFLUID,X2,FLOWRT,NRUN,VFLOW,
     +               TIN,TOUT,TOSURF(31,8),TISURF(31,8),IP(32),KST(32)
     +        /TEMP1/ TWALL(31,8),AMPS(31,8),RESIS(31,8),POWERS(32),
     +               TPOWER
     +        /MAIN1/ IST,KOUNT,NSTN
     +        /GEOM1/ XAREA(31),R(31),LTP(32),LTH(32),DELZ(31),LHEAT,
     +               LTEST,LOD(31),DOUT,DIN,DELR,NODES,NSLICE,PI
C
       REAL*4 LTH,LTP,LTEST,LHEAT,H,HAV,HTCOFF,HEF,HMIX,LOD
C
C ----------------------------------------------
C ----- SET FLAG FOR STATION OUTPUT CONTROL -----
C ----------------------------------------------
C
       ATST=NSTN/9.
       IFST=INT(ATST)+1
C
C ---------------------------------------
C ----- PRINT RUN NUMBER & TUBE DATA -----
C ---------------------------------------
C
      IF(IPICK.EQ.2)GO TO 1
C
C ----- ENGLISH UNITS -----
C
      WRITE(6,2001)NRUN
```

```
C
C ----- PRINT FLUID-TYPE DESCRIPTION -----
C
      IF(MFLUID.EQ.1)THEN
         WRITE(6,2003)
            ELSE
            WRITE(6,2004)X2
      ENDIF
C
C ----- PRINT TUBE DATA -----
C
      IGW=GW
      IREYN=REYNO
      IFXA=QFLXAV
      IQEX=QGEXPT
      IQBL=QBALC
C
      WRITE(6,2016)VFLOW,RMFL,IGW,VEL,TROOM,TIN,TOUT,IREYN,PRNO,
     +             TAMPS,VOLTS,IFXA,IQEX,IQBL,QPCT
C
      GO TO 2
C
C ----- S.I. UNITS -----
C
    1 SRMFL=RMFL*.126
      FLOWRT=63.09*FLOWRT
      SITR=(TROOM-32.0)/1.8
      SITIN=(TIN-32.0)/1.8
      SITOUT=(TOUT-32.0)/1.8
      SITR=(TROOM-32.0)/1.8
      SIGW=GW/737.33806
      SIQAV=QFLXAV*3.154591
      SIQG=QGEXPT*0.2930711
      SIQBAL=QBALC*0.2930711
      SIVEL=VEL*.3048
      WRITE(6,2001)NRUN
C
C ----- PRINT FLUID-TYPE DESCRIPTION -----
C
      IF(MFLUID.EQ.1)THEN
         WRITE(6,2003)
            ELSE
            WRITE(6,2004)X2
      ENDIF
C
C ----- PRINT TUBE DATA -----
C
      IGW=SIGW
      IREYN=REYNO
      IFXA=SIQAV
      IQEX=SIQG
      IQBL=SIQBAL
C
      WRITE(6,2018)FLOWRT,SRMFL,IGW,SIVEL,SITR,SITIN,SITOUT,IREYN,PRNO,
     +             TAMPS,VOLTS,IFXA,IQEX,IQBL,QPCT
```

```
C
C ------------------------------------------------
C ----- PRINT TUBE OUTSIDE SURFACE TEMPERATURES -----
C ------------------------------------------------
C
    2 IF(IPICK.EQ.2)GO TO 8
C
C ----- ENGLISH UNITS -----
C
      DO 5 K=1,NSTN
      IF(IP(K).EQ.2)THEN
         TOSURF(K,3)=TOSURF(K,2)
         TOSURF(K,2)=0.0
         TOSURF(K,4)=0.0
        ELSE
    5 ENDIF
C
      WRITE(6,2005)
      DO 7 ICNT=1,IFST
       KMIN=1+(ICNT-1)*9
       KMAX=KMIN+8
       IF(NSTN.LT.KMAX)KMAX=NSTN
        DO 6 IPR=1,IPMAX
         IF(IPR.EQ.1)WRITE(6,2006)(KST(K),K=KMIN,KMAX)
         IF(IPR.EQ.1 .AND. KMAX.LT.(KMIN+8))WRITE(6,2007)
C
         IF((IPR.EQ.2.OR.IPR.EQ.4).AND.ICNT.EQ.IFST.AND.IGO.EQ.1)THEN
            WRITE(6,2008)IPR,(TOSURF(IST,IPR),IST=KMIN,KMAX-5)
               ELSE
               WRITE(6,2008)IPR,(TOSURF(IST,IPR),IST=KMIN,KMAX)
         ENDIF
C
    6    CONTINUE
    7  CONTINUE
C
      IF(IGO.EQ.1)WRITE(6,2002)
      GO TO 13
C
C ----- S.I. UNITS -----
C
    8 DO 9 IST=1,NSTN
         IPP= IP(IST)
         DO 9 IPR=1,IPP
           SITOC(IST,IPR) = (TOSURF(IST,IPR)-32.0)/1.8
    9 CONTINUE
C
      DO 10 K=1,NSTN
      IF(IP(K).EQ.2)THEN
         SITOC(K,3)=SITOC(K,2)
         SITOC(K,2)=0.0
         SITOC(K,4)=0.0
        ELSE
   10 ENDIF
C
      WRITE(6,2009)
```

```
      DO 12 ICNT=1,IFST
       KMIN=1+(ICNT-1)*9
       KMAX=KMIN+8
       IF(NSTN.LT.KMAX)KMAX=NSTN
        DO 11 IPR=1,IPMAX
         IF(IPR.EQ.1)WRITE(6,2006)(KST(K),K=KMIN,KMAX)
         IF(IPR.EQ.1 .AND. KMAX.LT.(KMIN+8))WRITE(6,2007)
C
         IF((IPR.EQ.2.OR.IPR.EQ.4).AND.ICNT.EQ.IFST.AND.IGO.EQ.1)THEN
            WRITE(6,2008)IPR,(SITOC(IST,IPR),IST=KMIN,KMAX-5)
               ELSE
               WRITE(6,2008)IPR,(SITOC(IST,IPR),IST=KMIN,KMAX)
         ENDIF
C
   11  CONTINUE
   12  CONTINUE
      IF(IGO.EQ.1)WRITE(6,2002)
C
C ------------------------------------------------------------
C ----- PRINT INSIDE SURFACE TEMPERATURES TO OUTPUT FILE -----
C ------------------------------------------------------------
   13 WRITE(6,2017)NRUN
      IF(IPICK.EQ.2)GO TO 17
C
C ----- ENGLISH UNITS -----
C
      DO 14 K=1,NSTN
      IF(IP(K).EQ.2)THEN
         TISURF(K,3)=TISURF(K,2)
         TISURF(K,2)=0.0
         TISURF(K,4)=0.0
        ELSE
   14 ENDIF
C
      WRITE(6,2010)
      DO 16 ICNT=1,IFST
       KMIN=1+(ICNT-1)*9
       KMAX=KMIN+8
       IF(NSTN.LT.KMAX)KMAX=NSTN
        DO 15 IPR=1,IPMAX
         IF(IPR.EQ.1)WRITE(6,2006)(KST(K),K=KMIN,KMAX)
         IF(IPR.EQ.1 .AND. KMAX.LT.(KMIN+8))WRITE(6,2007)
C
         IF((IPR.EQ.2.OR.IPR.EQ.4).AND.ICNT.EQ.IFST.AND.IGO.EQ.1)THEN
            WRITE(6,2008)IPR,(TISURF(IST,IPR),IST=KMIN,KMAX-5)
               ELSE
               WRITE(6,2008)IPR,(TISURF(IST,IPR),IST=KMIN,KMAX)
   15    ENDIF
C
   16  CONTINUE
C
      GO TO 22
C
C ----- S.I. UNITS -----
C
```

```
   17 DO 18 IST=1,NSTN
         IPP= IP(IST)
         DO 18 IPR=1,IPP
            SITIS(IST,IPR)=(TISURF(IST,IPR)-32.0)/1.8
   18 CONTINUE
C
      DO 19 K=1,NSTN
      IF(IP(K).EQ.2)THEN
         SITIS(K,3)=SITIS(K,2)
         SITIS(K,2)=0.0
         SITIS(K,4)=0.0
        ELSE
   19 ENDIF
C
      WRITE(6,2011)
      DO 21 ICNT=1,IFST
       KMIN=1+(ICNT-1)*9
       KMAX=KMIN+8
       IF(NSTN.LT.KMAX)KMAX=NSTN
        DO 20 IPR=1,IPMAX
         IF(IPR.EQ.1)WRITE(6,2006)(KST(K),K=KMIN,KMAX)
         IF(IPR.EQ.1 .AND. KMAX.LT.(KMIN+8))WRITE(6,2007)
C
         IF((IPR.EQ.2.OR.IPR.EQ.4).AND.ICNT.EQ.IFST.AND.IGO.EQ.1)THEN
            WRITE(6,2008)IPR,(SITIS(IST,IPR),IST=KMIN,KMAX-5)
               ELSE
               WRITE(6,2008)IPR,(SITIS(IST,IPR),IST=KMIN,KMAX)
   20    ENDIF
C
   21  CONTINUE
C
C -----------------------------------------------
C ----- PRINT REYNOLDS NUMBERS TO OUTPUT FILE -----
C -----------------------------------------------
   22 DO 29 K=1,NSTN
      IF(IP(K).EQ.2)THEN
         IREN(K,1)=INT(REN(K,1))
         IREN(K,3)=INT(REN(K,2))
         IREN(K,2)=0
         IREN(K,4)=0
        ELSE
         DO 28 L=1,IPMAX
   28    IREN(K,L)=INT(REN(K,L))
   29 ENDIF
C
      WRITE(6,2014)
      DO 31 ICNT=1,IFST
       KMIN=1+(ICNT-1)*9
       KMAX=KMIN+8
       IF(NSTN.LT.KMAX)KMAX=NSTN
        DO 30 IPR=1,IPMAX
         IF(IPR.EQ.1)WRITE(6,2006)(KST(K),K=KMIN,KMAX)
         IF(IPR.EQ.1 .AND. KMAX.LT.(KMIN+8))WRITE(6,2007)
C
         IF((IPR.EQ.2.OR.IPR.EQ.4).AND.ICNT.EQ.IFST.AND.IGO.EQ.1)THEN
```

```
          WRITE(6,2015)IPR,(IREN(IST,IPR),IST=KMIN,KMAX-5)
             ELSE
             WRITE(6,2015)IPR,(IREN(IST,IPR),IST=KMIN,KMAX)
   30     ENDIF
C
   31  CONTINUE
      IF(IGO.EQ.1)WRITE(6,2012)
C
C ------------------------------------------------
C ----- PRINT INSIDE HEAT FLUXES TO OUTPUT FILE -----
C ------------------------------------------------
C
   33 WRITE(6,2017)NRUN
      IF(IPICK.EQ.2)GO TO 38
C
C ----- ENGLISH UNITS -----
C
      DO 35 K=1,NSTN
      IF(IP(K).EQ.2)THEN
         IDFLX(K,1)=INT(QFLXID(K,1))
         IDFLX(K,3)=INT(QFLXID(K,2))
         IDFLX(K,2)=0
         IDFLX(K,4)=0
        ELSE
         DO 34 L=1,IPMAX
   34    IDFLX(K,L)=INT(QFLXID(K,L))
   35 ENDIF
C
      WRITE(6,2020)
      DO 37 ICNT=1,IFST
       KMIN=1+(ICNT-1)*9
       KMAX=KMIN+8
       IF(NSTN.LT.KMAX)KMAX=NSTN
        DO 36 IPR=1,IPMAX
         IF(IPR.EQ.1)WRITE(6,2006)(KST(K),K=KMIN,KMAX)
         IF(IPR.EQ.1 .AND. KMAX.LT.(KMIN+8))WRITE(6,2007)
C
         IF((IPR.EQ.2.OR.IPR.EQ.4).AND.ICNT.EQ.IFST.AND.IGO.EQ.1)THEN
          WRITE(6,2021)IPR,(IDFLX(IST,IPR),IST=KMIN,KMAX-5)
             ELSE
             WRITE(6,2021)IPR,(IDFLX(IST,IPR),IST=KMIN,KMAX)
   36    ENDIF
C
   37  CONTINUE
C
      GO TO 44
C
C ----- S.I. UNITS -----
C
   38 DO 39 IST=1,NSTN
         IPP= IP(IST)
         DO 39 IPR=1,IPP
          SIQIN(IST,IPR)=QFLXID(IST,IPR)*3.15491
   39 CONTINUE
C
```

```
      DO 41 K=1,NSTN
      IF(IP(K).EQ.2)THEN
         IDFLX(K,1)=INT(SIQIN(K,1))
         IDFLX(K,3)=INT(SIQIN(K,2))
         IDFLX(K,2)=0
         IDFLX(K,4)=0
        ELSE
         DO 40 L=1,IPMAX
   40    IDFLX(K,L)=INT(SIQIN(K,L))
   41 ENDIF
C
      WRITE(6,2022)
      DO 43 ICNT=1,IFST
       KMIN=1+(ICNT-1)*9
       KMAX=KMIN+8
       IF(NSTN.LT.KMAX)KMAX=NSTN
        DO 42 IPR=1,IPMAX
         IF(IPR.EQ.1)WRITE(6,2006)(KST(K),K=KMIN,KMAX)
         IF(IPR.EQ.1 .AND. KMAX.LT.(KMIN+8))WRITE(6,2007)
C
         IF((IPR.EQ.2.OR.IPR.EQ.4).AND.ICNT.EQ.IFST.AND.IGO.EQ.1)THEN
            WRITE(6,2021)IPR,(IDFLX(IST,IPR),IST=KMIN,KMAX-5)
               ELSE
               WRITE(6,2021)IPR,(IDFLX(IST,IPR),IST=KMIN,KMAX)
   42    ENDIF
C
   43  CONTINUE
C
C -----------------------------------------------------
C ---- PRINT PERIPHERAL HEAT TRANSFER COEFFICIENTS -----
C -----------------------------------------------------
C
   44 IF(IPICK.EQ.2)GO TO 49
C
C ----- ENGLISH UNITS -----
C
      DO 46 K=1,NSTN
      IF(IP(K).EQ.2)THEN
         IHCOF(K,1)=INT(HTCOFF(K,1))
         IHCOF(K,3)=INT(HTCOFF(K,2))
         IHCOF(K,2)=0
         IHCOF(K,4)=0
        ELSE
         DO 45 L=1,IPMAX
   45    IHCOF(K,L)=INT(HTCOFF(K,L))
   46 ENDIF
C
      WRITE(6,2023)
      DO 48 ICNT=1,IFST
       KMIN=1+(ICNT-1)*9
       KMAX=KMIN+8
       IF(NSTN.LT.KMAX)KMAX=NSTN
        DO 47 IPR=1,IPMAX
         IF(IPR.EQ.1)WRITE(6,2006)(KST(K),K=KMIN,KMAX)
         IF(IPR.EQ.1 .AND. KMAX.LT.(KMIN+8))WRITE(6,2007)
```

```
C
         IF((IPR.EQ.2.OR.IPR.EQ.4).AND.ICNT.EQ.IFST.AND.IGO.EQ.1)THEN
            WRITE(6,2021)IPR,(IHCOF(IST,IPR),IST=KMIN,KMAX-5)
               ELSE
               WRITE(6,2021)IPR,(IHCOF(IST,IPR),IST=KMIN,KMAX)
   47    ENDIF
C
   48  CONTINUE
C
C
      GO TO 55
C
C ----- S.I. UNITS -----
C
   49 DO 50 IST=1,NSTN
         IPP= IP(IST)
         DO 50 IPR=1,IPP
            SIHP(IST,IPR)=HTCOFF(IST,IPR)*5.678263
   50 CONTINUE
C
      DO 52 K=1,NSTN
      IF(IP(K).EQ.2)THEN
         IHCOF(K,1)=INT(SIHP(K,1))
         IHCOF(K,3)=INT(SIHP(K,2))
         IHCOF(K,2)=0
         IHCOF(K,4)=0
        ELSE
         DO 51 L=1,IPMAX
   51    IHCOF(K,L)=INT(SIHP(K,L))
   52 ENDIF
C
      WRITE(6,2024)
      DO 54 ICNT=1,IFST
       KMIN=1+(ICNT-1)*9
       KMAX=KMIN+8
       IF(NSTN.LT.KMAX)KMAX=NSTN
        DO 53 IPR=1,IPMAX
         IF(IPR.EQ.1)WRITE(6,2006)(KST(K),K=KMIN,KMAX)
         IF(IPR.EQ.1 .AND. KMAX.LT.(KMIN+8))WRITE(6,2007)
C
         IF((IPR.EQ.2.OR.IPR.EQ.4).AND.ICNT.EQ.IFST.AND.IGO.EQ.1)THEN
            WRITE(6,2021)IPR,(IHCOF(IST,IPR),IST=KMIN,KMAX-5)
               ELSE
               WRITE(6,2021)IPR,(IHCOF(IST,IPR),IST=KMIN,KMAX)
   53    ENDIF
C
   54  CONTINUE
C
C ------------------------------------------------
C ----- PRINT SUMMATION DATA FOR OUTPUT FILE -----
C ------------------------------------------------
C
   55 IF(IGO.EQ.1)WRITE(6,2012)
      WRITE(6,2028)NRUN
C
```

```
      WRITE(6,2029)
      DO 56 J=1,NSTN
      IF(IPICK.EQ.2)H(J)=H(J)*5.678263 IF(IPICK.EQ.2)TBULK(J)=(TBULK(J)-
      32.0)/1.8 IF(IPICK.EQ.2)TAVG(J)=(TAVG(J)-32.0)/1.8
      LRENO=INT(RENO(J))
      IGRNO=INT(GRNO(J))
      IHAV=INT(H(J))
      WRITE(6,2030)KST(J),LOD(J),TBULK(J),LRENO,PR(J),SNUS(J),IGRNO,
     +          VISBW(J),SHTHB(J),IHAV
   56 CONTINUE
C
C ----- PRINT NOTE GIVING UNITS -----
C
      IF(IPICK.EQ.1)WRITE(6,2032)
      IF(IPICK.EQ.2)WRITE(6,2034)
C
C ---------------------------------------------------
C -----      PRINT OUTPUT DATA FOR USE WITH:    -----
C ----- "MARQ" - A CURVE-FITTING ROUTINE        -----
C ----- "SIGMAPLOT" OR SIMULAR GRAPHICS PACKAGES -----
C ---------------------------------------------------
C
      DO 57 I=1,NSTN
       WRITE(9,2033)SNUS(I),RENO(I),PR(I),GRNO(I),LOD(I),VISBW(I),H(I),
     +           TBULK(I),TAVG(I)
   57      CONTINUE
C
C ----- PRINT END OF FILE FLAG TO FILE FOR USE WITH "MARQ" -----
C
            EOFF=0.00
            WRITE(9,2033)EOFF,EOFF,EOFF,EOFF,EOFF,EOFF,EOFF,EOFF,EOFF C
      RETURN
C
 2001 FORMAT(//,18X,'*',41('-'),'*',/32X,'RUN NUMBER ',I4)
 2002 FORMAT(/////////////)
 2003 FORMAT(25X,'TEST FLUID IS DISTILLED WATER',/18X,'*',41('-'),'*')
 2004 FORMAT(19X,'MASS FRACTION OF ETHYLENE GLYCOL =',F8.4,/18X,'*',
     +      41('-'),'*')
 2005 FORMAT(////20X,'OUTSIDE SURFACE TEMPERATURES - DEGREES F')
 2006 FORMAT(//,8X,I2,6X,I2,6X,I2,6X,I2,6X,I2,6X,I2,6X,I2,6X,I2,6X,I2,/)
 2007 FORMAT(' ')
 2008 FORMAT(3X,I1,1X,9F8.2)
 2009 FORMAT(////20X,'OUTSIDE SURFACE TEMPERATURES - DEGREES C')
 2010 FORMAT(///20X,'INSIDE SURFACE TEMPERATURES - DEGREES F')
 2011 FORMAT(///20X,'INSIDE SURFACE TEMPERATURES - DEGREES C')
 2012 FORMAT(////)
 2013 FORMAT(4X,9F8.2)
 2014 FORMAT(///20X,'REYNOLDS NUMBER AT THE INSIDE TUBE WALL')
 2015 FORMAT(3X,I1,9I8)
 2016 FORMAT(//,18X,'VOLUMETRIC FLOW RATE =',F9.2,3X,'GPM',
     +      /18X,'MASS FLOW RATE',7X,'=',F9.1,3X,'LBM/HR',
     +      /18X,'MASS FLUX',12X,'=',I9,3X,'LBM/(SQ.FT-HR)',
     +      /18X,'FLUID VELOCITY',7X,'=',F9.2,3X,'FT/S',
     +      /18X,'ROOM TEMPERATURE',5X,'=',F9.2,3X,'F',
     +      /18X,'INLET TEMPERATURE',4X,'=',F9.2,3X,'F',
```

```
     +       /18X,'OUTLET TEMPERATURE',3X,'=',F9.2,3X,'F',
     +       /18X,'AVERAGE RE NUMBER',4X,'=',I9,
     +       /18X,'AVERAGE PR NUMBER',4X,'=',F9.2,
     +       /18X,'CURRENT TO TUBE',6X,'=',F9.1,3X,'AMPS',
     +       /18X,'VOLTAGE DROP IN TUBE =',F9.2,3X,'VOLTS',
     +       /18X,'AVERAGE HEAT FLUX',4X,'=',I9,3X,'BTU/(SQ.FT-HR)'
     +       /18X,'Q=AMP*VOLT',11X,'=',I9,3X,'BTU/HR',
     +       /18X,'Q=M*C*(T2-T1)',8X,'=',I9,3X,'BTU/HR',
     +       /18X,'HEAT BALANCE ERROR',3X,'=',F9.2,3X,'%')
 2017 FORMAT(//31X,'*',15('-'),'*',/32X,'RUN NUMBER ',I4,/31X,'*',
     +         15('-'),'*')
 2018 FORMAT(//,18X,'VOLUMETRIC FLOW RATE =',F9.2,3X,'CC/SEC',
     +       /18X,'MASS FLOW RATE',7X,'=',F9.1,3X,'KG/SEC',
     +       /18X,'MASS FLUX',12X,'=',I9,3X,'KG/(SQ.M-SEC)',
     +       /18X,'FLUID VELOCITY',7X,'=',F9.2,3X,'M/S',
     +       /18X,'ROOM TEMPERATURE',5X,'=',F9.2,3X,'C',
     +       /18X,'INLET TEMPERATURE',4X,'=',F9.2,3X,'C',
     +       /18X,'OUTLET TEMPERATURE',3X,'=',F9.2,3X,'C',
     +       /18X,'AVERAGE RE NUMBER',4X,'=',I9,
     +       /18X,'AVERAGE PR NUMBER',4X,'=',F9.2,
     +       /18X,'CURRENT TO TUBE',6X,'=',F9.1,3X,'AMPS',
     +       /18X,'VOLTAGE DROP IN TUBE =',F9.2,3X,'VOLTS',
     +       /18X,'AVERAGE HEAT FLUX',4X,'=',I9,3X,'W/(SQ. M)'
     +       /18X,'Q=AMP*VOLT',11X,'=',I9,3X,'W',
     +       /18X,'Q=M*C*(T2-T1)',8X,'=',I9,3X,'W',
     +       /18X,'HEAT BALANCE ERROR',3X,'=',F9.2,3X,'%')
 2020 FORMAT(///22X,'INSIDE SURFACE HEAT FLUXES BTU/HR/FT2')
 2021 FORMAT(3X,I1,9I8)
 2022 FORMAT(///22X,'INSIDE SURFACE HEAT FLUXES W PER SQ.M.')
 2023 FORMAT(///14X,'PERIPHERAL HEAT TRANSFER COEFFICIENT BTU/',
     +       '(SQ.FT-HR-F)')
 2024 FORMAT(///16X,'PERIPHERAL HEAT TRANSFER COEFFICIENT W/(SQ.M. K)')
 2027 FORMAT(///18X,'AVERAGE HEAT TRANSFER COEFFICIENT-W/(SQ.M. K)') 2028
 FORMAT(//31X,'*',15('-'),'*',/32X,'RUN NUMBER ',I4,/36X,
     +       'SUMMARY',/31X,'*',15('-'),'*',)
 2029 FORMAT(///8X,'X/D',4X,'TBULK',4X,'RE',6X,'PR',
     +       6X,'NU',6X,'GR',5X,'MUB/',3X,'HTOP/',4X,'H',/,22X,
     +       'NUMBER',2X,'NUMBER',2X,'NUMBER',2X,'NUMBER',3X,'MUW',4X,
     +       'HBOT',4X,'AVG',/)
 2030 FORMAT(2X,I2,2F8.2,I8,F8.2,F8.1,I8,F7.3,F7.2,I8)
 2032 FORMAT(//,13X,'NOTE: TBULK IS GIVEN IN DEGREES "F"',/,
     +           19X,'H(AVG) IS GIVEN IN BTU/(FT^2*HR*F)')
 2033 FORMAT(3F10.3,F15.3,3F10.3,2F8.2)
 2034 FORMAT(//,13X,'NOTE: TBULK IS GIVEN IN DEGREES "C"',/,
     +           19X,'H(AVG) IS GIVEN IN W/(M^2*K)')
C
      END
```

APPENDIX C

COMPUTER PROGRAMS

CALHT
GETAVE
CORR.DAT
INTERP
DELTA.DAT
GPMCALC
ETH
REDUCE30
RED96

```
{$I C:\TURBO\UTIL2 } (CALHT)
{ THIS PROGRAM READS INPUT FROM A DATA FILE PRODUCED BY ACQCOMP, AVERAGES }
{ OVER THE NUMBER OF SAMPLES PROVIDED IN THE FILE FOR EACH CHANNEL, USES  }
{ THESE AVERAGES IN CALCULATING AN AVERAGE TEMP' OVER THE LENGTH OF THE   }
{ TUBE, CALCULATES A DEVIATION FROM THIS AVERAGE (T.C.AVE - TUBEAVE), AND }
{ OUTPUTS THIS DEVIATION DATA INTO A FILE.                                }


CONST MAX = 97;
VAR
   INDAT:TEXT;   { FILE SPECIFIER: I/O }
   DATE,FNAMEIN,FNAMEOUT:LINE80; { FILE VARIABLE: I/O } DAT:ARRAY[1..100,1..MAX] OF REAL; {
   ARRAY FOR INPUT DATA } DT,TEMP:ARRAY[1..MAX] OF REAL;{ ARRAYS TEMP FOR AVERAGES AND DT FOR
   DEV'S }
   NUMREP,   { NUMBER OF SAMPLES }
   NUMCH:INTEGER; { NUMBER OF CHANNELS }
   TIME:STRING[10]; { DUMMY VAR }
   ST:STRING[3]; { DUMMY VAR }
   ST1:STRING[2]; { DUMMY VAR }
   TAVE:REAL;  { AVE TEMP OVER ENTIRE TUBE }
   Q:CHAR;
   STN:ARRAY[1..35] OF INTEGER;


PROCEDURE READ_IT; { THIS PROCEDURE READS DATA FROM A FILE PRODUCED BY          }
{ ACQCOMP. THE NUMBER OF CHANNELS IS DETERMINED BY THE NUMBER OF TEMP'S        } { IN THE FIRST
LINE OF DATA ( THE PRELIMINARY SH' IS SKIPPED BY THE PROGRAM). } { DATA IS THEN READ INTO THE
ARRAY 'DAT' UNTIL AN EOF SEQUENCE IS DETECTED AT }
{ WHICH POINT INPUT IS TERMINATED. A COUNTER IS USED IN THE FIRST LINE OF       }
{ INPUT TO DETERMINE THE NUMBER OF CHANNELS IN THE FILE AND ANOTHER IS USED    } { TO DETERMINE
THE NUMBER OF SAMPLES IN THE FILE (# LINES OF DATA). THE VALUE } { OF THESE COUNTERS IS OUTPUT
TO SCREEN FOR THE USER'S CONVENIENCE AS A CHECK } { **NOTE: THE DUMMY VARIABLES TIME, ST1 AND
ST2 ARE FORMATTED READ STATEMENTS }
{ WHICH READ IN COMMA'S AND SPACES WHICH ACQCOMP INSERTS INTO ITS OUTPUT        }
{ FILES FOR SOME UNGODLY REASON - THESE VARS WERE SIMPLY VARIED UNTIL THE       }
{ PROGRAM READ THE FILES PROPERLY.                                              }

   VAR I,J:INTEGER;

   BEGIN
   CLRSCR;
   REPEAT            { PROMPT FOR SOURCE FILENAME }
   PROMPT_NAME('Enter source filename: ',FNAMEIN);
   UNTIL FILE_EXISTS(FNAMEIN);
   ASSIGN(INDAT,FNAMEIN);
   RESET(INDAT);
   READLN(INDAT);     { SKIP THE PRELIMINARY ITEMS }
   READLN(INDAT);
   READLN(INDAT);
   READLN(INDAT);
   J:=1;              { INITIALIZE # SAMPLE COUNTER }
   I:=0;              { INITIALIZE # CHANNEL COUNTER }
   READ(INDAT,TIME);  { READ HEADER INTO DUMMY VARIABLE }
```

```
   REPEAT
   I:=I+1;
   READ(INDAT,ST1,DAT[J,I]); { ST1 READS IN THE COMMA BTWN TEMP'S }
   UNTIL i=96;(DAT[I,J] = 0.0;} { EACH LINE ENDS WITH A '0.0' }
   NUMCH:=I;  { THE I+1 INPUT IS THE '0.0': NUMCH = I-1 }
   WRITELN('Number of Channels = ',numch:2);
   WRITELN;
   READLN(INDAT);
   REPEAT
   J:=J+1;      { INCREMENT # SAMPLE COUNTER }
   GOTOXY(1,24); { OUTPUT # OF SAMPLE BEING READ TO SCREEN }
   WRITE('Sample ',J:3);
   READ(INDAT,TIME);
   FOR I:=1 TO NUMCH DO READ(INDAT,ST1,DAT[J,I]);
   READLN(INDAT);
   UNTIL (SEEKEOF(INDAT)) OR (J = 100);
   CLOSE(INDAT);
   NUMREP:=J;   { # OF SAMPLES = NUMREP }
   GOTOXY(1,24); { OUTPUT # OF SAMPLE BEING READ TO SCREEN }
   WRITELN('Number of Samples = ',NUMREP:3);
   END;   { OF READ_IT }

PROCEDURE STATION;  { SET UP STATION NUMBERS FOR OUTPUT }
   VAR I:INTEGER;
   BEGIN
   STN[2]:=1;
   FOR I:=3 TO 23 DO STN[I]:=I;
   REPEAT
   I:=I+2;
   STN[I]:=I;
   UNTIL I >=31;
   END;

PROCEDURE REDUCE_IT; { THIS PROCEDURE PERFORMS THE DATA REDUCTION. AN        }
{ AVERGAE IS CALCULATED FOR EACH CHANNEL AND EACH OF THESE AVERAGES ARE USED }
{ TO CALCULATE AN AVERAGE OVER THE ENTIRE LENGTH OF THE TUBE.  A DEVIATION   }
{ FROM THIS AVERAGE IS THEN CALCULATED FOR EACH CHANNEL.                     }
{ THIS DEVIATION IS THE ACTUAL TEMP - AVERAGE TEMP                           }

   VAR I,J:INTEGER;

   BEGIN
   FOR I:=1 TO NUMCH DO { CALC AVE FOR EACH CHANNEL }
      BEGIN
      TEMP[I]:=0.0;
      FOR J:=1 TO NUMREP DO TEMP[I]:=TEMP[I] + DAT[J,I];
      TEMP[I]:=(TEMP[I] / NUMREP);
      END;
   TAVE:=0.0;
   FOR I:=3 TO NUMCH DO TAVE:=TAVE + TEMP[I]; { CALC TUBE AVE }
   tave:=tave / (numch - 2);
   FOR I:=1 TO NUMCH DO DT[I]:=TEMP[I] - TAVE;{ CALC DEVIATION }
   FOR I:=1 TO 2 DO DT[I]:=0.0;
```

```
  END;

PROCEDURE WRITE_IT; { THIS PROCEDURE OUTPUTS DATA IN A FORMAT COMPATIBLE }
{ WITH THE PROGRAM GETAVE.PAS.                                            }

  VAR
    I,J,K:INTEGER;

  BEGIN
  REPEAT
  PROMPT_NAME(
    'Enter destination filename for Calibration Data ('+FNAMEIN+'): ', FNAMEOUT);
  UNTIL FILE_NAME_VALID(FNAMEOUT);
  ASSIGN(INDAT,FNAMEOUT);
  REWRITE(INDAT);
  WRITELN(INDAT,'Date of Acquisition = ',DATE);
  WRITELN(INDAT,'Source File: ',FNAMEIN);
  WRITELN(INDAT,'Tube Average = ',TAVE:12:4);  { TUBE AVERAGE }
  {WRITELN(INDAT,'              Tin          Tout');
  WRITELN(INDAT,'      ',DT[1]:12:4,' ',DT[2]:12:4);} { TIN AND TOUT }

  WRITELN(INDAT,'Station #      A            C            E            G'); I:=2;
  K:=1;
  REPEAT            { STATIONS 2 - 22 }
  K:=K+1;
  WRITE(INDAT,STN[K]:5,' ');
  FOR J:=1 TO 4 DO
    BEGIN
    I:=I+1;
    WRITE(INDAT,DT[I]:12:4,' ');
    END;
  WRITELN(INDAT);
  UNTIL I >= 86;
  k:=k-1;
  REPEAT            { STATIONS 23 - 31 }
  K:=K+2;
  WRITE(INDAT,STN[K]:5,' ');
  FOR J:=1 TO 2 DO
    BEGIN
    I:=I+1;
    WRITE(INDAT,DT[I]:12:4,'            ');
    END;
  WRITELN(INDAT);
  UNTIL I >= 96;
  CLOSE(INDAT);
  END;  { OF WRITE_IT }

BEGIN       { MAIN PROGRAM }
PROMPT_NAME('Enter Date of Data Acquisition: ',DATE);
REPEAT
READ_IT;    { CALL INPUT PROCEDURE }
REDUCE_IT;  { CALL DATA REDUCTION PROCEDURE }
STATION;
```

```
CLRSCR;
WRITELN;
WRITELN;
WRITELN;
WRITELN('Number of Channels = ',NUMCH:12);
WRITELN;
WRITELN('Number of Samples = ',NUMREP:12);
WRITELN;
WRITELN('Test Section Inlet Temp = ',TEMP[1]:12:4);
WRITELN;
WRITELN('Test Section Outlet Temp = ',TEMP[2]:12:4);
WRITELN;
WRITELN;
WRITE_IT;   { CAL OUTPUT PROCEDURE }
PROMPT_CHAR('Reduce another (y/n)? ',Q);
UNTIL Q = 'N';
END.        { MAIN PROGRAM }
```

```
{$I C:\TURBO\UTIL2 }
{ PROGRAM GETAVE.PAS }
{ THIS PROGRAM IS USED DURING THE CALIBRATION OF THERMOCOUPLES USED IN }
{ CONJUNCTION WITH THE MAC-14. SEVERAL RUNS ARE PERFORMED WITH ACQCOMP }
{ AND REDUCED WITH CALHT.PAS. THE DATA PRODUCED WITH CALHT IS THEN      }
{ INPUT TO THIS PROGRAM. THE INPUT DATA IS AVERAGED INTO CALIBRATION    }
{ CORRECTIONS FOR THE PROGRAM REDUCE96.PAS. THIS PROGRAM WAS WRITTEN    }
{ BY K.D.HOWARD IN TURBOPASCAL 3.0.                                     }

TYPE DAT = RECORD
          DT:ARRAY[1..20] OF REAL; { DELTA TEMP. FROM CALHT }
          SD,AVEDT:REAL; { SD = DEVIATION FROM AVE DT (SEE REDUCTION PROC) }
                         { AVEDT = AVERAGE DELTA TEMP }
          END; { OF RECORD }

VAR
   STNUM:ARRAY[1..31] OF INTEGER;
   TEMP:ARRAY[1..96] OF DAT; { ARRAY OF RECOERD 'DAT }
   K,I,J,FN:INTEGER;{ NUMSAMPLE = # SAMPLES INPUT }
   DATE,DATE_SOURCE,FNAME:LINE80;  { FILE VARIABLE: I/O }
   STN:STRING[8];
   TIME:STRING[15];
   INDAT:TEXT;    { FILE SPECIFIER: I/O }
   Q:CHAR;
   SQ,AVETEMP:REAL;
   ST:STRING[2];

BEGIN  { MAIN PROGRAM }
FN:=0; { INITIALIZE SAMPLE (FILE) # COUNTER }
PROMPT_NAME('Enter date of calibration: ',DATE);
REPEAT
FN:=FN + 1; { INCREMENT SAMPLE COUNTER }
STR(FN,ST); { CONVERT INTEGER VALUE TO STRING VALUE }
REPEAT
PROMPT_NAME('Enter source filename #'+ST+': ',FNAME); { PROMPT FOR FILENAME }
UNTIL FILE_EXISTS(FNAME); { CHECK THAT FILE EXISTS }
ASSIGN(INDAT,FNAME);
RESET(INDAT);
READLN(INDAT);
READLN(INDAT);
READLN(INDAT,TIME,AVETEMP); { READ AVERAGE TEMP }
{READLN(INDAT);
READLN(INDAT,TEMP[1].DT[FN],TEMP[2].DT[FN]);} { READ DELTA FOR TIN AND TOUT }
I:=2;   { INITIALIZE STATION # }
READLN(INDAT);
REPEAT
READ(INDAT,STN);
FOR J:=1 TO 4 DO { FOR STATIONS 2-22 READ 4 DELTAS }
   BEGIN
   I:=I+1;
   READ(INDAT,TEMP[I].DT[FN]);
   END;
READLN(INDAT);
UNTIL I >= 86;
```

```
REPEAT          { FOR STATIONS 23-31 READ 2 DELTAS }
READ(INDAT,STN);
FOR J:=1 TO 2 DO
   BEGIN
   I:=I + 1;
   READ(INDAT,TEMP[I].DT[FN]);
   END;
READLN(INDAT);
UNTIL I >= 96;
CLOSE(INDAT);
PROMPT_CHAR('Read another file (y/n)? ',Q); { ANOTHER SAMPLE? }
UNTIL Q = 'N';
FOR I:=3 TO 96 DO { CALCULATE AVE DELTA FOR EACH CHANNEL }
   WITH TEMP[I] DO
      BEGIN
      AVEDT:=0.0;
      SQ:=0.0;
      FOR J:=1 TO FN DO
         BEGIN
         AVEDT:=AVEDT + DT[J];
         SQ:=SQ + SQR(DT[J]);
         END;
{ AVERAGE DELTA = SUM OF DELTA (I=1 TO # SAMPLES) / # SAMPLES                }
{ DEVIATION FROM AVE = SD = SUM OF SQRT( SQR( AVE DELTA - DELTA[I]/ # SAMPLES }
      AVEDT:=AVEDT / FN;
      FOR J:=1 TO FN DO SQ:=SQR(DT[J] -AVEDT);
      SD:=SQRT(SQ / FN);
      END;
REPEAT    { SEND CORRECTION DATA TO FILE }
STNUM[1]:=1;
STNUM[2]:=3;
FOR I:=3 TO 31 DO STNUM[I]:=STNUM[I-1] +1;
PROMPT_NAME('Enter destination filename for averaged data: ',FNAME);
UNTIL FILE_NAME_VALID(FNAME);
ASSIGN(INDAT,FNAME);
REWRITE(INDAT);
WRITELN(INDAT,'Heat Transfer test section');
WRITELN(INDAT,'Correction data from calibration on: ',date);
WRITELN(INDAT,'ST#     A            C            E            G');
K:=0;
I:=2;
REPEAT
K:=K+1;
WRITE(INDAT,STNUM[K]:2,' ');
FOR J:=1 TO 4 DO
   BEGIN
   I:=I+1;
   WRITE(INDAT,TEMP[I].AVEDT:12:6,' ');
   END;
WRITELN(INDAT);
UNTIL I >= 86;
K:=K+1;
REPEAT
WRITE(INDAT,STNUM[K]:2,' ');
```

```
FOR J:=1 TO 2 DO
   BEGIN
   I:=I+1;
   WRITE(INDAT,TEMP[I].AVEDT:12:6,'            ');
   END;
K:=K+2;
WRITELN(INDAT);
UNTIL I >= 96;
CLOSE(INDAT);
REPEAT    { SEND DEVIATION DATA TO FILE }
PROMPT_NAME('Enter destination filename for SD data: ',FNAME);
UNTIL FILE_NAME_VALID(FNAME);
ASSIGN(INDAT,FNAME);
REWRITE(INDAT);
WRITELN(INDAT,TEMP[1].SD:12:6,' ',TEMP[2].SD:12:6);
I:=2;
REPEAT
FOR J:= 1 TO 4 DO
   BEGIN
   I:=I+1;
   WRITE(INDAT,TEMP[I].SD:12:6);
   END;
WRITELN(INDAT);
UNTIL I >= 86;
REPEAT
FOR J:=1 TO 2 DO
   BEGIN
   I:=I+1;
   WRITE(INDAT,TEMP[I].SD:12:6);
   END;
WRITELN(INDAT);
UNTIL I >= 96;
CLOSE(INDAT);
END.  { OF MAIN PROGRAM }
```

File CORR.DAT* (created by GETAVE used in RED96)
Heat Transfer test section
Correction data from calibration on: 7-24-89

	LOCATION#			
ST#	1	2	3	4
1	-0.016464	-0.048764	-0.055691	-0.041300
3	-0.005155	-0.071364	-0.057036	-0.058936
4	-0.025827	-0.055200	-0.052555	-0.044009
5	-0.032064	-0.089473	-0.036845	-0.016882
6	-0.012755	-0.019809	-0.060109	-0.008127
7	0.002236	-0.030236	-0.021618	-0.012155
8	-0.006473	-0.015018	-0.028336	-0.005000
9	0.014173	-0.027309	0.049873	0.079564
10	0.093973	0.049473	0.074536	0.120227
11	0.110982	0.434136	0.105255	0.105064
12	0.115845	0.103191	0.001400	0.067718
13	0.054018	-0.015945	-0.236464	-0.200000
14	-0.233382	-0.210818	-0.270564	-0.191364
15	-0.160173	-0.177318	-0.147500	-0.114573
16	-0.058545	-0.070364	-0.060991	-0.026536
17	-0.003355	-0.048173	-0.026809	0.006927
18	0.033300	0.010955	0.023436	0.029927
19	0.062818	0.009064	0.028518	0.042036
20	0.065127	0.038282	0.049673	0.088645
21	0.092945	0.039382	0.065673	0.095627
22	0.128982	0.068955	0.107518	0.135045
23	0.113682		0.136891	
25	0.163945		0.076755	
27	0.103064		0.056973	
29	0.016527		-0.005009	
31	-0.039291		-0.220664	

* CORR.DAT is generated without headings.

```
10 '                    PROGRAM INTERP.BAS
20 '                    DOUG STRICKLAND
30 '                    SEPTEMBER 12, 1989
40 '
50 CLS: Z1=0
60 DIM NUM(49,6),TRUTEMP(49)
70 PRINT"     THIS PROGRAM USES TEMPERATURE DIFFERENCE INFORMATION"
80 PRINT"(DELTA=Ti-Ttrue) TO CORRECT FOR INLET AND EXIT THERMOCOUPLE"
90 PRINT"BULK TEMPERATURE READINGS USING AN INTERPOLATION SCHEME ASSUMING"
100 PRINT"THE DIFFERENCES FOLLOW A LINEAR PATTERN (although they do not)."
110 PRINT"THE PROGRAM OUTPUTS A CORRECTION VALUE WHICH SHOULD BE SUBTRACTED"
120 PRINT"FROM THE INPUT TEMPERATURE TO YIELD A CORRECTED TEMPERATURE OF "
130 PRINT"THE FORM: "
140 PRINT"                   Tcorr = Tinput - CORRECTION"
150 PRINT:PRINT"     ALSO IN THIS PROGRAM IS CURVE FIT DATA PERTAINING TO THE"
160 PRINT"WELDER.  THE USER MAY IF DESIRED CORRECT THE DATA READ AT THE WELDER"
170 PRINT"BY SELECTING THAT OPTION.  THE PROGRAM RETURNS THE CORRECTED VALUE."
180 PRINT:
190 PRINT"      THE LIMITATIONS ARE GIVEN AS FOLLOWS:
200 PRINT"              INLET TEMPERATURE        (75 < TIN < 115) F"
210 PRINT"              EXIT TEMPERATURE         (75 < TOUT < 149) F"
220 PRINT"              WELDER AMPERAGE          (155 < AMPS < 600) A"
230 PRINT:PRINT:PRINT
240 PRINT"      WHICH PARAMETER DO YOU WISH TO CORRECT?"
250 INPUT"TO CORRECT TEMPERATURE PRESS 1 & ENTER, FOR AMPS PRESS 2 & ENTER";PC
260 IF PC > 1 GOTO 740
270 PRINT:PRINT:PRINT
280 '
290 IF Z1>0 THEN IF PC >1 GOTO 310
300 'DIM NUM(49,6),TRUTEMP(49)
310 OPEN "I",#1,"DELTA.DAT"
320 '
330 CLS:PRINT"WHICH THERMOCOUPLE DO YOU NEED A CORRECTION FOR?"
340 INPUT"PRESS 1 & RETURN FOR BULK INLET, PRESS 2 & RETURN FOR BULK EXIT";TC
350 PRINT:PRINT
360 INPUT"WHAT IS THE TEMPERATURE OF OPERATION FOR THE THERMOCOUPLE";TOP
370 PRINT:PRINT
380 '
390 FOR I=1 TO 49
400      FOR J=1 TO 6
410              INPUT#1,NUM(I,J)
420              TRUTEMP(I)=NUM(I,1)
430 '            PRINT TOP,NUM(I,1)
440              IF TOP = NUM(I,1) THEN IF J=6 GOTO 620
450              IF NUM(I,1) > TOP THEN IF J=6 GOTO 480
460      NEXT J
470  NEXT I
480 '
490 IF TC=2 GOTO 560
500 AVG1=(NUM(I-1,4)-NUM(I,4))
510 AVG1A=(TOP-NUM(I-1,1))/(NUM(I-1,1)-NUM(I,1))
520 DEL1=NUM(I-1,4)+AVG1A*AVG1
530 CLS:PRINT:PRINT
```

```
540 PRINT"THE CORRECTION FOR TEMP=";TOP" IS FOUND TO BE=";DEL1
550 GOTO 700
560 AVG2=(NUM(I-1,2)-NUM(I,2))
570 AVG2A=(TOP-NUM(I-1,1))/(NUM(I-1,1)-NUM(I,1))
580 DEL2=NUM(I-1,2)+AVG2A*AVG2
590 CLS:PRINT:PRINT
600 PRINT"THE CORRECTION FOR TEMP=";TOP" IS FOUND TO BE=";DEL2
610 GOTO 710
620 IF TC=2 GOTO 670
630 DEL1=NUM(I,4)
640 CLS:PRINT:PRINT
650 PRINT"THE CORRECTION FOR TEMP=";TOP" IS FOUND TO BE=";DEL1
660 GOTO 700
670 DEL2=NUM(I,2)
680 CLS:PRINT:PRINT
690 PRINT"THE CORRECTION FOR TEMP=";TOP" IS FOUND TO BE=";DEL2:GOTO 710
700 TCORR=TOP-DEL1: GOTO 720
710 TCORR=TOP-DEL2
720 PRINT"        THE CORRECTED TEMPERATURE TCORR=";TCORR
730 PRINT:PRINT:PRINT:GOTO 790
740 CLS:PRINT:PRINT:PRINT
750 INPUT"       WHAT IS THE WELDER AMPEREAGE READING (amps)";AMPS
760 AMPCORR=14.536838# + AMPS *.937169
770 PRINT:PRINT:PRINT"THE CORRECTED AMPERAGE READING=";AMPCORR
780 PRINT:PRINT:PRINT:PRINT
790 INPUT"IF YOU WANT TO RUN AGAIN, PRESS 1 & ENTER";Z
800 IF Z=2 GOTO 820
810 CLS:CLOSE #1:Z1=1:GOTO 250
820 END
```

FILE DELTA.DAT CALLED BY INTERP.BAS

TEMP.	DELTA 1	DELTA 2	DELTA 3	DELTA 4	DELTA 5
77.5917	-1.191696	-1.691696	-1.291695	-2.191696	-.6916962
79.0536	-1.053597	-1.753594	-1.053597	-2.053597	-.653595
79.2857	-.785698	-1.385696	-.785698	-1.985695	-.4856949
80.2321	-.8321	-1.432098	-.8321	-2.232102	-.5321045
80.4107	-.8106995	-1.610695	-.6106949	-2.110695	-.5106964
81.178	-.3779984	-1.578003	-.4780045	-1.478005	-.0780029
82.2127	-.512703	-1.512703	-.4126969	-1.312698	.0873032
83.14	-.2399979	-1.339996	-.9400024	-1.739998	-.2399979
83.6065	.1935043	-1.106499	-1.306496	-1.306496	.0934982
90.0719	.9281006	-.0718994	.3281021	.2281037	1.628098
95.0045	-.2044983	-1.204498	-.1044998	-.8045044	.5954971
100.2676	-.0676041	-1.267601	-.0676041	-1.067604	.5324021
101.163	-.663002	-1.563004	-.163002	-1.463005	.0369949
101.8794	.8205948	-.7794037	-.1794052	-1.179405	.3205948
103.2227	.0773010	-1.122704	-.5227051	-1.322701	-.1227036
104.1183	-.5183029	-1.318298	-.5183029	-1.7183	-.2182999
104.692	.1080017	-1.292	.4079971	-1.292	.2080002
105.7677	.0323029	-1.367699	-.4676972	-1.567703	.2322998
106.3409	.4591065	-.7408981	.3591004	-.5408936	.559105
107.4155	-.3154984	-1.615494	-.5154953	-1.615494	.1845017
107.7739	-.1739044	-1.073906	-.573906	-1.4739	.1260986
108.3476	-.4476013	-1.347603	.1523972	-1.647606	-.0475998
109.6028	-.2027969	-.8027955	-.6027985	-1.802795	-.1027985
111.3972	-.1972046	-1.4972	-.2972031	-1.597199	-.1972046
111.8275	-.1275024	-.9274979	.0725021	-1.027496	.5725021
113.0646	-.4645996	-1.764595	-.7645951	-1.764595	-.0645981
113.603	.2970047	-.8029938	-.6029968	-1.202995	.1970062
115.3982	-.0981979	-1.198204	.0018005	-1.398201	.2017975
116.476	.1240006	-1.175995	.3240051	-1.276001	.223999
117.194	.2060013	-1.094002	.1060028	-1.194	.5059967
118.8096	.2903977	-1.209602	-.5095978	-1.209602	.1903992
119.7078	-.3078003	-1.907799	.3921967	-1.407799	.0922012
120.9293	-.1292954	-1.3293	-.4292984	-1.429298	.2706986
122.7624	-.3623963	-1.062401	-.5624008	-1.562401	.0376053
124.0212	.3787995	-.9212036	-.1212006	-.5212021	.5787964
125.1001	-.4001007	-.7000961	-1.100098	-1.400101	-.0000992
126.179	-.1790009	-.8789978	.1210022	-1.278999	.3209992
127.6176	-.3175964	-.6175995	-.3175964	-1.317596	.0823974
129.0561	-.3561096	-.5561066	-.7561035	-1.35611	.0438995
129.956	.2440033	-.9559936	-.5559998	-1.355988	-.1559906
131.756	-.4559937	-1.755997	-.8560028	-1.556	-.2559967
132.656	.8439941	.1439972	-.5559998	-.6560059	.7439881
134.8167	.8833008	-.6166993	1.083298	-.5166932	1.183304
139.861	.5390015	-.6609955	1.039001	-.4609986	1.339005
143.8291	.5708923	.0708923	.1708984	-.3291016	1.070892
145.9934	-.0933991	-1.293396	.0066071	-1.493393	.0066071
147.0756	-.2756043	-1.175613	.3243866	-1.275604	.5243988
147.9781	-.5781098	-.6781006	-1.278107	-1.478104	-.1781006
149.7836	1.016403	-.4835968	.7164001	-1.183594	.5164032

```
10 '                          PROGRAM GPMCALC.BAS
20 '                          DOUG STRICKAND
30 '                          FEBUARY 28,1989
40 '
50 '    THIS PROGRAM WILL USE AN INPUT OF EITHER FREQUENCY OR FLOWRATE
60 '  AND FOR A SPECIFIED TURBINEMETER (LARGE OR SMALL) AND CALCULATE THE
70 '  UNSPECIFIED VALUE OVER THE RANGE OF GPM OR FREQUENCY AVAILABLE.
80 '  BECAUSE EACH FLUID CONCENTRATION REQUIRES A SEPARATE CURVE FIT, IT
90 '  WILL BE NECESSARY TO SPECIFY WHICH CONCENTRATION OF FLUID THE USER
100 '  WISHES TO EVALUATE.
110 '
120 CLS:PRINT"          THREE FLUID CONCENTRATIONS ARE AVAILALE"
130     PRINT"                  1:  100 DISTILLED WATER"
140     PRINT"                  2:  100% ETHYLENE GLYCOL"
150     PRINT"                  3:   60% ETHYLENE GLYCOL"
160     PRINT"                       40% DISTILLED WATER" :PRINT
170 INPUT"       ENTER THE NUMBER DEFINING THE CONCENTRATION OF INTEREST";CON
180 IF CON=1 GOTO 240
190 IF CON=2 GOTO 1150
200 IF CON=3 GOTO 2030
210 '-------------------------------------------------------------
220 '   AT THE NEXT CONTROL LINE CALCULATIONS FOR PURE WATER BEGIN
230 '-------------------------------------------------------------
240 C=(4.28958E-03+4.30351E-03+4.28176E-03)/3
250 C1=(6.952761E-02+.0692793+.068818)/3
260 K=-(.021509+.026951+.017774)/3
270 K1=-(.13566+.084517+.033494)/3
280 CLS
285 PRINT"  FOR A MIXTURE OF 100% DISTILLED WATER....."
290 PRINT"  YOU CAN EITHER CALCULATE FLOWRATE FROM FREQ. OR VICE VERSA"
300 PRINT
310 INPUT"  ENTER  '1'  TO CALCULATE GPM, ENTER  '2'  TO CALCULATE FREQ.";W
320 IF W=1 THEN CLS: GOTO 690
330 CLS
340 PRINT"  WHICH FLOWMETER DO YOU WANT FREQUENCY DATA FOR?"
350 PRINT
360 INPUT" ENTER '1' FOR SMALL METER, ENTER '2' FOR LARGE METER";V
370 IF V=1 THEN CLS: GOTO 540
380 CLS
390 INPUT" WHAT IS THE FLOWRATE FOR THE LARGE METER (GPM)";GPM
400 HF=((GPM-K1)/C1)
410 IF GPM<3.72 GOTO 500
420 IF GPM>10.25 GOTO 500
430 CLS
440 PRINT USING"           FREQUENCY = ###.#### HERTZ";HF
450 PRINT
460 PRINT USING"        FOR A FLOWRATE OF ##.#### GPM";GPM
470 PRINT
480 GOTO 1100
490 CLS
500 PRINT
510 PRINT"  THE FLOWRATE INPUT IS NOT AVAILABLE FOR THE LARGE METER."
520 PRINT
```

```
530 GOTO 1050
540 INPUT"  WHAT IS THE FLOWRATE FOR THE SMALL METER";GPM
550 SF=((GPM-K)/C)
560 IF GPM<.56 GOTO 650
570 IF GPM>4.61 GOTO 650
580 CLS
590 PRINT USING"              FREQUENCY = ####.#### HERTZ";SF
600 PRINT
610 PRINT USING"             FOR A FLOWRATE OF #.#### GPM";GPM
620 PRINT
630 GOTO 1100
640 CLS
650 PRINT
655 PRINT"  FOR A MIXTURE OF 100% DISTILLED WATER....."
660 PRINT"  THE FLOWRATE INPUT IS NOT AVAILABLE FOR THE SMALL METER"
670 PRINT
680 GOTO 1050
690 PRINT"  WHICH FLOWMETER DO YOU REQUIRE GPM DATA FOR? "
700 PRINT
710 INPUT"  ENTER '1' FOR SMALL METER, ENTER '2' FOR LARGE METER";A
720 PRINT
730 IF A=1 THEN CLS: GOTO 920
740 CLS
750 INPUT"  ENTER THE FREQUENCY FOR THE LARGE TURBINEMETER";LF
760 IF LF<55 GOTO 860
770 IF LF>155 GOTO 860
780 GPM=C1*LF+K1
790 CLS
800 PRINT
810 PRINT USING"           FLOWRATE= ##.#### GPM";GPM
820 PRINT
830 PRINT USING" FOR A FREQUENCY OF  ###.### HERTZ";LF
840 GOTO 1080
850 CLS
860 PRINT" THE FREQUENCY INPUT IS NOT IN THE AVAILABLE LARGE METER RANGE"
870 PRINT
880 INPUT" DO YOU WISH TO TRY AGAIN (YES=1 NO=2)";B
890 IF B=2 THEN CLS: GOTO 2880
900 GOTO 240
910 PRINT
920 INPUT"  ENTER THE FREQUENCY FOR THE SMALL TURBINEMETER IN Hz";SF
930 IF SF<135 GOTO 1020
940 IF SF>1080 GOTO 1020
950 GPM=C*SF+K
960 CLS
970 PRINT
980 PRINT USING"                 FLOWRATE =##.#### GPM";GPM
990 PRINT
1000 PRINT USING"      FOR A FREQUENCY OF  ####.#### HERTZ";SF
1010 GOTO 1080
1020 CLS
1030 PRINT"  THE FREQUENCY INPUT IS NOT AVAILABLE FOR THE SMALL METER"
1040 PRINT
```

```
1050 INPUT" DOU YOU WISH TO TRY AGAIN  (YES=1 NO=2)";B1
1060 IF B1=2 THEN CLS: GOTO 2880
1070 GOTO 240
1080 PRINT
1090 PRINT
1100 INPUT" TO RUN THE PROGRAM AGAIN, ENTER '1', TO STOP ENTER '2'";Z
1110 IF Z=1 GOTO 240 ELSE GOTO 2880
1120 '-----------------------------------------------------------------------
1130 ' AT THE NEXT CONTROL LINE CALCULTATIONS FOR PURE ETHYLENE GLYCOL BEGIN
1140 '-----------------------------------------------------------------------
1150 C=.00411: C1=.0614121: K=.054708: K1=.064849
1160 CLS
1165 PRINT"  FOR A MIXTURE OF 100% ETHYLENE GLYCOL....."
1170 PRINT"  YOU CAN EITHER CALCULATE FLOWRATE FROM FREQ. OR VICE VERSA"
1180 PRINT
1190 INPUT"  ENTER  '1'  TO CALCULATE GPM, ENTER  '2'  TO CALCULATE FREQ.";W
1200 IF W=1 THEN CLS: GOTO 1570
1210 CLS
1220 PRINT" WHICH FLOWMETER DO YOU WANT FREQUENCY DATA FOR?"
1230 PRINT
1240 INPUT" ENTER '1' FOR SMALL METER, ENTER '2' FOR LARGE METER";V
1250 IF V=1 THEN CLS: GOTO 1420
1260 CLS
1270 INPUT" WHAT IS THE FLOWRATE FOR THE LARGE METER (GPM)";GPM
1280 HF=((GPM-K1)/C1)
1290 IF GPM<3.15 GOTO 1380
1300 IF GPM>6.6 GOTO 1380
1310 CLS
1320 PRINT USING"           FREQUENCY = ###.#### HERTZ";HF
1330 PRINT
1340 PRINT USING"        FOR A FLOWRATE OF ##.#### GPM";GPM
1350 PRINT
1360 GOTO 1980
1370 CLS
1380 PRINT
1390 PRINT" THE FLOWRATE INPUT IS NOT AVAILABLE FOR THE LARGE METER."
1400 PRINT
1410 GOTO 1930
1420 INPUT" WHAT IS THE FLOWRATE FOR THE SMALL METER";GPM
1430 SF=((GPM-K)/C)
1440 IF GPM<.621 GOTO 1530
1450 IF GPM>3.43 GOTO 1530
1460 CLS
1470 PRINT USING"             FREQUENCY = ####.#### HERTZ";SF
1480 PRINT
1490 PRINT USING"            FOR A FLOWRATE OF #.#### GPM";GPM
1500 PRINT
1510 GOTO 1980
1520 CLS
1530 PRINT
1540 PRINT" THE FLOWRATE INPUT IS NOT AVAILABLE FOR THE SMALL METER"
1550 PRINT
1560 GOTO 1930
```

```
1570 PRINT"  WHICH FLOWMETER DO YOU REQUIRE GPM DATA FOR? "
1580 PRINT
1590 INPUT"  ENTER '1' FOR SMALL METER, ENTER '2' FOR LARGE METER";A
1600 PRINT
1610 IF A=1 THEN CLS: GOTO 1800
1620 CLS
1630 INPUT"  ENTER THE FREQUENCY FOR THE LARGE TURBINEMETER";LF
1640 IF LF<49.5 GOTO 1740
1650 IF LF>105 GOTO 1740
1660 GPM=C1*LF+K1
1670 CLS
1680 PRINT
1690 PRINT USING"          FLOWRATE= ##.#### GPM";GPM
1700 PRINT
1710 PRINT USING" FOR A FREQUENCY OF  ###.### HERTZ";LF
1720 GOTO 1960
1730 CLS
1740 PRINT" THE FREQUENCY INPUT IS NOT IN THE AVAILABLE LARGE METER RANGE"
1750 PRINT
1760 INPUT" DO YOU WISH TO TRY AGAIN (YES=1 NO=2)";B
1770 IF B=2 THEN CLS: GOTO 2880
1780 GOTO 1160
1790 PRINT
1800 INPUT"  ENTER THE FREQUENCY FOR THE SMALL TURBINEMETER IN Hz";SF
1810 IF SF<141 GOTO 1900
1820 IF SF>800 GOTO 1900
1830 GPM=C*SF+K
1840 CLS
1850 PRINT
1860 PRINT USING"                FLOWRATE =##.#### GPM";GPM
1870 PRINT
1880 PRINT USING"      FOR A FREQUENCY OF  ####.#### HERTZ";SF
1890 GOTO 1960
1900 CLS
1910 PRINT" THE FREQUENCY INPUT IS NOT AVAILABLE FOR THE SMALL METER"
1920 PRINT
1930 INPUT" DOU YOU WISH TO TRY AGAIN  (YES=1 NO=2)";B1
1940 IF B1=2 THEN CLS: GOTO 2880
1950 GOTO 1150
1960 PRINT
1970 PRINT
1980 INPUT" TO RUN THE PROGRAM AGAIN, ENTER '1', TO STOP ENTER '2'";Z
1990 IF Z=1 GOTO 1150 ELSE GOTO 2880
2000 '-----------------------------------------------------------------------
2010 ' THE NEXT CONTROL LINES GIVE ANSWERS FOR MIXED FLUID CONCENTRATION
2020 '-----------------------------------------------------------------------
2030 C=.004245: C1=.065902: K1=-.1618492 :K=.007308
2040 CLS
2045 PRINT"  FOR A MIXTURE CONTAINING 60% ETHYLENE GLYCOL....."
2050 PRINT"  YOU CAN EITHER CALCULATE FLOWRATE FROM FREQ. OR VICE VERSA"
2060 PRINT
2070 INPUT"  ENTER  '1'  TO CALCULATE GPM, ENTER  '2'  TO CALCULATE FREQ.";W
2080 IF W=1 THEN CLS: GOTO 2450
```

```
2090 CLS
2100 PRINT" WHICH FLOWMETER DO YOU WANT FREQUENCY DATA FOR?"
2110 PRINT
2120 INPUT" ENTER '1' FOR SMALL METER, ENTER '2' FOR LARGE METER";V
2130 IF V=1 THEN CLS: GOTO 2300
2140 CLS
2150 INPUT" WHAT IS THE FLOWRATE FOR THE LARGE METER (GPM)";GPM
2160 HF=((GPM-K1)/C1)
2170 IF GPM<2.67 GOTO 2260
2180 IF GPM>7.22 GOTO 2260
2190 CLS
2200 PRINT USING"           FREQUENCY = ###.#### HERTZ";HF
2210 PRINT
2220 PRINT USING"        FOR A FLOWRATE OF ##.#### GPM";GPM
2230 PRINT
2240 GOTO 2860
2250 CLS
2260 PRINT
2270 PRINT" THE FLOWRATE INPUT IS NOT AVAILABLE FOR THE LARGE METER."
2280 PRINT
2290 GOTO 2810
2300 INPUT" WHAT IS THE FLOWRATE FOR THE SMALL METER";GPM
2310 SF=((GPM-K)/C)
2320 IF GPM<.728 GOTO 2410
2330 IF GPM>3.98 GOTO 2410
2340 CLS
2350 PRINT USING"             FREQUENCY = ####.#### HERTZ";SF
2360 PRINT
2370 PRINT USING"            FOR A FLOWRATE OF #.#### GPM";GPM
2380 PRINT
2390 GOTO 2860
2400 CLS
2410 PRINT
2420 PRINT" THE FLOWRATE INPUT IS NOT AVAILABLE FOR THE SMALL METER"
2430 PRINT
2440 GOTO 2810
2450 PRINT"  WHICH FLOWMETER DO YOU REQUIRE GPM DATA FOR? "
2460 PRINT
2470 INPUT"  ENTER '1' FOR SMALL METER, ENTER '2' FOR LARGE METER";A
2480 PRINT
2490 IF A=1 THEN CLS: GOTO 2680
2500 CLS
2510 INPUT"  ENTER THE FREQUENCY FOR THE LARGE TURBINEMETER";LF
2520 IF LF<42.9 GOTO 2620
2530 IF LF>113 GOTO 2620
2540 GPM=C1*LF+K1
2550 CLS
2560 PRINT
2570 PRINT USING"           FLOWRATE= ##.#### GPM";GPM
2580 PRINT
2590 PRINT USING" FOR A FREQUENCY OF  ###.### HERTZ";LF
2600 GOTO 2840
2610 CLS
```

```
2620 PRINT" THE FREQUENCY INPUT IS NOT IN THE AVAILABLE LARGE METER RANGE"
2630 PRINT
2640 INPUT" DO YOU WISH TO TRY AGAIN (YES=1 NO=2)";B
2650 IF B=2 THEN CLS: GOTO 2880
2660 GOTO 2030
2670 PRINT
2680 INPUT"  ENTER THE FREQUENCY FOR THE SMALL TURBINEMETER IN Hz";SF
2690 IF SF<168 GOTO 2780
2700 IF SF>934 GOTO 2780
2710 GPM=C*SF+K
2720 CLS
2730 PRINT
2740 PRINT USING"                 FLOWRATE =##.#### GPM";GPM
2750 PRINT
2760 PRINT USING"      FOR A FREQUENCY OF  ####.#### HERTZ";SF
2770 GOTO 2840
2780 CLS
2790 PRINT" THE FREQUENCY INPUT IS NOT AVAILABLE FOR THE SMALL METER"
2800 PRINT
2810 INPUT" DOU YOU WISH TO TRY AGAIN  (YES=1 NO=2)";B1
2820 IF B1=2 THEN CLS: GOTO 2880
2830 GOTO 2030
2840 PRINT
2850 PRINT
2860 INPUT" TO RUN THE PROGRAM AGAIN, ENTER '1', TO STOP ENTER '2'";Z
2870 IF Z=1 GOTO 2030 ELSE GOTO 2880
2880 END
```

```
200  '
210  '  Jody R. Augustine
220  '  01-27-89
230  '
240  '  Thesis Program
250  '
260  '  Thermophysical Properties of Ethylene Glycol-Water Mixtures
270  '
280  '  Reference
290  '
300  '       D. Bohne, S. Fischer, and E. Obermeier:  "Thermal Conductivity,
310  '       Density, Viscosity, and Prandtl-Numbers of Ethylene Glycol-
320  '       Water Mixtures".  Ber. Bunsenges. Phys. Chem. 88, 742-744
330  '       (1984)
340  '
350  '  The program allows the user to input the mass concentration or
360  '  mass fraction of the ethylene glycol water-mixture along with
370  '  the bulk temperature of the mixture.  The wall temperature at
380  '  a specific thermocouple station may also be entered for the
390  '  purpose of obtaining a Grashof Number for natural convection
400  '  calculations.  The volumetric flow rate is also entered in order
410  '  to obtain an average tube velocity and a Reynolds Number.
420  '
430  '  The thermophysical property equations have a temperature range of
440  '  32 to 300 degrees Farenheit (0 to 150 degrees Celsius) at
450  '  atmospheric pressure.
460  '
470  '  The accuracy of the properties are as follows:
480  '
490  '       density          -+ 1%
500  '       thermal cond.    -+ 1%
510  '       abs. visc.       -+ 5%
520  '       Prandtl No.      -+ 5%
530  '
540  '  Variables ending with the letter (E) designate English units.
550  '
560  '
570  DEFDBL A-H,K-Z
580  RA = 1
590  CLS
600  '
610  INPUT "ENTER 1 FOR S.I. UNIT OUTPUT, RETURN FOR ENGLISH ";UN
620  '
630  INPUT "ENTER MASS CONCENTRATION (0 for water, 1 for pure eth. glyc.) ";X
640  IF X < 0 OR X > 1 THEN PRINT " MASS CONCENTRATION MUST BE BETWEEN ZERO AND ONE " : GOTO 630
650  '
660  '
670  IF UN = 1 THEN 910
680  '
690  '
700  INPUT "ENTER BULK TEMPERATURE (F) ";T
710  IF T < 32 OR  T > 300 THEN PRINT " BULK TEMPERATURE MUST BE BETWEEN 32 AND 300 DEGREES
FARENHEIT" : GOTO 700
```

```
720   '
730   '
740   INPUT "ENTER WALL TEMPERATURE (F) ";TW
750   IF TW <= T OR TW > 500 THEN PRINT " WALL TEMPERATURE MUST BE GREATER THAN THE BULK
TEMPERATURE AND LESS THAN 500 F" : GOTO 740
760   '
770   '
780   INPUT "ENTER FLOWRATE (gpm) ";FL
790   IF FL < 0 OR FL > 20 THEN PRINT " FLOWRATE MUST BE BETWEEN 0 AND 20 gpm" : GOTO 780
800   '
810   '
820   INPUT "ENTER INSIDE TUBE DIAMETER (in.), default (.622 in.) ";DIAM
830   IF DIAM < 0 OR DIAM > 3 THEN PRINT " DIAMETER MUST BE BETWEEN 0 AND 3 INCHES" : GOTO 820
840   '
850   '
860   IF DIAM = 0 THEN DIAM = .622
870   DIAM = DIAM/12
880   '
890   IF UN = 0 THEN GOTO 1100
900   '
910   INPUT "ENTER BULK TEMPERATURE (C) ";T
920   IF T<0 OR T>150 THEN PRINT " BULK TEMPERATURE MUST BE BETWEEN 0 AND 150 DEGREES CELSIUS" :
GOTO 910
930   '
940   '
950   INPUT "ENTER WALL TEMPERATURE (C) ";TW
960   IF TW<T OR TW>300 THEN PRINT " WALL TEMPERATURE MUST BE GREATER THAN THE BULK TEMPERATURE
AND LESS THAN 300 C" : GOTO 950
970   '
980   '
990   INPUT "ENTER FLOWRATE (m^3/min) ";FL
1000   IF FL<0 OR FL>1 THEN PRINT " FLOWRATE MUST BE GREATER THAN 0 AND LESS THAN 1 m^3/min" :
GOTO 990
1010   '
1020   '
1030   INPUT "ENTER THE TUBE INSIDE DIAMETER (mm), default (15.8 mm) ";DIAM
1040   IF DIAM<0 OR DIAM>100 THEN PRINT " TUBE I.D. MUST BE BETWEEN 0 AND 100 mm " : GOTO 1030
1050   '
1060   IF DIAM = 0 THEN DIAM = 15.8
1070   DIAM = DIAM * .0032808   ' converting mm to ft
1080   '
1090   FL = FL*264.17           ' converting m^3/min to gpm
1100   '
1110   PRINT
1120   PRINT
1130   IF RA = 0 THEN GOTO 1550
1140   '
1150   '
1160   DIM AD(3,3)             'dimensioning arrays for curve fit coefficients
1170   DIM D(3,3)
1180   DIM AV(3,3)
1190   DIM V(2,3)
1200   DIM V2(3)
```

```
1210  DIM AP(3,3)
1220  DIM P(2,3)
1230  DIM P2(3)
1240  '
1250  AD(1,1) = 1.0004          'curve fit coefficients for density
1260  AD(1,2) = .17659
1270  AD(1,3) = -.049214
1280  AD(2,1) = -1.2379E-04
1290  AD(2,2) = -9.9189E-04
1300  AD(2,3) = 4.1024E-04
1310  AD(3,1) = -2.9837E-06
1320  AD(3,2) = 2.4614E-06
1330  AD(3,3) = -9.5278E-08
1340  '
1350  AV(1,1) = .55164          'curve fit coefficients for viscosity
1360  AV(1,2) = 2.6492
1370  AV(1,3) = .82935
1380  AV(2,1) = -.027633
1390  AV(2,2) = -.031496
1400  AV(2,3) = .0048136
1410  AV(3,1) = 6.0629E-17
1420  AV(3,2) = 2.2389E-15
1430  AV(3,3) = 5.879E-16
1440  '
1450  AP(1,1) = 2.5735          'curve fit coefficients for Prandtl-Numbers
1460  AP(1,2) = 3.0411
1470  AP(1,3) = .60237
1480  AP(2,1) = -.031169
1490  AP(2,2) = -.025424
1500  AP(2,3) = .0037454
1510  AP(3,1) = 1.1605E-16
1520  AP(3,2) = 2.5283E-15
1530  AP(3,3) = 2.3777E-16
1540  '
1550  IF UN = 1 THEN GOTO 1580
1560  T = (5/9)*(T-32)             'converting bulk temperature to degrees C
1570  '
1580  FOR I = 1 TO 3
1590       FOR J = 1 TO 3
1600            D(I,J) = AD(I,J)*X^(J-1)*T^(I-1)
1610       NEXT J
1620  NEXT I
1630  DENS = D(1,1)+D(1,2)+D(1,3)+D(2,1)+D(2,2)+D(2,3)+D(3,1)+D(3,2)+D(3,3)
1640  '  units are g/cm^3
1650  '
1660  '
1670  PDRTA=   -1.2379E-04 -9.9189E-04*X +4.1024E-04*X*X
1680  PDRTB=   2*((-2.9837E-06*T +2.4614E-06*X*T -9.5278E-08*X*X*T))
1690  PDRT = PDRTA+PDRTB           ' part. deriv. of density wrpt temp.
1700  BETA = -(1/DENS)*(PDRT)      ' thermal expansion coefficient
1710  BETAF = (1/BETA)*1.8
1720  BETAE = 1/BETAF
1730  '
```

```
1740   FOR I = 1 TO 2
1750       FOR J = 1 TO 3
1760           V(I,J) = AV(I,J)*X^(J-1)*T^(I-1)
1770           V2(J) = AV(3,J)*X^(J-1)
1780       NEXT J
1790   NEXT I
1800   V3 = (V2(1)+V2(2)+V2(3))^.25*T*T
1810   VISC = V(1,1)+V(1,2)+V(1,3)+V(2,1)+V(2,2)+V(2,3) + V3
1820   ABVISC = EXP(VISC)
1830   '  units are mPa*s
1840   '
1850   '
1860   FOR I = 1 TO 2
1870       FOR J = 1 TO 3
1880           P(I,J) = AP(I,J)*X^(J-1)*T^(I-1)
1890           P2(J) = AP(3,J)*X^(J-1)
1900       NEXT J
1910   NEXT I
1920   P3 = (P2(1)+P2(2)+P2(3))^.25*T*T
1930   PRAN = P(1,1)+P(1,2)+P(1,3)+P(2,1)+P(2,2)+P(2,3) + P3
1940   PR = EXP(PRAN)
1950   '
1960   KH2O = .56276+.001874*T-.0000068*T*T
1970   KETH = .24511+.0001755*T-8.52E-07*T*T
1980   F = .6635-.3698*X-.000885*T
1990   K = (1-X)*KH2O + X*KETH - F*(KH2O-KETH)*(1-X)*X
2000   ' units are  W/m*K
2010   '
2020   '
2030   CP = PR*K/ABVISC       '  units are KJ/Kg*K
2040   '
2050   '
2060   DENSE = DENS*62.428            ' coverting from S.I. to English units
2070   ABVISCE = (ABVISC/1000)*.67197
2080   KE = K*.57818
2090   CPE = CP*.23901
2100   KINVISC = (ABVISC/DENS)
2110   KINVISCE = (ABVISCE/DENSE)
2120   '
2130   '
2140   T = T*1.8 + 32
2150   IF UN = 1 THEN TW = TW * 1.8 + 32
2160   GR = ((32.174*BETAE*DIAM^3*(TW-T))/(KINVISCE^2))/10000
2170   '
2180   VELE = (FL/7.48/((3.14159/4)*DIAM^2))/60
2190   VEL = VELE * .3048
2200   RE = VELE*(DIAM)/ KINVISCE/1000
2202   MFE = DENSE * ((3.141593/4)*DIAM^2) * VELE * 3600
2206   MF = DENS * 1000 * ((3.141593/4)*(DIAM*.3048)^2) * VEL
2210   '
2220   KINVISCE = KINVISCE * 3600 * 1000      ' scaling factors
2230   ABVISCE = ABVISCE * 3600
2240   BETA = BETA*10^3
```

```
2250   BETAE = BETAE * 10^3
2260   '
2270   TD = (K/(DENS*CP)) * 100     '  thermal diffusivity w/ scaling fact.
2280   TDE = (KE/(DENSE*CPE))*1000
2290   '
2300   IF UN = 1 THEN GOTO 2510
2310   '
2320   PRINT "Density  (lbm/ft^3)        =  ";USING"###.##";DENSE
2330   PRINT "                                                 Prandtl Number          =
";USING"###.##";PR
2340   PRINT "Abs. Vis. (lbm/ft*hr)      =  ";USING"###.##";ABVISCE
2350   PRINT "                                                 Grashof Number * 10^-4  =
";USING"###.##";GR
2360   PRINT "Kin. Vis. (ft^2/hr)*10^3   =  ";USING"###.##";KINVISCE
2370   PRINT "                                                 Velocity (ft/sec)       =
";USING"###.##";VELE
2380   PRINT "Sp. Heat  (Btu/lbm*F)      =  ";USING"###.###";CPE
2390   PRINT "                                                 Reynolds No. * 10^-3    =
";USING"###.###";RE
2400   PRINT "Th. Cond. (Btu/hr*ft*F)    =  ";USING"###.###";KE
2407   PRINT "                                                 Mass Flowrate (lbm/hr)  =
";USING"####";MFE
2420   PRINT "Beta  (1/F)  * 10^3        =  ";USING"###.###";BETAE
2430   PRINT
2440   PRINT "Th. Diff. (ft^2/hr) *10^3 =  ";USING"###.##";TDE
2450   PRINT
2460   PRINT
2470   '
2480   IF UN = 0 THEN GOTO 2680
2490   '
2500   '
2510   PRINT "Density  (Kg/m^3) * 10^-3 =  ";USING"###.###";DENS
2520   PRINT "                                                 Prandtl Number          =
";USING"###.##";PR
2530   PRINT "Abs. Vis. (Kg/m*s)*10^3    =  ";USING"###.##";ABVISC
2540   PRINT "                                                 Grashof Number * 10^-4  =
";USING"###.##";GR
2550   PRINT "Kin. Vis. (m^2/s) * 10^6   =  ";USING"###.##";KINVISC
2560   PRINT "                                                 Velocity (m/s)          =
";USING"###.##";VEL
2570   PRINT "Sp. Heat (KJ/Kg*K)         =  ";USING"###.##";CP
2580   PRINT "                                                 Reynolds No. * 10^-3    =
";USING"###.##";RE
2590   PRINT "Th. Cond. (W/m*K)          =  ";USING"###.###";K
2596   PRINT "                                                 Mass Flowrate (Kg/s)    =
";USING"#.###";MF
2610   PRINT "Beta  (1/C)  * 10^3        =  ";USING"###.###";BETA
2620   PRINT
2630   PRINT "Th. Diff. (m^2/s) * 10^8   =  ";USING"###.##";TD
2640   PRINT
2650   PRINT
2660   '
2670   '
```

```
2680  INPUT "RUN AGAIN?  (Return for yes, any number for no) ";RA
2690  IF RA = 0 THEN GOTO 590
2700  '
2710  '
2720  END
```

```
program reduce_30;

uses crt,util2;


TYPE
   TT = RECORD
        T:ARRAY[1..100] OF REAL;
        TAVE:REAL;
        ST:STRING[80];
        CH:INTEGER;
        Q:STRING[2];
        END;

VAR
   TEMP:ARRAY[1..30] OF TT;
   FNAMEIN,FNAMEOUT:LINE80;
   INDAT:TEXT;
   NUM,NS:INTEGER;
   QQ:CHAR;

PROCEDURE INITIAL;
   BEGIN
   TEMP[1].ST:='T Bulk Out = '; TEMP[2].ST:='TF out =   ';
   TEMP[3].ST:='T Bulk In =  '; TEMP[4].ST:='TF in =    ';
   TEMP[5].ST:='HEin =       '; TEMP[6].ST:='HEout =    ';
   TEMP[7].ST:='dP: Tin =  '; TEMP[8].ST:='dP: Tout = ';
   TEMP[9].ST:='1a =       '; TEMP[10].ST:='1e =       ';
   TEMP[11].ST:='4a =       '; TEMP[12].ST:='4e =       ';
   TEMP[13].ST:='6a =       '; TEMP[14].ST:='6e =       ';
   TEMP[15].ST:='9a =       '; TEMP[16].ST:='9e =       ';
   TEMP[17].ST:='13a =       '; TEMP[18].ST:='13e =       ';
   TEMP[19].ST:='17a =       '; TEMP[20].ST:='17e =       ';
   TEMP[21].ST:='19a =       '; TEMP[22].ST:='19e =       ';
   TEMP[23].ST:='21a =       '; TEMP[24].ST:='21e =       ';
   TEMP[25].ST:='23a =       '; TEMP[26].ST:='23e =       ';
   TEMP[27].ST:='27a =       '; TEMP[28].ST:='27e =       ';
   TEMP[29].ST:='31a =       '; TEMP[30].ST:='31e =       ';
   END;

PROCEDURE READ_IT;
   VAR
      K,I,J,JJ:INTEGER;
      QQ:CHAR;
      ST1:STRING[1];
      ST2:STRING[2];
   BEGIN
   PROMPT_INT('Enter number of channels per sample: ',NUM); PROMPT_NAME
   ('Enter source filename: ',FNAMEIN);
   CLRSCR;
   ASSIGN(INDAT,FNAMEIN);
   RESET(INDAT);
   FOR I:=1 TO 4 DO READLN(INDAT);
```

```
   k:=0;
   repeat
   k:=k+1;
   READLN(INDAT);
   FOR J:=1 TO NUM DO READLN(INDAT,ST2,TEMP[J].Q,ST1,TEMP[J].T[K]);
   until seekeof(indat);
   CLOSE(INDAT);
   ns:=k;
   WRITELN('Number of Samples = ',NS:3);
   WRITELN;
   WRITELN;
   WRITELN('Number of Channels = ',NUM:3);
   WRITELN;
   WRITELN;
   FOR I:=1 TO NUM DO
      BEGIN
      VAL(TEMP[I].Q,TEMP[I].CH,JJ);
      IF JJ <> 0 THEN
         PROMPT_CHAR('Error in input file!! Press any key to continue: ',QQ);
      END;
   END;

PROCEDURE REDUCE_IT;
   VAR I,J:INTEGER;
   BEGIN
   FOR I:=1 TO NUM DO WITH TEMP[I] DO
      BEGIN
      TAVE:=0.0;
      FOR J:=1 TO NS DO TAVE:=TAVE + T[J];
      TAVE:=TAVE / NS;
      END;
   END;

PROCEDURE WRITE_IT;
   VAR
      que:char;
      I,J,K:INTEGER;
   BEGIN
   QQ:='Y';
   PROMPT_CHAR('Write Average data to Screen (y/n)? ',QUE);
   IF QUE <> 'N' THEN
      BEGIN
      WRITELN;
      WRITELN;
      WRITELN(' Channel #        Label & Average');
      I:=0;
      REPEAT
      I:=I+1;
      WITH TEMP[I] DO
         BEGIN
         J:=CH;
         WRITELN('  ',CH:3,'            ',TEMP[J].ST,TAVE:8:4);
         END;
```

```
      UNTIL I >= NUM;
      WRITELN;
      WRITELN;
      END;
   PROMPT_CHAR('Write Average data to file (y/n)? ',QUE);
   IF QUE <> 'N' THEN
      BEGIN
      REPEAT
      PROMPT_NAME('Enter destination filename: ',FNAMEOUT);
      UNTIL FILE_NAME_VALID(FNAMEOUT);
      ASSIGN(INDAT,FNAMEOUT);
      REWRITE(INDAT);
      WRITELN(INDAT,' Channel #       Label & Average');
      WRITELN(INDAT);
      I:=0;
      REPEAT
      I:=I+1;
      WITH TEMP[I] DO
         BEGIN
         J:=CH;
         WRITELN(INDAT,'  ',CH:3,'           ',TEMP[J].ST,TAVE:8:4);
         END;
      WRITELN(INDAT);
      UNTIL I >= NUM;
      CLOSE(INDAT);
      END;
   END;

BEGIN
REPEAT
CLRSCR;
INITIAL;
READ_IT;
REDUCE_IT;
WRITE_IT;
PROMPT_CHAR('Reduce another (y/n)? ',QQ);
UNTIL QQ = 'N';
END.
```

```
program reduce;  (RED96)
uses crt,turbo3,UTIL2;
{ THIS PROGRAM READS INPUT FROM A DATA FILE PRODUCED BY ACQCOMP, AVERAGES }
{ OVER THE NUMBER OF SAMPLES PROVIDED IN THE FILE, AND PROVIDES           }
{ TWO FORMS OF OUTPUT. THE FIRST FORM OF OUTPUT OFFERED TO THE USER       }
{ IS COMPATIBLE WITH THE WALL TEMP GRADIENT CORRECTION PROGRAM AND THE    }
{ SECOND PRODUCES A DATA FILE FOR EACH STATION CONTAINING THE AVERAGE     }
{ CORRECTED VALUE AND ALL OF THE DATA FOR EACH T.C. AT THAT STATION. THE  }
{ PROGRAM APPENDS THE USER INPUT FILENAME WITH '.001' FOR STATION 1, ETC. }
{ AN I/O ERROR WILL OCCUR IF THE USER INPUT FILE NAME IS LONGER THAN 8    }
{ CHAR OR CONTAINS A PERIOD (.). THIS PROGRAM WAS WRITTEN BY K.D.HOWARD   }
{ IN TURBOPASCAL 3.0.                                                      }


CONST MAX = 97; STATIONS = 26;  { TOTAL # STATIONS ACTUALLY BEING USED }
VAR
   INDAT:TEXT;        { FILE SPECIFIER: I/O }
   I,run:INTEGER;
   FNAME:LINE80;      { FILE VARIABLE: I/O }
   DAT:ARRAY[1..100,1..MAX] OF REAL; { ARRAY FOR ACQCOMP FORMATTED DATA }
   CORR,TEMP:ARRAY[1..MAX] OF REAL; { TEMP = ARRAY FOR AVERAGED TEMP'S }
   { CORR = ARRAY OF CORRECTION VALUES FROM CALIBRATION } STNUM,NUMTC:ARRAY[1..31] OF
   INTEGER; { STATION NUMBER AND T.C. # ARRAY'S } NUMREP,{ NUMREP = # SAMPLES INPUT FOR
   EACH CHANNEL }
   NUMCH, { NUMCH = # OF CHANNELS INPUT }
   INDEX:INTEGER; { INDEX = FLUID INDEX: 1 FOR PURE H2O OR 2 FOR EG MIX } Z:ARRAY[1..31]
   OF REAL; { Z = ARRAY OF Z (OR X) LOCATIONS IN INCHES } TIME:STRING[10]; { DUMMY
   VARIABLE WHICH READS HEADER FROM EACH LINE INPUT } ST:STRING[3];
   ST1:STRING[2]; CONC,GPM,CURRENT,VOLTAGE,BTEMP,INTEMP,OUTTEMP,HEINTEMP,CWINTEMP,
   TIN,TOUT,ROOMTEMP:REAL;
   { RUN = RUN INDEX (ANY NUMERIC LABEL); CONC = MASS CONCENTRATION OF EG (%) }
   { GPM = FLOW RATE IN GPM; CURRENT = SOURCE CURRENT IN AMPS; VOLTAGE =      }
   { VOLTAGE IN VOLTS; BTEMP = BATH TEMP; INTEMP = INLET TEMP; OUTTEMP = EXIT }
   { TEMP; HEINTEMP = HEAT EXCHANGER INLET TEMP; CWINTEMP = COOLANT WATER     }
   { INLET TEMP; ROOMTEMP = ROOM TEMP; TIN = INLET BULK TEMP                  }
   { TOUT = OUTLET BULK TEMP.                                                  }
   Q:CHAR;

PROCEDURE READ_CORRDAT; { THIS PROCEDURE READS DATA FROM THE CORRECTION }
{ FACTOR DATA FILE 'C:CORR.DAT' .                                        }

   VAR I,J:INTEGER;

   BEGIN
   REPEAT
   FNAME:='C:CORR.DAT';
   IF NOT FILE_EXISTS(fname) THEN
      PROMPT_CHAR('CORR.DAT MUST BE PLACED IN THE ACQCOMP DIRECTORY! ',Q);
   UNTIL FILE_EXISTS(FNAME);
   ASSIGN(INDAT,FNAME);
   RESET(INDAT);
   FOR I:=1 TO 3 DO READLN(INDAT);
   I:=2;
   REPEAT
   READ(INDAT,J);
```

```
   FOR J:=1 TO 4 DO
      BEGIN
      I:=I+1;
      READ(INDAT,CORR[I]);
      END;
   READLN(INDAT);
   UNTIL I >= 86;
   REPEAT
   READ(INDAT,J);
   FOR J:=1 TO 2 DO
      BEGIN
      I:=I+1;
      READ(INDAT,CORR[I]);
      END;
   READLN(INDAT);
   UNTIL I = 96;
   CLOSE(INDAT);
   END;  { OF READ_CORRDAT }

PROCEDURE READ_IT; { THIS PROCEDURE READS DATA FROM A FILE PRODUCED BY        }
{ ACQCOMP. THE NUMBER OF CHANNELS IS DETERMINED BY THE NUMBER OF TEMP'S       }
{ IN THE FIRST LINE OF DATA ( THE PRELIMINARIES ARE SKIPPED BY THE PROGRAM. }
{ DATA IS THEN READ INTO THE ARRAY 'DAT' UNTIL AN EOF SEQUENCE IS DETECTED AT }
{ WHICH POINT INPUT IS TERMINATED. A COUNTER IS USED IN THE FIRST LINE OF     }
{ INPUT TO DETERMINE THE NUMBER OF CHANNELS IN THE FILE AND ANOTHER IS USED   }
{ TO DETERMINE THE NUMBER OF SAMPLES IN THE FILE (# LINES OF DATA). THE VALUE }
{ OF THESE COUNTERS IS OUTPUT TO SCREEN FOR THE USER'S CONVENIENCE AS A CHECK }
{ **NOTE: THE DUMMY VARIABLES TIME, ST1 AND ST2 ARE FORMATTED READ STATEMENTS }
{ WHICH READ IN COMMA'S AND SPACES WHICH ACQCOMP INSERTS INTO ITS OUTPUT      }
{ FILES FOR SOME UNGODLY REASON - THESE WERE SIMPLED VARIED UNTIL THE FILES   }
{ WERE INPUT PROPERLY.                                                        }

   VAR I,J:INTEGER;

   BEGIN
   CLRSCR;
   REPEAT              { PROMPT FOR SOURCE FILENAME }
   PROMPT_NAME('Enter source filename: ',FNAME);
   UNTIL FILE_EXISTS(FNAME);
   ASSIGN(INDAT,FNAME);
   RESET(INDAT);
   READLN(INDAT);      { SKIP THE PRELIMINARY ITEMS }
   READLN(INDAT);
   READLN(INDAT);
   READLN(INDAT);
   J:=1;               { INITIALIZE # SAMPLE COUNTER }
   I:=0;               { INITIALIZE # CHANNEL COUNTER }
   READ(INDAT,TIME);  { READ HEADER INTO DUMMY VARIABLE }
   FOR I:=1 TO 96 DO
      READ(INDAT,ST1,DAT[J,I]); { ST1 READS IN THE COMMA BTWM TEMP'S }
   NUMCH:=I;
   WRITELN('Number of Channels = ',numch:2);
   WRITELN;
   READLN(INDAT);
   REPEAT
```

```
J:=J+1;      { INCREMENT # SAMPLE COUNTER }
GOTOXY(1,24); { OUTPUT # OF SAMPLE BEING READ TO SCREEN }
WRITE('Sample ',J:3);
READ(INDAT,TIME);
FOR I:=1 TO NUMCH DO READ(INDAT,ST1,DAT[J,I]);
READLN(INDAT);
UNTIL (SEEKEOF(INDAT)) OR (J = 100);
CLOSE(INDAT);
NUMREP:=J;   { # OF SAMPLES = NUMREP }
WRITELN;
WRITELN;
PROMPT_CHAR('Data File has been read... Press any key to continue: ',Q);
END;   { OF READ_IT }

PROCEDURE INITIAL; { THIS PROCEDURE INITIALIZES CHANNEL NUMBER AND STATION    }
{ NUMBER ARRAYS; AND PROMPTS THE USER FOR TEST INFORMATION. THE USER IS       }
{ PROMPTED FOR ONE PIECE OF INFO AT A TIME; WHEN FINISHED THE IPUT DATA IS    }
{ OUTPUT TO SCREEN SUCH THAT THE USER MAY SEE THE INPUTS. IF THE DATA IS      }
{ INCORRECT THEN THE USER MAY REPEAT THE INPUT SESSION, OTHERWISE THE PROGRAM }
{ INITIALIZES SEVERAL VARIABLES AND PROCEEDS TO THE NEXT PTOCEDURE.           }

BEGIN
CLRSCR;
REPEAT
PROMPT_int('Enter run index (number): ',RUN);
PROMPT_INT('Enter fluid index (1-pure water or 2-EG): ',INDEX); PROMPT_REAL
('Enter mass concentration of Ethylene Glycol: ',CONC); PROMPT_REAL
('Enter flow rate in GPM: ',GPM);
PROMPT_REAL('Enter source current in Amps: ',CURRENT);
PROMPT_REAL('Enter source voltage in Volts: ',VOLTAGE);
PROMPT_REAL('Enter room temperature (deg. F): ',ROOMTEMP);
PROMPT_REAL('Enter Inlet bulk temperature: ',TIN);
PROMPT_REAL('Enter Outlet bulk temperature: ',TOUT);
CLRSCR;
WRITELN;
WRITELN;
WRITELN;
WRITELN;
WRITELN;
WRITELN;
WRITE('            ');
WRITELN('               Run Index = ',RUN:12);
WRITE('            ');
WRITELN('             Fluid Index = ',INDEX:12);
WRITE('            ');
WRITELN('           Mass Conc. EG = ',CONC:12:4);
WRITE('            ');
WRITELN('               Flow Rate = ',GPM:12:4);
WRITE('            ');
WRITELN('          Source Current = ',CURRENT:12:4);
WRITE('            ');
WRITELN('          Source Voltage = ',VOLTAGE:12:4);
WRITE('            ');
WRITELN('              Room Temp. = ',roomtemp:12:4);
WRITE('            ');
```

```
WRITELN('        Inlet Bulk Temp. = ',TIN:12:4);
WRITE('           ');
WRITELN('       Outlet Bulk Temp. = ',TOUT:12:4);
WRITELN;
WRITELN;
PROMPT_CHAR('Everything OK (y/n)? ',Q);
UNTIL Q <> 'N';

{ THIS SECTION SETS THE Z-LOCATION ALONG THE TUBE IN INCHES }
Z[1]:=2.0;Z[2]:=4.0;Z[3]:=6.0;Z[4]:=8.0;Z[5]:=12.0;Z[6]:=16.0;
Z[7]:=20.0;Z[8]:=24.0;Z[9]:=28.0;Z[10]:=32.0;Z[11]:=36.0;Z[12]:=40.0;
Z[13]:=44.0;Z[14]:=48.0;Z[15]:=52.0;Z[16]:=56.0;Z[17]:=60.0;Z[18]:=72.0;
Z[19]:=84.0;Z[20]:=96.0;Z[21]:=108.0;Z[22]:=120.0;Z[23]:=132.0;Z[24]:=144.0;
Z[25]:=156.0;Z[26]:=168.0;Z[27]:=180.0;Z[28]:=192.0;Z[29]:=204.0;
Z[30]:=216.0;Z[31]:=228.0;

{ THIS SECTION SETS TH NUMBER OF T.C.'S AT EACH STATION }
NUMTC[1]:=4;NUMTC[2]:=4;NUMTC[3]:=4;NUMTC[4]:=4;NUMTC[5]:=4;
NUMTC[6]:=4;NUMTC[7]:=4;NUMTC[8]:=4;NUMTC[9]:=4;NUMTC[10]:=4;
NUMTC[11]:=4;NUMTC[12]:=4;NUMTC[13]:=4;NUMTC[14]:=4;NUMTC[15]:=4;
NUMTC[16]:=4;NUMTC[17]:=4;NUMTC[18]:=4;NUMTC[19]:=4;NUMTC[20]:=4;
NUMTC[21]:=4;NUMTC[22]:=4;NUMTC[23]:=2;NUMTC[24]:=2;NUMTC[25]:=2;
NUMTC[26]:=2;NUMTC[27]:=2;NUMTC[28]:=2;NUMTC[29]:=2;NUMTC[30]:=2;
NUMTC[31]:=2;

{ THIS SECTION SETS THE STATION NUMBERS }
STNUM[1]:=1;STNUM[2]:=2;STNUM[3]:=3;STNUM[4]:=4;STNUM[5]:=5;
STNUM[6]:=6;STNUM[7]:=7;STNUM[8]:=8;STNUM[9]:=9;STNUM[10]:=10;
STNUM[11]:=11;STNUM[12]:=12;STNUM[13]:=13;STNUM[14]:=14;STNUM[15]:=15;
STNUM[16]:=16;STNUM[17]:=17;STNUM[18]:=18;STNUM[19]:=19;STNUM[20]:=20;
STNUM[21]:=21;STNUM[22]:=22;STNUM[23]:=23;STNUM[24]:=24;STNUM[25]:=25;
STNUM[26]:=26;STNUM[27]:=27;STNUM[28]:=28;STNUM[29]:=29;STNUM[30]:=30;
STNUM[31]:=31;

END; { OF INITIAL }

PROCEDURE REDUCE_IT; { THIS PROCEDURE COMPUTES A SIMPLE AVERAGE FOR EACH }
{ CHANNEL.                                                               }
   VAR I,J:INTEGER;
   BEGIN
   FOR I:=1 TO NUMCH DO
      BEGIN
      TEMP[I]:=0.0;
      FOR J:=1 TO NUMREP DO TEMP[I]:=TEMP[I] + DAT[J,I];
      TEMP[I]:=TEMP[I] / NUMREP - CORR[I];
      END;
   ROOMTEMP:=0.5*(TEMP[1] + TEMP[2]);
   END;  { OF REDUCE_IT }

PROCEDURE WRITE_IT;  { THIS PROCEDURE OUTPUTS THE REDUCED DATA IN TWO FORMS: }
{ 1) IN FORMAT COMPATABLE WITH THE TUBE WALL TEMP' GRADIENT CORRECTION       }
{ PROGRAM; AND 2) A SERIES OF FILES - ONE FOR EACH STATION - CONTAINING DATA }
{ FOR EACH STATION. THE LATTER FILE IS INTENDED FOR ANALYZING THE CHARACTER- }
{ ISTICS OF THE T.C.'S PER STATION. A THE DIGIT CODE IS APPENDED TO EACH OF  }
{ FILES INDICATING THE STATION NUMBER. NOTE THAT AN I/O ERROR WILL OCCUR IF  }
```

```
{ USER INPUTS MORE THAN 8 CHAR AND/OR A PERIOD '.' FOR THE FILENAME IN THE   }
{ LATTER CASE.                                                                }

   VAR
      I,J,K:INTEGER;
      FNAME1:LINE80;

   BEGIN
   {BEGIN SECTION FOR 'CORRECTION PROGRAM' FORMATTED OUTPUT }
   REPEAT
   PROMPT_NAME('Enter destination filename for H.T. Data: ',FNAME);
   UNTIL FILE_NAME_VALID(FNAME);
   ASSIGN(INDAT,FNAME);
   REWRITE(INDAT);
   WRITELN(INDAT,RUN:4,STATIONS:3);
   WRITELN(INDAT,
      INDEX:2,CONC:7:2,GPM:7:4,CURRENT:7:2,VOLTAGE:7:2, TIN:7:2,TOUT:7:2,ROOMTEMP:7:2);
   K:=2;
   FOR I:=1 TO 1 DO
      BEGIN
      WRITE(INDAT,STNUM[I]:3,NUMTC[I]:3,Z[I]:9:2);
      FOR J:=1 TO 4 DO
         BEGIN
         K:=K+1;
         WRITE(INDAT,TEMP[K]:7:2);
         END;
      END;
   WRITELN(INDAT);
   FOR I:=3 TO 22 DO
      BEGIN
      WRITE(INDAT,STNUM[I]:3,NUMTC[I]:3,Z[I]:9:2);
      FOR J:=1 TO 4 DO
         BEGIN
         K:=K+1;
         WRITE(INDAT,TEMP[K]:7:2);
         END;
      WRITELN(INDAT);
      END;
   I:=21;
   REPEAT
   I:=I+2;
   WRITE(INDAT,STNUM[I]:3,NUMTC[I]:3,Z[I]:9:2);
   FOR J:=1 TO 2 DO
      BEGIN
      K:=K+1;
      WRITE(INDAT,TEMP[K]:7:2);
      END;
   WRITELN(INDAT);
   UNTIL I = 31;
   for i:=1 to 7 do write(indat,'0');
   writeln(indat);
   CLOSE(INDAT);
   { BEGIN SECTION FOR DATA BY STATION: MUCH CODE REQUIRED }
   PROMPT_CHAR('Do you wish to file ''data by Station'' (y/n)? ',Q);
   IF Q = 'Y' THEN
```

```
BEGIN
REPEAT
PROMPT_NAME('Enter destination filename for Data by Station: ',FNAME); UNTIL
FILE_NAME_VALID(FNAME);
FNAME1:=FNAME+'.TIN';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; TIN AND TOUT');
WRITELN(INDAT,'AVE=>       ',TEMP[1]:12:4,' ',TEMP[2]:12:4);
WRITELN(INDAT,'TIME        TIN         TOUT');
FOR I:=1 TO NUMREP DO WRITELN(INDAT,DAT[I,1]:12:4,
                        ' ',DAT[I,2]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.001';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 1');
WRITE(INDAT,'AVE=>       ');
FOR I:=3 TO 6 DO WRITE(INDAT,TEMP[I]:12:4,' ');
WRITELN(INDAT);
WRITELN(INDAT,'TIME        A           C             E            G');
FOR I:=1 TO NUMREP DO
   WRITELN(INDAT,DAT[I,3]:12:4,' ',DAT[I,4]:12:4,
        ' ',DAT[I,5]:12:4,' ',DAT[I,6]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.003';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 3');
WRITE(INDAT,'AVE=>       ');
FOR I:=7 TO 10 DO WRITE(INDAT,TEMP[I]:12:4,' ');
WRITELN(INDAT);
WRITELN(INDAT,'TIME        A           C             E            G');
FOR I:=1 TO NUMREP DO
   WRITELN(INDAT,DAT[I,7]:12:4,' ',DAT[I,8]:12:4,
        ' ',DAT[I,9]:12:4,' ',DAT[I,10]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.004';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 4');
WRITELN(INDAT,'A            C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,11]:12:4,' ',DAT[I,12]:12:4,' ',DAT[I,13]:12:4,' ',
        DAT[I,14]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.005';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 5');
WRITELN(INDAT,'A            C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,15]:12:4,' ',DAT[I,16]:12:4,' ',DAT[I,17]:12:4,' ',
        DAT[I,18]:12:4);
CLOSE(INDAT);
```

```
FNAME1:=FNAME+'.006';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 6');
WRITELN(INDAT,'A             C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,19]:12:4,' ',DAT[I,20]:12:4,' ',DAT[I,21]:12:4,' ',
        DAT[I,22]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.007';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 7');
WRITELN(INDAT,'A             C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,23]:12:4,' ',DAT[I,24]:12:4,' ',DAT[I,25]:12:4,' ',
        DAT[I,26]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.008';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 8');
WRITELN(INDAT,'A             C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,27]:12:4,' ',DAT[I,28]:12:4,' ',DAT[I,29]:12:4,' ',
        DAT[I,30]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.009';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 9');
WRITELN(INDAT,'A             C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,31]:12:4,' ',DAT[I,32]:12:4,' ',DAT[I,33]:12:4,' ',
        DAT[I,34]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.010';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 10');
WRITELN(INDAT,'A             C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,35]:12:4,' ',DAT[I,36]:12:4,' ',DAT[I,37]:12:4,' ',
        DAT[I,38]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.011';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 11');
WRITELN(INDAT,'A             C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,39]:12:4,' ',DAT[I,40]:12:4,' ',DAT[I,41]:12:4,' ',
        DAT[I,42]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.012';
```

```
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 12');
WRITELN(INDAT,'A             C             E            G');
FOR I:=1 TO NUMREP DO
   WRITELN(INDAT,DAT[I,43]:12:4,' ',DAT[I,44]:12:4,' ',DAT[I,45]:12:4,' ',
        DAT[I,46]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.013';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 13');
WRITELN(INDAT,'A             C             E            G');
FOR I:=1 TO NUMREP DO
   WRITELN(INDAT,DAT[I,47]:12:4,' ',DAT[I,48]:12:4,' ',DAT[I,49]:12:4,' ',
        DAT[I,50]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.014';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 14');
WRITELN(INDAT,'A             C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,51]:12:4,' ',DAT[I,52]:12:4,' ',DAT[I,53]:12:4,' ',
        DAT[I,54]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.015';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 15');
WRITELN(INDAT,'A             C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,55]:12:4,' ',DAT[I,56]:12:4,' ',DAT[I,57]:12:4,' ',
        DAT[I,58]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.016';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 16');
WRITELN(INDAT,'A             C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,59]:12:4,' ',DAT[I,60]:12:4,' ',DAT[I,61]:12:4,' ',
        DAT[I,62]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.017';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 17');
WRITELN(INDAT,'A             C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,63]:12:4,' ',DAT[I,64]:12:4,' ',DAT[I,65]:12:4,' ',
        DAT[I,66]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.018';
ASSIGN(INDAT,FNAME1);
```

```
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 18');
WRITELN(INDAT,'A            C             E            G');
FOR I:=1 TO NUMREP DO
   WRITELN(INDAT,DAT[I,67]:12:4,' ',DAT[I,68]:12:4,' ',DAT[I,69]:12:4,' ',
        DAT[I,70]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.019';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 19');
WRITELN(INDAT,'A            C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,71]:12:4,' ',DAT[I,72]:12:4,' ',DAT[I,73]:12:4,' ',
        DAT[I,74]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.020';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 20');
WRITELN(INDAT,'A            C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,75]:12:4,' ',DAT[I,76]:12:4,' ',DAT[I,77]:12:4,' ',
        DAT[I,78]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.021';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 21');
WRITELN(INDAT,'A            C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,79]:12:4,' ',DAT[I,80]:12:4,' ',DAT[I,81]:12:4,' ',
        DAT[I,82]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.022';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 22');
WRITELN(INDAT,'A            C             E            G');
FOR I:=1 TO NUMREP DO
  WRITELN(INDAT,DAT[I,83]:12:4,' ',DAT[I,84]:12:4,' ',DAT[I,85]:12:4,' ',
        DAT[I,86]:12:4);
CLOSE(INDAT);
FNAME1:=FNAME+'.023';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 23');
WRITELN(INDAT,'A            E');
FOR I:=1 TO NUMREP DO
   WRITELN(INDAT,DAT[I,87]:12:4,' ',DAT[I,88]:12);
CLOSE(INDAT);
FNAME1:=FNAME+'.025';
ASSIGN(INDAT,FNAME1);
REWRITE(INDAT);
WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 25');
```

```
      WRITELN(INDAT,'A            E');
      FOR I:=1 TO NUMREP DO
         WRITELN(INDAT,DAT[I,89]:12:4,' ',DAT[I,90]:12);
      CLOSE(INDAT);
      FNAME1:=FNAME+'.027';
      ASSIGN(INDAT,FNAME1);
      REWRITE(INDAT);
      WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 27');
      WRITELN(INDAT,'A            E');
      FOR I:=1 TO NUMREP DO
         WRITELN(INDAT,DAT[I,91]:12:4,' ',DAT[I,92]:12);
      CLOSE(INDAT);
      FNAME1:=FNAME+'.029';
      ASSIGN(INDAT,FNAME1);
      REWRITE(INDAT);
      WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 29');
      WRITELN(INDAT,'A            E');
      FOR I:=1 TO NUMREP DO
         WRITELN(INDAT,DAT[I,93]:12:4,' ',DAT[I,94]:12);
      CLOSE(INDAT);
      FNAME1:=FNAME+'.031';
      ASSIGN(INDAT,FNAME1);
      REWRITE(INDAT);
      WRITELN(INDAT,'RUN INDEX = ',RUN:10,'; STATION 31');
      WRITELN(INDAT,'A            E');
      FOR I:=1 TO NUMREP DO
         WRITELN(INDAT,DAT[I,95]:12:4,' ',DAT[I,96]:12);
      CLOSE(INDAT);
      END;
   END; { OF WRITE_IT }

BEGIN         { MAIN PROGRAM }
REPEAT
READ_IT;      { CALL INPUT PROCEDURE }
INITIAL;      { CALL INITIALIZATION PROCEDURE }
PROMPT_CHAR('Read Calibration Data from CORR.DAT (y/n)? ',Q);
IF Q <> 'N' THEN READ_CORRDAT { CALL CORRECTION FACTOR INPUT PROCEDURE }
   ELSE FOR I:=1 TO 96 DO CORR[I]:=0.0;
CORR[1]:=0.0;
CORR[2]:=0.0;
REDUCE_IT;    { CALL DATA REDUCTION PROCEDURE }
CLRSCR;
WRITELN;
WRITELN;
WRITELN;
WRITELN('Number of Channels = ',NUMCH:12);
WRITELN;
WRITELN('Number of Samples = ',NUMREP:12);
WRITELN;
WRITELN('Test Section Inlet Temp = ',TEMP[1]:12:4);
WRITELN;
WRITELN('Test Section Outlet Temp = ',TEMP[2]:12:4);
WRITELN;
WRITELN;
WRITE_IT;     { CALL OUTPUT PROCEDURE }
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PROMPT_CHAR('Reduce another file (y/n)? ',Q);
UNTIL Q = 'N';
END.          { OF MAIN PROGRAM }
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APPENDIX D

CORRELATION COMPARISON SUPPLEMENT

This appendix presents the data assembled from a comparison of the established correlations of laminar mixed convection, turbulent forced convection, and the mixed convection heat transfer equations of Chen (square-edged inlet). The comparisons are presented in the form of Tables XIII and XIV, giving maximum and absolute average percent deviations against our data runs.

The 82 data runs collected were split into two regions for comparison; the laminar and lower transition flow region and the upper transition and turbulent flow region. By splitting the data into these regions, the laminar and lower transition comparisons occurred over a maximum of 57 data runs. Similarly the upper transition and turbulent region correlations were compared with 33 of the data sets, leaving a remainder of eight runs being evaluated in both regions (see Chapter III).

The tables present first the investigators name and then the date the respective correlation was established such that the equation can be identified in Tables I through IV, of Chapter I.

TABLE XIII

LAMINAR AND LOWER TRANSITION FLOW FORCED AND MIXED COMPARISONS

COMPARISON OF INVESTIGATOR(S): SIEDER & TATE (1936)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
2679	-31.0	6.5
2681	-30.4	16.4
2674	-38.0	14.3
2673	-35.0	10.8
2671	-30.0	15.4
2007	-46.0	23.2
2667	-44.0	15.8

COMPARISON OF INVESTIGATOR(S): MORI ET AL (1966)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
2652	-30.8	27.8
2662	-30.9	25.2
2657	-29.2	23.7
2658	-25.2	19.4
2614	18.0	10.2
2683	9.1	5.2
2010	-23.5	10.2

TABLE XIII CONTINUED

COMPARISON OF INVESTIGATOR(S): COLBURN (1933)		
RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
2682	30.9	11.7
2677	26.2	12.1
2679	25.2	13.9
2681	-25.4	10.2
2676	33.0	15.9
2675	24.7	11.9
2674	18.3	9.9
2673	-19.1	9.0
2603	-37.7	19.0
2671	21.0	8.7
2009	-40.2	14.5
2666	-31.8	11.3
2668	28.1	10.2
2007	-23.4	10.0
2667	-28.3	12.2
2006	-23.8	10.5
2004	22.3	7.2
2002	-23.0	6.4
2005	-28.9	9.1

COMPARISON OF INVESTIGATOR(S): COLBURN (1933)		
RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
2682	-31.8	17.0
2677	-28.2	17.4
2676	-40.7	17.9
2675	28.7	16.5
2674	34.2	23.2
2671	31.4	21.7
2009	-30.2	11.9
2666	-31.9	16.8
2007	35.2	21.9

TABLE XIII CONTINUED

COMPARISON OF INVESTIGATOR(S): GNIELINSKI (1976)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
2655	4.4	2.6
2656	4.8	2.7
1018	13.1	9.4
2662	- 4.6	2.5
2657	4.2	3.0
2658	5.5	3.4
1019	17.8	10.7
2614	-10.1	4.4
2683	18.3	8.8
2609	-14.7	2.9
2613	28.2	8.3
2684	25.7	16.9
2608	17.1	4.6
2011	15.5	5.0
2010	28.7	7.3

COMPARISON OF INVESTIGATOR(S): FRIEND & METZNER (1958)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
2655	19.6	14.5
2656	20.8	15.4
1018	14.7	9.5
2662	31.6	16.6
2656	22.5	15.4
2658	23.8	18.5
1019	22.2	14.8
2614	22.4	15.5
2001	27.9	12.8

COMPARISON OF INVESTIGATOR(S): CHURCHILL (1977b)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
2655	28.8	24.5
2656	27.0	25.4
2662	31.6	26.8
2657	33.0	25.4

TABLE XIII CONTINUED

COMPARISON OF INVESTIGATOR(S): CHEN (1988a)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
2682	17.8	5.6
2677	22.4	7.6
2679	48.1	20.5
2681	27.5	10.3
2676	11.9	4.0
2675	24.0	7.4
2603	13.9	7.1
2671	38.1	13.9
2604	-13.4	8.5
2666	26.8	7.8
2668	35.0	12.8
2007	-20.7	11.4
2667	40.5	16.2
2606	-17.7	10.8
2006	-20.6	11.4
2605	-15.9	9.0
2004	-23.5	12.6
2002	-21.5	11.5
2005	-20.0	10.2
2003	-19.2	9.7
2001	-19.7	5.8

COMPARISON OF INVESTIGATOR(S): CHEN (1988b)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
2655	89.4	8.5
2652	83.1	11.6
2656	84.9	8.1
1018	44.6	4.9
2662	84.2	7.4
2686	86.2	3.7
2657	82.5	7.7
2658	78.1	7.1
1019	38.9	7.1
2614	68.5	6.5
2685	72.7	6.1
2683	65.1	8.4
2691	66.4	15.7
2609	54.2	4.8
2613	55.2	11.4
2692	59.2	23.2
2684	60.5	23.4

TABLE XIII CONTINUED

COMPARISON OF INVESTIGATOR(S): CHEN (1988d)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
2652	35.0	5.2
2656	23.0	9.5
2662	21.2	10.7
2657	-18.2	12.4
2658	-21.9	15.2
2614	-23.0	15.9
2683	-30.2	20.1
2609	-22.3	17.2
2613	-35.7	24.9
2684	-27.6	17.2
2608	-26.7	18.4
2612	-25.2	5.6
2009	-37.7	19.7
2604	21.2	11.0
2666	24.0	7.7
2668	25.6	8.8
2606	-19.4	12.2
2605	-17.8	10.2
2004	-37.7	26.0
2002	-34.9	21.9
2005	-22.9	13.4
2003	-12.3	5.5
2001	38.4	27.8

TABLE XIV

UPPER TRANSITION AND TURBULENT FLOW FORCED
CONVECTION COMPARISONS

COMPARISON OF INVESTIGATOR(S): SIEDER & TATE (1936)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
1005	-10.1	6.6
1006	-14.3	9.1
1004	- 9.1	5.7
1007	17.9	9.4
1014	- 6.9	3.1
1008	-11.1	7.6
1015	- 8.7	4.1
1016	-12.5	7.6
1009	- 4.3	1.8
2654	-18.0	14.3
1010	- 6.7	2.8
1012	- 6.0	2.0
2663	-17.7	13.9
1013	- 4.3	2.1
2659	-11.5	8.8
2601	-11.6	7.4
2664	-16.0	6.8
2602	-10.2	6.0
2660	- 9.5	6.4
2610	-12.8	8.2
2665	-13.6	6.8
1017	12.3	9.0
2661	- 7.1	3.4
2655	- 6.3	2.4
2656	- 5.6	2.3
1018	16.6	12.1
2662	- 5.2	2.5
2657	4.6	2.6

TABLE XIV CONTINUED

COMPARISON OF INVESTIGATOR(S): DITTUS-BOELTER (1930)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
1005	11.5	3.0
1006	7.3	3.7
1004	6.5	3.5
1007	30.1	21.5
1014	13.9	9.2
1008	6.6	2.9
1015	11.0	6.5
1016	- 6.2	2.6
1009	12.4	9.1
2654	7.1	4.5
1010	10.0	6.4
1012	11.2	7.8
2663	- 3.4	1.9
1013	14.2	9.4
2659	15.0	12.0
2601	15.0	10.9
2664	5.8	3.1
2602	16.8	12.2
2660	17.0	14.5
2610	13.2	9.6
2665	10.9	6.1
1017	22.8	18.6
2661	20.6	17.4
2655	23.5	18.9
2656	24.5	19.7
1018	27.0	21.7
2662	26.0	20.9
2657	25.7	21.0

TABLE XIV CONTINUED

COMPARISON OF INVESTIGATOR(S): SLEICHER & ROUSE (1975)		
RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
1005	22.9	12.6
1006	17.1	10.1
1004	15.9	12.5
1014	23.8	17.8
1008	15.1	10.0
1015	18.7	14.0
1016	9.6	6.4
1009	19.2	15.8
2654	12.8	9.2
1010	16.4	13.1
1012	17.7	14.6
2663	5.8	3.4
1013	20.6	16.8
2659	16.5	13.7
2601	17.3	13.8
2664	8.2	5.2
2602	19.0	15.0
2660	18.3	16.0
2610	15.6	12.4
2665	12.8	8.8
2661	21.0	18.5
2655	23.6	19.7
2656	24.6	20.5
2662	26.0	21.7
2657	25.7	21.8

TABLE XIV CONTINUED

COMPARISON OF INVESTIGATOR(S): McADAMS (1954)		
RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
1005	11.5	3.0
1006	7.3	3.7
1004	9.2	3.5
1007	30.1	21.5
1014	13.9	9.2
1008	6.6	2.9
1015	11.0	6.5
1016	- 6.2	2.6
1009	12.8	9.1
2654	7.1	4.5
1010	10.0	6.4
1012	11.2	7.8
2663	3.4	1.9
1013	14.2	9.4
2659	14.3	12.0
2601	15.0	10.9
2664	6.2	3.1
2602	16.8	12.2
2660	17.1	14.5
2610	13.2	9.6
2665	10.9	6.3
1017	22.8	18.6
2661	20.6	17.4
2655	23.5	18.9
2656	24.5	19.7
1018	27.0	21.7
2662	26.0	20.9
2657	25.7	21.1

TABLE XIV CONTINUED

COMPARISON OF INVESTIGATOR(S): McADAMS (1954b)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
1005	11.5	3.0
1006	6.0	3.7
1004	9.2	3.5
1007	30.1	21.5
1014	14.9	9.2
1008	6.5	2.9
1015	11.0	6.5
1016	- 6.2	3.7
1009	12.7	9.1
2654	7.1	4.5
1010	10.0	6.4
1012	11.2	7.8
2663	- 4.0	1.9
1013	14.2	9.4
2659	15.0	12.0
2601	15.0	10.9
2664	5.8	3.1
2602	16.8	12.2
2660	17.1	14.5
2610	13.2	9.6
2665	23.5	6.3
1017	22.8	18.6
2661	20.6	17.4
2655	23.5	18.9
2656	24.5	19.7
1018	26.4	21.7
2662	26.0	20.9
2657	25.7	21.0

TABLE XIV CONTINUED

COMPARISON OF INVESTIGATOR(S): NUSSELT (1931)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
1005	24.6	12.9
1006	26.8	11.2
1004	27.0	13.8
1008	28.4	12.4
1016	24.8	10.3
1009	36.5	19.9
2654	16.1	5.9
1010	33.0	17.1
1012	33.2	18.4
2663	-11.7	5.5
1013	35.4	20.4
2659	20.1	7.7
2601	24.8	8.7
2664	10.4	5.7
2602	26.8	9.8
2660	24.2	9.7
2610	23.2	8.3
2665	14.8	7.2
2661	28.7	12.3
2655	31.4	13.7
2656	32.2	14.5
2662	33.6	15.8
2657	33.4	16.0

TABLE XIV CONTINUED

COMPARISON OF INVESTIGATOR(S): vonKRAMAN (1934)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
1005	22.2	11.3
1006	15.5	8.8
1004	18.9	10.8
1007	39.3	28.7
1014	21.5	16.2
1008	15.1	9.6
1015	17.1	13.2
1016	7.7	5.3
1009	17.6	14.2
2654	-23.1	21.6
1010	13.4	11.3
1012	15.4	11.7
2653	25.0	22.4
2663	-29.4	25.5
1013	16.5	13.2
2659	-25.3	23.5
2601	-20.5	18.2
2664	-28.4	23.8
2602	-19.6	17.2
2660	-23.9	22.1
2610	-21.1	18.6
2665	-26.6	21.0
1017	24.3	21.3
2661	-22.6	20.8
2655	-21.2	20.4
2652	19.8	1.9
2656	-21.5	19.8
1018	27.4	24.0
2662	-20.7	18.8
2657	-20.8	18.9

TABLE XIV CONTINUED

COMPARISON OF INVESTIGATOR(S): GNEILINSKI (1976)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
1005	27.0	17.0
1006	22.0	14.5
1004	23.1	16.4
1014	28.0	21.7
1008	17.6	12.7
1015	21.6	16.7
1016	7.0	4.3
1009	15.3	12.4
2654	7.3	2.5
1010	10.7	8.2
1012	10.5	6.8
2663	-10.6	7.2
1013	10.2	6.1
2659	- 3.5	1.5
2601	- 2.7	1.1
2664	-11.5	7.2
2602	- 3.4	1.0
2660	- 3.3	0.5
2610	- 6.5	3.4
2665	-12.1	6.5
1017	14.9	9.8
2661	- 3.8	1.7
2655	- 4.8	2.7
2656	- 4.8	2.7
1018	13.1	9.4
2662	- 4.6	2.4
2657	- 5.3	3.0

TABLE XIV CONTINUED

COMPARISON OF INVESTIGATOR(S): FRIEND & METZNER (1958)		
RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
1005	- 4.8	3.1
1006	-10.7	5.3
1004	- 6.5	2.7
1007	21.6	13.5
1014	7.2	3.3
1008	- 8.8	5.1
1015	- 6.2	2.6
1016	-11.9	7.6
1009	- 5.2	2.1
2654	- 3.4	2.1
1010	- 7.0	3.3
1012	- 7.0	2.8
2663	- 6.4	2.6
1013	- 6.2	2.6
2659	10.3	7.2
2601	10.7	6.3
2664	6.9	3.7
2602	12.8	7.7
2660	12.5	9.7
2610	8.7	4.9
2665	12.8	7.1
1017	11.0	6.8
2661	16.5	12.8
2655	19.6	14.5
2656	20.7	15.4
1018	14.6	9.5
2662	22.3	16.6
2657	22.2	16.9

TABLE XIV CONTINUED

COMPARISON OF INVESTIGATOR(S): CHURCHILL (1977b)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
1005	26.5	16.9
1006	21.9	14.3
1004	22.2	16.0
1014	27.1	21.9
1008	17.6	13.0
1015	22.5	17.4
1016	12.5	7.8
1009	22.3	18.0
2654	15.2	11.5
1010	19.4	15.0
1012	20.6	17.0
2663	8.6	6.4
1013	25.2	19.2
2659	20.0	17.1
2601	21.3	17.5
2664	11.8	8.6
2602	21.9	19.0
2660	22.2	19.7
2610	19.9	16.5
2665	17.1	12.8
2661	25.7	22.8
2655	28.8	24.5
2656	29.9	25.4
2662	30.8	26.8
2657	31.9	27.0

TABLE XIV CONTINUED

COMPARISON OF INVESTIGATOR(S): CHEN (1988b)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
1009	105.6	9.8
2654	190.0	6.0
1010	98.0	4.9
1012	76.0	2.1
2663	128.6	12.0
1013	65.0	1.8
2659	118.7	10.1
2601	100.8	8.4
2664	113.2	11.3
2602	104.5	6.4
2660	111.6	8.9
2610	99.0	10.7
2687	105.4	6.4
2665	97.1	9.2
1017	51.0	4.7
2661	96.9	8.5
2655	89.4	8.5
2652	83.1	11.6
2656	84.9	8.1
1018	44.6	4.9
2662	84.2	7.4
2686	86.2	3.7
2657	82.5	7.7

TABLE XIV CONTINUED

COMPARISON OF INVESTIGATOR(S): CHEN (1988c)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
1005	21.3	10.3
1006	14.3	7.2
1004	18.4	10.2
1007	37.1	26.8
1008	12.0	5.5
1015	14.1	9.0
1016	6.6	2.2
1009	14.3	8.1
2654	-11.1	6.9
1010	10.6	5.0
1012	7.9	4.9
2663	-11.9	8.5
1013	10.6	6.1
2659	-7.1	4.1
2601	-6.5	3.0
2664	-10.0	6.3
2602	6.6	2.3
2660	-5.3	2.3
2610	-8.1	4.1
2665	-8.0	3.0
1017	17.7	13.7
2661	7.1	1.8
2655	9.4	2.5
2656	10.1	2.8
1018	21.1	16.1
2662	11.3	3.7
2657	11.1	3.7

COMPARISON OF INVESTIGATOR(S): CHEN (1988d)

RUN#	MAX %DEV.	AVE. ABSOLUTE %DEV.
2660	156.9	27.1
2610	153.9	23.7
1017	80.1	30.0
2661	119.6	9.0
2655	101.9	8.5
2652	80.0	5.2
2656	97.2	9.5
1018	61.8	14.1
2662	92.6	10.7
2657	87.9	12.4
2658	78.1	7.1

VITA

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Candidate for the Degree of

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