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SINGLE PHASE HEAT TRANSFER ENHANCEMENT BY DOUBLY AUGMENTED TUBES

by

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TABLE OF CONTENTS

Page

Page

LIST OF FIGURES

LIST OF TABLES

Chapter I

INTRODUCTION

Heat exchangers are devices which are being used in a wide range of engineering applications. Many types of heat exchangers have been developed for use at such varied levels of technological advances and sizes as steam power plants, chemical processing plants, building heating and air conditioning, household refrigerators, radiators for space vehicles, and many other applications. Over the past few decades economic and energy considerations have motivated investigators to explore the use of different techniques to augment the heat transfer and improve the performance of heat exchangers. The main benefit of these attempts is to reduce the size of heat exchangers and thereby reduce their cost. In a report published by Bergles [1] in 1975, it was estimated that the investment in heat exchanger equipment in the U.S was about one billion dollars per year. Ten to twenty percent reduction in capital cost could produce savings of the order of 100 million dollars.

During the past 50 years, numerous techniques have been tried to improve the heat transfer in heat exchangers. These techniques can be broadly classified into active and passive techniques. Mechanical stirring of fluids, heat transfer surface vibration, electrostatic fields and jet impingement are some of the active techniques. Since active techniques of augmenting the heat transfer require external power, passive techniques are in general preferred because they do not require external power. Treatment and/or roughening of the heat transfer surfaces, extended heat transfer surfaces and swirl flow device inserts are among the passive techniques that have been used successfully in augmenting the heat transfer. These passive techniques are easy to incorporate, relatively simple to operate and pose less problems in maintenance. Commercial production of heat exchanger tubes which includes one of these passive techniques are in vogue. One of the factors which is limiting the use of some of these techniques is the lack of reliable design correlations for predicting heat transfer and pressure drop. Although several of these augmentation techniques have been incorporated in a few industrial applications, considerable amount of testing is needed before most of these techniques can be put into full scale industrial use.

All the investigations that have been conducted to date on enhancement of the heat transfer of heat exchangers were mainly directed toward augmenting the heat transfer on the inside or outside surfaces of the inner tubes of the exchangers. No references could be located that were concerned with heat transfer enhancement by augmenting the heat transfer of both inside and outside surfaces. This is the main concern of the present study. Such a study will contribute to the knowledge in an area of heat transfer enhancement of heat exchangers which has not been explored before.

Chapter II

LITERATURE SURVEY

A bibliography of world literature on augmentation was published by Bergles et al. [2]. Over 3000 references were cited in this bibliography. The references were grouped under two major classifications:

1. Techniques of Augmentation: Active, requiring external power, or passive, requiring no external power.
2. Modes of heat transfer.

All references related to these techniques (active and passive) were listed in this bibliography. Considering the vast scope of augmentation, the present literature survey had to be limited to reference of particular interest to the present study, namely:

1. Single phase forced convection heat transfer inside internally finned tubes.
2. Single phase forced convection heat transfer in annuli with rough surfaces.

INTERNALLY FINNED TUBES

In recent years, manufacturing techniques have been developed to produce a wide variety of internally finned tubes. Numerous experimental and analytical studies on the effects of these tubes on heat transfer and pressure drop in single-phase flow were conducted by different investigators.

Watkinson et al. [3,4,5] experimentally investigated the heat transfer and pressure drop with internally finned tubes of different diameters and fin geometries in single-phase flow. An enhancement of heat transfer as high as 170% at the same Reynolds number and up to 80% at constant pumping power was reported in [3] for turbulent flow of water in internally finned tubes. Enhancement in heat transfer decreased with Reynolds number in the range 10,000 to 150,000. Heat transfer enhancement over smooth tube values at a Reynolds number of 50,000, based on inside diameter, varied from 17 to 95% in turbulent air flow [4]. At the same Reynolds number, heat transfer enhancement varied from 150 to 87% for turbulent water flow [3]. Heat transfer increased with tighter spiralling. The effect of tighter spiralling decreased with increase in Reynolds number. Also, at constant pumping power, the performance of spiral fin tubes increased with the inter-fin spacing to pitch ratio.

Heat transfer and pressure drop measurements were made for laminar oil flow in [5]. At a Reynolds number of 500, heat transfer was enhanced over smooth tube values by 8 to 224% depending on tube geometry.

Carnavos et al. [6,7,8,9] conducted experiments to cool air, and to heat water inside tubes with intenal fins. In the study of cooling of air in longitudinal internally finned tubes, Russell and Carnavos [6] concluded that the heat transfer enhancement was roughly equal to the heat transfer area increase. The increase in friction factors were in the range of 80 to 100% of the square of the heat transfer area increase. In the study of air, Carnavos [7] used 21 tubes with integral spiral and longitudinal fins. The finned tubes were reported to perform better than smooth tubes by factors of 1.2-2.0 at constant pumping power. He also presented correlation equations to predict heat transfer and friction factor for internally finned tubes.

In [8], Carnavos experimentally investigated the heat transfer performance of five composite tubes for cooling air in turbulent flow. These tubes were made by mechanically coupling in parallel individual copper tubes having continuous integral spiral and longitudinal fins. He reported that at constant pumping power, the composite tubes performed better than smooth tubes by factors of 1.7 to 10. Correlations to predict heat transfer and pressure drop were also presented. In [9], Carnavos experimentally investigated the heat transfer performance of heating water and/or a 50%-50% ethylene glycol-water solution in turbulent flow by tubes having internal spiral and longitudinal fins and found that these tubes performed better than smooth tubes. Correlating equations were presented for heat transfer and Fanning friction factor that described air, water and ethylene glycol-water data to within ±10%.

Bergles et al. [10] presented heat transfer and fluid friction data for a variety of tubes having internal fins, with water as the coolant. Short spiralled fins produced an increase in heat transfer above the smooth tube values. This increase was over 200% for equal flow conditions and up to 170% at equal pumping power.

For laminar flow of water and ethylene glycol in internally finned tubes, Bergles [11] concluded that there could be a larger increase in heat transfer (100-1600%) for modest pressure drop increases (30%).

Compared to the experimental investigations, analytical studies of heat transfer and pressure drop inside internally finned tubes are limited.

Hu and Chang [12] studied the heat transfer of fully developed laminar flow in internally finned tubes analytically. The fins considered had fictitious zero thickness. They concluded that the presence of internal fins improved the heat transfer performance for laminar flow more than for turbulent flow. They also concluded that the Nusselt number for laminar flow in tubes with optimum number of internal fins of particular height can surpass that for turbulent flow in finless tubes.

Nandakumar and Masilayah [13] used the finite element method of analysis to solve the momentum equation describing the laminar fluid flow in a finned tube. They developed empirical correlations for friction factor and Nusselt number for a wide range of fin parameters. In a later paper [14], they obtained the heat transfer coefficients for forced convection fully developed laminar flow in an internally finned tube with axial uniform heat flux and with peripherally uniform temperature, using the finite element method. They concluded that the Nusselt number of a triangular finned tube was higher than that of a finless tube, and was a strong function of fin length and fin thickness.

Soliman and Feingold [15] conducted an analysis of fully developed laminar flow in internally finned tubes by solving the momentum equation using infinite series coefficients matching method to obtain the velocity profile and friction factor. They reported that the values of the dimensionless velocity at any location within the tube depended mainly on the number of fins and their height, and to a lesser degree on the fin half-angle. In a later paper [16], they analytically solved the energy equation for fully developed laminar flow to obtain the temperature profile and heat transfer. It was conluded that the Nusselt number increases with the increase of the number of fins up to a critical fin number beyond which a reversal of trend occurs.

Rowley and Patankar [17] presented a numerical study for the laminar flow and heat transfer in tubes with internal circuferential fins. The results werre presented for a range of Reynolds number and for various values of geometrical parameters of the fins. Results showed that increase in heat transfer could be achieved only for high Prandtl number fluids.

AUGMENTATION OF OUTSIDE SURFACES

Not much work has been done in considering the effect of augmentation on the outside surface of an inner tube of a heat exchanger on shell side heat transfer. Bergles [18] collected data given by Maeda [19] to derive individual heat transfer coefficients for external flow over turbotec tubing. The annular gap was not specified and a gap of approximately the same as the tube diameter was assumed. The outer profile of the turbotec tube increased the shell side heat transfer coefficient by as much as 100%.

Fedynskii [20] studied the effect of transverse turbulizing fins on heat transfer rate during the flow of water in an annular space. There was a 75% increase in heat transfer with trianular turbulizing fins on the outer surface of the inner tube.

An analytical method for the transformation of turbulent Stanton numbers measured in the annular channel geometry was developed by Hudina [21]. This method was based on the solution of a simplified differential energy transport equation for turbulent flow.

Other references relevant to the present investigation are reviewed later in this work. These are the references dealing with performance evaluation of augmented versus unaugmented heat transfer surfaces.

Chapter III

EXPERIMENTAL INVESTIGATION

The primary objectives of this experimental investigation were:

1. To take heat transfer and pressure drop data during cooling of liquid R-113 inside six different double pipe, counterflow heat exchangers with R-113 flowing inside the inner tube and the cooling water flowing in the annulus. The heat exchangers consisted of two sets with three exchangers in each set. Within each set the inner tube of each exchanger had the same outside diameter. However, in one exchanger the inner tube had internal fins on the inside and a smooth surface on the outside. In the second exchanger the inner tube had knurls on the outside surface and a smooth surface on the inside. In the third exchanger the inner tube had the same fins of the first exchanger on the inside and the same knurls of the second exchanger on the outside. This tube is what is identified as a doubly augmented tube. The two sets of exchangers differed in the outside diameter of the inner tube and the number of fins.
2. To compare the heat transfer enhancements of the heat exchangers with doubly augmented tubes with the enhancement in the heat exchangers with singly augmented inside tubes under the same flow conditions or constraints.
3. To identify conditions under which double augmentation could be beneficial compared to inside versus outside augmentation in enhancing the heat transfer.

3.1 TEST FACILITY

Figure 3.1 shows a photographic view of the test facility. This facility was designed to be flexible enough to allow the test heat

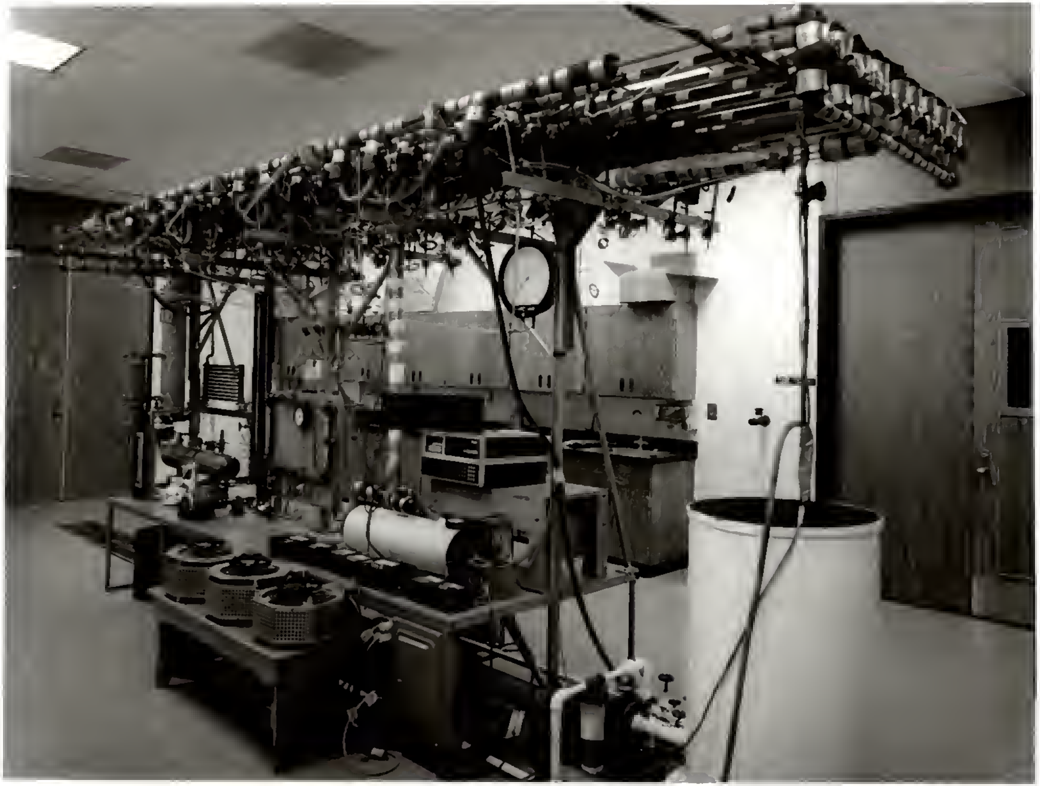

Fig. 3.1 Photographic view of the entire test facility.

exchangers mounted on the test facility to be used to study the effects of doubly augmented tubes on the enhancement of heat transfer during heating or cooling of single phase fluids as well as during condensation inside the inner tubes. This facility has a closed loop for R-113, and an open flow loop for the cooling water.

3.2 R-113 FLOW LOOP

Figure 3.2 shows a schematic diagram of the R-113 flow loop. It included the following components:

1. A liquid circulating gear pump.
2. An electrically heated tube.
3. An after cooler.
4. A liquid receiver.
5. Liquid flow meters.
6. A liquid drier.

In addition to the above components, six test heat exchangers were mounted horizontally in parallel as shown in Fig. 3.2. Each test heat exchanger consisted of four subsections. Each subsection was a double pipe counterflow heat exchanger.

3.2.1 Test Heat Exchanger Construction and Instrumentation

Each heat exchanger was constructed of four subsections, each of approximately 2 feet in length. Each subsection was a counter flow tube-in-tube heat exchanger where the test fluid flowed inside the inner tube and the cooling water flowed in the annulus. The four subsections were constructed from one single inner tube inside which R-113 flowed, and around which four equal lengths of copper tubing, with equal spacing in between, were soldered to the outside of the test tube using the proper

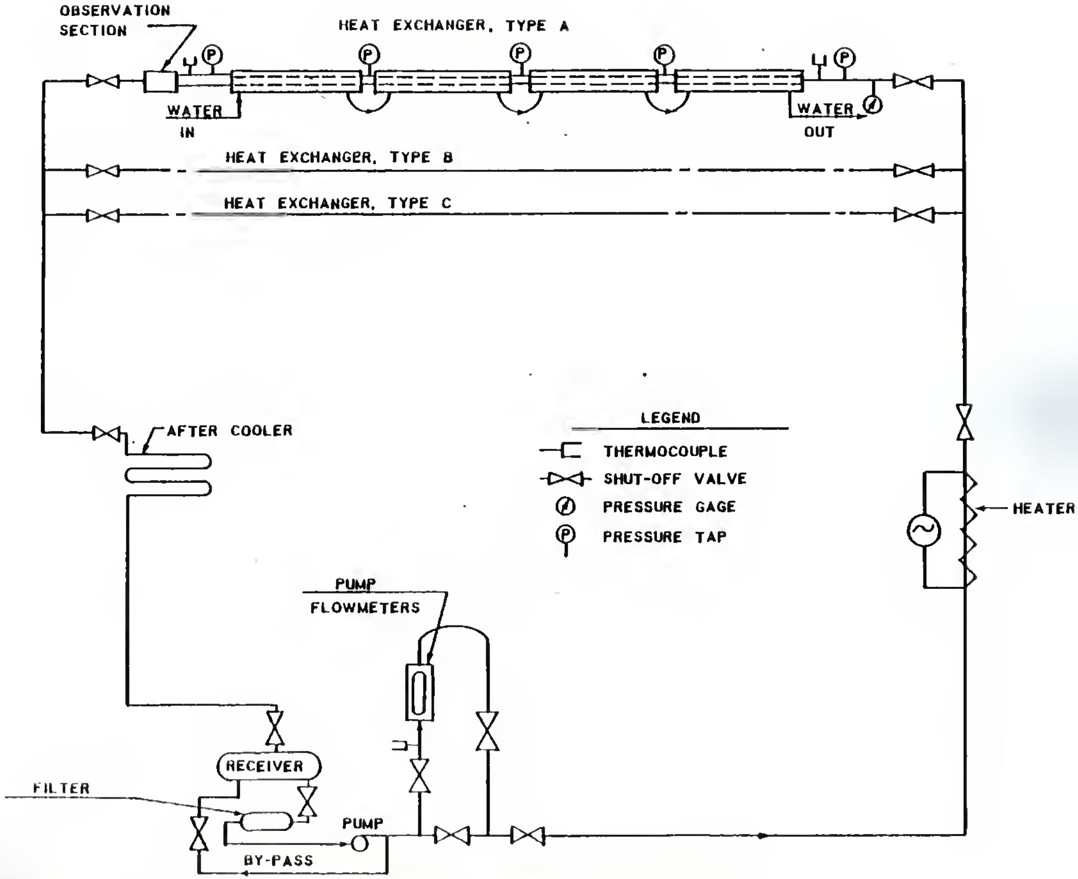

Fig. 3.2 A schematic diagram of the R-113 circuit.

fittings and connections. This arrangement provided the four subsections of each exchanger. The reason for constructing the heat exchangers out of four subsections each is the fact that such construction allows the flexibility of measuring the heat transfer during partial or complete condensation if the heat exchangers were used as condensers. Each outer tube of the four subsections was provided with coolant inlet and outlet connections and connections to allow the insertion of thermocouple leads to measure the outside surface temperature of the inner tube. The length of each subsection was 0.61 m (24 in) between coolant's inlet and outlet. After the coolant left a subsection it was directed by plastic tubing to the next subsection. At the exit of each exchanger an observation section, made of high pressure clear glass, was installed to observe the flow pattern of the condensing fluid. The inside diameter of each glass section was selected equal to the inside nominal diameter of the inner tube. The outer tube for each exchanger was selected to have an inside diameter equal to the outside diameter of the inner tube plus 0.025 m (1 in).

Each exchanger was instrumented to measure the inlet and outlet temperatures of the cooling water of each subsection, the inlet pressure and temperature, and the exit temperature of R-113 of each test exchanger. All temperatures were measured by copper-constantan thermocouples. Four thermocouples were attached to the outside surface of the inner tube of each test subsection to measure its temperature. At two axial locations 0.152 m (6 in) from the coolant's inlet and outlet, two thermocouples were attached to the surface, one at the top and the other at the bottom of the inner tube. Five pressure taps were provided to measure the pressure drop across each subsection and the pressure at the inlet to the heat exchanger. Figure 3.3 shows a schematic diagram of the heat exchanger. Figure 3.4

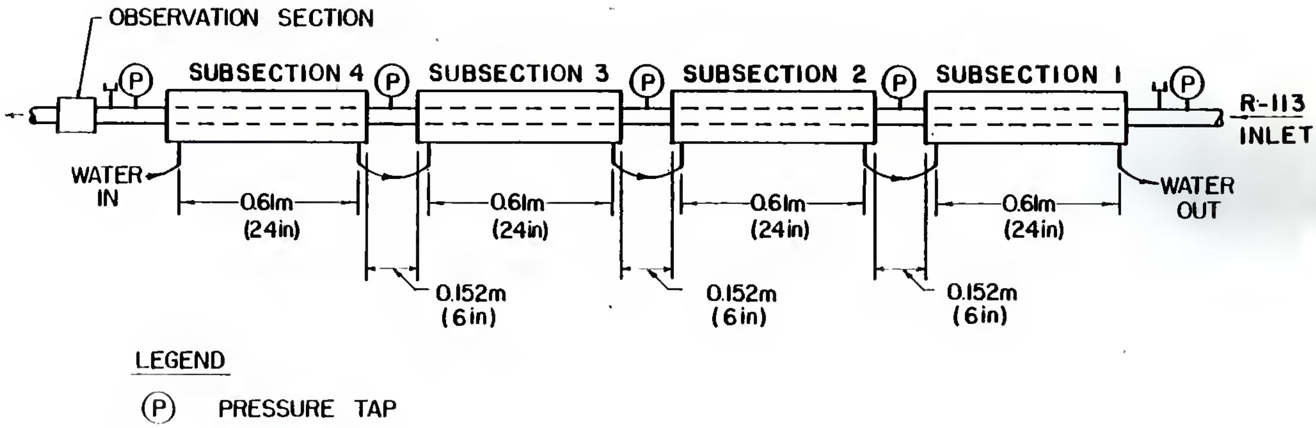

Fig. 3.3 A schematic diagram of the test heat exchanger.

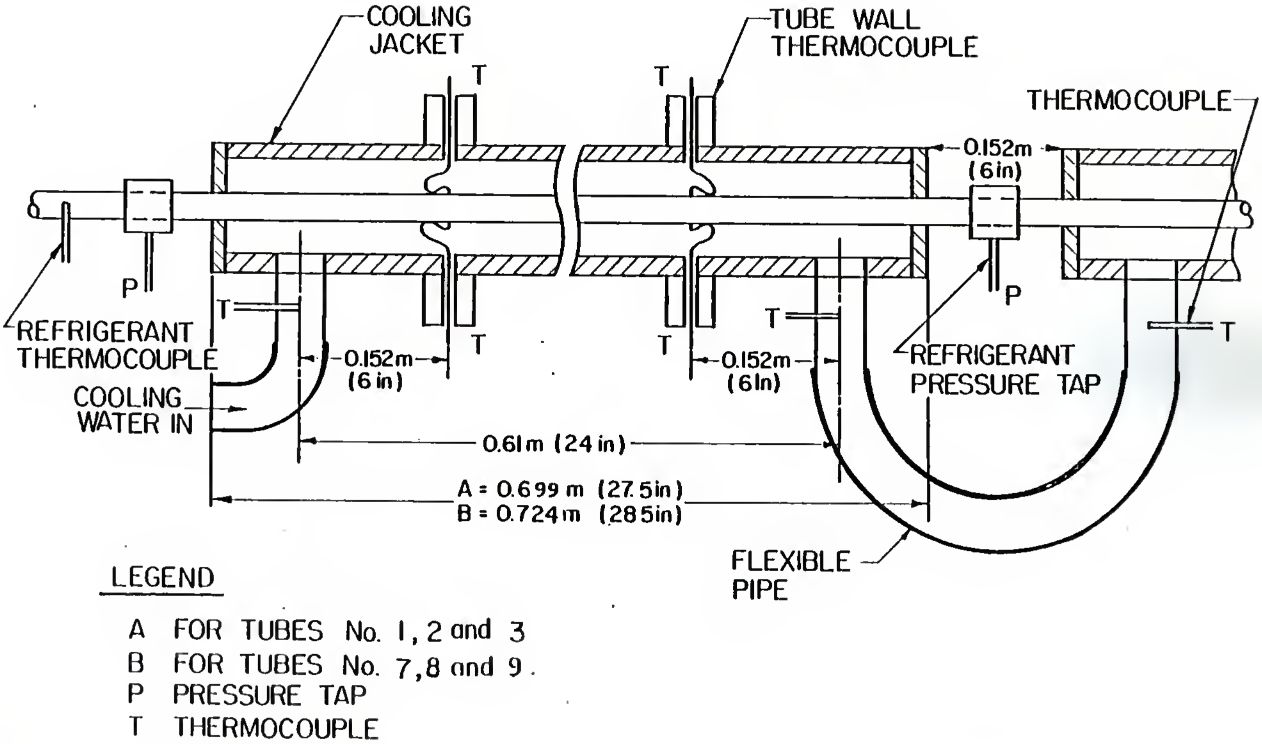

Fig. 3.4 Construction details of a subsection of the test heat exchangers.

shows the construction details of a subsection of the heat exchanger.

To accommodate any expansion due to heating during the experimental runs each exchanger was attached to a flexible copper vibration eliminator. It was placed after the observation section.

All temperatures were measured by copper-constantan B and S 24 gage thermocouples. All thermocouple junctions were calibrated at the boiling water and melting ice temperatures. Inlet pressure of the heat exchanger was measured by a Heise pressure gage type H22635. The pressure drop between any two pressure taps was measured by a Foxboro Differential Pressure (D/P) Cell Type 13A. At each pressure tap location a 0.002 m (1/16 in) diameter hole was drilled at the bottom of the tube. A compression fitting was soldered to the tube and a short 0.003 m (1/18 in) diameter copper tube was then attached to the compression fitting. The tube was then attached to the D/P cell. Figure 3.5 shows a schematic diagram for the piping system of the pressure drop measurements.

Average wall temperatures of the inner tube of each subsection of the heat exchangers were measured by four thermocouples. Two thermocouples were attached at the top and the other two at the bottom as shown in Fig. 3.4. An axial distance of 0.30 m (12 in.) was maintained between the two thermocouple stations located at 0.152m(6 in.) from the inlet and outlet of the coolant (water).

The construction detail of each thermocouple junction is shown in Fig. 3.6. The location of each thermocouple junction was marked. At these locations, a conical groove, slightly larger than the bead of a thermocouple, was drilled. The bead was then silver brazed in its place. Fast drying epoxy (a product of Armstrong company type A-36/B-36) was applied to the silver brazed portion of the thermocouple. A short length of nylon heat shrink tubing was slid from the other end of the thermocouple

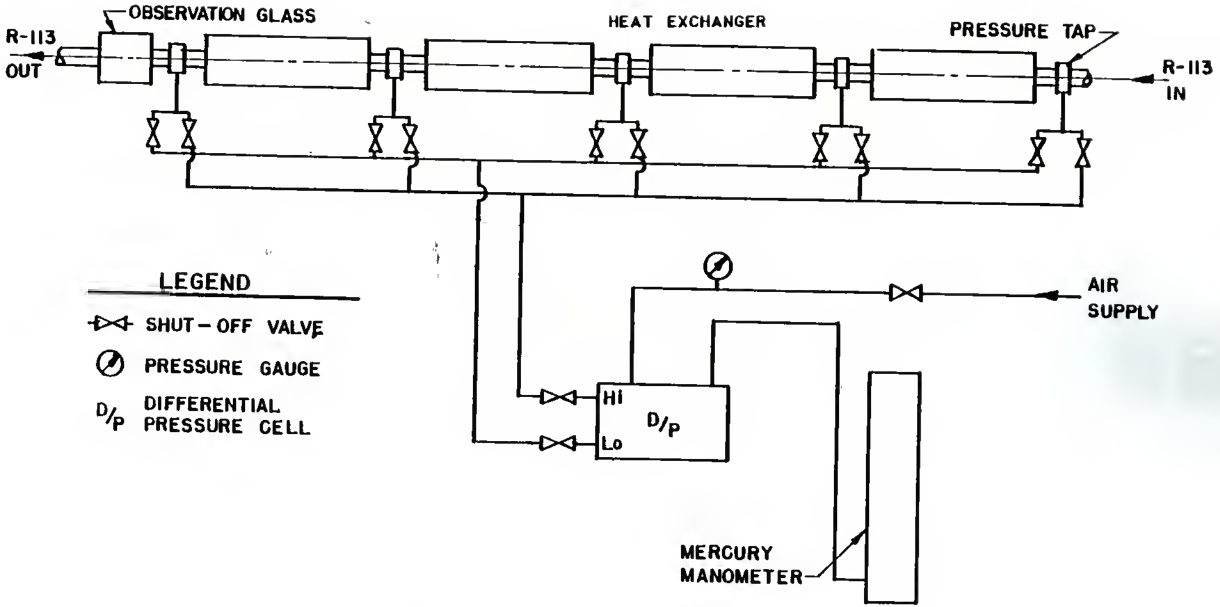

Fig. 3.5 Schematic diagram for pressure drop measurement.

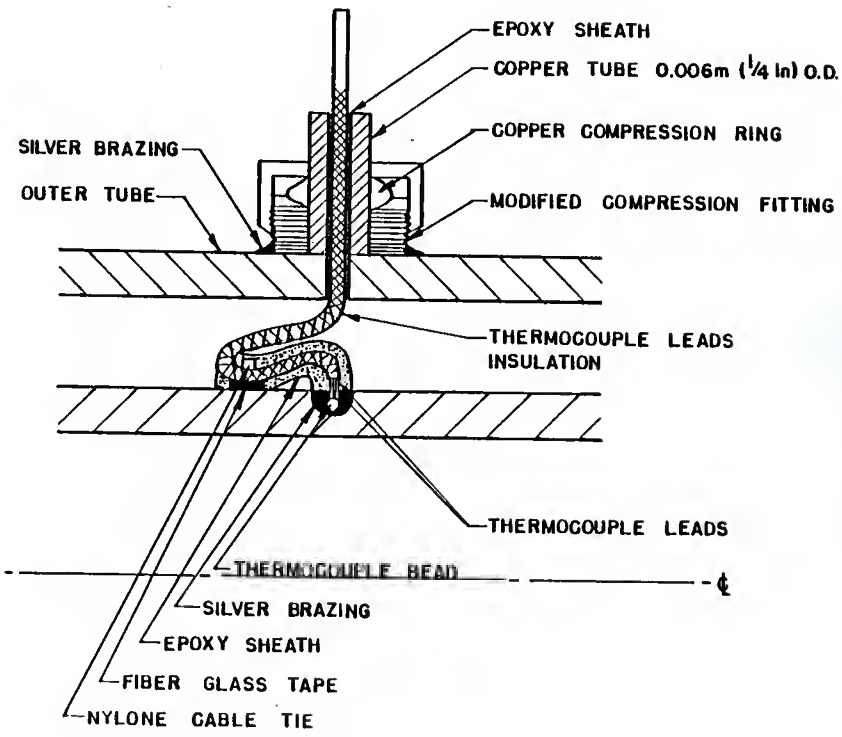

Fig. 3.6 Thermocouple mounting detail.

towards the junction until it touched the outside surface of the heat exchanger's inner tube and covered the junction. More epoxy was added to secure the nylon tube to the outside surface. This arrangement ensured the elimination of contact between the thermocouple junction and cooling water. A thin strip of fiber glass insulating tape was wrapped around the inner tube next to the thermocouple junction. The thermocouple leads were then tied to the inner tube of the heat exchanger using a nylon cable tie over the fiber glass tape. The fiber glass tape protected the thermocouple leads from getting heated at the portion where they were tied to the inner tube. The nylon tie prevented the detachment of the thermocouple junction if it was accidentally pulled from the other end. The tube was then cleaned in accordance with the procedure given by Pence [22]. The procedure used was the following:

> A solution of equal parts of ethyl alcohol and a 50 percent solution of sodium hydroxide was prepared. The solution was then heated to 85°C. This solution was applied to the outer surface of the inner tube. The tube was then rinsed with tap water. It was then rinsed again and dried by blowing compressed air.

Each thermocouple junction was checked for good contact using a digital multimeter. The thermocouple leads were then led through the outside tube as shown in Fig. 3.4. The hole between the leads and the copper tube attached to the outer tube of the subsection was filled with epoxy (a product of Armstrong company type A-34/B-34). The present arrangement was chosen as it provided the best compromise between the practical assembly and the minimization of thermocouple conduction error.

The six heat exchangers tested differed in the geometry and the condition of the inside and outside surfaces of the inner tube. They were selected in 2 sets of three tubes each. In each set the outside diameter of the inner tube was the same. However, within every set one tube had spiral fins on the inside and a smooth surface on the outside, Tube A in Fig. 3.7, the second tube had knurls on the outside and a smooth surface on the inside, Tube B in Fig. 3.7. The third tube had the same fins of Tube A on the inside and the same knurls of tube B on the outside, Tube C in Fig. 3.7. This third tube is what is called a doubly augmented tube. Table 3.1 gives the geometric parameters of the tubes tested. Figure 3.8 gives a schematic view of the various geometric parameters given in Table 3.1. The tubes were provided by Noranda Metal Industries.

Table 3.2 gives the lengths of the different subsections of the various heat exchangers tested. It is to be noticed that the heat exchangers were given the numbers 1,2,3,7,8 and 9. The reason for such numbering scheme was due to the fact that there were other heat exchangers 4,5 and 6 mounted on the test facility that were not used in the present study.

The flow rate of R-113 leaving the circulating pump was passed through a filter drier and then through one of two Fisher and Porter variable area flow meters. The flow meters were mounted vertically in parallel to be used individually or simultaneously to measure the flow rate of R-113. One had a range of 0.0 to 0.35 g.p.m. and the other 0.0 to 0.51 g.p.m. The temperature of R-113 entering the flow meters was also measured.

The heating section of R-113 mounted between the flow meters and the heat exchangers was a 0.016 m (5/8 in) diameter of copper pipe 1.829 m (6 ft) long. It was wrapped along its length with teflon tape in order to insulate the heating element from the surface of the tube. Heating was accomplished by a ribbon-type Chromel heating element, which was wound

Fig. 3.7 Photograph of the inner tubes tested.

TABLE 3.1. Geometric Parameters of the Experimental Tubes*

Heat Exchanger No.	1	2	3	7	8	9
Type of inner tube	A	B	C	A	B	C
Material	Cu	Cu	Cu	Cu	Cu	Cu
Outside surface	Smooth	Knurled	Knurled	Smooth	Knurled	Knurled
Inside surface	Finned	Smooth	Finned	Finned	Smooth	Finned
No. of fins, n	24	——	24	38	——	38
Outside diameter, D_o	0.625	0.624	0.618	0.875	0.830	0.830
Inside diameter, D_i	0.525	0.500	0.494	0.750	0.705	0.705
Equivalent diameter, D_e	0.509	0.500	0.477	0.733	0.705	0.686
Hydraulic diameter, D_h	0.271	0.500	0.246	0.373	0.705	0.338
Fin height, b	0.025	——	0.025	0.025	——	0.025
Wall thickness	0.0500	0.0620	0.0625	0.0625	0.0625	0.0625
Fin height/inside diameter	0.0476	——	0.0506	0.0333	——	0.0355
Actual flow area, A_{fa}**	0.2054	0.1963	0.1806	0.4242	0.3904	0.3728
Nominal flow area, A_{fn}**	0.2165	0.1963	0.1917	0.4417	0.3904	0.3904
Core flow area, A_{fc}**	0.1772	0.1963	0.1548	0.3848	0.3904	0.3370
Actual area, A_a***	3.0350	1.5708	2.9376	4.5501	2.2148	4.4088
Nominal area, A_n***	1.6493	1.5708	1.5519	2.3562	2.2148	2.2148

TABLE 3.1. Geometric Parameters of the Experimental Tubes*

Heat Exchanger No.	1	2	3	7	8	9
Fin thickness, t	0.025	———	0.025	0.025	———	0.025
Helix angle, α	30°	———	30°	30°	———	30°
Height of knurls, e	———	0.0075	0.0105	———	0.0225	0.0225

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

** area in in^2

*** area in in^2/in

Table 3.2. Dimensions for each subsection of the heat exchangers*.

Heat Exchanger No.	1	2	3	7	8	9
Length of subsection I	27.69	27.69	27.44	28.88	27.81	28.94
Length of subsection II	27.62	27.75	27.50	28.88	28.69	27.88
Length of subsection III	27.81	27.38	27.38	28.88	27.69	27.94
Length of subsection IV	27.63	27.13	27.44	29.00	27.81	27.75
Water jacket inner diameter, D_j	1.50	1.50	1.50	2.00	2.00	2.00

* all lengths and diameters are in inches.

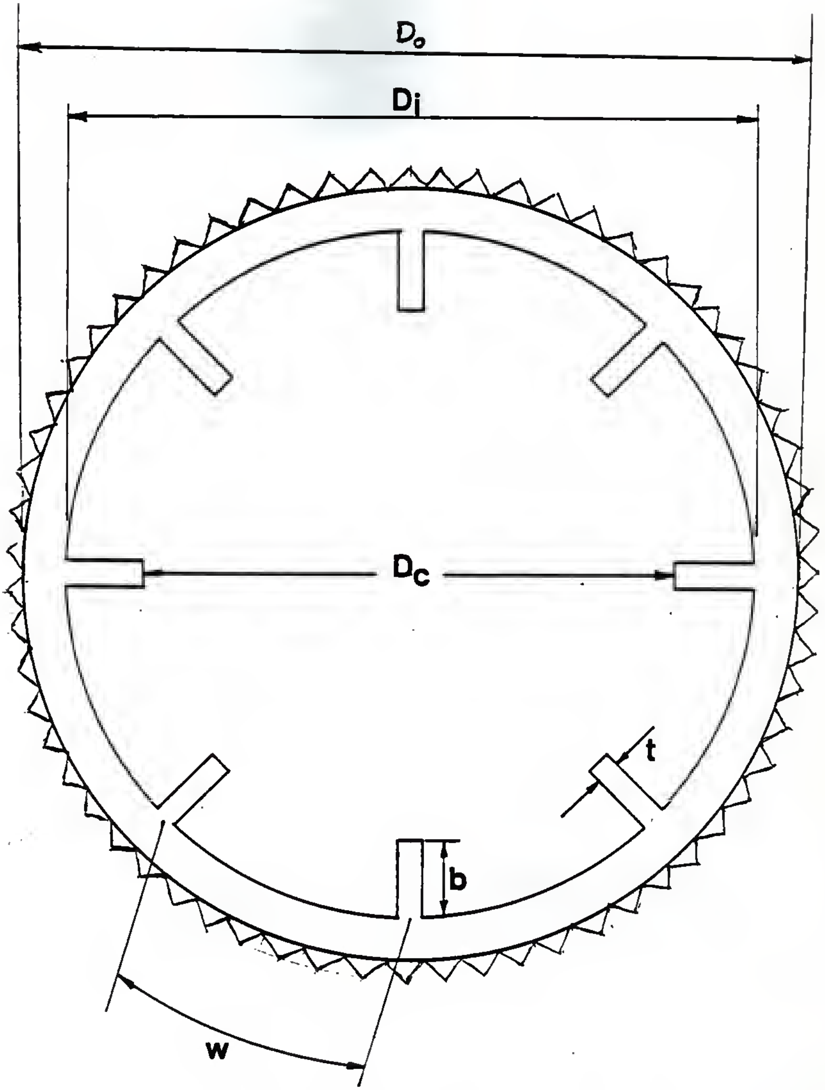

Fig. 3.8 Cross-section of a doubly augmented tube.

uniformly on the teflon tape. The heating element was then coated with a layer of epoxy resin to hold it onto the tube, and a second layer of teflon tape was again wound over the resin. The entire test section was then insulated with 0.064 m (2.5 in) thick fiberglass insulation to prevent any heat losses to the atmosphere. Power input to the heating coil was controlled by a powerstat variable transformer. The power input to the heating coil was measured by measuring the current and voltage across the heating coil. Appendix A lists additional details about the components used in the refrigerant flow circuit.

Each heat exchanger was individually leak-proof tested under pressurized and evacuated conditions. Also, the entire assembly of the test facility was leak-proof tested under pressurized and evacuated conditions. Finally, the heat exchangers were insulated with rubber insulation to avoid heat transfer to ambient air. The rubber insulation was 0.064 m (2.5 in.) thick.

3.3 COOLING WATER LOOP

Figure 3.9 shows a schematic diagram of the cooling water flow loop. City water was supplied to a mixing tank from the water main. Water was pumped through a circulating pump to the heat exchanger. Two Brooks Rotameters were mounted vertically in parallel to measure water flow rate through the heat exchanger. One had a range of 0.0 to 3.0 g.p.m., and the other had a range of 0.0 to 21.0 g.p.m. The cooling water loop was designed to permit the control of water flow rate and varying its inlet temperature to the heat exchanger. The control of inlet water temperature was achieved by passing steam through the mixing tank. City water was also passed through the after cooler and the pressure taps. Appendix A lists additional details of the components used in the cooling water circuit.

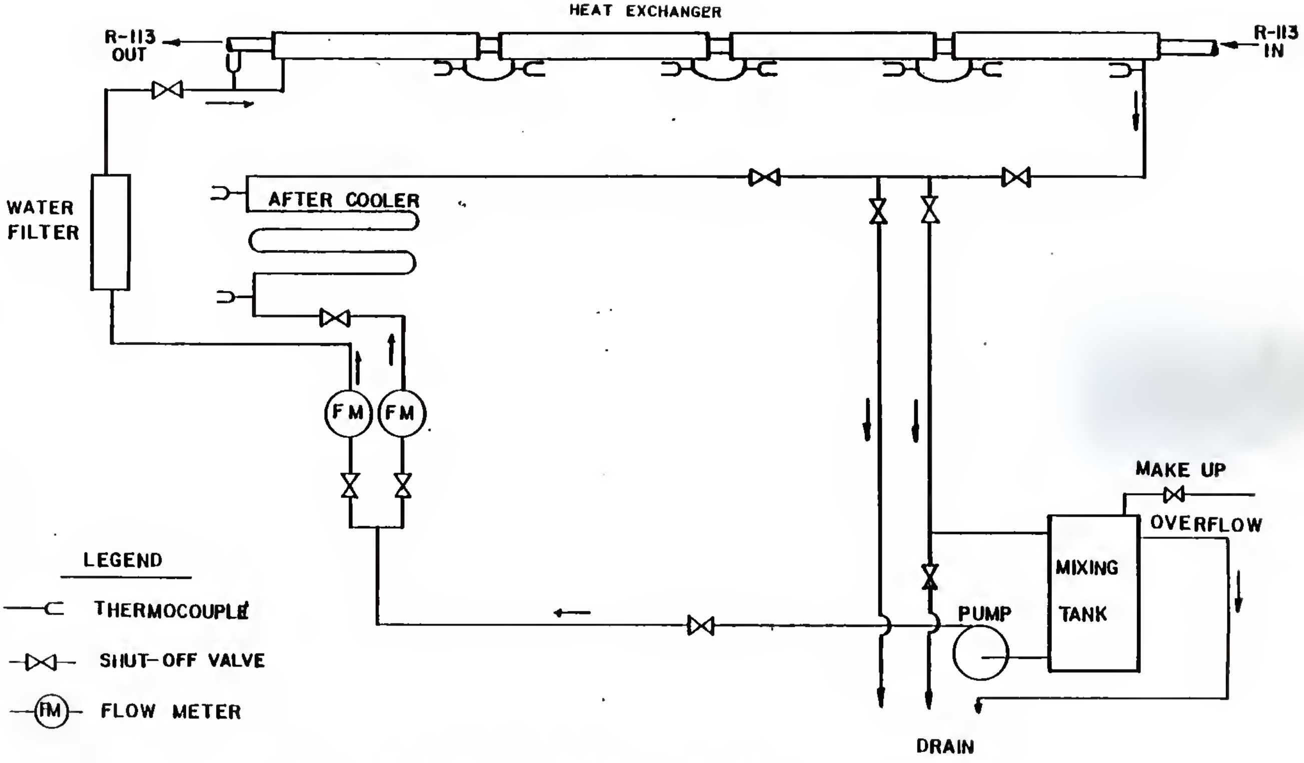

Fig. 3.9 A schematic diagram of cooling water circuit.

3.4 OPERATION OF TEST FACILITY AND DATA ACQUISITION

After the construction, instrumentation and assembly of the test facility, the entire system was evacuated down to 1000 microns of mercury pressure by a vacuum pump. The system was then charged with R-113. The process of evacuation and charging was repeated three times to get rid of the non-condensables from the system.

Each experimental run was started by filling the water mixing tank with water to the overflow level. The circulating water pump was then put into operation and a desired flow rate was maintained.

Liquid R-113 circulating pump was started and R-113 was pumped through the heat exchanger at the desired flow rate. Electric power was applied to the heating tube to maintain a constant temperature of R-113 at the inlet of the heat exchanger. For all experimental runs, steady state condition was considered established when the readings of all temperatures and flow rates remained steady for a period of at least one hour. The following measurements were taken for each experimental run:

1. Water flow rate through the heat exchanger, in m^3/s (g.p.m.).
2. Inlet and outlet water temperatures of each subsection, in ^oC (^oF).
3. R-113 flow rate through the heat exchanger, in m^3/s (g.p.m.).
4. Inlet and outlet temperatures of R-113, in ^oC (^oF).
5. Inlet pressure of R-113 at inlet to the heat exchanger, in kPa (psia).
6. The temperature of sixteen thermocouples attached to the outer wall of the inner tube of the heat exchanger, in ^oC (^oF).
7. The temperature of R-113 leaving the circulating pump, in ^oC (^oF).

All thermocouples wires were attached to an Esterline Angus programmable data acquisition system capable of printing all temperatures at specified intervals. Additional information about the data acquisition system are given in Appendix A.

Heat balance error of ±5% was set as the criterion for accepting any experimental run. The heat balance error was calculated as the ratio of the difference between heat gain rate of cooling water and the rate of heat loss by R-113 to the rate of heat gain by cooling water. The heat gain rate of cooling water was calculated from its flow rate and its temperature rise. The rate of heat loss of R-113 was calculated from its enthalpy change between the inlet and outlet to the heat exchanger.

The heat balance error for the experimental runs rarely exceeded ±5%. However, a few runs with heat balance error slightly higher than the above limits were also included in the data reported in this study.

The ranges of experimental parameters covered in this study were dictated by the limitations of the existing experimental set up. The ranges of the experimental parameters are listed in Table 3.3.

TABLE 3.3. Ranges of Experimental Parameters Covered

Refrigerant's Mass Flux (based on nominal inside area)	$3.46(0.71\times10^5) - 14.55(2.98\times10^5)$ $kg/s\text{-}m^2$ $(lbm/hr\text{-}ft^2)$
Overall Heat Transfer Conductance	22.79(43.20) - 111.07(210.54) W/ ^oC (Btu / hr ^oF)
Overall Heat Transfer Rate	358.5(1223.3) - 1054.4(3597.8) W (Btu/hr)
Inlet Refrigerant temperature	26.4(79.5) - 46.4(115.5) ^oC (^oF)
Inlet Coolant temperature	13.9(57.0) - 19.6(67.3) ^oC (^oF)
Refrigerant Reynolds Number	2787.4 - 7826.2
Coolant Reynolds Number	747.47 - 3433.69
Refrigerant Prandtl Number	7.806 - 8.906
Coolant Prandtl Number	6.868 - 8.324

Chapter IV

EXPERIMENTAL RESULTS

4.1 INTRODUCTION

A total of 300 heat transfer experimental runs were made for all heat exchangers tested. The parameters that were controlled during these runs were the flow rate of R-113 and its inlet temperature, and the flow rate of the cooling water and its inlet temperature. The heat transfer data reduction was accomplished by a computer program written for a Zenith microcomputer, model Z-150. The program is given in Appendix B. The reduced data of heat transfer are given in Appendix C, Tables C.1 through C.5. Each experimental run was identified by four digits xxxx. Starting from left to right the first digit refers to the heat exchanger's number which immediately identifies the type of inner tube it had by referring to Table 3.1. The second digit refers to the flow constraints applied to that particular exchanger. For example:

0 refers to the set of runs where the water flow rate and the inlet temperature of water and R-113 were fixed while the flow rate of R-113 varied.

1 refers to the set of runs having the same constraints as runs of group 0 except that the flow rate of water was different.

2 refers to the set of runs with fixed water flow rate and inlet temperature and a fixed rate of heat transfer in the heat exchanger.

3 refers to the set of runs where the flow rate of R-113 and the inlet temperatures of water and R-113 were fixed while the water flow rate varied.

4 refers to the set of runs having the same constraints as group 3 except that the flow rate of R-113 was different.

The last two digits in a given run designate the run number within a group

TABLE 4.1. Experimental Constraints for each Experimental Set.

Set No.	0		1		2		3		4	
Heat Exchanger No.	1 to 3	7 to 9	1 to 3	7 to 9	1 to 3	7 to 9	1 to 3	7 to 9	1 to 3	7 to 9
Inlet R-113 temperature, °C	40	40	40	40	---	---	40	40	40	40
Inlet water temperature, °C	15	15	15	15	15	15	15	15	15	15
Water flow rate, GPM	2.5	3.4	1.5	1.7	1.5	1.65	---	---	---	---
R-113 flow rate, GPM	---	---	---	---	---	---	.275	.500	.275	.500
Heat Flux, Btu/hr-ft^2	---	---	---	---	1200	1000	---	---	---	---

--- means varied within a set

of the same constraints. Table 4.1 gives a summary of the numerical values of the experimental constraints of each set of runs.

It is significant to point out the fact that since the performance of each heat exchanger depends on the type of its inner tube, each inner tube will be identified by the same number as the heat exchanger throughout the discussion of the results. For example, results of tube x refers to the results of the inner tube of heat exchanger x.

The inlet pressure of R-113 into all heat exchangers during all experimental runs was in the range of 110.32 kPa (16 psia) to 124.11 kPa (18 psia). Since within such a range the pressure has no effect on the heat transfer, its value for each individual run was not reported in the tables of Appendix C. Due to the fact that the pressure drop of R-113 for the individual subsections were very small thus making the credibility of the measurements questionable, the pressure drops of R-113 of the entire heat exchanger, between its inlet and exit, are reported. Table C.6 of Appendix C gives the pressure drop data for all heat exchangers. They were taken during isothermal runs of R-113 at different flow rates. It is to be be noticed that the experimental runs of pressure drop were designated by three digits, xxx, only. Beginning from left to right, the first digit designates the heat exchanger number and the next two digits designate the run number for a given heat exchanger.

During the heat transfer data reduction the inside heat transfer coefficient h_i, the outside heat transfer coefficient h_o of the inner tube of every heat exchanger, as well as its overall heat transfer rate (UA) were sought. These were obtained as follows:

The overall heat transfer rate (Q) is given by:

$$Q = UA\ (LMTD) \qquad (4\text{-}1)^*$$

where LMTD is the overall logarithmic mean temperature difference given by:

$$LMTD = \frac{(T_{Ri}-T_{wao}) - (T_{Ro}-T_{wai})}{\ln\{(T_{Ri}-T_{wao})/(T_{Ro}-T_{wai})\}} \qquad (4\text{-}2)$$

Q was determined from the rate of energy gain of the cooling water given by:

$$Q = m_{wa}\ Cp_{wa}\ (T_{wao}-T_{wai}) \qquad (4\text{-}3)$$

The outside heat transfer coefficient was determined from:

$$Q = h_o\ A_o\ (LMTD)_o \qquad (4\text{-}4)$$

where

$$(LMTD)_o = (T_{wao}-T_{wai})/\ln\{(\bar{T}_w-T_{wai})/(\bar{T}_w-T_{wao})\} \qquad (4\text{-}5)$$

$\bar{T}_w$ is the average outside surface temperature of the inner tube obtained from the arithmetic mean of the sixteen thermocouples attached to the outside surface of the inner tube of the heat exchanger. The inside heat transfer coefficient was then obtained from the following equation:

$$1/h_i = (D_i/D_o)\{A_o/(UA) - 1/h_o - (D_o/2k)\ln(D_o/D_i)\} \qquad (4\text{-}6)$$

All inside and outside heat transfer coefficients h_i and h_o respectively were based on the nominal inside and outside diameters D_i and D_o irrespective of the condition of the surface.

The friction coefficient f for pressure drop measurements was calculated from:

$$f = \Delta p\ /\ (L/2D_i)\rho\ V^2 \qquad (4\text{-}7)$$

---

* See nomenclature for the definition of symbols.

A sample of the data reduction computation for one of the runs is given in Appendix D. Appendix E gives the experimental uncertainity in the computation of the overall heat transfer rate UA. The uncertainity was estimated to be $\pm 17.9\%$.

4.2 HEAT TRANSFER RESULTS

The effects of inside augmentation, outside augmentation, and double augmentation on the inside heat transfer coefficient, outside heat transfer coefficient, and the overall heat transfer coefficient, respectively, will be discussed in this section.

4.2.1 Inside Heat Transfer Coefficients

Figures 4.1, 4.2 and 4.3 show plots of the inside heat transfer coefficients h_i versus Reynolds number of R-113 for the experimental sets 0,1 and 2 respectively for tubes 1,2 and 3. As was expected, tubes 1 and 3 which had internally finned tubes had a higher heat transfer coefficient compared to tube 2 which had a smooth inside surface. Also, for all tubes, h_i increased with the increase of Reynolds number of R-113. Also, irrespective of the condition of the outside surface of the tube, tubes 2 and 3 had the same inside heat transfer coefficient for the same Reynolds number. Figures 4.4, 4.5 and 4.6 show similar plots for tubes 7,8 and 9. The results are similar to those for tubes 1,2 and 3. However, the inside heat transfer coefficients for tubes 1,2 and 3 are higher than those for tubes 7,8 and 9 for the same Reynolds number of R-113. This is due to the fact that the inside nominal diameter of the first set is smaller than the inside nominal diameter of the second set. Figure 4.7 combines the results of Figs. 4.1,4.2 and 4.3 while Fig. 4.8 combines the results of Figs. 4.4,4.5 and 4.6 for the data sets 0,1 and 2. The effects of internal fins

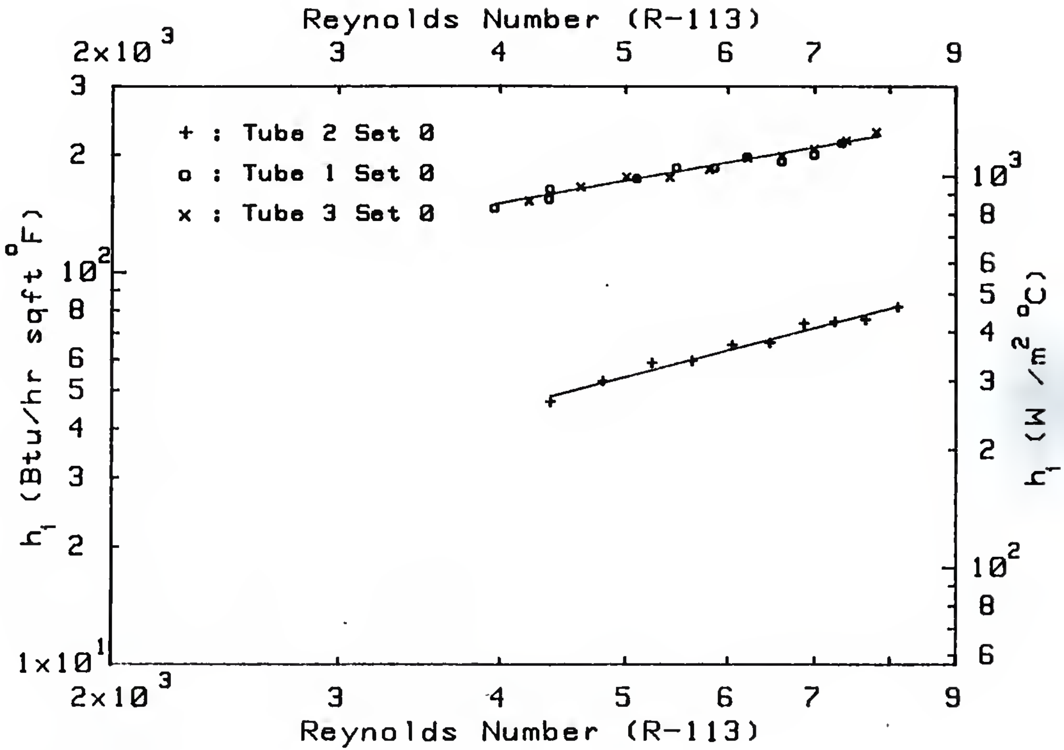

Fig. 4.1 Experimental overall inside heat transfer coefficient versus Reynolds number of R-113, tubes 1,2 and 3, set 0.

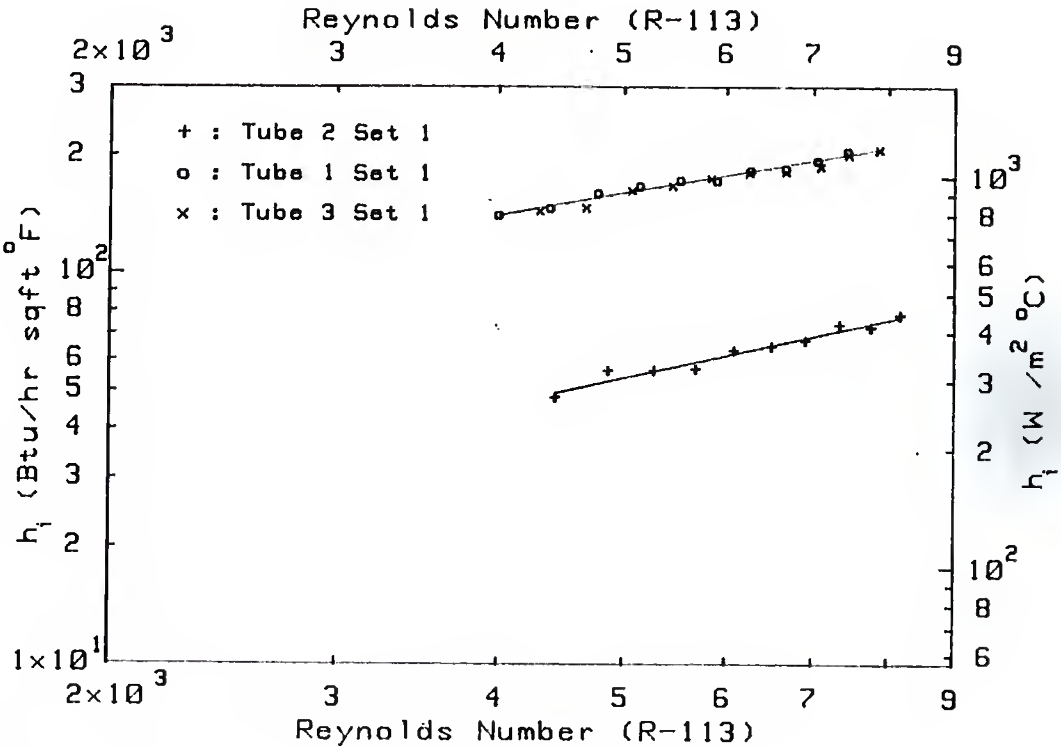

Fig. 4.2 Experimental overall inside heat transfer coefficient versus Reynolds number of R-113, tubes 1,2 and 3, set 1.

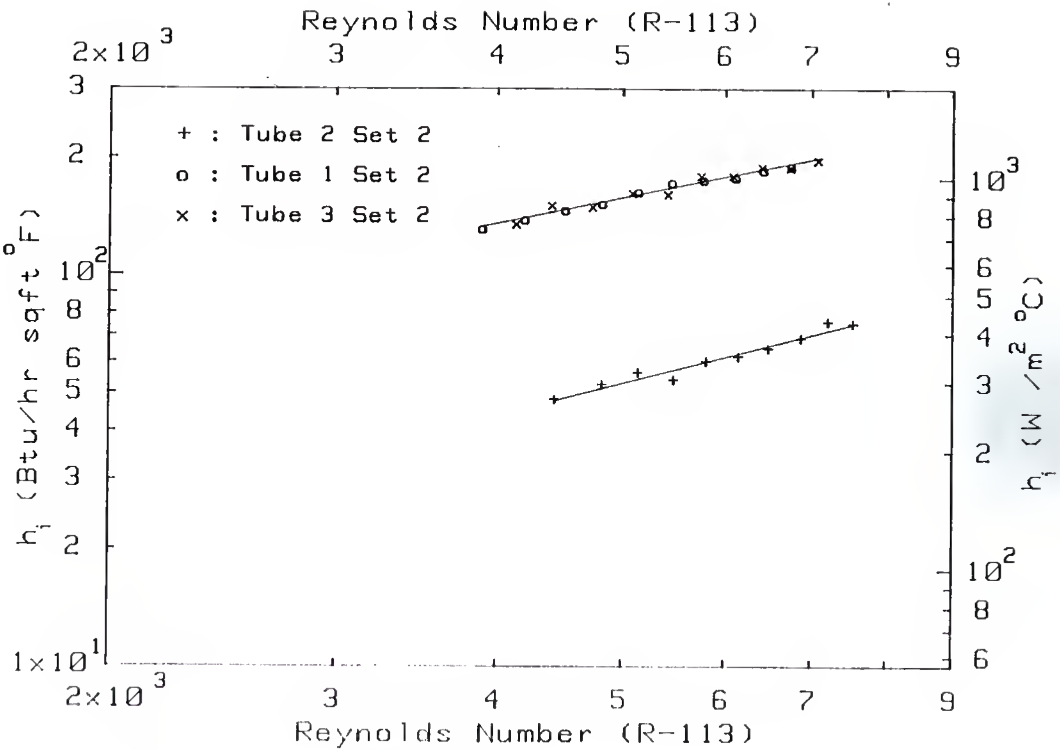

Fig. 4.3 Experimental overall inside heat transfer coefficient versus Reynolds number of R-113, tubes 1,2 and 3, set 2.

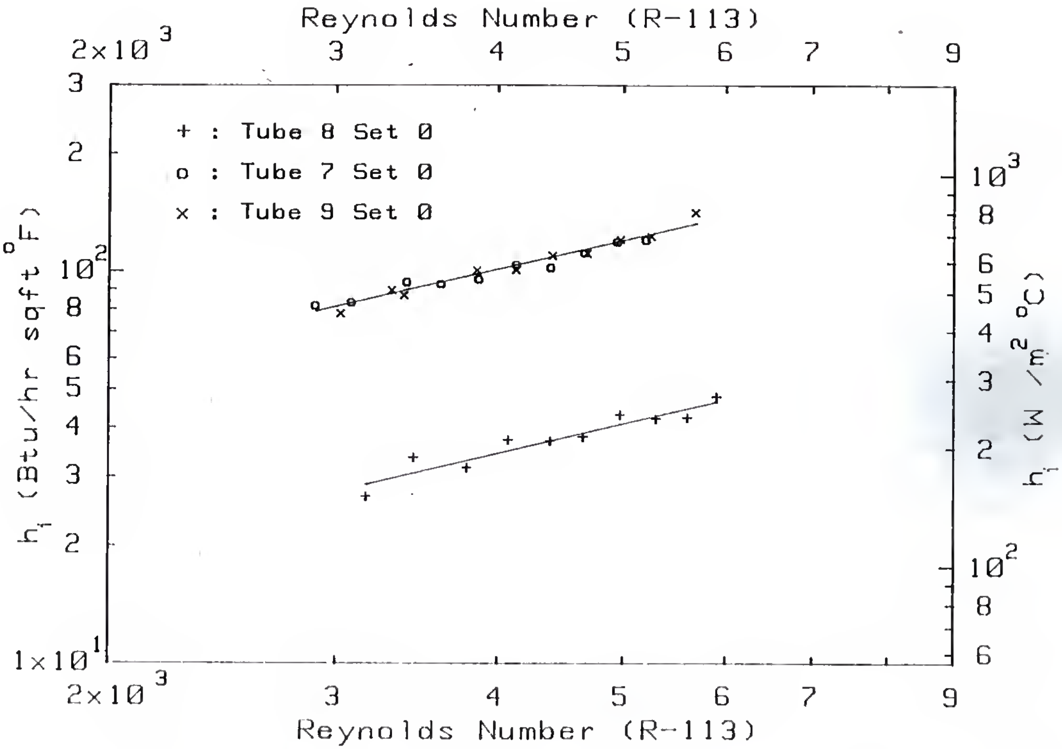

Fig. 4.4 Experimental overall inside heat transfer coefficient versus Reynolds number of R-113, tubes 7,8 and 9, set 0.

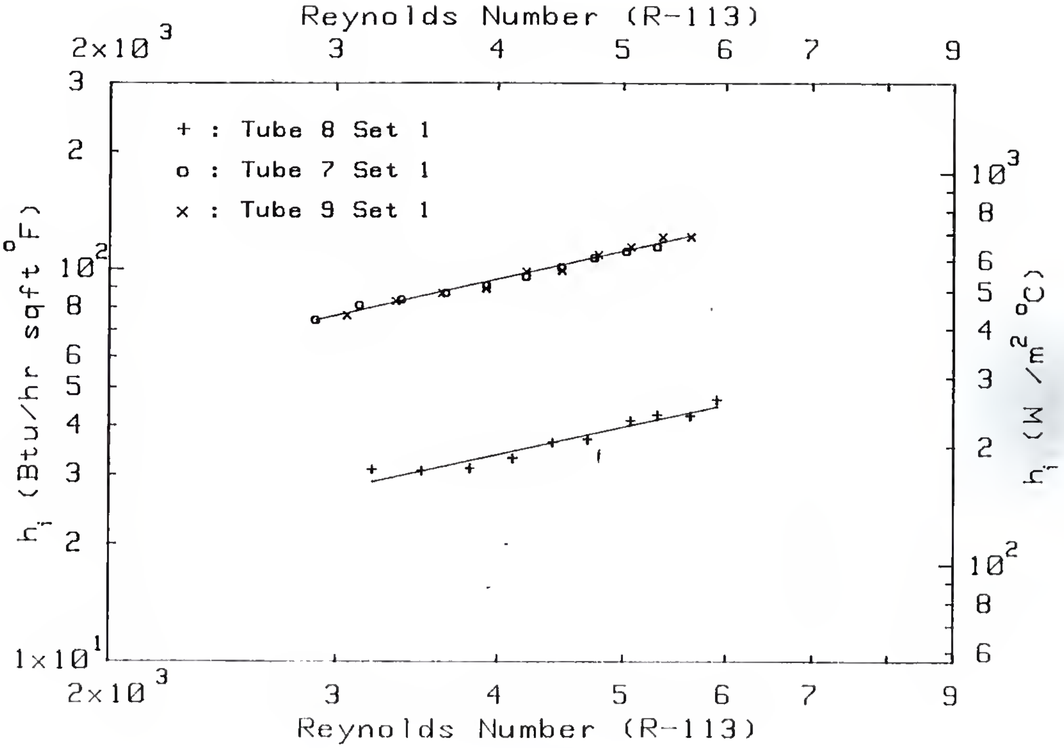

Fig. 4.5 Experimental overall inside heat transfer coefficient versus Reynolds number of R-113, tubes 7,8 and 9, set 1.

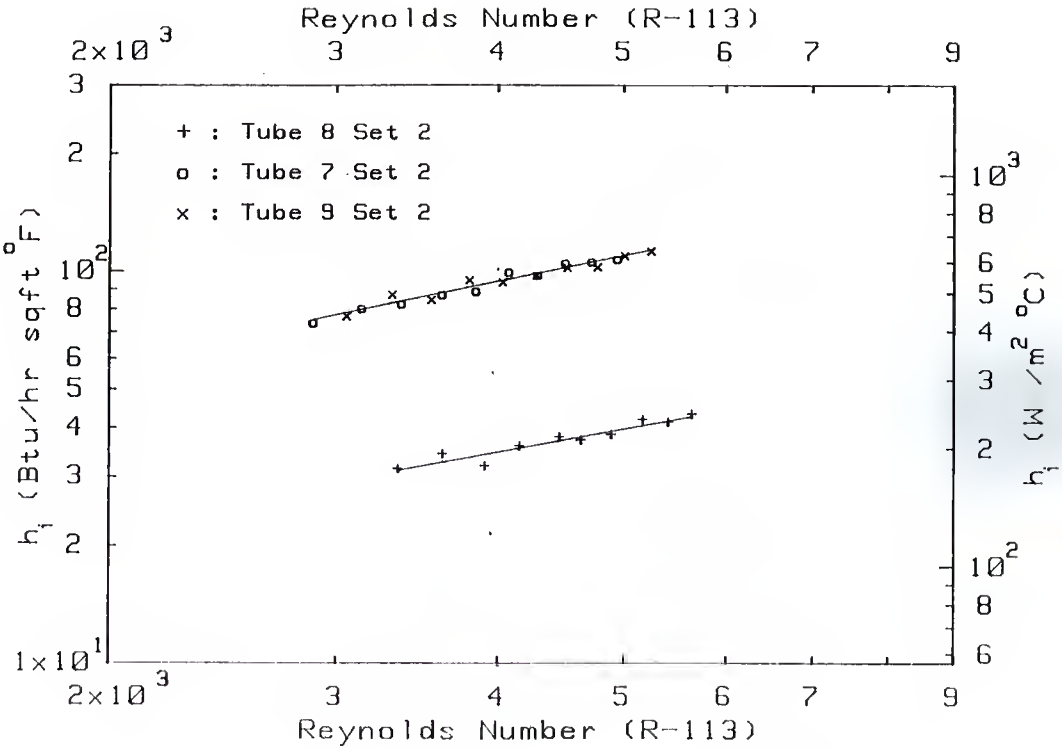

Fig. 4.6 Experimental overall inside heat transfer coefficient versus Reynolds number of R-113, tubes 7,8 and 9, set 2.

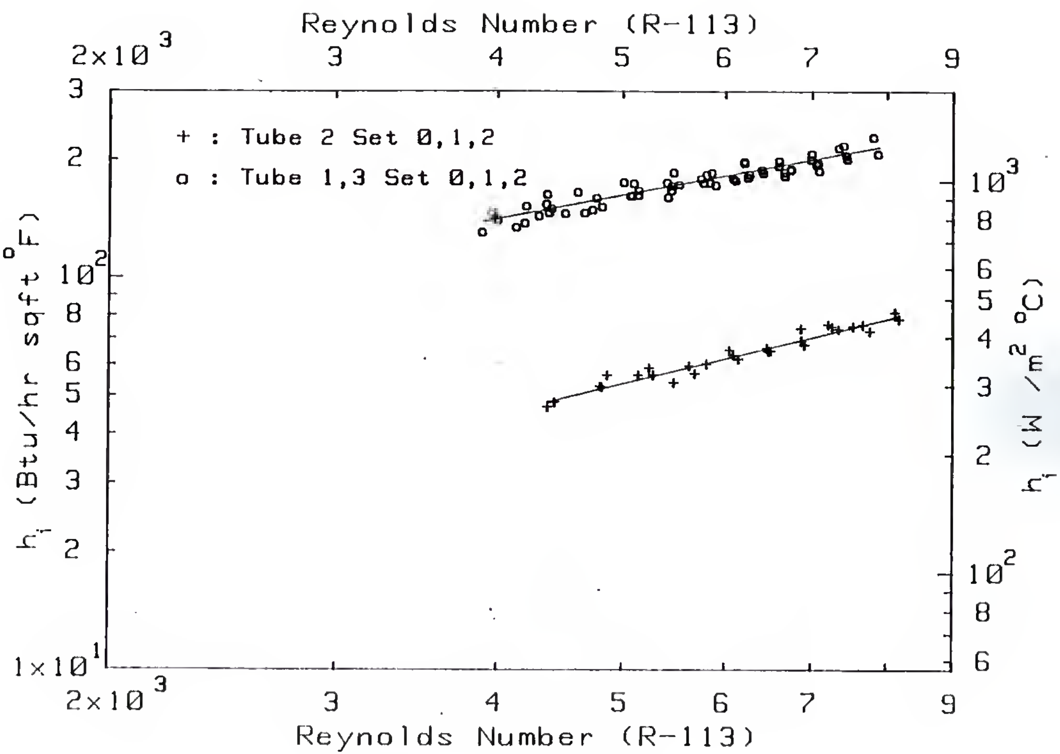

Fig. 4.7 Experimental overall inside heat transfer coefficient versus Reynolds number of R-113, tubes 1,2 and 3, sets 0,1 and 2.

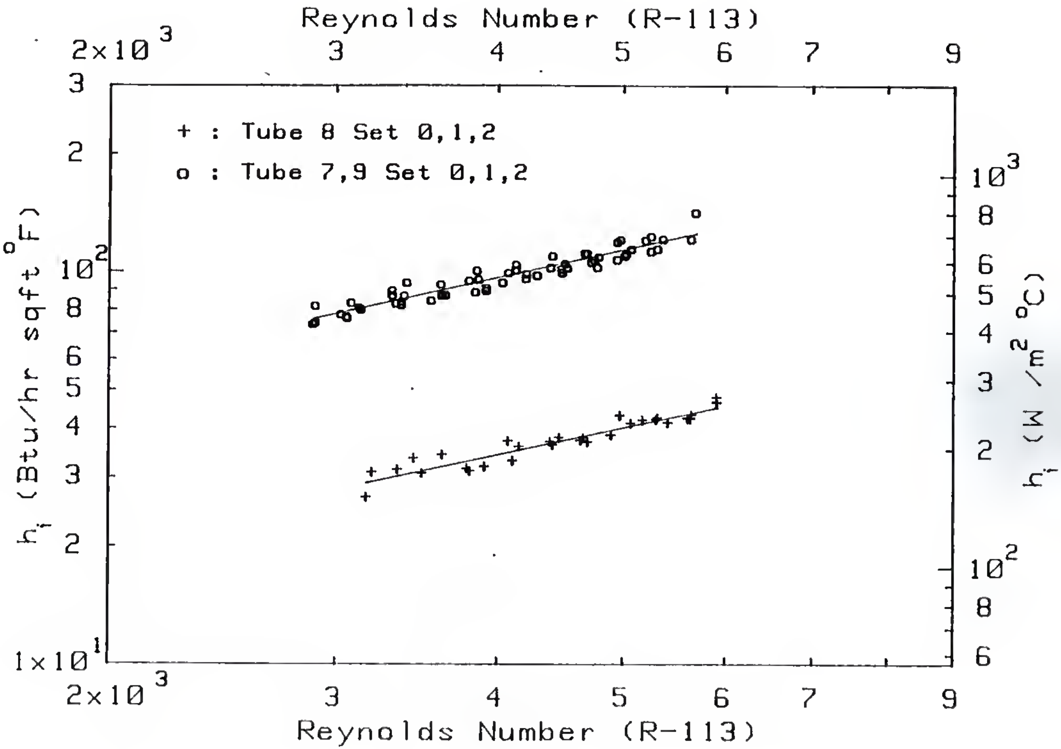

Fig. 4.8 Experimental overall inside heat transfer coefficient versus Reynolds number of R-113, tubes 7,8 and 9, sets 0,1 and 2.

on the enhancement of the inside heat transfer coefficient will be discussed in a later chapter.

4.2.2 Outside Heat Transfer Coefficient

The effects of outside roughness (knurls) on the outside heat transfer coefficient of the inner tubes are given in Figs. 4.9 through 4.14. The experimental outside heat transfer coefficients of experimental data sets 3 and 4 in which the flow rate of R-113 was fixed while the coolant (water) flow rate was varied were plotted. Figures 4.9 and 4.10 include the results of tubes 1,2 and 3 while Figs. 4.11 and 4.12 include the results of tubes 7,8 and 9. It is obvious that the tubes with outside knurls had a higher heat transfer coefficient than the tubes with smooth surfaces. Also, in all the figures, h_o increased with the increase of Reynolds number. Figure 4.13 combines the data of Figs. 4.9 and 4.10, while Fig. 4.14 combines the data of Figs. 4.11 and 4.12. Due to the fact that the range of Prandtl number of water covered in this study varied very slightly, the results of h_o could be plotted versus Reynolds number. It is also to be observed that h_o of tubes 2 and 3 (with knurls) were the same for the same Reynolds number of water. Also, h_o for tubes 8 and 9 were the same at the same Reynolds number. Although the range of Reynolds number of water for tubes 1,2 and 3 and tubes 7,8 and 9 did not overlap, by a little extrapolation of regression lines in Figs. 4.13 and 4.14, it may be seen that the outside heat transfer coefficient h_o is higher for tube 1 than for tube 7 at the same Reynolds number of water. Similarly, the outside heat transfer coefficient for tubes 2 and 3 is higher than for tubes 8 and 9 respectively. This is because tubes 1,2 and 3 had a lower hydraulic diameter than tubes 7,8 and 9, and thus for the same Reynolds number of

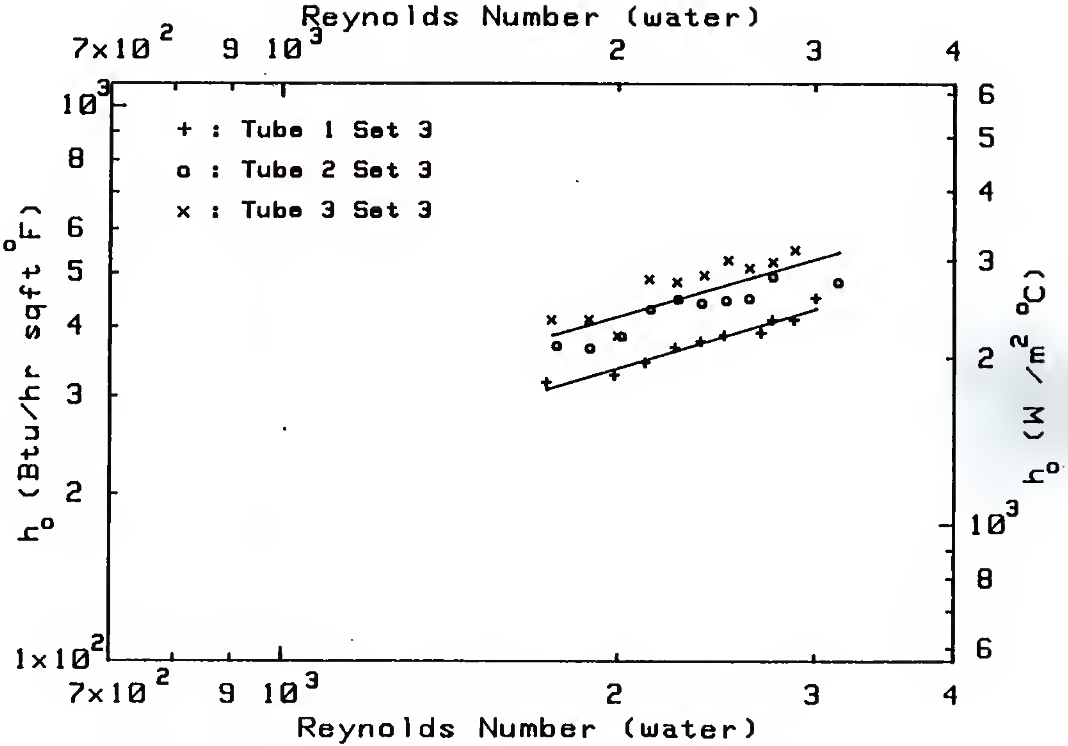

Fig. 4.9 Experimental overall outside heat transfer coefficient versus Reynolds number of water, tubes 1,2 and 3, set 3.

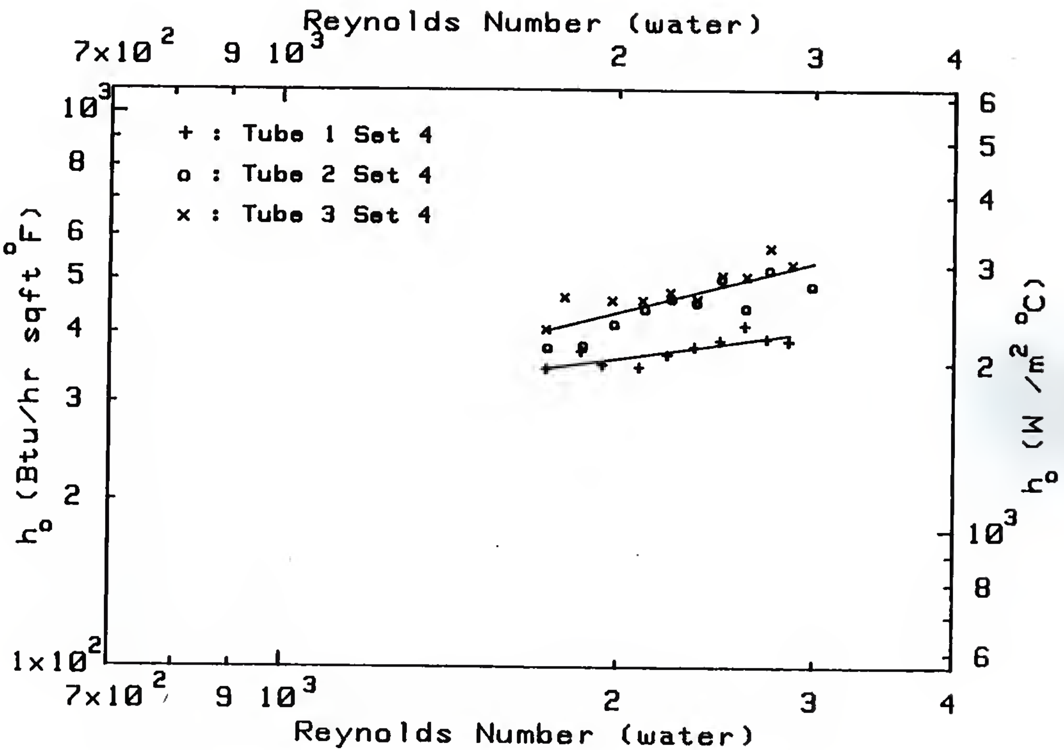

Fig. 4.10 Experimental overall outside heat transfer coefficient versus Reynolds number of water, tubes 1,2 and 3, set 4.

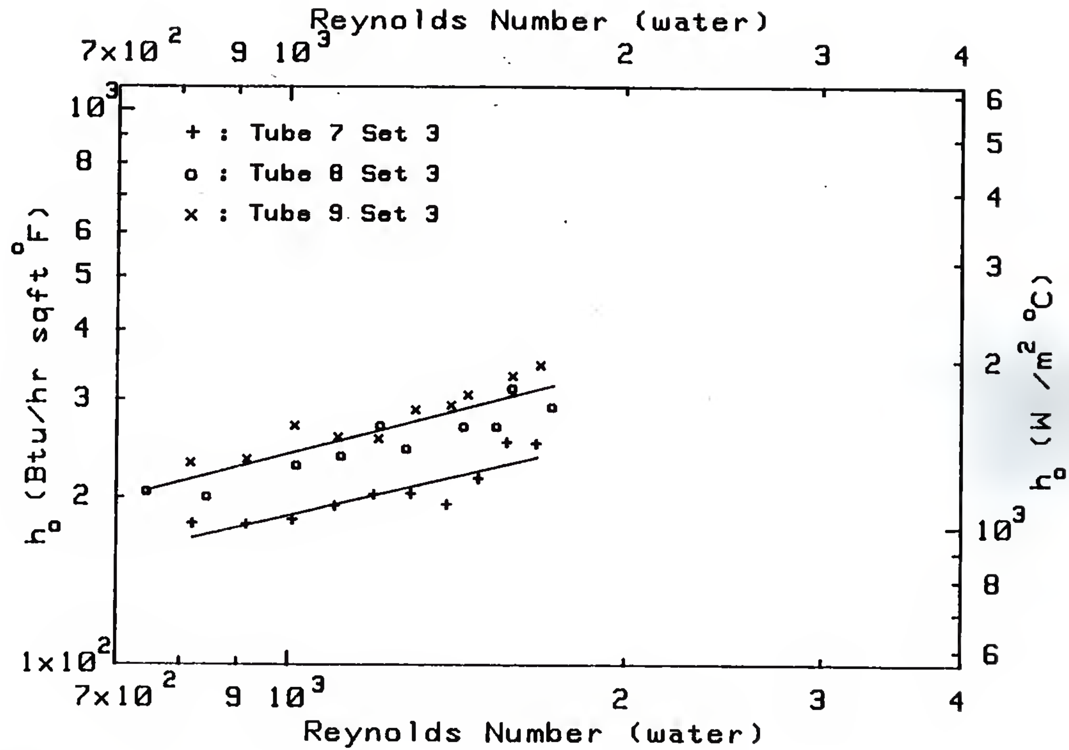

Fig. 4.11 Experimental overall outside heat transfer coefficient versus Reynolds number of water, tubes 7,8 and 9, set 3.

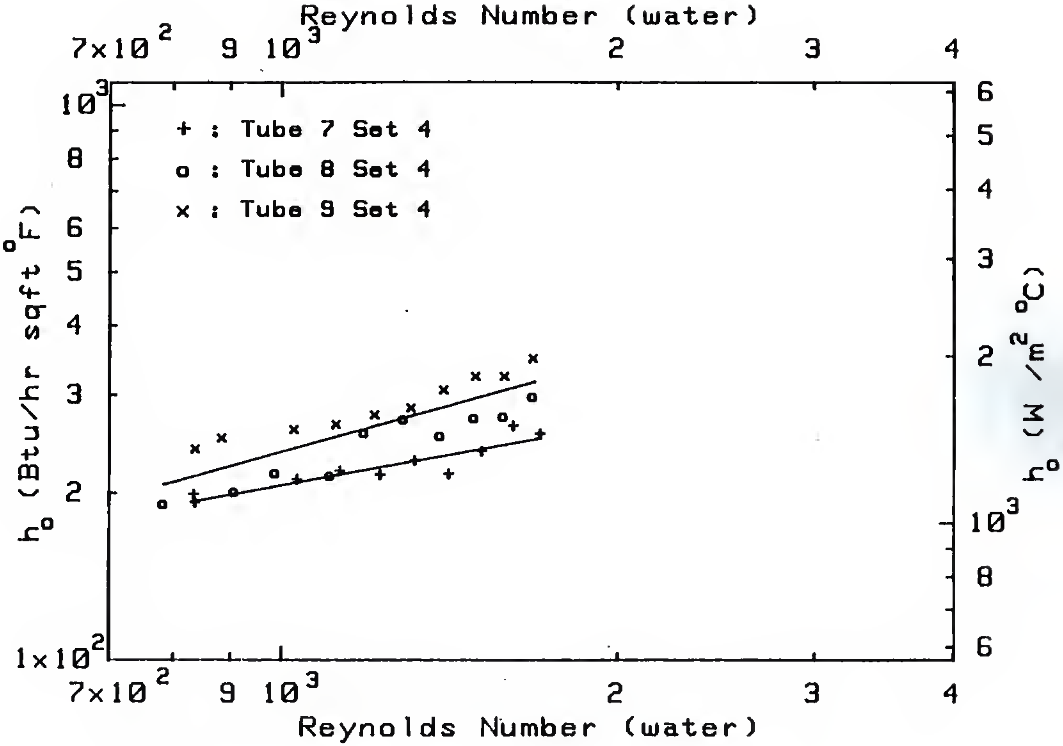

Fig. 4.12 Experimental overall outside heat transfer coefficient versus Reynolds number of water, tubes 7,8 and 9, set 4.

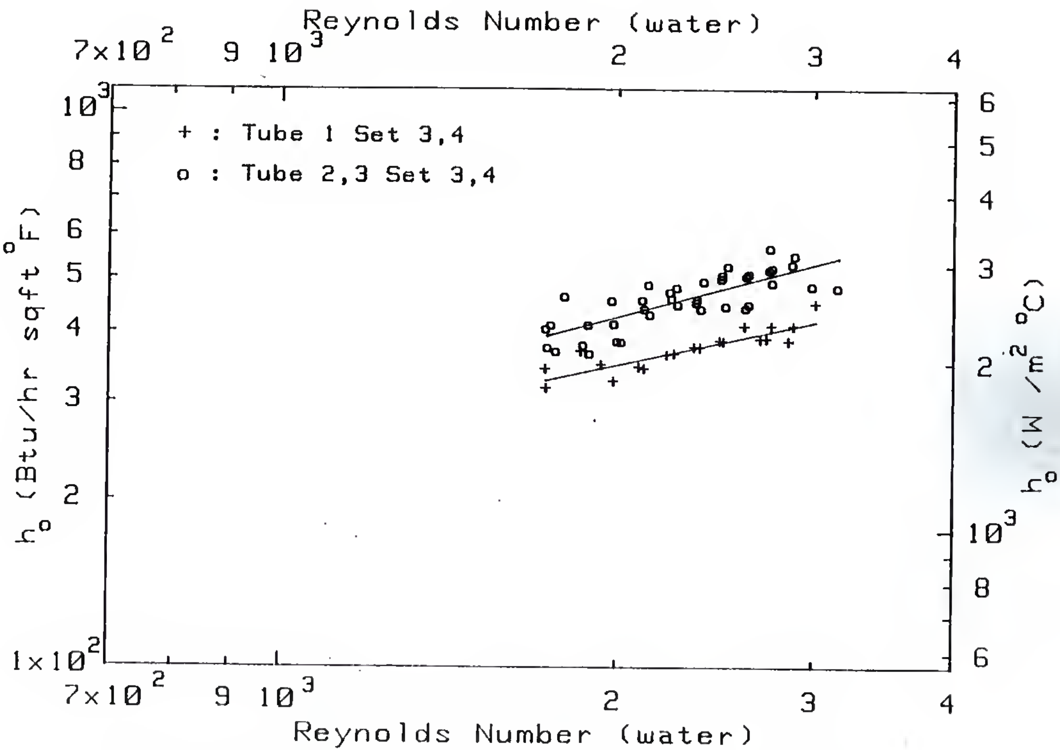

Fig. 4.13 Experimental overall outside heat transfer coefficient versus Reynolds number of water, tubes 1,2 and 3, sets 3 and 4.

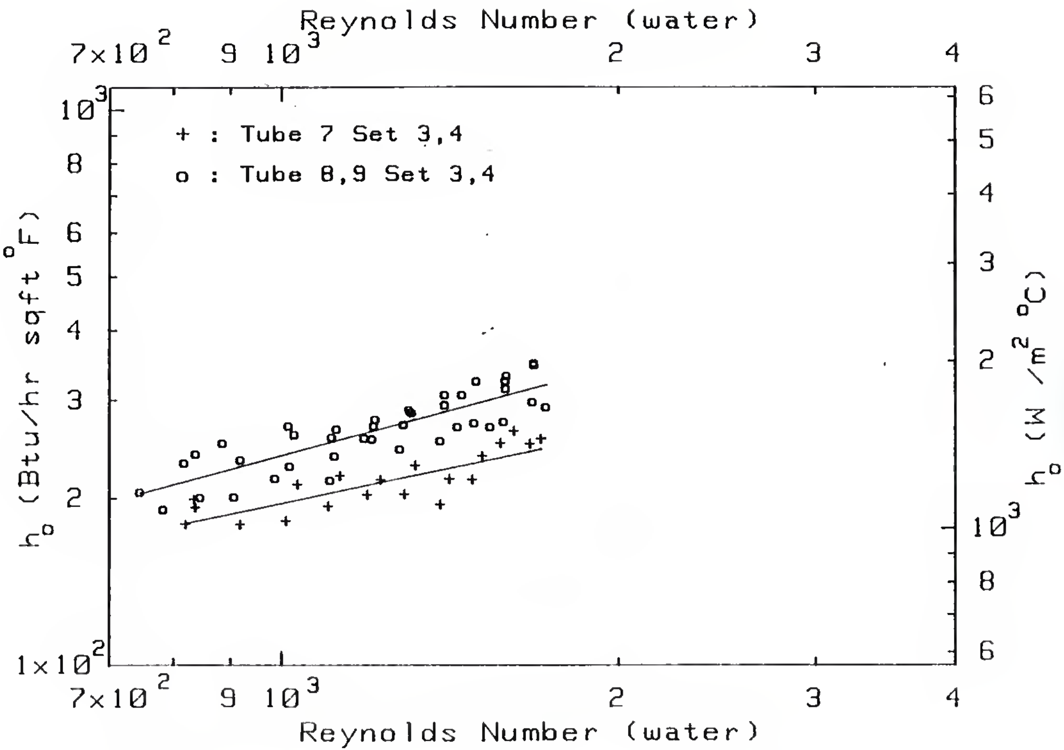

Fig. 4.14 Experimental overall outside heat transfer coefficient versus Reynolds number of water, tubes 7,8 and 9, sets 3 and 4.

water, velocity of water in tubes 1,2 and 3 would be higher than in tubes 7,8 and 9 resulting in higher outside heat transfer coefficients.

It is to be pointed out that all lines drawn in Figs. 4.1 through 4.14 as well as the remaining figures of this chapter are the regression lines obtained by the least square regression analysis. The quantitative improvement in the outside heat transfer coefficient by the addition of the knurls will be discussed in a later chapter.

4.2.3 Overall Heat Transfer Coefficient

The effect of enhancing the inside and/or the outside heat transfer coefficient of the inner tube of the heat exchanger is usually reflected in the resulting overall heat transfer coefficient U. Usually the overall heat transfer coefficient can be based on either the inside or the outside surface areas of the inner tube. Usually, the choice of the area is based on the side which has the controlling heat transfer resistance. In the following section comparisons were made between the overall rate of heat transfer per unit temperature difference (UA) for the different tubes tested.

Figures 4.15,4.16 and 4.17 show plots of UA versus the Reynolds number of R-113 for experimental data sets 0,1 and 2 respectively, for tubes 1,2 and 3. Figures 4.18,4.19 and 4.20 show similar plots for tubes 7,8 and 9. Two observations are worth noticing in these figures. First, tubes 1 and 3, and tubes 7 and 9 had the same UA for the same Reynolds number of R-113. Second, UA in all figures showed a definite increase with Reynolds number of R-113 for all tubes and for all data sets.

Figures 4.21 and 4.22 show plots of UA versus Reynolds number of water for experimental data sets 3 and 4 respectively for tubes 1,2 and 3. Figures 4.23 and 4.24 show similar plots for tubes 7,8 and 9. The results

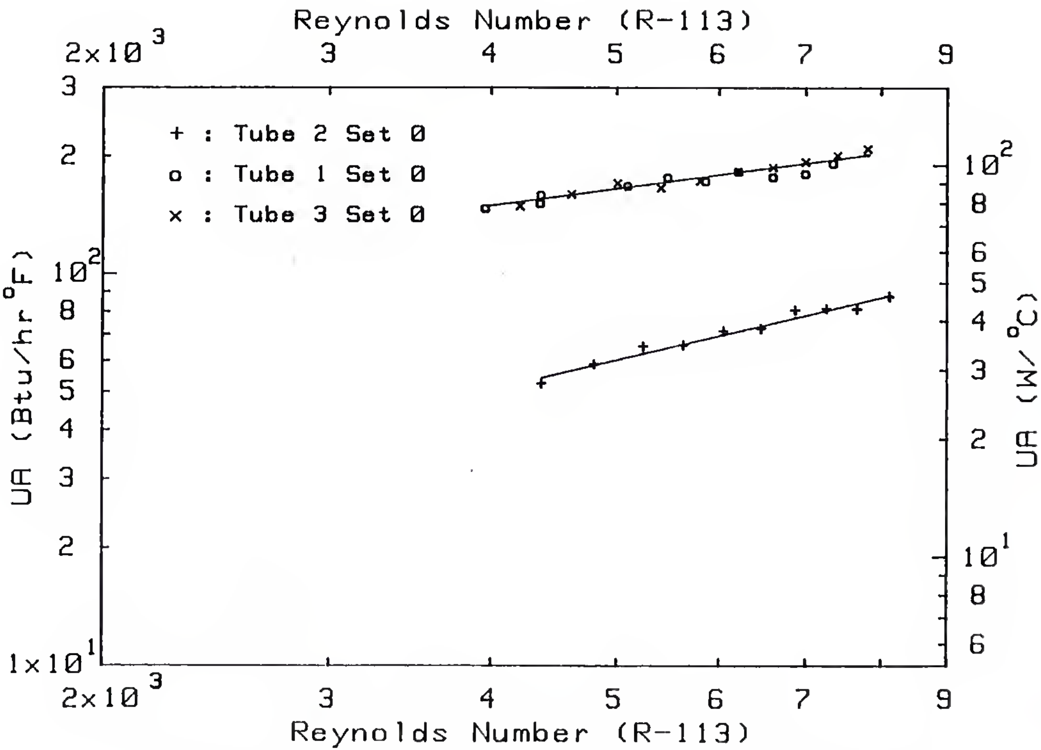

Fig. 4.15 Experimental overall heat transfer conductance (UA) versus Reynolds number of R-113, tubes 1,2 and 3, set 0.

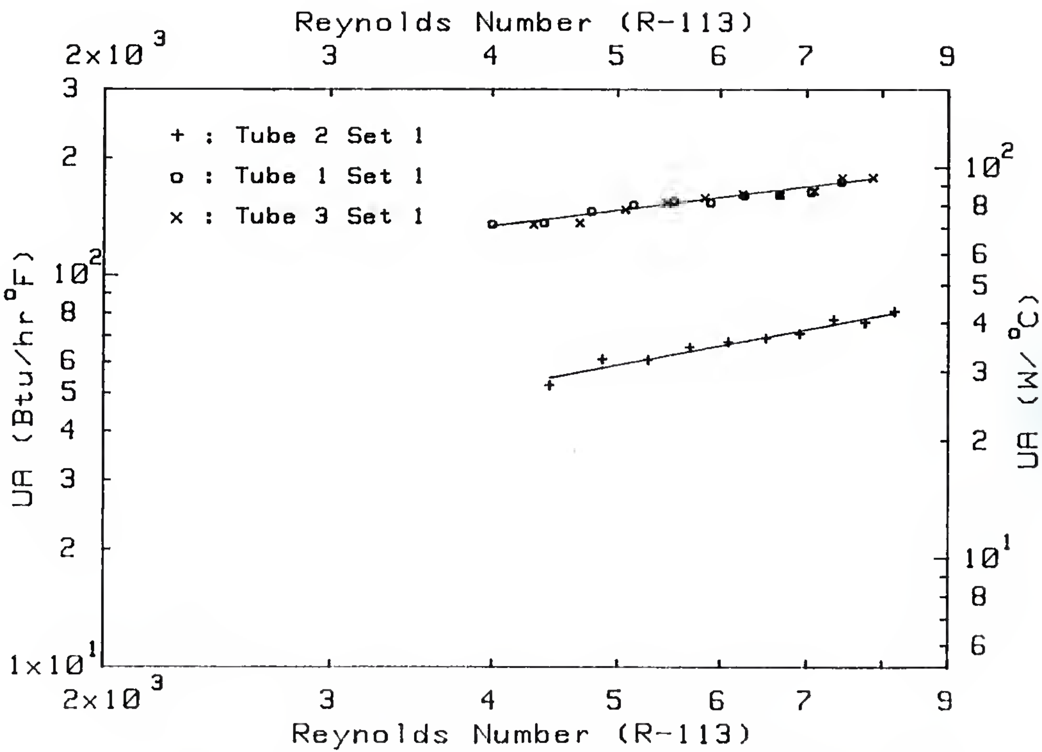

Fig. 4.16 Experimental overall heat transfer conductance (UA) versus Reynolds number of R-113, tubes 1,2 and 3, set 1.

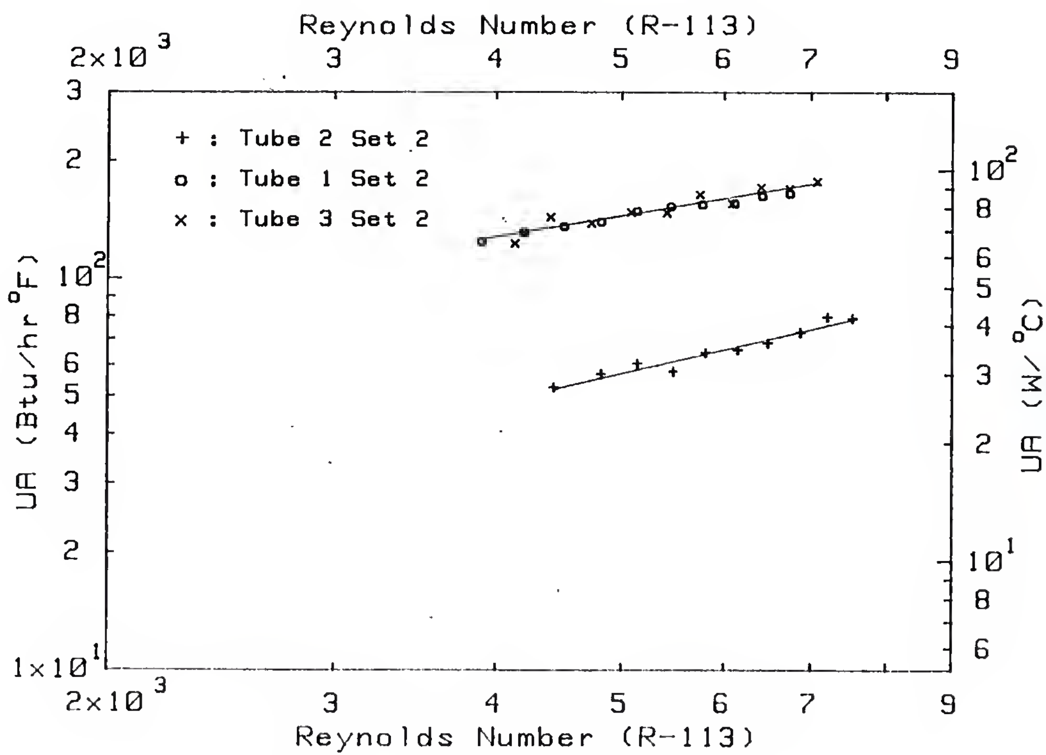

Fig. 4.17 Experimental overall heat transfer conductance (UA) versus Reynolds number of R-113, tubes 1,2 and 3, set 2.

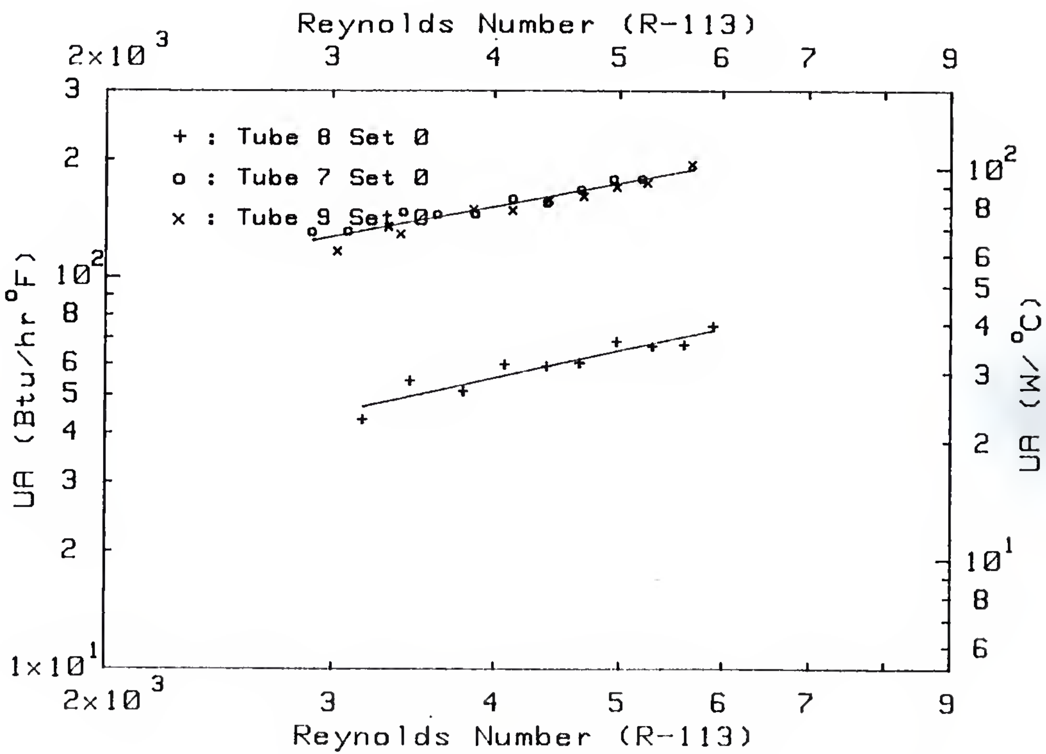

Fig. 4.18 Experimental overall heat transfer conductance (UA) versus Reynolds number of R-113, tubes 7,8 and 9, set 0.

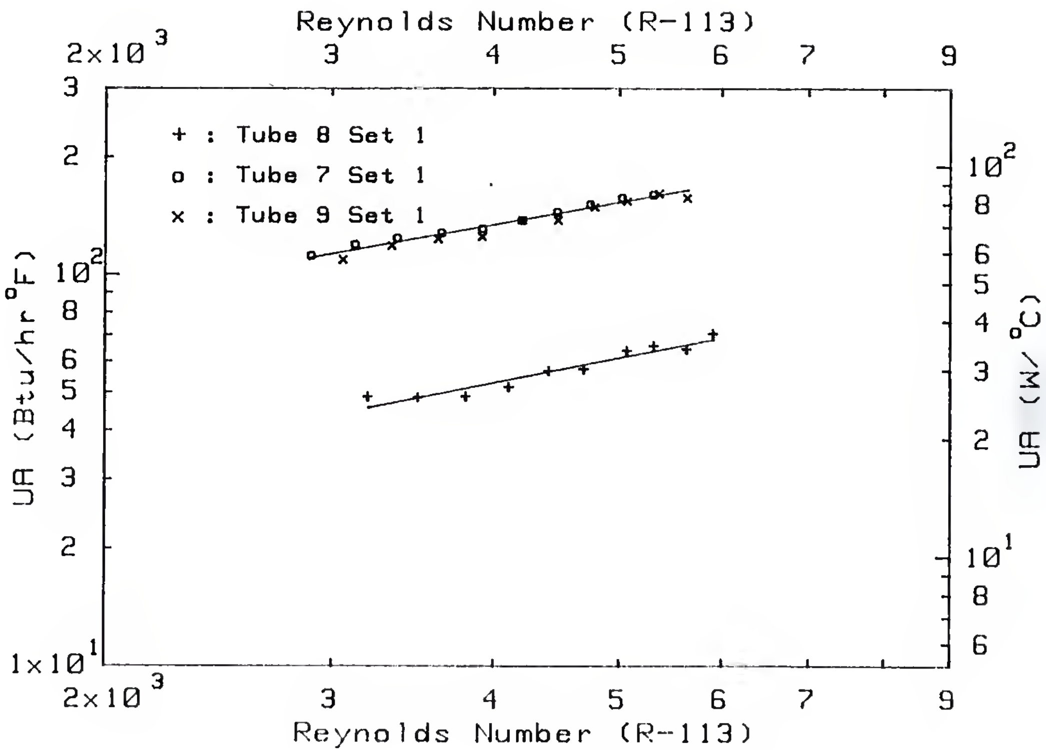

Fig. 4.19 Experimental overall heat transfer conductance (UA) versus Reynolds number of R-113, tubes 7,8 and 9, set 1.

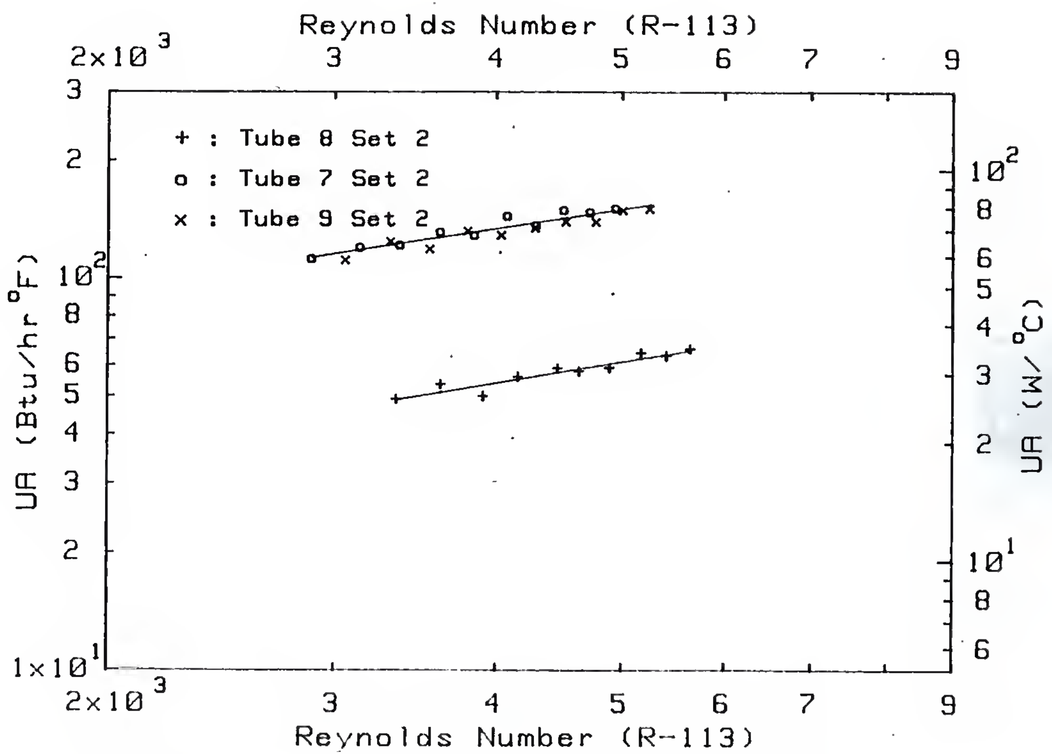

Fig. 4.20 Experimental overall heat transfer conductance (UA) versus Reynolds number of R-113, tubes 7,8 and 9, set 2.

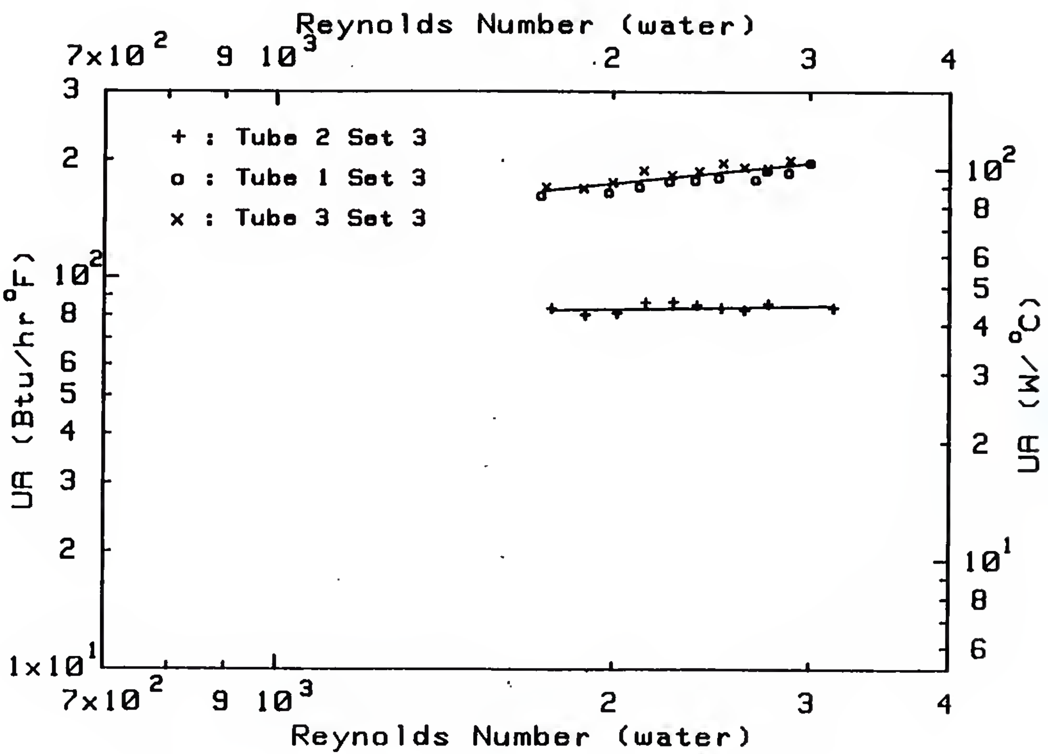

Fig. 4.21 Experimental overall heat transfer conductance (UA) versus Reynolds number of water, tubes 1,2 and 3, set 3.

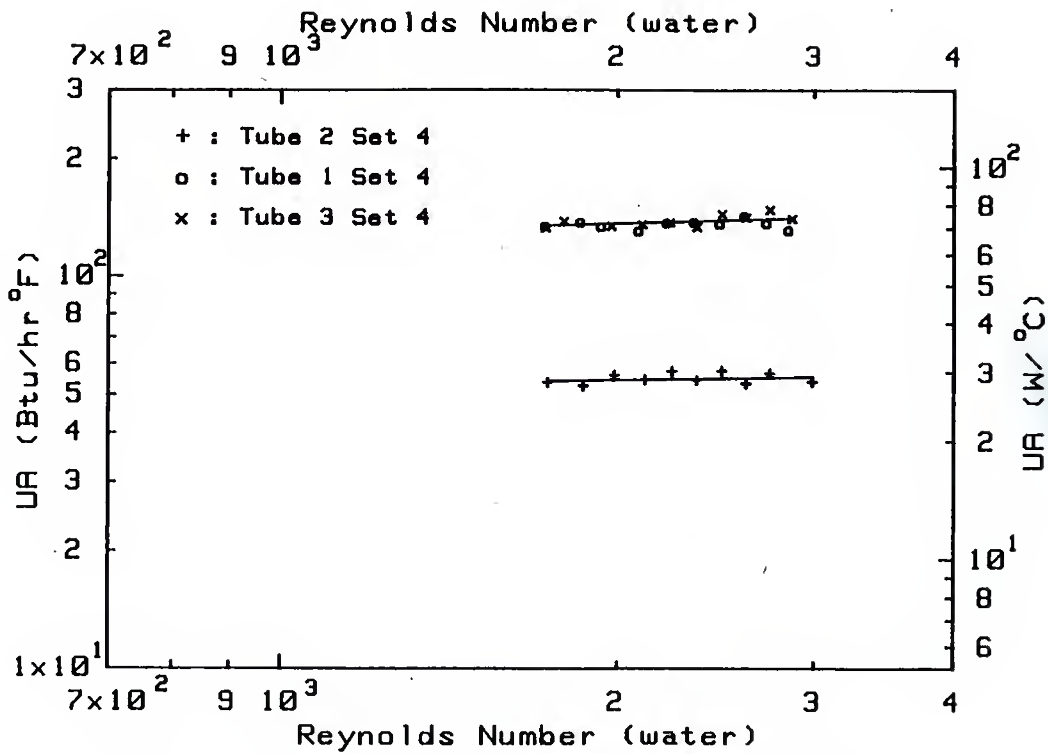

Fig. 4.22 Experimental overall heat transfer conductance (UA) versus Reynolds number of water, tubes 1,2 and 3, set 4.

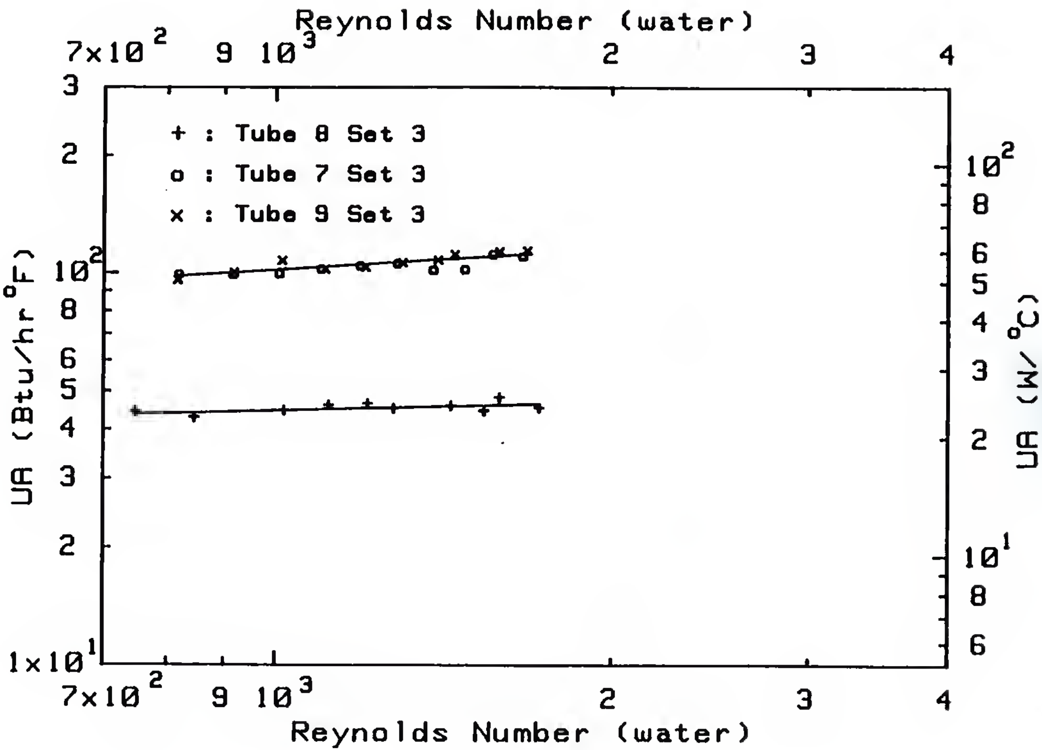

Fig. 4.23 Experimental overall heat transfer conductance (UA) versus Reynolds number of water, tubes 7,8 and 9, set 3.

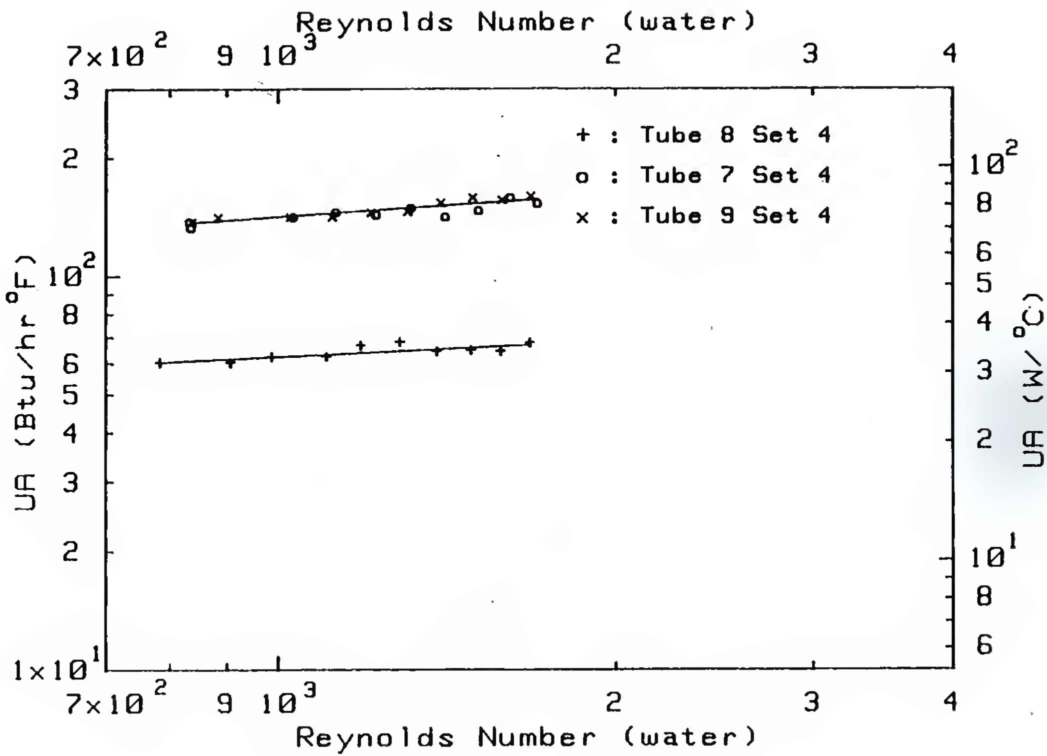

Fig. 4.24 Experimental overall heat transfer conductance (UA) versus Reynolds number of water, tubes 7,8 and 9, set 4.

also show that tubes 1 and 3, and tubes 7 and 9 had the same value of UA for the same Reynolds number of water. However, the rate of increase of UA with the Reynolds number of water is less than the rate of its increase with Reynolds number of R-113. The reason for such results is the fact that the outside heat transfer coefficient of the cooling water of the smooth surface is about three times larger than the inside heat transfer coefficient of R-113 for the smooth surface. As a result, the enhancement of the inside heat transfer coefficient by internally finned tubes or by increasing Reynolds number of R-113 is more effective in increasing UA than increasing the outside heat transfer coefficient by surface roughness (knurls) or increasing the Reynolds number of water. More will be said about UA in a later chapter.

4.3 PRESSURE DROP RESULTS

As mentioned earlier, pressure drop measurements were made for the flow of R-113 inside the tubes. The friction coefficient was calculated from Eq. (4-7). Figure 4.25 shows a plot of the friction coefficient f versus Reynolds number of R-113 for tubes 2 and 8 which had smooth inside surface. Figure 4.26 shows a similar plot for tubes 1,3,7 and 9 which were internally finned. As expected, the friction coefficient for internally finned tubes is higher than for smooth tubes. In a later chapter a qualitative analysis will be made on the benefits of increasing the heat transfer by internally finned tubes and penalties of increasing the pressure drop.

It is significant to point out that in this study, the characteristic dimension used in defining the Reynolds number for the flow inside finned or smooth tubes was the nominal inside diameter as if the fins did not exist (see Fig. 3.8). D_i was also used in calculating the inside mass flux

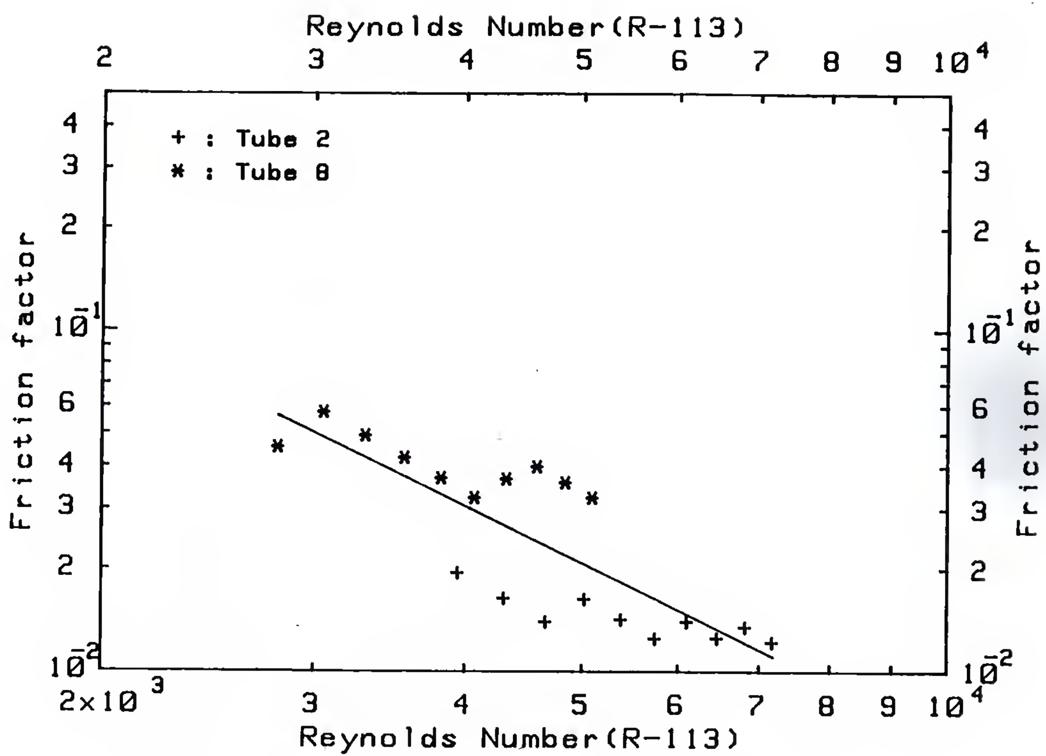

Fig. 4.25 Experimental friction factor versus Reynolds number of R-113, tubes 2 and 8.

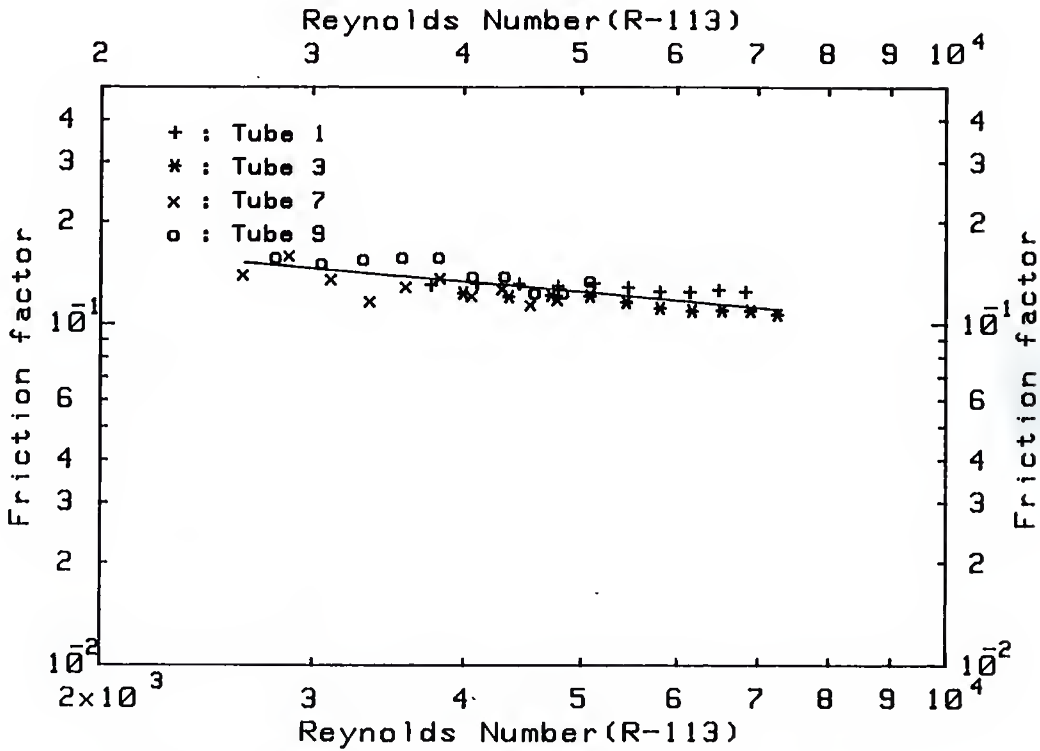

Fig. 4.26 Experimental friction factor versus Reynolds number of R-113, tubes 1,3,7 and 9.

of R-113 and the inside Reynolds number. The characteristic dimension used in calculating the Reynolds number and Nusselt number of water was the hydraulic diameter of the annulus.

Chapter V

CORRELATION OF EXPERIMENTAL DATA

5.1 INTRODUCTION

Figures 5.1 and 5.2 show plots of $Nu/Pr^{0.4}$ versus Reynolds number for inside and outside heat transfer coefficients of tubes tested. It is to be recalled that tubes 2 and 8 were smooth on the inside while tubes 1,3,7 and 9 were internally finned. Also, tubes 1 and 7 were smooth on the outside while tubes 2,3,8 and 9 were knurled on the outside. An attempt was made in the present study to develop design correlations for predicting the heat transfer and pressure drop during single phase flow inside internally finned tubes. Before developing these new correlations, existing correlation are discussed in the following section.

5.2 EXISTING HEAT TRANSFER CORRELATIONS

All existing heat transfer and pressure drop correlations were developed by applying modifying factors to smooth tube correlations to bring about the best agreement between the predictions and experimental measurements. Such modifying factors usually include certain geometric parameters which are pertinent to the geometry of the finned tube.

Watkinson et al. [3] developed the following heat transfer correlation for internally finned tubes for heating water in turbulent flow. Data were taken for Reynolds number in the range of 10,000 to 150,000. Smooth tube data were predicted by the Sieder-Tate equation:

$$Nu = 0.026 \; Re^{0.8} \; Pr^{1/3} \; (\mu_{wa}/\mu_{wall})^{0.14} \tag{5-1}$$

For finned tubes, for $0.12<b/D<0.52$, $9.2<P/D<79.2$ and for the range $5000<Re<100,000$, the above equation was modified to obtain the following equation:

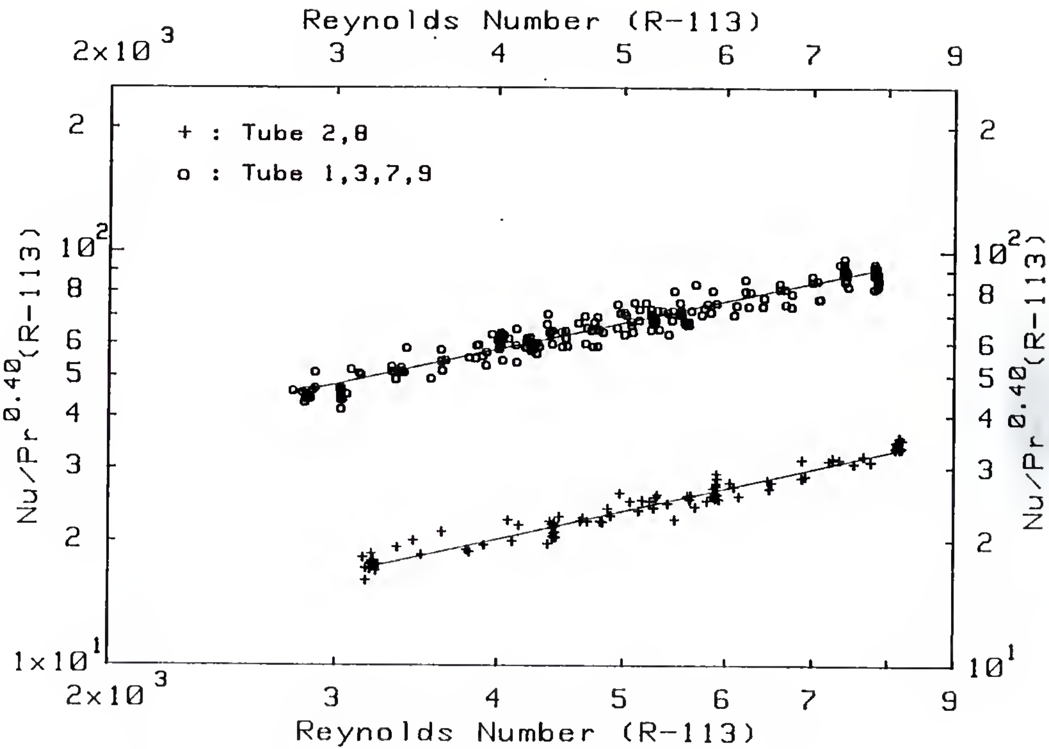

Fig. 5.1 Experimental $Nu/Pr^{0.4}$(R-113) versus Reynolds number of R-113, tubes 1,2,3,7,8 and 9

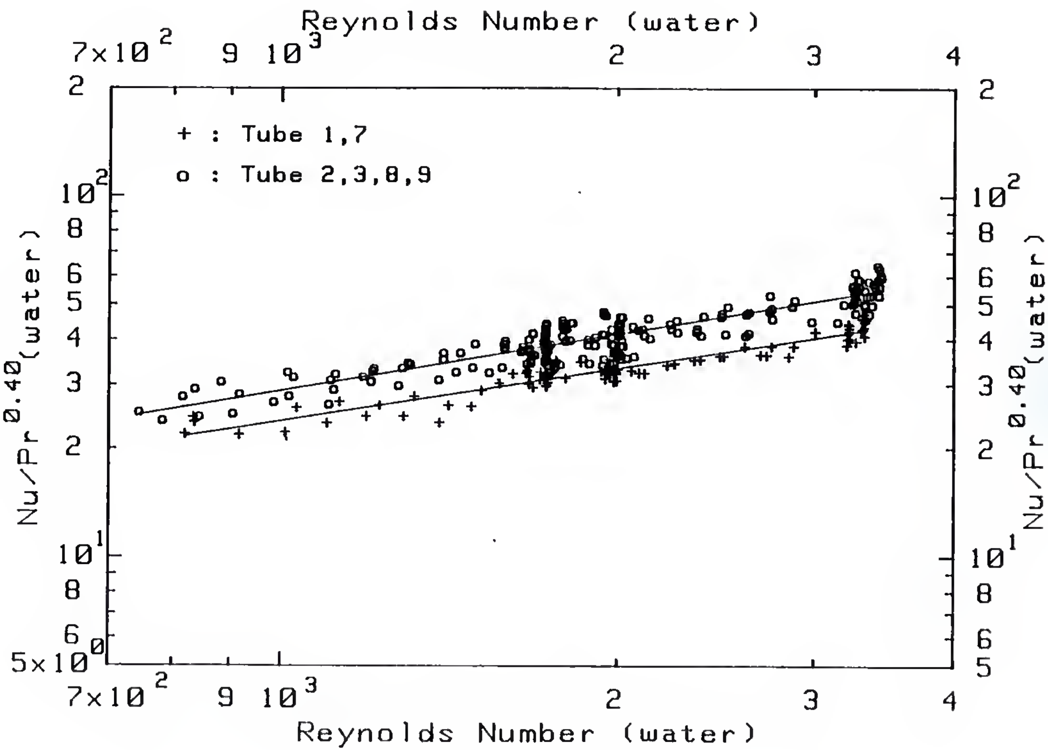

Fig. 5.2 Experimental Nu/Pr$^{0.4}$(water) versus Reynolds number of water, tubes 1,2,3,7,8 and 9

$$\frac{Nu_h}{Pr^{1/3}\,(\mu_{wa}/\mu_{wall})^{0.14}} = 0.369\ Re_h^{0.63}\,(P/D_h)^{-0.27}(b/D_h)^{0.21} \qquad (5\text{-}2)$$

For the same tubes, Watkinson et al. [4] developed the following correlations for heating air in turbulent flow:

For smooth tubes:

$$Nu = 0.039\ Re^{0.75}Pr^{0.4}\,(T_{bulk}/T_{wall})^{0.5} \qquad (5\text{-}3)$$

For internally finned tubes the correlation was:

$$\frac{Nu_h}{Pr^{0.4}\left(\frac{T_{bulk}}{T_{wall}}\right)^{0.5}} = 0.242Re_h^{0.645}\left(\frac{P}{D_h}\right)^{-2.95/Re_h}\left(\frac{b}{D_h}\right)^{0.0045Re} \qquad (5\text{-}4)$$

Watkinson et al. [5] developed a heat transfer correlation for internally finned tubes in laminar oil flow. Prandtl number range of 180 to 250 and Reynolds number (based on inside diameter) range of 500 to 3000 were covered. Smooth tube results were correlated by the equation of Kern and Othmer [23] for Reynold numbers below 1300 (which marked the beginning of the transition zone). This equation was:

$$\frac{Nu}{2.25Pr^{1/3}\left[\frac{D}{L}\right]^{1/3}\left[\frac{\mu}{\mu_{wa}}\right]^{0.14}\left[\frac{1+0.01Gr^{1/3}}{\log Re}\right]} = 1.86\ Re^{1/3} \qquad (5\text{-}5)$$

This correlation for spirally finned tubes for $0.0031<b/P<0.0383$, $14<Re<1300$ and $Gz>50$ was:

$$\frac{Nu}{2.25Pr^{1/3}\left[\frac{D_h}{L}\right]^{1/3}\left[\frac{\mu}{\mu_{wa}}\right]^{0.14}\left[\frac{1+0.01Gr_h^{1/3}}{\log Re_h}\right]} = 19.2\left(\frac{b}{P}\right)^{0.5} Re_h^{0.26} \qquad (5\text{-}6)$$

Experimental heat transfer data for cooling air inside internally finned tubes during turbulent flow were taken by Carnavos [7]. He developed the following correlation:

$$Nu_h/Pr^{0.4} = 0.023\ Re_h^{0.80}\ F_1^{0.10}\ F_2^{0.50}\ F_3^{3} \tag{5-7}$$

where F_1,F_2 and F_3 were modifying factors defined as follows:

$$F_1 = A_{fa}/A_{fc} \tag{5-8}$$

$$F_2 = A_n/A_a \tag{5-9}$$

$$F_3 = \mathrm{Sec}\ \alpha \tag{5-10}$$

where

A_{fa} = actual flow area, $\pi D_e^2/4$, m^2 (ft^2)

A_{fc} = open core free flow area, $\pi D_c^2/4$, m^2 (ft^2)

A_n = nominal heat transfer area based on D_i as if fins were not present, m^2/m (ft^2/ft)

A_a = actual heat transfer area, m^2/m (ft^2/ft)

A_{fn} = nominal flow area based on D_i as if fins were not present, $\pi D_i^2/4$, m^2 (ft^2)

In the above definitions, the equivalent diameter D_e and the core diameter D_c are defined by:

$$\pi D_e^2/4 = \pi D_i^2/4 - nbt/\cos\alpha \tag{5-11}$$

$$\pi D_c^2/4 = \pi (D_i - 2b)^2/4 \tag{5-12}$$

Also, a hydraulic diameter was defined by:

$$D_h = 4A_{fa}/A_a \tag{5-13}$$

On the basis of the above definitions it can be shown that:

$$F_1 = A_{fa}/A_{fc} = (D_e/D_i)^2/[1 - (2b/D_i)]^2$$
$$= [1 - (4nbt)/(\pi D_i^2 \cos\alpha)]/[1 - 2b/D_i]^2 \tag{5-14}$$

and

$$F_2 = A_n/A_a = (D_i D_h/D_e^2) = \pi D_i/[\pi D_i + 2nb/\cos\alpha] \qquad (5\text{-}15)$$

See Fig. 3.8 for the identification of all the geometric parameters.

Since the range of experimental Reynolds number covered in the present study was in the transition range $2700 < Re_R < 7850$, none of the existing heat transfer correlations correlated the data of this investigation.

5.3 NEW HEAT TRANSFER CORRELATION

Over the range of Reynolds number of R-113 covered, and by using the same modifiers F_1,F_2 and F_3 suggested by Carnavos [9], the following correlation was developed using the least square regression analysis:

$$Nu/Pr^{0.4} = 0.07\ Re^{0.6795}\ F_1^{-1.5016}\ F_2^{1.7646}\ F_3^{16.5476} \qquad (5\text{-}16)$$

Figure 5.3 shows a comparison between the predictions of Eq. (5-16) and the experimental measurements. It can be seen that 98% of the data points fall within the $\pm 10\%$ disagreement zone. Several observations on the above correlation need to be made.

1. The characteristic dimension in Reynold and Nusselt numbers in Eq. (5-16) were based on the nominal diameter D_i rather than the hydraulic diameter D_h. This is due to the fact that the experimentally measured inside heat transfer coefficient was also based on the nominal diameter D_i. One can argue the validity of using D_i versus D_h. However, in the final analysis, it appears that both characteristic lengths can be used, and the difference can be taken care of by the exponents of the modifying factors.
2. Equation (5-16) applies for internally smooth as well as internally finned tubes for the Reynolds number range of $2700 < Re_R < 7850$ which

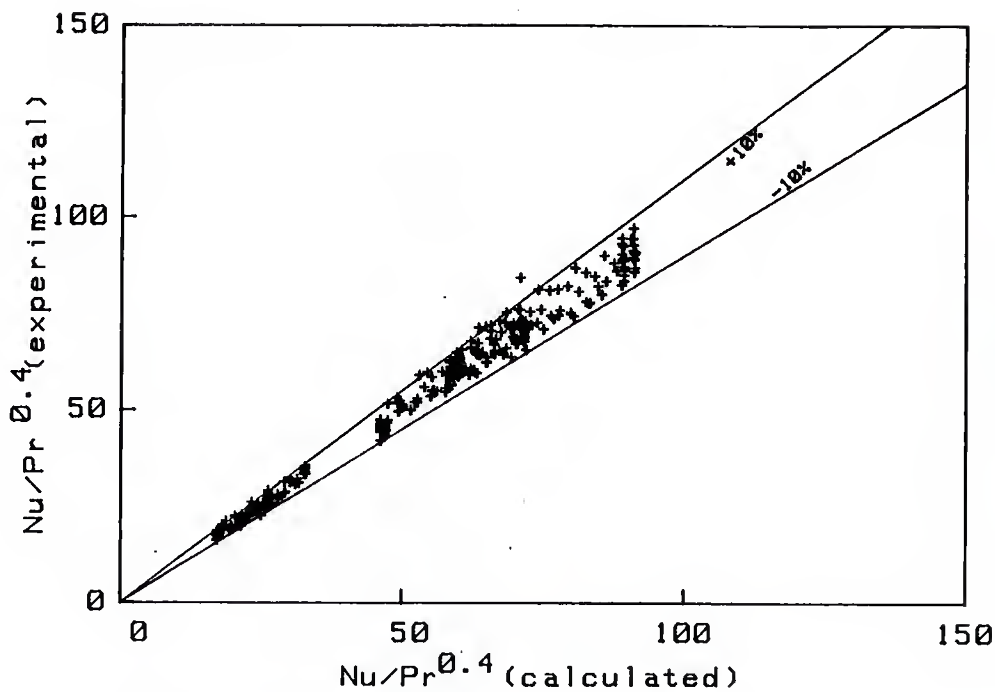

Fig. 5.3 Comparison of experimental and calculated value of $Nu/Pr^{0.4}$ (R-113).

is in the transition flow range and also valid for values of F_1, F_2 and F_3 given in Table 5.1. F_1, F_2 and F_3 are all equal to 1.0 for smooth tubes.

5.4 PRESSURE DROP CORRELATIONS

For turbulent flow of water inside internally finned tubes for the ranges of $5000<Re_h<75000$ and $9.2<\mathbb{P}/D_h<79.2$, Watkinson et al. [3] developed the following correlation for the friction coefficient:

$$f = \frac{0.614}{Re_h^{0.39} (\mathbb{P}/D_h)^{0.2}} \tag{5-17}$$

For turbulent flow of air where $10000<Re_h<75000$ and $9<\mathbb{P}/D_h<80$ Watkinson et al. [4] developed the following correlation:

$$f = \frac{0.0546}{Re_h^{0.15} (\mathbb{P}/D_h)^{0.24}} \tag{5-18}$$

For laminar oil flow, Watkinson et al. [5] correlated their data with:

$$f = 16.4 \, (D_h/D_i)^{1.4} / Re_h \tag{5-19}$$

Carnavos [7,9] correlated data for pressure drop in tubes for heating water in turbulent flow and cooling air in turbulent flow with the following equation:

$$f = \frac{0.046}{Re_h^{0.2} \, F_4^{0.5} \, (\mathrm{Sec}\ \alpha)^{0.75}} \tag{5-20}$$

where

$$F_4 = A_{fa}/A_{fn} = (D_e/D_i)^2 = [1 - 4nbt/\pi \, D_i^2 \cos\alpha] \tag{5-21}$$

TABLE 5.1. Values of F_1, F_2, F_3 and F_4 for the tubes tested.

Tube No.	1	2	3	7	8	9
F_1	1.158	1.0	1.166	1.102	1.0	1.106
F_2	0.543	1.0	0.528	0.518	1.0	0.502
F_3	1.155	1.0	1.155	1.155	1.0	1.155
F_4	0.949	1.0	0.942	0.960	1.0	0.955

Due to the fact that the range of Reynolds number of R-113 tested in the present study was in the transition flow range, the experimental pressure drop data could not be correlated with any of the above correlations.

Using the least square regression analysis and the pressure drop measurements, the following friction factor coefficient was developed for finned tubes:

$$f_d = \frac{330.634}{Re^{1.13036}} F_3^{4.7231} F_4^{-7.3889} \tag{5-19}$$

A plot of experimental value of friction factor and friction factors predicted by the above equation is given in Fig. 5.4. The above equation predicted 62% of the data points to within ±15%. Most of the data points which are outside the agreement zone were taken at low flow rates which made the accuracy of the pressure drop measurements subject to question.

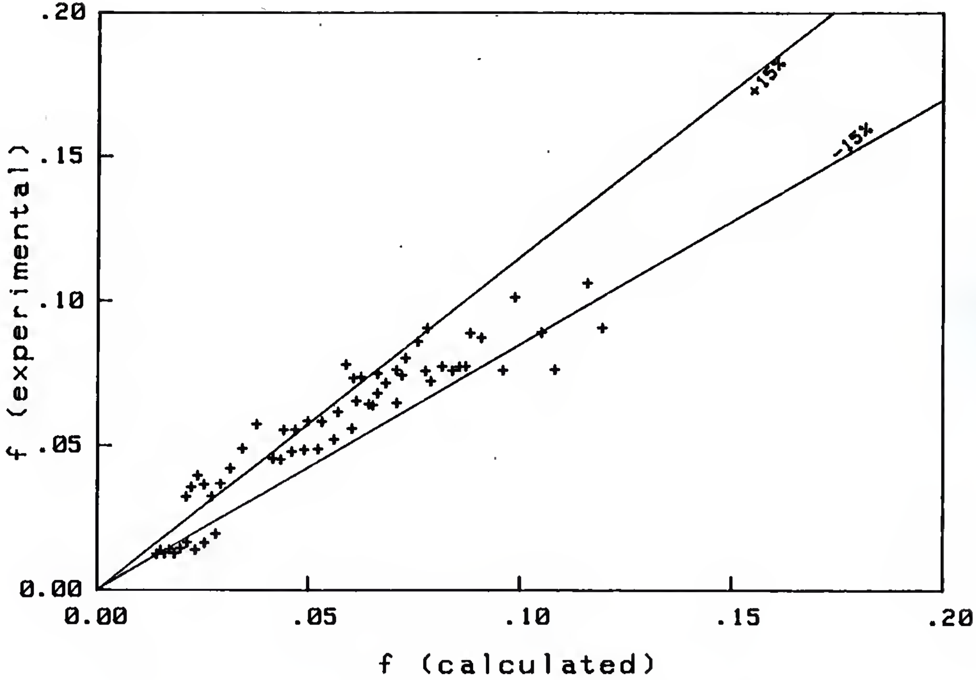

Fig. 5.4 Comparison of experimental and calculated value of friction factor.

Chapter VI

PERFORMANCE EVALUATION OF DIFFERENT TUBES TESTED

6.1 INTRODUCTION

In the present investigation three types of tubes were tested, namely, A,B and C which are shown in Fig. 3.7. An attempt was made in this chapter to compare the performance of the different heat exchangers with different types of inner tubes. These comparisons were made to determine the effect of inside augmentation alone (internal fins), outside augmentation alone (roughness created by knurls) and double augmentation on the performance of the heat exchangers. Performance evaluation criteria suggested by different investigators as well as new ones were used in performing the task of this evaluation. The ultimate objective was to determine certain guidelines under which internal versus external augmentation as well as double augmentation could be used in designing more efficient heat exchangers.

6.2 PERFORMANCE INDICES

Any performance index should be directed towards a certain objective. Depending on the objective, each evaluation index is determined by calculating the ratio of certain parameters of interest for the augmented and smooth surfaces, subject to certain constraints. A similar analysis is done for heat exchangers as well.

Bergles et al. [24] developed eight performance evaluation criteria which define the merits of surfaces with promoters to augment heat transfer in single-phase flow. Four of these criteria were directed towards the use of promoters for improvement of existing heat exchangers. Therefore, the basic geometry was fixed. The remaining four criteria were directed towards evaluating the advantages of using promoters in the design of a new heat exchanger. Therefore, the length and number of tubes were unrestricted and the objective was to reduce the exchanger's size. A summary of these criteria is shown in Table 6.1. Some of the constraints of these criteria are difficult to realize experimentally. However, criterion number 1 in the table was used in evaluating internal fins and external roughness in improving the inside and outside heat transfer coefficients respectively, of the tubes tested.

6.2.1 Effects of Internal Fins

Figures 6.1 and 6.2 show plots of the ratio of the inside heat transfer coefficients for internally finned tubes (tubes 1 and 3) to the same for smooth tube (tube 2). These ratios were determined under the constraints of same geometry (length L and diameter D), same mass flow rate m_{wa} and inlet temperature T_{wai} of water, the inlet temperature of R-113 T_R and its flow rate expressed in terms of Reynolds numbers. The lines representing this ratio were reproduced from Figs. 4.1 and 4.2 by determining the ratio of h_i from the regression lines of these figures at the same Reynolds number. This approach was dictated by the fact that all constraints stated above could be controlled except Reynolds number. However, a limited number of experimental points could be found which satisfied all the constraints. These points are shown on the same figures. Figures 6.3 and 6.4 are plots of the ratio of h_i of internally finned tubes (tubes 7 and 9) to h_i for the smooth tube (tube 8) under the same constraints of Figs. 6.1 and 6.2. Figures 6.3 and 6.4 were reproduced from Figs. 4.4 and 4.5 respectively. A few experimental points which satisfied all the constraints are also included in these figures. The results in Figs. 6.1 through 6.4 are summarized in the following:

1. Over the range of Reynolds number of R-113 in this study (4000 to 7850

	Fixed					Objective		
Criterion Number	Basic Geometry	Flow Rate	Pressure Drop	Pumping Power	Heat Duty	Increase Heat Transfer	Reduce Pumping Power	Reduce Exchanger Size
1	x	x				x		
2	x		x			x		
3	x			x		x		
4	x				x		x	
5				x	x			x
6			x		x			x
7		x			x			x
8		x	x		x			x

Table 6.1. Summary of Performance Evaluation Criteria of Bergles et al. [24]

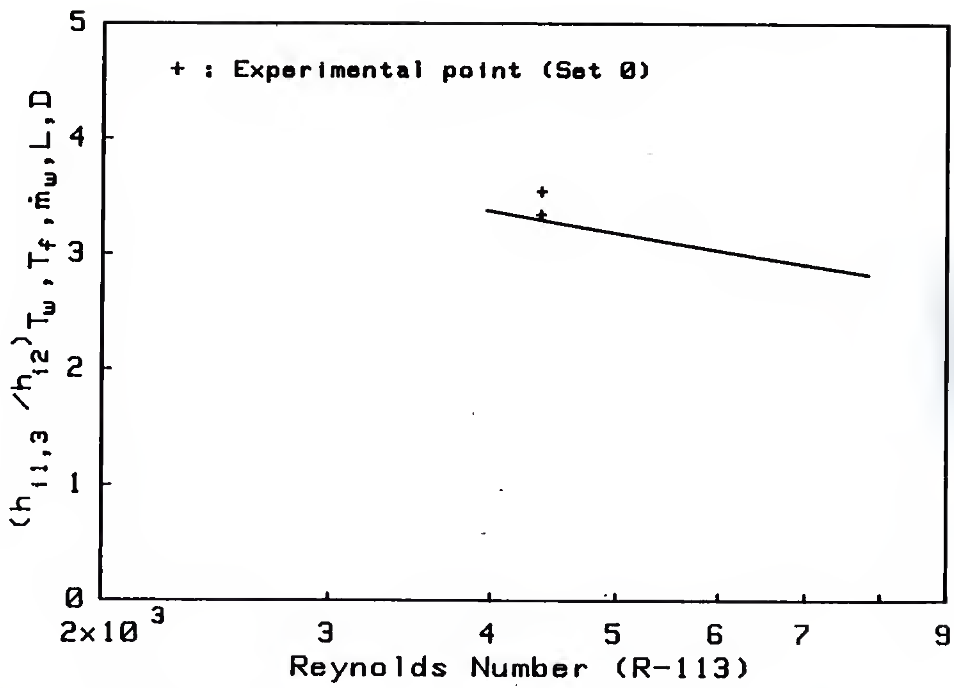

Fig. 6.1 Ratio of inside heat transfer coefficients for inside finned and inside smooth inner tubes versus Reynolds number of R-113, set 0, tubes 1,2 and 3.

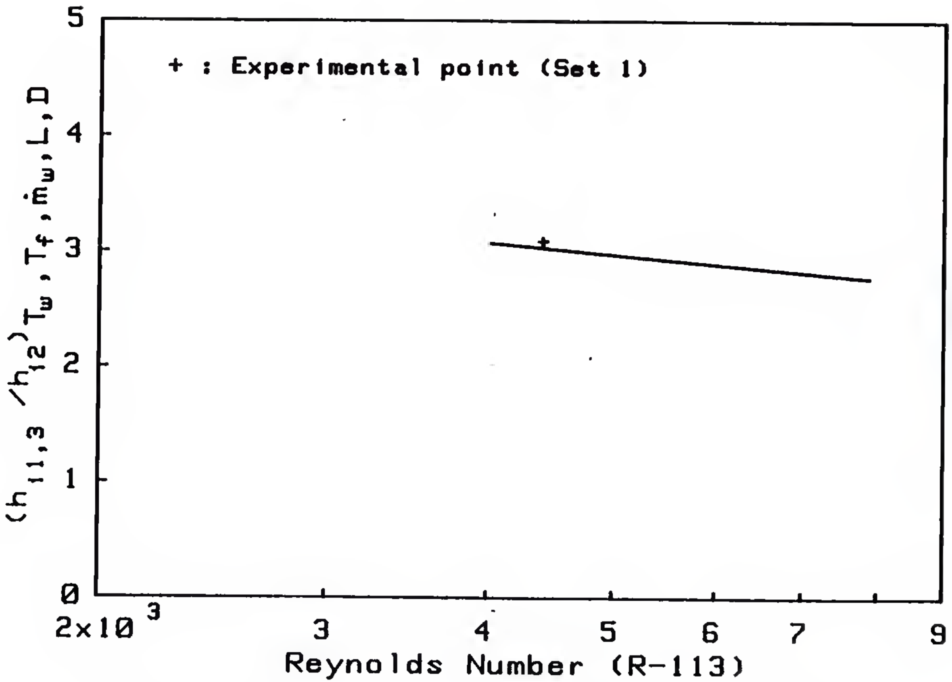

Fig. 6.2 Ratio of inside heat transfer coefficients for inside finned and inside smooth inner tubes versus Reynolds number of R-113, set 1, tubes 1,2 and 3.

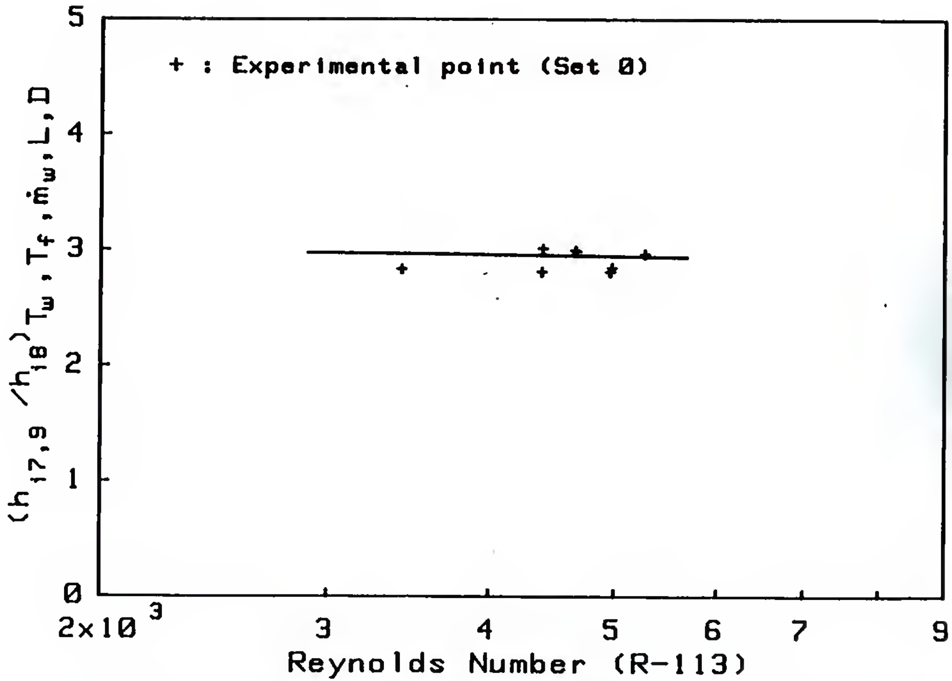

Fig. 6.3 Ratio of inside heat transfer coefficients for inside finned and inside smooth inner tubes versus Reynolds number of R-113, set 0, tubes 7,8 and 9.

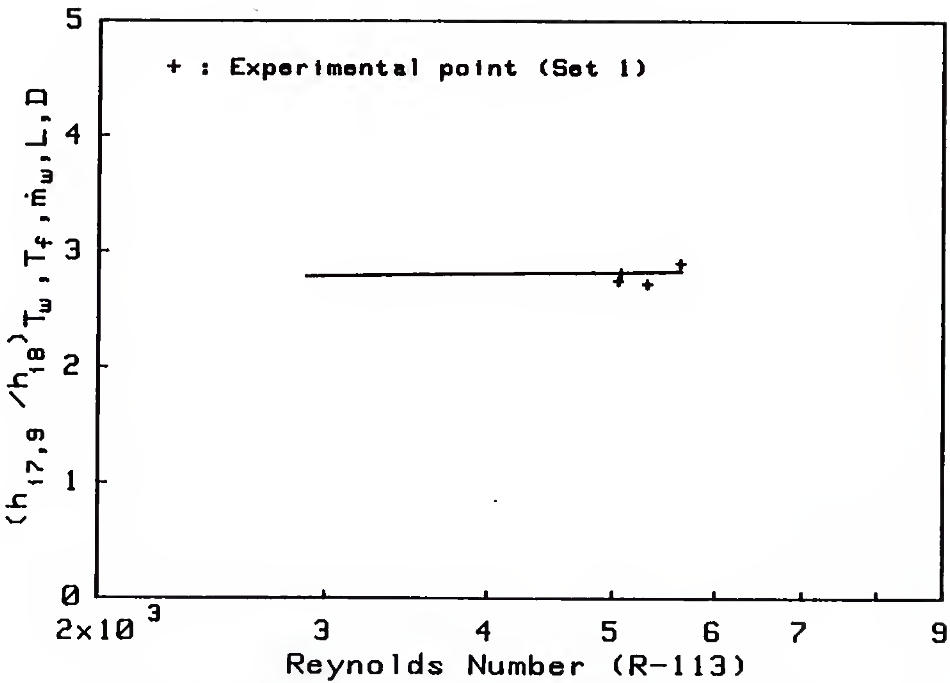

Fig. 6.4 Ratio of inside heat transfer coefficients for inside finned and inside smooth inner tubes versus Reynolds number of R-113, set 1, tubes 7,8 and 9.

for tubes 1,2 and 3 and 2700 to 6000 for tubes 7,8 and 9), the inside heat transfer coefficient for internally finned tubes is about three times that for smooth tubes, on a nominal area basis.

2. For tubes 1,2 and 3, $h_{i,aug}/h_{i,smooth}$ decreased with the increase in Reynolds number of R-113. For tubes 7,8 and 9, this ratio increased very slightly with the increase in Reynolds number of R-113.
3. In the range of experimental data, inlet temperatures of R-113 and water and mass flow rate of water had no effect on the ratio of inside heat transfer coefficents. This conclusion was substantiated by the fact that data for sets 0 and 1 could be combined into one plot since the ratios of inside heat transfer coefficients were about the same for these data sets.
4. Outside surface augmentation has no effect on the inside heat transfer coefficients.

6.2.2 Effects of Outside Knurling

Figures 6.5 and 6.6 show plots of the ratio of the outside heat transfer coefficient h_o of the knurled tubes (tubes 2 and 3) to the outside heat transfer coefficient of the smooth tube (tube 1) for data sets 3 and 4 respectively. The plotted lines of Figs. 6.5 and 6.6 were generated from the ratio of the regression lines in Figs. 4.9 and 4.10 respectively. The ratios were determined under the constraints of same geometric parameters (D and L), same inlet water temperature T_{wai}, same water flow rate m_{wa} and same inlet temperature and flow rate of R-113. Experimental data points are also shown in the figures. Figures 6.7 and 6.8 show similar plots for tubes 7,8 and 9 with the same constraints of Figs. 6.5 and 6.6. The results can be summarized in the following:

1. The knurls increased the outside heat transfer coefficient by about 20% under the constraints stated above.

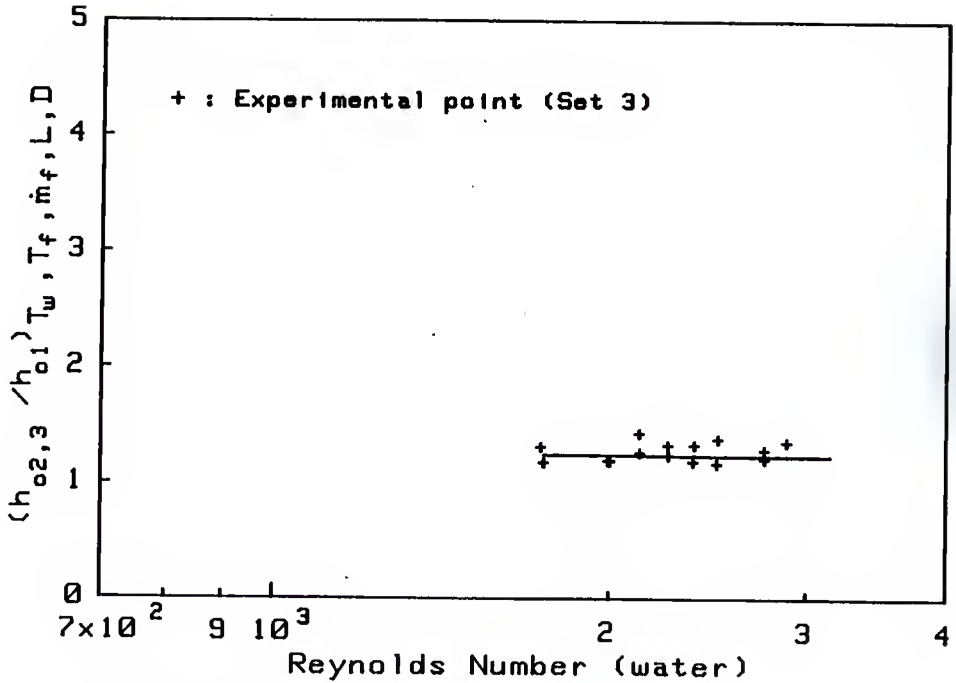

Fig. 6.5 Ratio of outside heat transfer coefficients for outside knurled and outside smooth tubes versus Reynolds number of water, tubes 1,2 and 3, set 3.

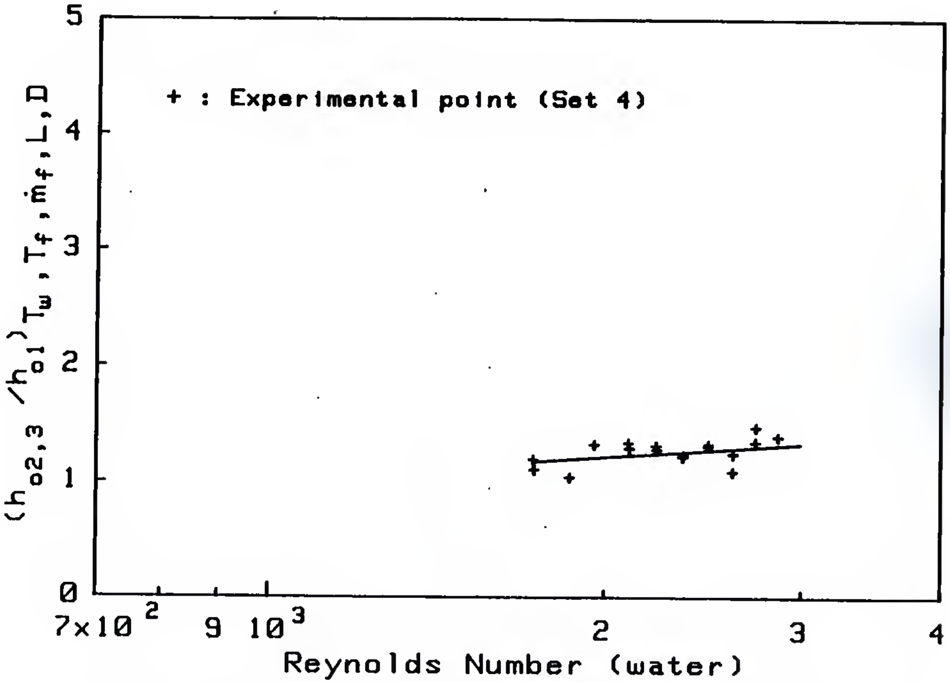

Fig. 6.6 Ratio of outside heat transfer coefficients for outside knurled and outside smooth tubes versus Reynolds number of water, tubes 1,2 and 3, set 4.

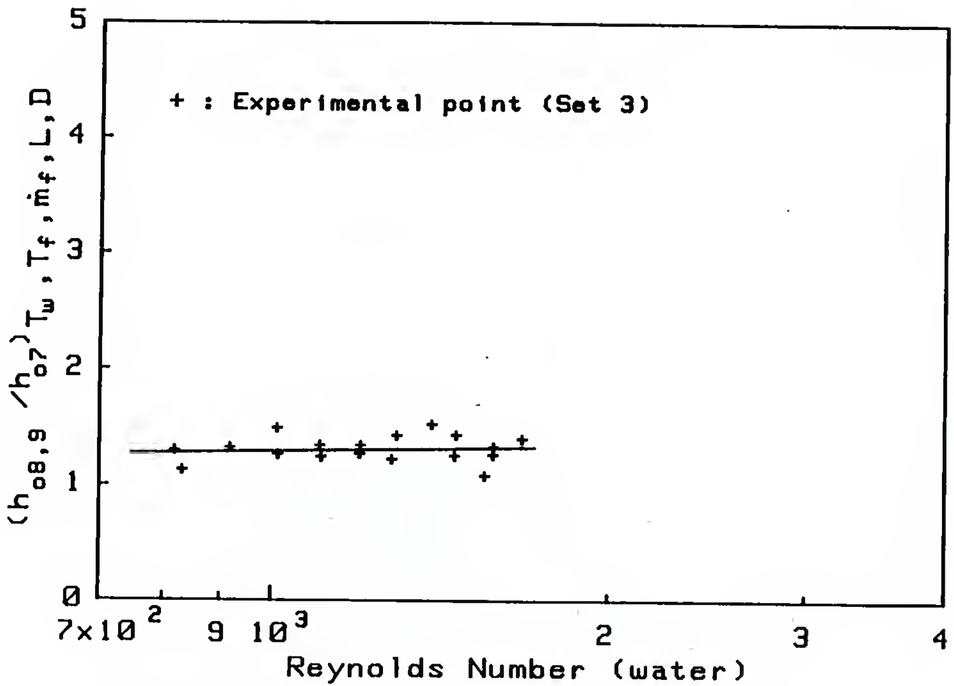

Fig. 6.7 Ratio of outside heat transfer coefficients for outside knurled and outside smooth tubes versus Reynolds number of water, tubes 7,8 and 9, set 3.

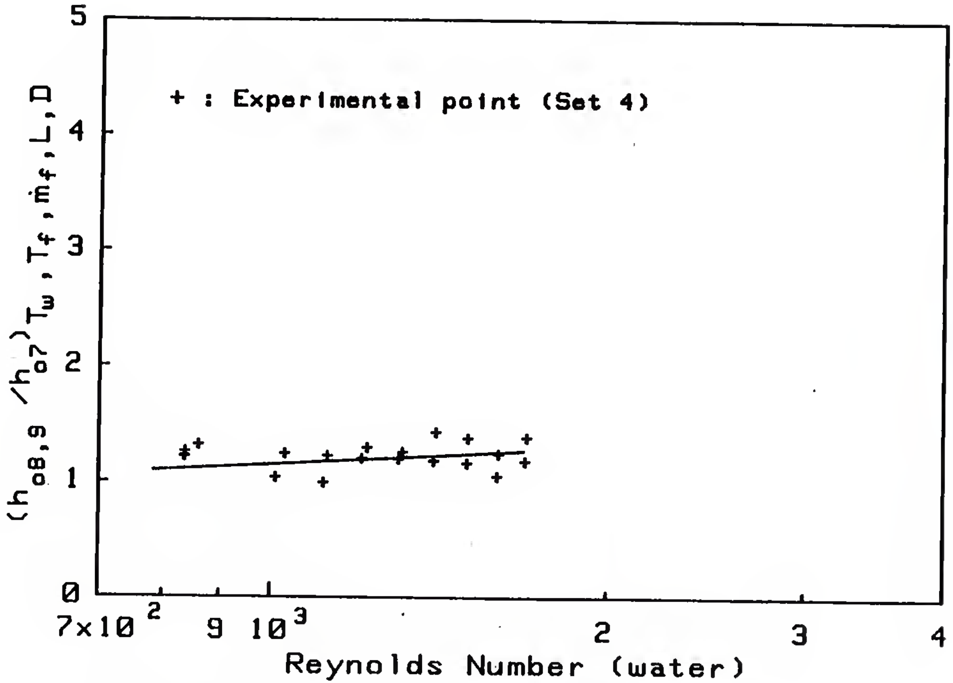

Fig. 6.8 Ratio of outside heat transfer coefficients for outside knurled and outside smooth tubes versus Reynolds number of water, tubes 7,8 and 9, set 4.

2. In the range of Reynolds number of water in this experimental investigation, $h_{o,aug}/h_{o,smooth}$ increased slightly with the increase in Reynolds number of water.

3. There is no effect of flow rate of R-113 on $h_{o,aug}/h_{o,smooth}$.

6.2.3 Comparisons of Overall Heat Conductance UA of Heat Exchangers Tested

To compare the overall heat transfer performance of the heat exchangers tested, ratios of their overall conductances were compared under similar constraints. Figures 6.9 and 6.10 show plots of UA ratios of internally finned tubes (tubes 1 and 3) to the internally smooth tube (tube 2) versus Reynolds number of R-113, under the same constraints used in comparing the inside and outside heat transfer coefficients alone. The lines drawn in these figures were determined from the ratio of the values obtained from regression lines of individual plots of UA at the same Reynolds number. On the same figures actual experimental points are also plotted. Figures 6.11 and 6.12 are similar plots for heat exchangers 7,8 and 9. Also, Figs. 6.13 through 6.16 show similar plots for the ratios of UA of the different heat exchangers plotted versus the Reynolds number of cooling water. Figures 6.9 through 6.12 were plotted mainly to evaluate the effects of augmentation by internal fins while Figs. 6.13 to 6.16 were plotted to evaluate the effects of outside augmentation on the overall conductance UA of the heat exchanger. The following conclusions can be drawn from these plots:

1. The ratio of UA for tubes with inside augmentation to UA for tubes without inside augmentation is about 2.5. The ratio slightly decreased with Reynolds number as can be seen in Figs. 6.9 through 6.12. Whether the tube had outside knurls or not, it had no effect no effect on the ratio. These results confirm the fact that augmentation

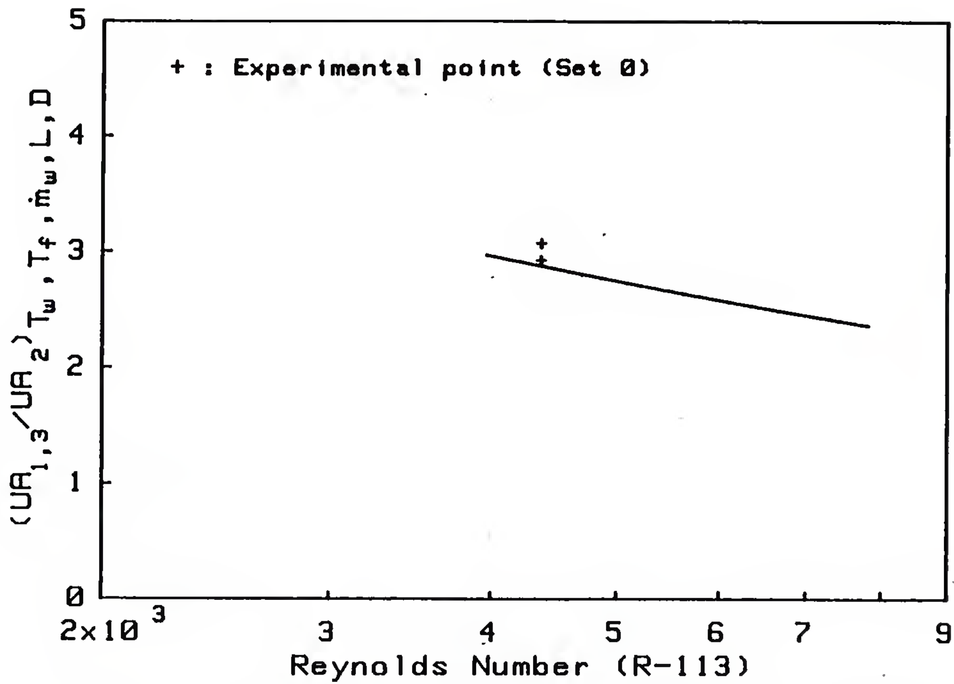

Fig. 6.9 Ratio of heat duty (UA) for inside augmented and inside smooth inner tubes versus Reynolds number of R-113, Set 0, tubes 1,2 and 3.

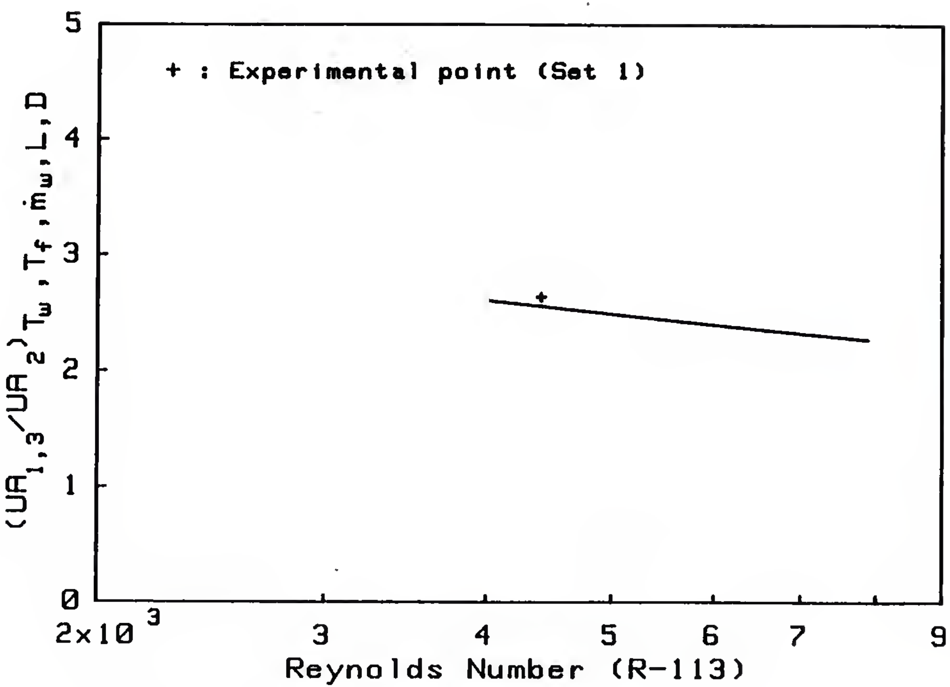

Fig. 6.10 Ratio of heat duty (UA) for inside augmented and inside smooth inner tubes versus Reynolds number of R-113, Set 1, tubes 1,2 and 3.

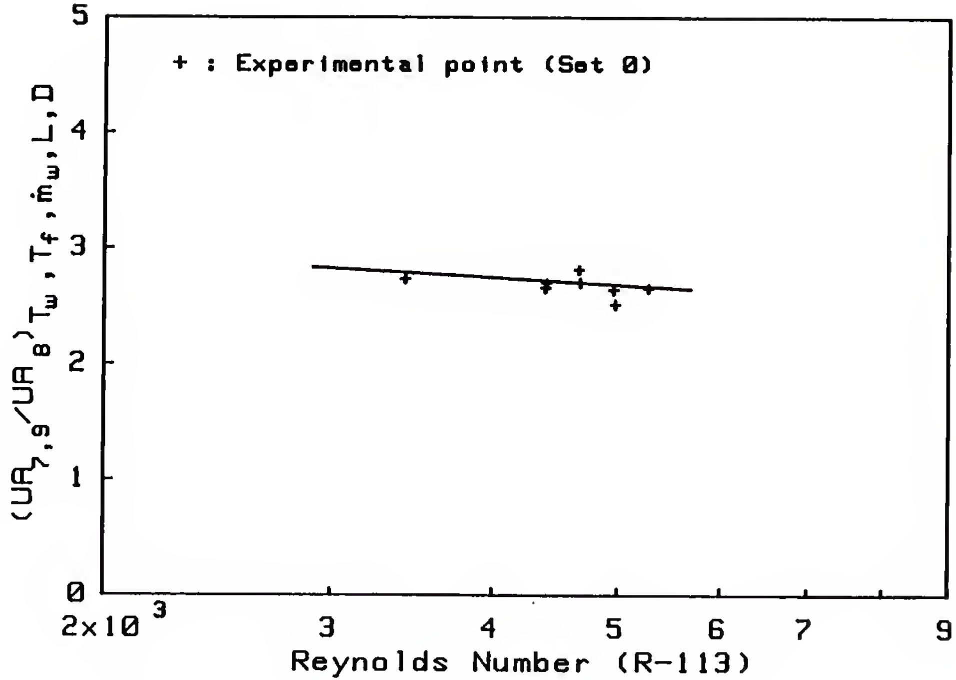

Fig. 6.11 Ratio of heat duty (UA) for inside augmented and inside smooth inner tubes versus Reynolds number of R-113, Set 0, tubes 7,8 and 9.

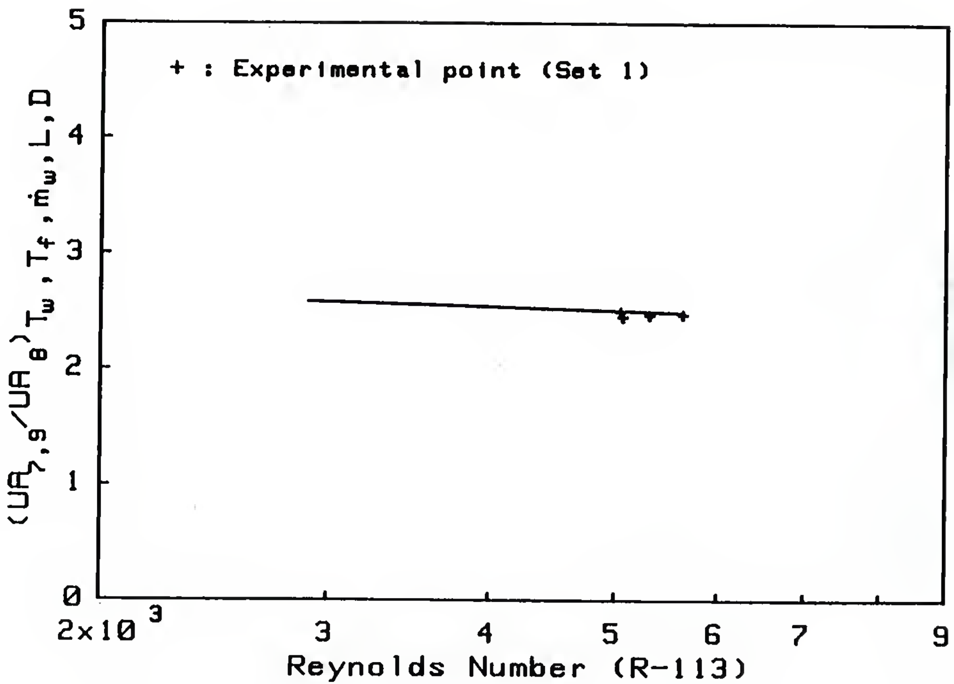

Fig. 6.12 Ratio of heat duty (UA) for inside augmented and inside smooth inner tubes versus Reynolds number of R-113, Set 1, tubes 7,8 and 9.

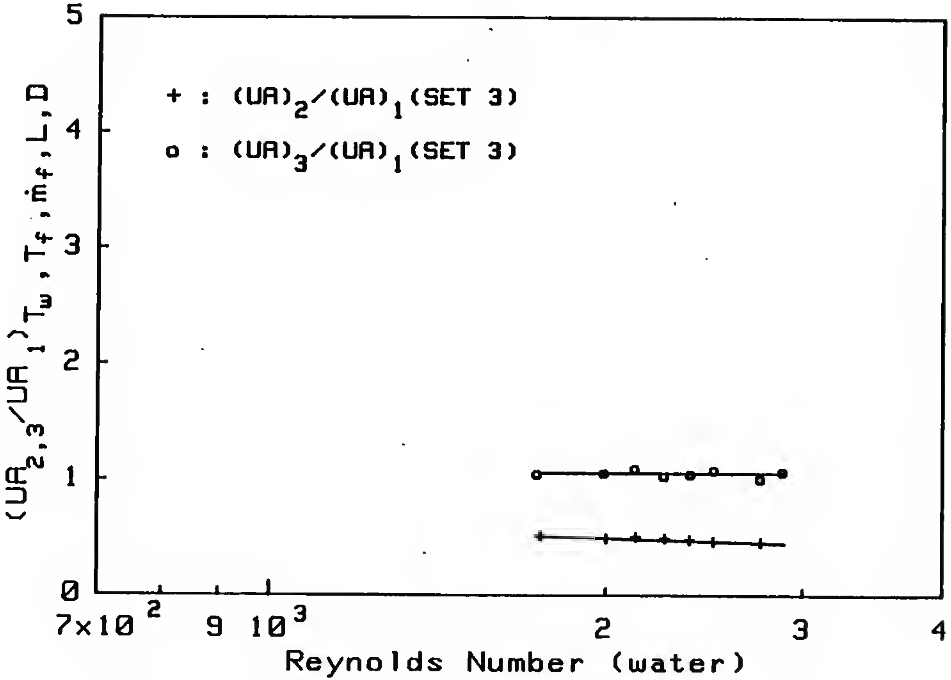

Fig. 6.13 Ratio of heat duty (UA) for tubes augmented on the outside to tubes smooth on the outside versus Reynolds number of water, tubes 1,2 and 3, set 3.

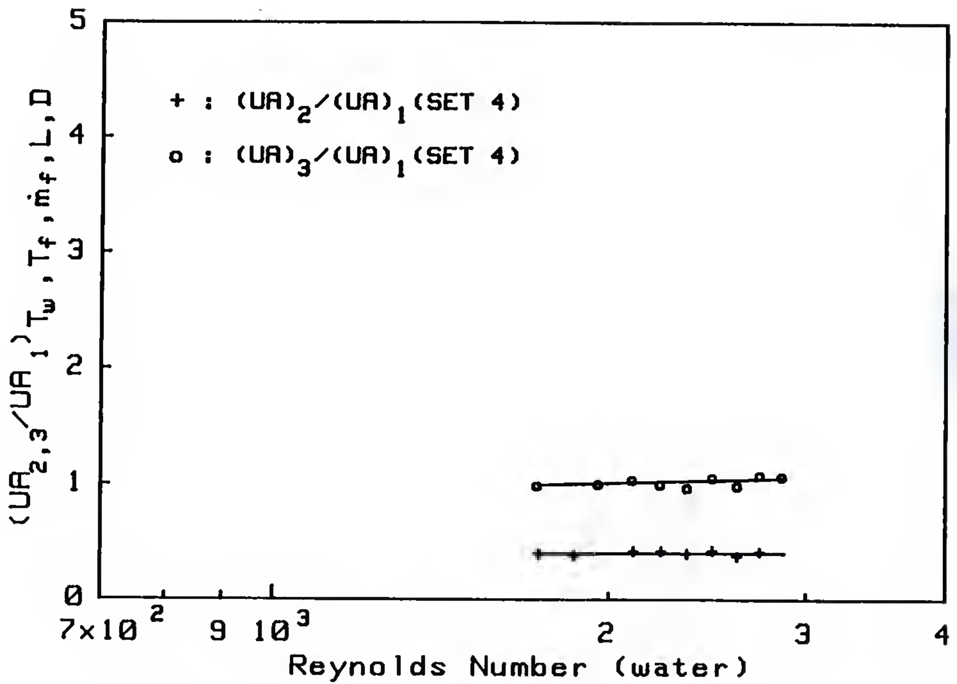

Fig. 6.14 Ratio of heat duty (UA) for tubes augmented on the outside to tubes smooth on the outside versus Reynolds number of water, tubes 1,2 and 3, set 4.

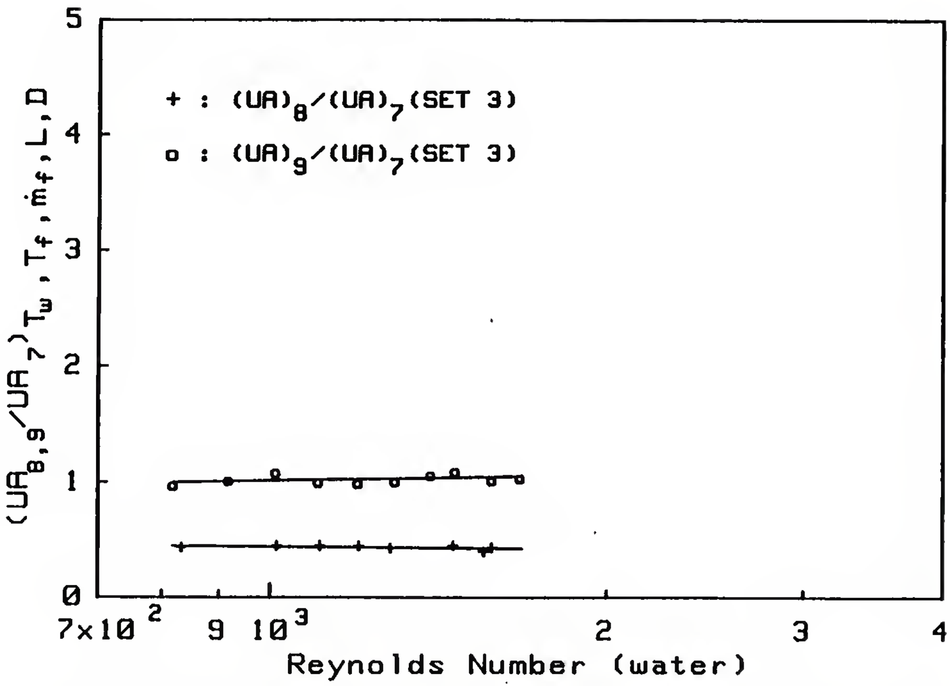

Fig. 6.15 Ratio of heat duty (UA) for tubes augmented on the outside to tubes smooth on the outside versus Reynolds number of water, tubes 7,8 and 9, set 3.

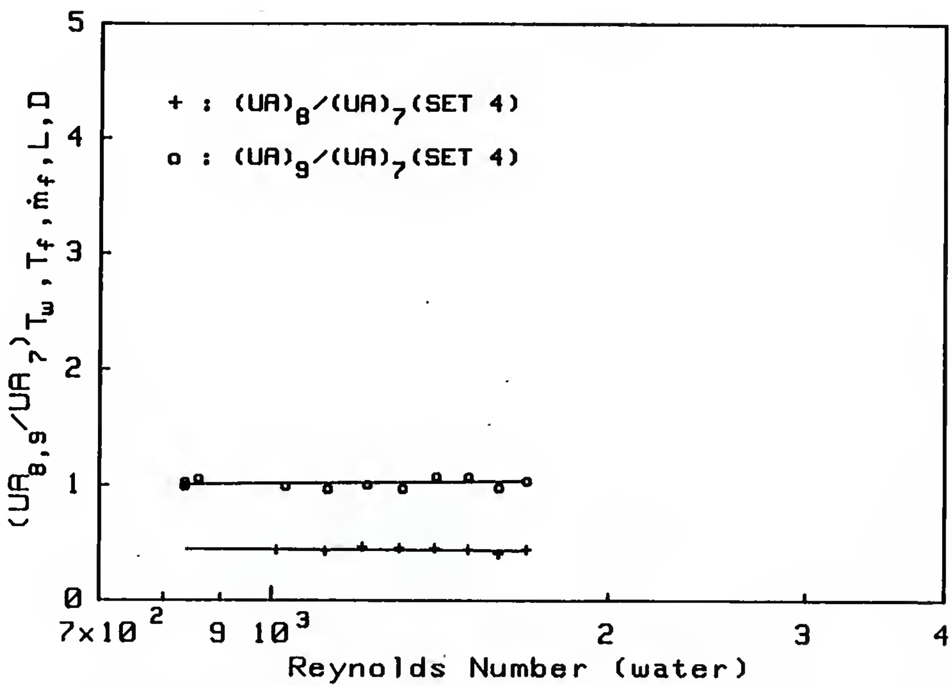

Fig. 6.16 Ratio of heat duty (UA) for tubes augmented on the outside to tubes smooth on the outside versus Reynolds number of water, tubes 7,8 and 9, set 4.

is most beneficial when applied to the side of the tubes which has the lower heat transfer coefficient. In the present study, h_o is about three times h_i for smooth outside and inside surfaces.

2. Figures 6.13 through 6.16 which give UA ratios for the various tubes tested confirm the observation made before. The ratios of $(UA)_2/(UA)_1$, Figs. 6.13 and 6.14, and $(UA)_8/(UA)_7$, Figs. 6.15 and 6.16, are less than 1.0 because the increase in (UA) by internally finned tubes alone was larger than the increase in UA by outside knurls alone.

6.3 PERFORMANCE EVALUATION OF SINGLY AND DOUBLY AUGMENTED TUBES FOR HEAT EXCHANGER APPLICATIONS

Webb and Eckert [25] developed equations to define the performance advantage of roughened tubes in heat exchanger design, relative to smooth tubes of equal diameter. Three rough tube applications were presented:

1. To obtain increased heat transfer exchange capacity under the constraints of equal pumping power and heat transfer area.
2. To reduce heat transfer surface area for the same heat transfer rate and pumping power.
3. To reduce friction power for the same heat transfer rate and heat transfer surface area.

This analysis was extended by Webb and Scott [26] to compare the performance of an internally finned tube heat exchanger with that of an exchanger having an internally smooth tube. The criteria applied to internally finned tubes were:

1. Reduced tube material volume for equal pumping power and heat duty (UA).

2. Increased heat duty (UA) for equal pumping power and total length of heat exchanger tubing.
3. Reduced pumping power for equal heat duty and total length of heat exchanger tubing.

By varying the internal fin geometric parameters, the performance benefits were studied. The calculations were based on a Prandtl number of 3. It was concluded that it was possible to have a material saving of 49% using internal fins having a 30 degree helix angle, for equal pumping power and heat duty. The heat exchanger heat duty (UA) could be increased by 35-40% for equal pumping power and total tubing length. Pumping power reduction of 42% may be obtained by using 5 fins 2mm in height with zero helix angle.

Webb [27] extended the work of Bergles [24] and Webb et al. [25,26] to establish criteria applicable to single phase flow in tubes. The equations included the effect of shell side enhancement and fouling and were applicable to internally roughned and internally finned tubes. Four design cases were presented:

1. Reduced heat exchanger material volume.
2. Increased heat duty.
3. Reduced log-mean temperature difference.
4. Reduced pumping power.

Recently, Marner et al. [28], proposed a uniform format for presenting pressure drop and heat transfer data for enhanced surfaces to enable evaluating the performance of various surfaces. A standardized data format using the inside and outside envelope diameter as the basis for presenting the various geometrical flow, and heat transfer parameters for all tubular enhanced surfaces was proposed and discussed.

A different view for rating heat exchanger performance was proposed by Bejan [29]. He proposed that heat exchanger losses be evaluated in terms of one single quantity, namely, its rate of irreversibility or rate of entropy production. Bejan and Pfister [30] used entropy generation as a measure of the relative merit of heat transfer augmentation techniques relative to one another.

In the present section the approach developed by Webb and Eckert [25] will be adopted to evaluate the performance of singly and doubly augmented tubes for heat exchanger applications.

For a heat exchanger with an inner tube which is smooth on the inside as well as on the outside, the following equation can be written if the wall conduction and fouling resistances are ignored:

$$\frac{1}{(UA)_s} = \frac{1}{h_{is}A_{is}} + \frac{1}{h_{os}A_{os}} \tag{6-1}$$

or

$$\frac{1}{(UA)_s} = \frac{1}{h_{os}A_{os}} \left(1 + \frac{h_{os}}{h_{is}} B \right) \tag{6-2}$$

where

$$B = A_o/A_i \tag{6-3}$$

Similarly, for a heat exchanger with the inner tube augmented on the inside only, one can write:

$$\frac{1}{(UA)_{ia}} = \frac{1}{h_{ia}A_{ia}} \left(1 + \frac{h_{ia}}{h_{os}} \cdot \frac{1}{B} \right) \tag{6-4}$$

For a heat exchanger with the inner tube augmented on the outside only:

$$\frac{1}{(UA)_{oa}} = \frac{1}{h_{oa}A_{oa}} \left(1 + \frac{h_{oa}}{h_{is}} B \right) \tag{6-5}$$

or

$$\frac{1}{(UA)_{oa}} = \frac{1}{h_{is}A_{is}} \left(1 + \frac{h_{is}}{h_{oa}} \frac{1}{B} \right) \tag{6-6}$$

For a heat exchanger with the inner tube doubly augmented:

$$\frac{1}{(UA)_{da}} = \frac{1}{h_{ia}A_{ia}} \left(1 + \frac{h_{ia}}{h_{oa}} \frac{1}{B} \right) \tag{6-7}$$

or

$$\frac{1}{(UA)_{da}} = \frac{1}{h_{oa}A_{oa}} \left(1 + \frac{h_{oa}}{h_{ia}} B \right) \tag{6-8}$$

From the above equations, the following four equations that compare different heat exchangers can be obtained:

$$\frac{(UA)_{ia}}{(UA)_{is}} = \frac{h_{ia}}{h_{is}} \left(\frac{1 + (h_{is}/h_{os})(1/B)}{1 + (h_{ia}/h_{is})(h_{is}/h_{os})(1/B)} \right) \tag{6-9}$$

$$\frac{(UA)_{oa}}{(UA)_{os}} = \frac{1 + (h_{is}/h_{os})(1/B)}{1 + (h_{is}/h_{oa})(1/B)} \tag{6-10}$$

$$\frac{(UA)_{da}}{(UA)_{ia}} = \frac{1 + (h_{ia}/h_{os})(1/B)}{1 + (h_{ia}/h_{oa})(1/B)} \tag{6-11}$$

$$\frac{(UA)_{da}}{(UA)_{oa}} = \frac{1 + (h_{oa}/h_{is})(B)}{1 + (h_{oa}/h_{ia})(B)} \tag{6-12}$$

Equations (6-9) through (6-12) were applied for heat exchangers with inner tubes (augmented or unaugmented) having the same length and nominal inside and outside diameters. Also, all heat exchangers have the same fluid flow rate and equal entering fluid temperatures.

For the tubes used in this study, B was about 1.22. For this value of B, the following graphs were plotted:

1. $(UA)_{ia}/(UA)_{is}$ versus h_{is}/h_{os} with h_{ia}/h_{is} as a parameter (Figure 6.17)

2. $(UA)_{ia}/(UA)_{is}$ versus h_{ia}/h_{is} with h_{is}/h_{os} as a parameter (Figure 6.18)
3. $(UA)_{oa}/(UA)_{os}$ versus h_{is}/h_{os} with h_{is}/h_{oa} as a parameter (Figure 6.19)
4. $(UA)_{oa}/(UA)_{os}$ versus h_{is}/h_{oa} with h_{is}/h_{os} as a parameter (Figure 6.20)
5. $(UA)_{da}/(UA)_{ia}$ versus h_{ia}/h_{os} with h_{ia}/h_{oa} as a parameter (Figure 6.21)
6. $(UA)_{da}/(UA)_{ia}$ versus h_{ia}/h_{oa} with h_{ia}/h_{os} as a parameter (Figure 6.22)
7. $(UA)_{da}/(UA)_{oa}$ versus h_{oa}/h_{is} with h_{oa}/h_{ia} as a parameter (Figure 6.23)
8. $(UA)_{da}/(UA)_{oa}$ versus h_{oa}/h_{ia} with h_{oa}/h_{is} as a parameter (Figure 6.24)

From these figures, the following conclusions can be drawn:

1. Figure 6.17 shows that $(UA)_{ia}/(UA)_{is}$ decreases with the increase in h_{is}/h_{os} for constant h_{ia}/h_{is}. It also shows that the rate of decrease of $(UA)_{ia}/(UA)_{is}$ with the increase in h_{is}/h_{os} is higher (hence less beneficial) at smaller values of h_{is}/h_{os} (less than 1).
2. Figure 6.18 shows that $(UA)_{ia}/(UA)_{is}$ increases with the increase in h_{ia}/h_{is} at constant h_{is}/h_{os}. It also shows that the rate of increase in $(UA)_{ia}/(UA)_{is}$ diminishes with the increase in h_{ia}/h_{is}.
3. Figure 6.19 shows that $(UA)_{oa}/(UA)_{os}$ increases linearly with the increase in h_{is}/h_{os} at constant h_{is}/h_{oa}.
4. Figure 6.20 shows that $(UA)_{oa}/(UA)_{os}$ decreases with the increase in h_{is}/h_{oa} for a fixed value of h_{is}/h_{os}. It also shows that $(UA)_{oa}/(UA)_{os}$ decreases at a greater rate as h_{is}/h_{oa} increases for smaller values of h_{is}/h_{oa} (less than 1.5).
5. Figure 6.21 shows that $(UA)_{da}/(UA)_{ia}$ increases linearly with the increase in h_{ia}/h_{os} for constant h_{ia}/h_{oa}. It also shows that the rate of decrease in $(UA)_{da}/(UA)_{ia}$ with the increase in h_{ia}/h_{os} is larger at smaller values of h_{ia}/h_{os} (less than 2).
6. Figure 6.22 shows that $(UA)_{da}/(UA)_{ia}$ decreases with the increase in h_{ia}/h_{oa} at fixed value of h_{ia}/h_{os}.

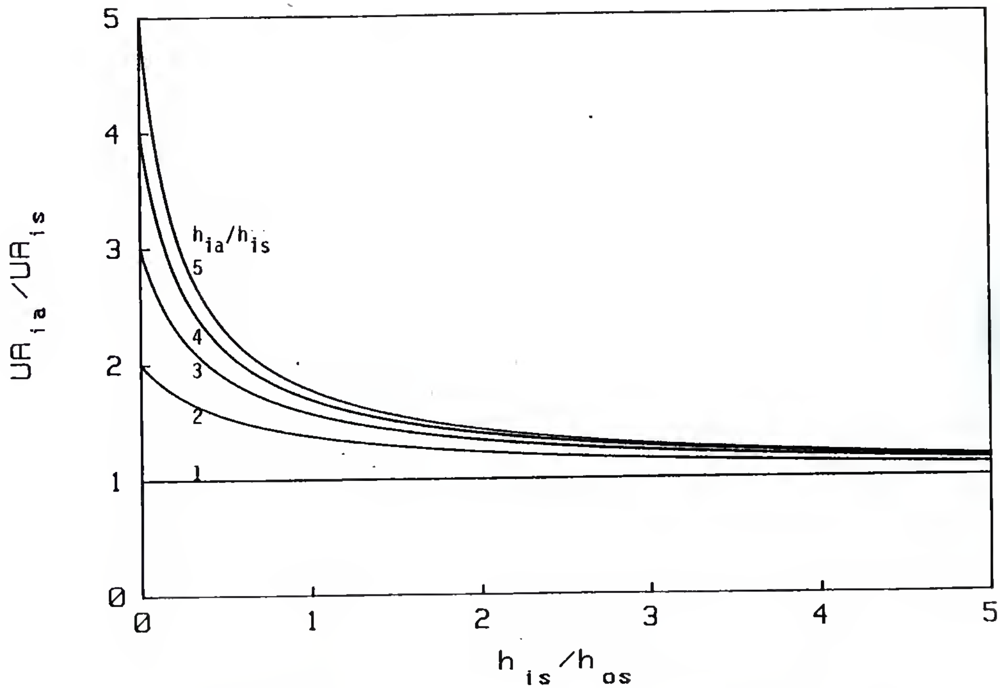

Fig. 6.17 UA_{ia}/UA_{is} versus h_{is}/h_{os} at various values of h_{ia}/h_{is}.

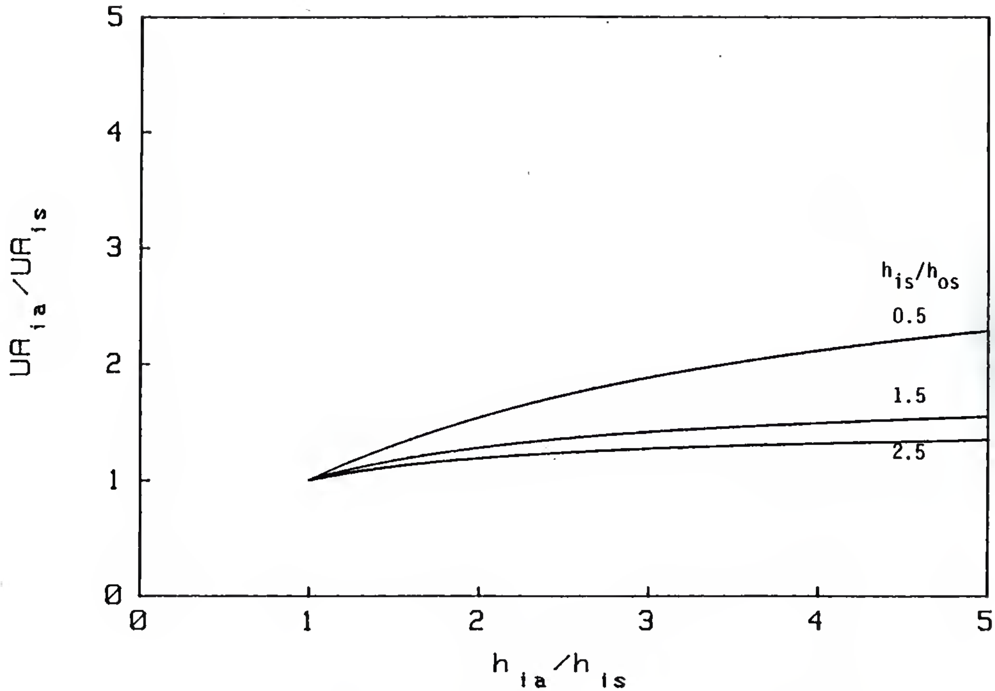

Fig. 6.18 UA_{ia}/UA_{is} versus h_{ia}/h_{is} at various values of h_{is}/h_{os}.

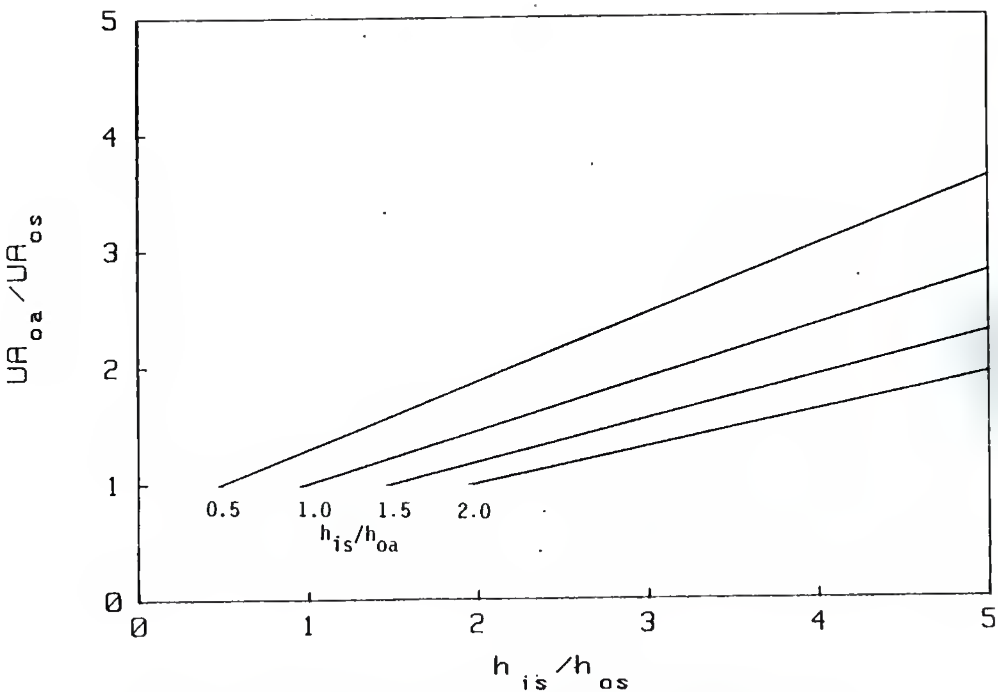

Fig. 6.19 UA_{oa}/UA_{os} versus h_{is}/h_{os} at various values of h_{is}/h_{oa}.

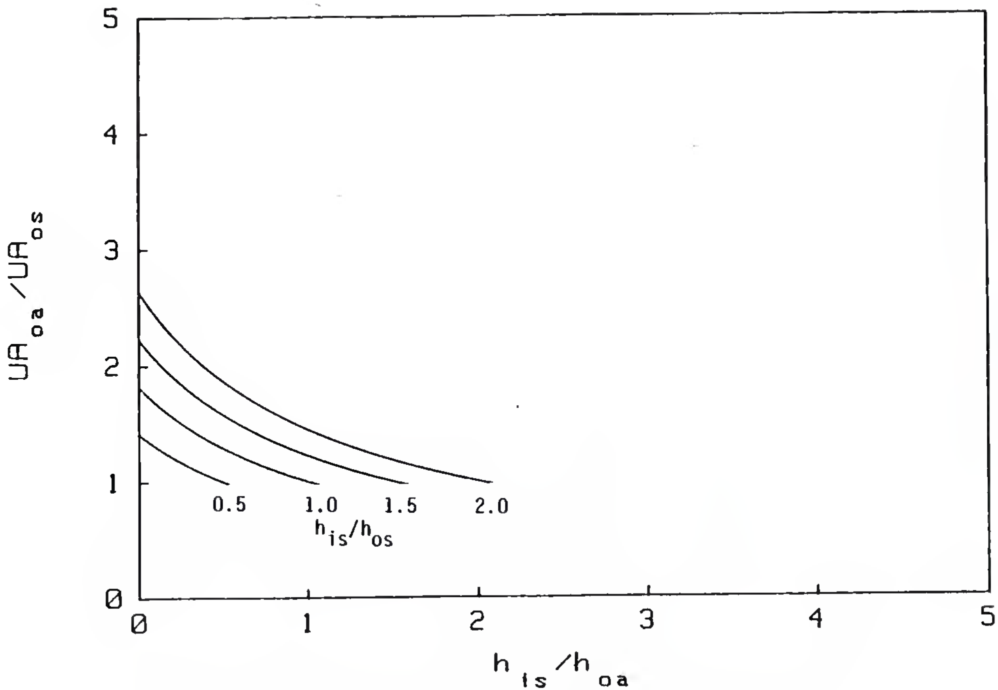

Fig. 6.20 UA_{oa}/UA_{os} versus h_{is}/h_{oa} at various values of h_{is}/h_{os}.

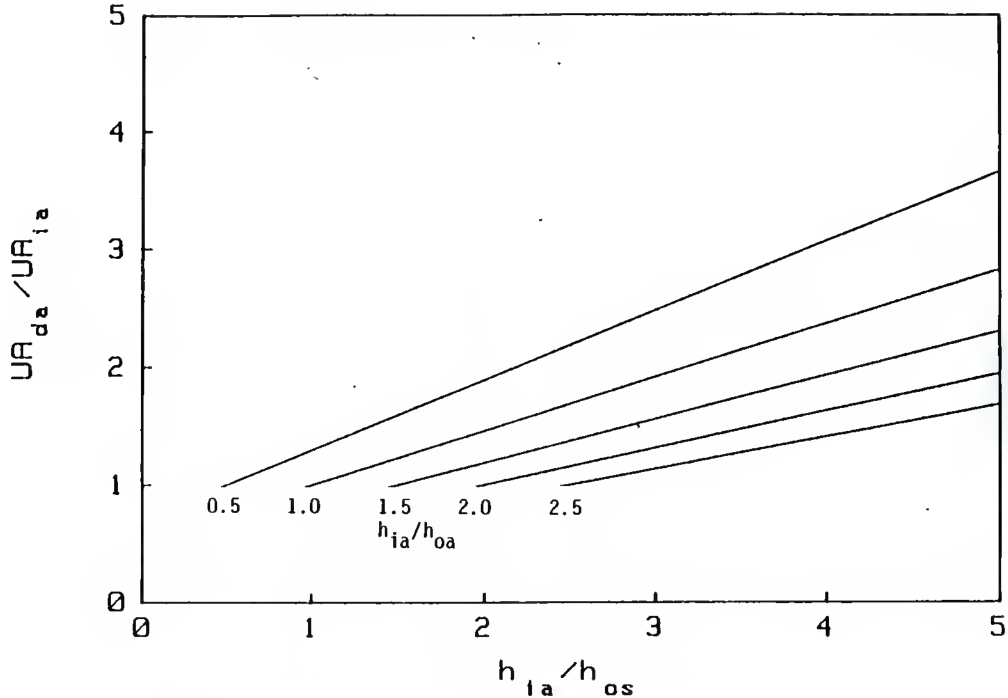

Fig. 6.21 UA_{da}/UA_{ia} versus h_{ia}/h_{os} at various values of h_{ia}/h_{oa}.

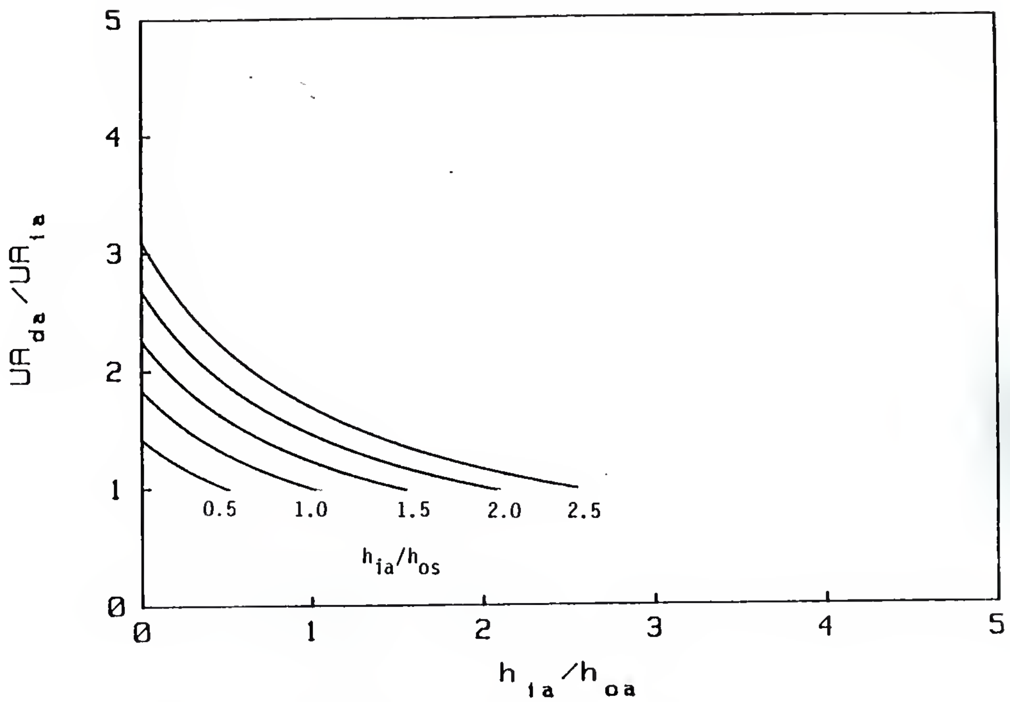

Fig. 6.22 UA_{da}/UA_{ia} versus h_{ia}/h_{oa} at various values of h_{ia}/h_{os}.

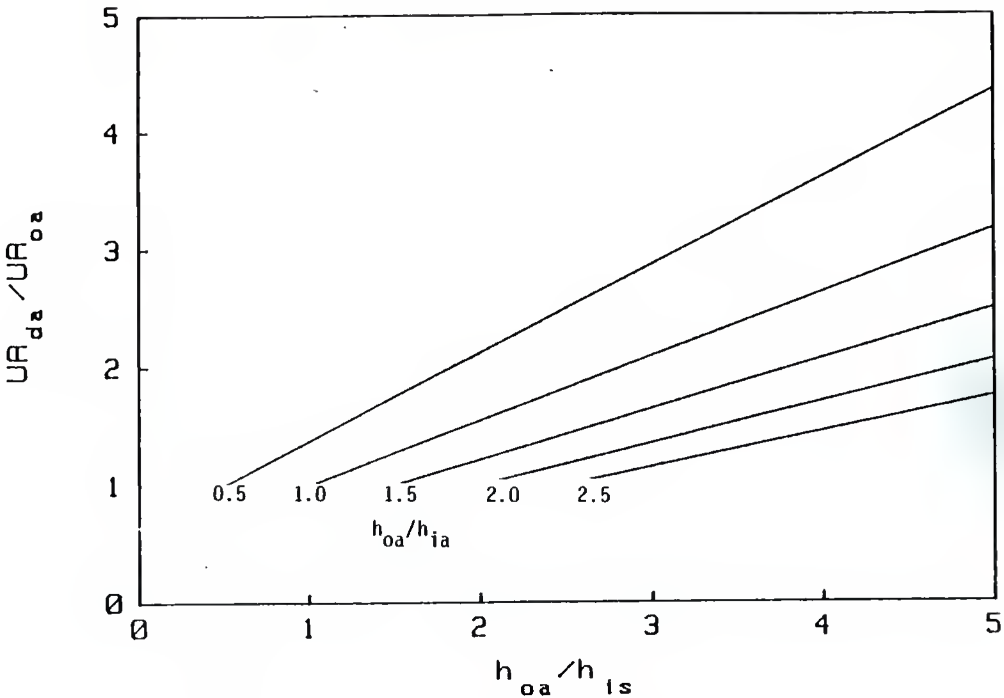

Fig. 6.23 UA_{da}/UA_{oa} versus h_{oa}/h_{is} at various values of h_{oa}/h_{ia}.

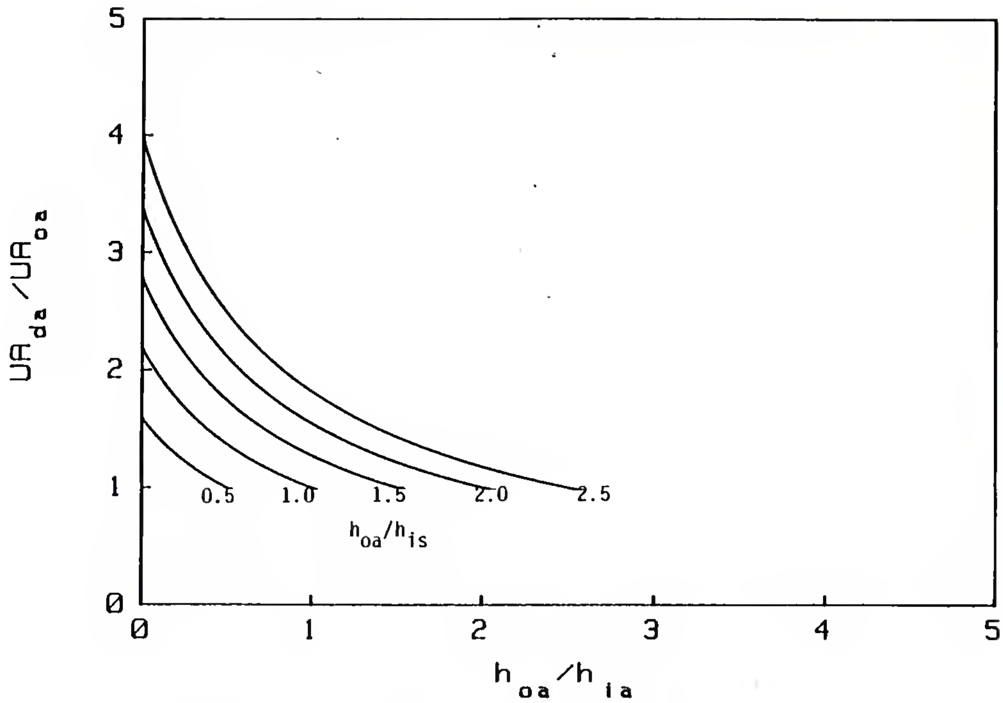

Fig. 6.24 UA_{da}/UA_{oa} versus h_{oa}/h_{ia} at various values of h_{oa}/h_{is}.

7. Figure 6.23 shows that $(UA)_{da}/(UA)_{oa}$ increases linearly with the increase in h_{oa}/h_{is} for constant h_{oa}/h_{ia}.
8. Figure 6.24 shows that $(UA)_{da}/(UA)_{oa}$ also decreases with the increase in h_{oa}/h_{ia} at constant h_{oa}/h_{is}.

In general, it can be concluded that it is beneficial to augment the inner surface of tubes if the ratio of the inside to the outside heat transfer coefficients is smaller than 1.5 (Fig. 6.17). It is not beneficial to keep increasing the inside heat transfer coefficient (by augmentation of the inner surface) since there is only a marginal increase in UA_{ia}/UA_{is} for a value of $h_{ia}/h_{is}>3.0$ (Fig. 6.18). An increase in UA_{ia}/UA_{is} with the increase in h_{ia}/h_{is} at fixed values of h_{is}/h_{os} is explained by the fact that an increase in h_{ia}/h_{is} at a fixed value of h_{is}/h_{os} requires that h_{ia} must increase. An increase in h_{ia} increases UA_{ia} (overall heat conductance).

Similar conclusions may be drawn for outside augmentation (Figs. 6.19 and 6.20). A decrease in UA_{oa}/UA_{os} with an increase in h_{is}/h_{oa} at fixed values of h_{is}/h_{os} (Fig. 6.20) is due to the fact that a decrease in h_{oa} for fixed h_{is}/h_{os} results in an increase in h_{is}/h_{oa}. As expected, a decrease in h_{oa} results in a decrease in UA_{oa}/UA_{os}.

The above conclusions also apply for doubly augmented tubes. UA_{da}/UA_{ia} must decrease with an increase in h_{ia}/h_{oa} for fixed values of h_{ia}/h_{os} because an increase in h_{ia}/h_{oa} with h_{ia} held constant means that h_{oa} must decrease resulting in a decrease in UA_{da} (Fig. 6.22). Figure 6.22 can also be used to predict the increase in overall heat transfer conductance (UA) for a given increase in outside heat transfer coefficient due to augmentation. For a fixed value of h_{ia}/h_{os}, the rate of increase in overall heat transfer conductance (UA) with an increase in the outside heat transfer coefficient is given by the negative of the slope of the curve corresponding to the chosen value of h_{ia}/h_{os}. It is easily seen that the rate of increase in UA_{da}/UA_{ia} is highest for values of $h_{ia}/h_{oa}<2.0$.

Similarly, UA_{da}/UA_{oa} must decrease with the increase in h_{oa}/h_{ia} for fixed values of h_{oa}/h_{is} as shown in Fig. 6.24. Increasing h_{oa}/h_{ia} and keeping h_{oa}/h_{is} constant means that h_{ia} should be reduced resulting in a reduction of UA_{da}/UA_{oa}. Figure 6.24 can be used to predict the increase in overall heat transfer conductance (UA) with an increase in the inside heat transfer coefficient by augmentation of the inner surface. In this case, the rate of increase in UA_{da}/UA_{oa} is highest for values of $h_{oa}/h_{ia}<2.0$.

6.4 PERFORMANCE EVALUATION AT CONSTANT PUMPING POWER

Among the criteria of performance evaluation suggested by Webb et al. [25,26,27], only the following two criteria are applicable to this investigation:

1. Increased heat duty (UA) for equal pumping power for heat exchangers with the same geometric parameters.
2. Reduced pumping power for equal heat duty for heat exchangers with the same geometric parameters.

The first criterion is considered in this section and the second criterion is considered in the next section.

An equation for increased heat duty at constant pumping power was developed. Consider two heat exchangers having the same geometric parameters (same nominal diameter and same length). These heat exchangers can have any type of inner/outer surfaces for the inner tubes. When comparing the two heat exchangers, let one of them be considered as a reference heat exchanger. The following notations will apply to the inner tube of the heat exchanger and will be adopted for the analysis that follows:

1. h = heat transfer coefficient
2. A = heat transfer area
3. Subscript 'o' refers to the outside surface
4. Inside surface is identified by no subscript
5. A second subscript 's' denotes the reference heat exchanger
6. The heat exchanger to be compared to the reference heat exchanger will have no second subscript.

As an example, h_s represents the inside heat transfer coefficient of the reference heat exchanger while h represents the inside heat transfer coefficient of the heat exchanger being compared. Similarly, h_{os} represents the outside heat transfer coefficient of the reference heat exchanger while h_o would be the outside heat transfer coefficient of the heat exchanger being compared. A_s represents the inside heat transfer area for the reference heat exchanger and A represents the inside heat transfer area of the heat exchanger being compared. Similarly, A_{os} is the outside heat transfer area for the reference heat exchanger and A_o is the outside heat transfer area of the heat exchanger to be compared.

If the wall and fouling resistances are ignored, since they are usually small compared to the inside and outside film coefficients, then the following equations can be written for the reference heat exchanger and the other heat exchanger being compared which will be called the test heat exchanger in the remainder of this chapter.

$$\frac{1}{U_sA_s} = \frac{1}{h_sA_s} + \frac{1}{h_{os}A_{os}} \tag{6-13}$$

and

$$\frac{1}{UA} = \frac{1}{hA} + \frac{1}{h_oA_o} \tag{6-14}$$

These equations can be re-written as:

$$\frac{1}{U_sA_s} = \frac{1}{h_sA_s}(1 + r) \tag{6-15}$$

and

$$\frac{1}{UA} = \frac{1}{hA}\left[1 + r\left(\frac{h_{os}}{h_s}\right)\left(\frac{A_{os}}{A_s}\right)\left(\frac{h}{h_o}\right)\left(\frac{A}{A_o}\right)\right] \tag{6-16}$$

$$\text{where} \quad r = (h_s/h_{os})(A_s/A_{os}) \tag{6-17}$$

Dividing Eq. (6-15) by Eq. (6-16) yields:

$$\frac{UA}{U_sA_s} = \frac{1 + r}{(h_s/h)(A_s/A) + r(h_{os}/h_o)(A_{os}/A_o)} \tag{6-18}$$

It can be shown that:

$$h_s/h = (St_s/St)(G_s/G) \tag{6-19}$$

Thus, Eq. (6-18) becomes:

$$\frac{UA}{U_sA_s} = \frac{1 + r}{(St_s/St)(G_s/G)(A_s/A) + r(h_{os}/h_o)(A_{os}/A_o)} \tag{6-20}$$

It is convenient to define:

$$G^* = G_s/G \tag{6-21}$$

Hence:

$$\frac{UA}{U_sA_s} = \frac{1 + r}{(St_s/St)(A_s/A)G^* + r(h_{os}/h_o)(A_{os}/A_o)} \tag{6-22}$$

This equation is a more general one than the one developed by Webb and Scott [26]. However, it is similar to the equation developed by Webb [27] except that the fouling and wall resistances are neglected.

The pumping power is calculated by multiplying the volume flow rate of the tube-side fluid and its pressure drop across the heat exchanger. The pumping power required to pump the fluid through the inner tube of the two heat exchangers can be calculated as follows:

$$P = \Delta p\, Q = 4\, f\, (L/D)(\rho\, V^2/2)\, Q \tag{6-23}$$

and

$$P_s = \Delta p_s\, Q_s = 4\, f_s\, (L/D_s)\, (\rho\, V_s{}^2/2)\, Q_s \tag{6-24}$$

It is to be recalled that the subscript 's' denotes a reference heat exchanger and an abscence of the subscript 's' designates the test heat exchanger.

By dividing Eq. (6-23) by Eq. (6-24) and simplifying, the following equation is obtained:

$$P/P_s = (f/f_s)(A/A_s)(1/G^*)^3 \tag{6-25}$$

Equations (6-22) and (6-25) can be used to compare any two heat exchangers. For comparing the heat exchangers at constant pumping power, Eq. (6-25) reduces to :

$$(f/f_s)(A/A_s)(1/G^*)^3 = 1 \tag{6-26}$$

The pumping power in the above equations refer to the pumping power of the fluid flowing inside the inner tube. This is the power required to pump R-113 in the present study.

When comparing two heat exchangers there are two possibilities:

1. The inner surfaces of the inner tubes are the same in the two heat exchangers.

2. The inner surfaces of the inner tube are different in the two heat exchangers. For example, one can be augmented while the second can be smooth.

In the case where the inner surfaces of the inner tubes of the two heat exchangers are the same, pumping power for the same mass flow rate of tube side fluid is the same. However, in the second case, pumping power is not the same for the same mass flow rate of tube side fluid in the two heat exchangers. As an example, the pressure drop is higher for internally finned tubes than for tubes with smooth inner surface at the same mass/volume flow rate.

The above two cases are considered seperately because of this inherent difference.

6.4.1 Inner tubes with same inside surfaces

As mentioned earlier, the pumping power of the reference and test heat exchangers is the same only when the mass flow rate of the fluid on the tube side is the same for the two heat exchangers. Also, the outside surface of the inner tube has no effect on the pumping power of the tube side fluid. Thus, this analysis can be applied in comparing heat exchangers with the same inside surfaces for the inner tubes but with different kinds of augmentation on the outside.

If the mass flux inside the inner tube is based on the nominal diameter then G^* for this case is unity. For the same pumping power the following also applies:

$$P/P_s = f/f_s = St/St_s = G^* = 1 \qquad (6\text{-}27)$$

Thus Eq. (6-22) reduces to:

$$UA/U_sA_s = (1 + r)/[1 + r(h_{os}/h_o)] \qquad (6\text{-}28)$$

where

$$r = (h_s/h_{os})(A_s/A_{os}) \qquad (6\text{-}17)$$

Equation (6-28) is the governing equation to compare the heat rate at constant pumping power for heat exchangers with inner tubes having the same inside surface.

In this investigation, tubes 1,3,7 and 9 had internal fins. Also, tubes 1 and 7 had smooth outside surfaces while tubes 3 and 9 had knurled outside surfaces. From the constraint of similar geometry, tubes 1 and 3 or tubes 7 and 9 could be compared. From the experimental measurements of tubes 1 and 3, an average value of h_{os}/h_o from Figs. 6.5 and 6.6 was about 0.75. An average value of h_s/h_{os} from Table C.3 was calculated to be 0.47. From the geometric parameters of the tubes, A_s/A_{os} was about 0.82 for tubes 1 and 3. For tubes 7 and 9, average values of h_{os}/h_o was 0.83 (Figs. 6.7 and 6.8), h_s/h_{os} was 0.25 (Table C.3) and A_s/A_{os} was 0.83. Thus the heat conductance ratio UA/U_sA_s calculated from Eq. (6-28) was 1.07 for tubes 1 and 3 (UA_3/UA_1) and 1.03 for tubes 7 and 9 (UA_9/UA_7). As expected, these ratios are greater than unity.

An attempt was also made to verify these results by actual computation of the pumping power. The pumping power was calculated for all the experimental runs by multiplying the volume flow rate with the corresponding pressure drop. For runs with the same pumping power, UA/U_sA_s ratios were calculated from the experimental results of these runs. It was found that UA_3/UA_1 was 1.07 and UA_9/UA_7 was 0.90 for the same pumping power. Although UA_3/UA_1 was greater than 1.0, UA_9/UA_7 was found to be less than 1. The reason for the ratio being less than 1.0 for tubes 7 and 9 was the fact that these tubes had larger nominal diameters and had very small pressure drops. Therefore, measurements were subject to greater error.

Equation (6-28) can also be used to predict how UA/U_sA_s varies with the outside surface of the inner tubes expressed in the ratio h_{os}/h_o and the ratio h_s/h_{os} of the inside to the outside heat transfer coefficients of the reference tube.

i) Effect of varying h_s/h_{os} while keeping h_{os}/h_o fixed

This case can be viewed as comparing two heat exchangers with the same inside surfaces but different outside surfaces for the inner tubes at different flow rates of the tube side fluid. Actually, it does not matter how h_s/h_{os} is varied while keeping h_{os}/h_o constant. It is important however, to determine theoretically how UA/U_sA_s varies with h_s/h_{os}.

Figure 6.25 shows the variation of UA/U_sA_s as a function of h_s/h_{os} at various values of h_{os}/h_o.

Certain interesting conclusions can be made. For values of h_{os}/h_o less than unity (cases where the reference inner tube has smooth outside surface and the test heat exchanger has an inner tube with augmented outside surface), UA/U_sA_s increases with increase in h_s/h_{os}.

2) Effect of varying h_{os}/h_o while keeping h_s/h_{os} fixed.

This case can be viewed as comparing different heat exchangers with different kinds of augmentations on the outside surface for the inner tube. Also, a comparison can be made for two heat exchangers with different flow rates of the fluid on the annular side in the test heat exchanger.

Figure 6.26 shows the effect of changing h_{os}/h_o on UA/U_sA_s at different values of h_s/h_{os}. UA/U_sA_s decreases with the increase in h_{os}/h_o. The rate of decrease in UA/U_sA_s is higher for lower values of

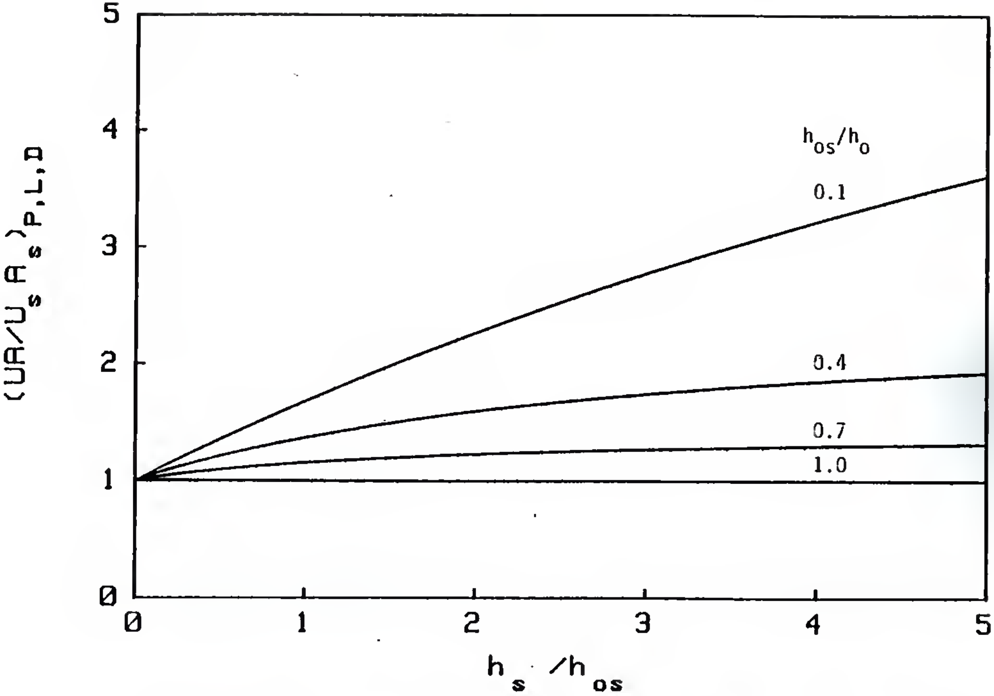

Fig. 6.25 Ratio of heat duty (UA) of augmented to smooth inner tubes of heat exchangers at constant pumping power and fixed geometry versus h_s/h_{os} at constant h_{os}/h_o.

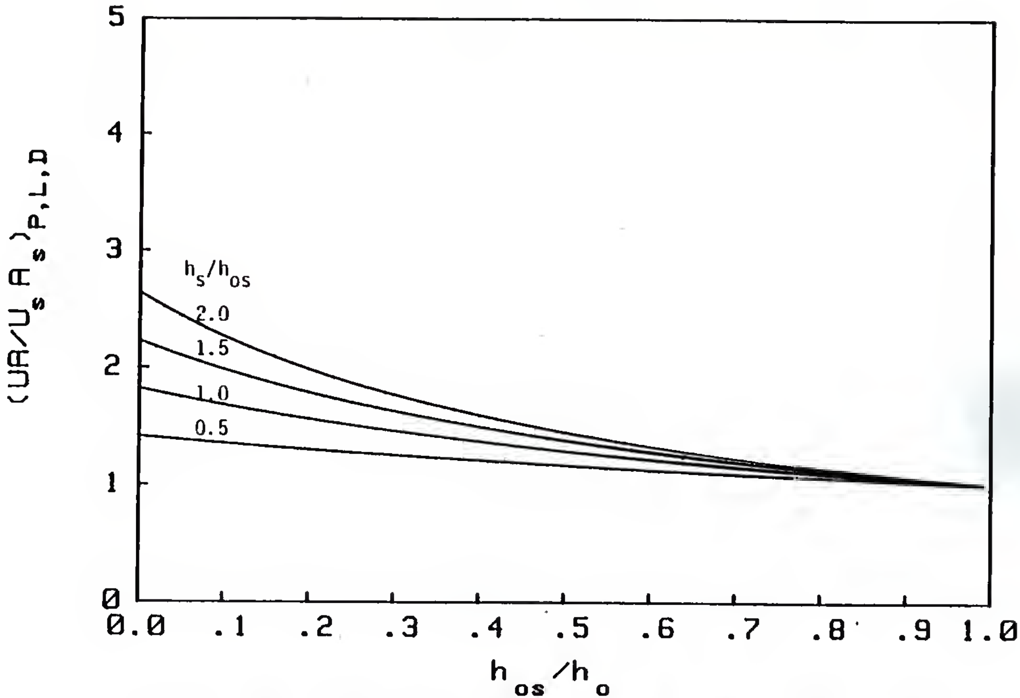

Fig. 6.26 Ratio of heat duty (UA) of augmented to smooth inner tubes of heat exchangers at constant pumping power and fixed geometry versus h_{os}/h_o at constant h_s/h_{os}.

h_{os}/h_o (less than 0.5).

In general, it may be concluded that outside augmentation for the inner tube can be beneficial in increasing the overall heat conductance (UA) as long as h_s/h_{os} is less than 3.0.

6.4.2 Inner tubes with different inside surfaces

If the inner surfaces of the inner tubes are different, the pumping power required to pump tube side fluid through the two heat exchangers is different for the same mass flow rate.

Since the comparisons are being made at constant pumping power, Eq. (6-26) is still valid. However, for reasons mentioned earlier Eq. (6-27) is not. Equation (6-22) is also valid for this case. Equations (6-26) and (6-22) must be solved simultaneously to derive the governing equation for this case. Since the two heat exchangers being compared have inner tubes with different inner surfaces, correlation equations are necessary to determine St_s/St and G^* in Eq. (6-22) and f/f_s in Eq. (6-26).

Many correlation equations were discussed in an earlier chapter for internally finned tubes. Correlation equations developed in this study for internally finned tubes were used in this analysis. Hence the analysis that follows is applicable to internally finned tubes only. The correlation equations for heat transfer and pressure drop are of the following form:

$$Nu = c\, Re^a\, Pr^{0.4}\, F_1^{\,e}\, F_2^{\,f}\, F_3^{\,g} \tag{6-29}$$

$$f = d\, Re^b\, F_3^{\,\ell}\, F_4^{\,m} \tag{6-30}$$

where a, b, c, d, e, f, g, ℓ and m are all constants and F_1, F_2, F_3 and F_4 are parameters of the tubes defined in chapter 5.

The above equations can be written for the reference and test heat exchangers. It can then be shown that:

$$Nu/Nu_s = (Re/Re_s)^a F_1^e F_2^f F_3^g \tag{6-31}$$

$$f/f_s = (Re/Re_s)^b F_3^{\ell} F_4^m \tag{6-32}$$

Therefore the following equation can be derived:

$$G^* (St_s/St) = (D_s/D)^{a-1} (G^*)^a F_1^{-e} F_2^{-f} F_3^{-g} \tag{6-33}$$

Combining Eqs. (6-26) and (6-32) it can be shown that:

$$G^* = [(D/D_s)^b F_3^{\ell} F_4^m]^{1/(3+b)} \tag{6-34}$$

Substituting Eqs. (6-33) and (6-34) into Eq. (6-22), the following equation is obtained:

$$\frac{UA}{U_sA_s} = \frac{1 + r}{(A_s/A)(D_s/D)^{a-1}[(D/D_s)^b F_3^{\ell} F_4^m]^{a/(3+b)} F_1^{-e} F_2^{-f} F_3^{-g} + rh_{os}A_{os}/h_oA_o} \tag{6-35}$$

In this study, all calculations are based on the nominal diameter. Thus:

$$D_s/D = A_s/A = A_{os}/A_o = 1 \tag{6-36}$$

Hence Eq. (6-35) reduces to:

$$\frac{UA}{U_sA_s} = \frac{1 + r}{F_3^{\{[a.\ell/(3+b)] - g\}} F_4^{[a.m/(3+b)]} F_1^{-e} F_2^{-f} + r(h_{os}/h_o)} \tag{6-37}$$

Values of a,b,c,d,e,f,g,ℓ and m were obtained from the correlations developed in chapter 5 and are given in Table 6.2. It is to be emphasized that Eq. (6-37) is valid for comparing two heat exchangers with one of them having an inner tube with smooth inside surface and the other having an internally finned inner tube. Also, the constants in Table 6.2 are only

TABLE 6.2. Values of constants in correlation equations.

Constant	Experimental value
a	0.680
b	-1.130
c	0.070
d	330.634
e	-1.502
f	1.765
g	16.548
ℓ	4.723
m	-7.389

applicable to single phase flow for $2700<Re_R<7850$.

Equation (6-37) was used to evaluate UA/U_sA_s for tubes 2 and 3, and tubes 8 and 9 at constant pumping power. Values of F_1, F_2, F_3 and F_4 are given in Table 5.1. Since the outside surfaces are knurled in each pair:

$$h_{os}/h_o = 1 \tag{6-38}$$

Over the range of Reynolds number tested, the value of r calculated from Eq. (6-17) using average values from Table C.3 was 0.11 for tube 2 and 0.07 for tube 8. The UA ratios as calculated from Eq. (6-37) is 1.68 for tubes 2 and 3 (UA_3/UA_2) and 1.66 for tubes 8 and 9 (UA_9/UA_8).

From the experimental measurements, UA_3/UA_2 was found to be 3.0 and UA_9/UA_8 was 1.95 at constant pumping power. Equation (6-37) does not seem to predict the heat duty ratio accurately for tubes 2 and 3. The reason can be attributed to the fact that using correlation equations of the form given in Eqs. (6-29) and (6-30) assumes that the regression lines for the smooth and finned tubes are parallel. This was not the case for the friction factors. However, the correlation lines of $Nu/Pr^{0.4}$(R-113) versus Reynolds number of R-113 were parallel for the finned tubes and tubes with smooth inner surfaces.

Equation (6-37) can still predict qualitatively UA/U_sA_s at constant pumping power for internally finned tubes with different outside surface conditions. It was used to show the effect of h_{os}/h_o and h_s/h_{os} on UA/U_sA_s.

i) Effect of varying h_{os}/h_o while keeping h_s/h_{os} constant.

The comparison is equivalent to comparing heat exchangers with various kinds of augmentation on the outside. A plot of UA/U_sA_s against h_{os}/h_o at different values of h_s/h_{os} is shown in Fig. 6.27.

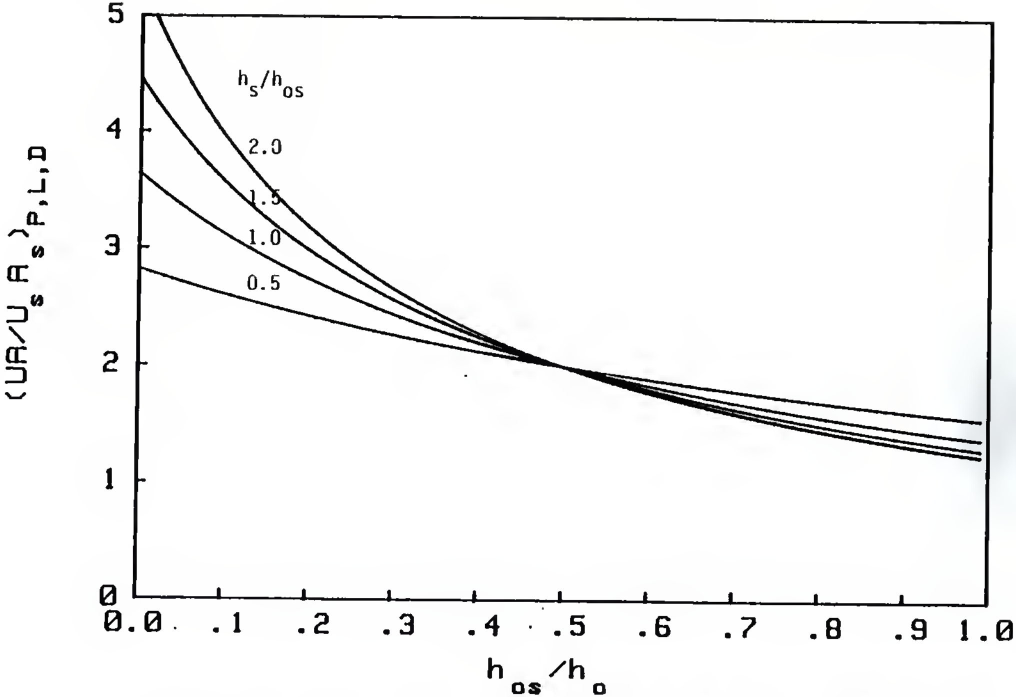

Fig. 6.27 Ratio of heat duty (UA) for a heat exchanger with augmented inner tubes to that of a heat exchanger with smooth inner tubes at constant pumping power and fixed geometry versus h_{os}/h_o at constant h_s/h_{os}.

The figure shows that UA/U_sA_s decreases with the increase in h_{os}/h_o for tubes with the same geometry. The rate of decrease in UA/U_sA_s is higher at lower values of h_{os}/h_o (less than 0.5) because the outside heat transfer coefficients do not have any effect on pumping power. Curves of UA/U_sA_s versus h_{os}/h_o at various values of h_s/h_{os} intersect at a value of $h_{os}/h_o = 0.5$.

The reason for the curves to intersect at 0.5 is that for the tubes studied in this investigation, the average value of $F_3{\{[\ell.a/(3+b)] - g\}}F_4{[a.m/(3+b)]}F_1{-e}F_2{-f}$ in Eq. (6-37) was 0.5 and hence UA/U_sA_s was equal to 2.0 (Eq. 6-37) for all values of h_s/h_{os} when h_{os}/h_o was equal to 0.5.

ii) Effect of varying h_s/h_{os} while keeping h_{os}/h_o constant.

This case compares two heat exchangers where the reference heat exchanger has a tube with a smooth inner surface and the test heat exchanger has an inner tube with internal fins on the inside.

A plot of UA/U_sA_s versus h_s/h_{os} is shown in Fig. 6.28. The results show that UA/U_sA_s increased with the increase in h_s/h_{os} for $h_{os}/h_o<0.5$ and decreases with the increase in h_s/h_{os} for $h_{os}/h_o>0.5$. Thus augmentation on the outside is most beneficial if $h_o>2h_{os}$. Also, the UA ratios are greater for smaller values of h_s/h_{os}.

6.5 PERFORMANCE EVALUATION AT CONSTANT UA

In this case UA/U_sA_s is unity. Heat exchangers are compared on the basis of pumping power for the same heat duty (UA). Equation (6-22) becomes:

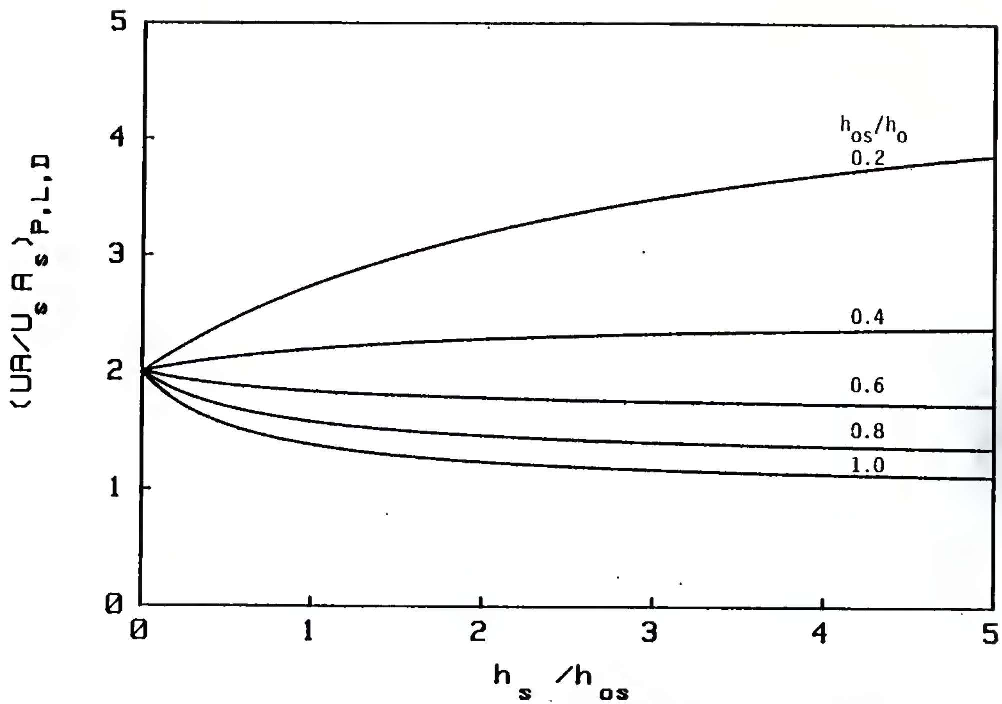

Fig. 6.28 Ratio of heat duty (UA) for a heat exchanger with augmented inner tubes to that of a heat exchanger with smooth inner tubes at constant pumping power and fixed geometry versus h_s/h_{os} at constant h_{os}/h_o.

$$\frac{1 + r}{(St_s/St)(A_s/A)G^* + r(h_{os}/h_o)(A_{os}/A_o)} = 1 \tag{6-39}$$

This can be solved for G^*:

$$G^* = \{1 + r\,[1 - (h_{os}/h_o)(A_{os}/A_o)]\}\ (St/St_s)(A/A_s) \tag{6-40}$$

Substituting Eq. (6-40) into Eq. (6-25) yields:

$$P/P_s = (f/f_s)(A/A_s)\{(St_s/St)(A_sA)/[1+r(1-h_{os}A_{os}/h_oA_o)]\}^3 \tag{6-41}$$

Substituting Eqs. (6-29) and (6-30) into Eq. (6-41) yeilds:

$$P/P_s=(A/A_s)(D/D_s)F_3{}^{\ell}F_4{}^{m}\ \{1+r(1-h_{os}A_{os}/h_oA_o)(A/A_s)(D_s/D)^{1-a}F_1{}^{e}F_2{}^{f}F_3{}^{g}\}^{(-3-b)/a} \tag{6-42}$$

Since correlation equations were used to develop the above equation, this equation is valid for the case of a reference heat exchanger having an inner tube with smooth inner surface and the test heat exchanger having an inner tube which is internally finned.

Since the correlations used were based on the nominal diameters and areas, Eq. 6.42 reduces to:

$$P/P_s = F_3{}^{\ell}F_4{}^{m}\{[1+r(1-h_{os}/h_o)]F_1{}^{e}F_2{}^{f}F_3{}^{g}\}^{(-3-b)/a} \tag{6-43}$$

Equation (6-43) can be used to show the change in the pumping power ratio as h_{os}/h_o is changed. The plot of P/P_s versus h_{os}/h_o at constant heat rate per unit temperature difference is shown in Fig. 6.29.

Pumping power ratio at constant heat duty (UA) increases with the increase in h_{os}/h_o. This means that at constant h_s/h_{os}, if h_{os}/h_o is decreased, for example, by increasing the shell side fluid flow rate, the pumping power ratio P/P_s must decrease for the same heat duty (UA).

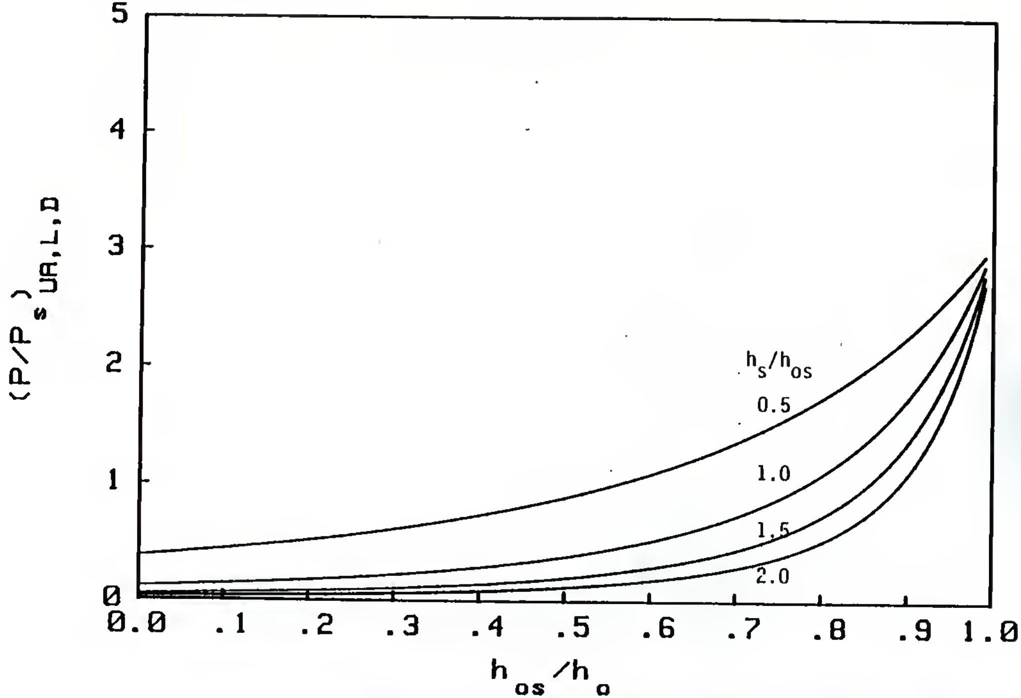

Fig. 6.29 Ratio of pumping power for heat exchanger with augmented inner tubes to that of a heat exchanger with smooth inner tubes, at constant heat duty (UA) and fixed geometry, versus h_{os}/h_o at constant h_s/h_{os}.

Increase in P/P_s with increase in h_{os}/h_o is highest for larger values of h_{os}/h_o (greater than 0.5).

Figure 6.30 shows the effect of h_s/h_{os} on P/P_s at constant heat duty and fixed geometry for different values of h_{os}/h_o. At a constant value of h_{os}/h_o, P/P_s decreases (and thus is beneficial) with the increase in h_s/h_{os}. This benefit is greater for smaller values of h_s/h_{os}. Also, the rate of decrease in P/P_s with the increase in h_s/h_{os} is greater for smaller values of h_{os}/h_o (less than 0.6).

Comparing two heat exchangers with inner tubes having the same outside surface results in h_{os}/h_s=1.0, Eq. (6-43) reduces to:

$$P/P_s = F_3^{\ell} F_4^{m} \, [F_1^{e} F_2^{f} F_3^{g}]^{(-3-b)/a} \qquad (6\text{-}44)$$

Pumping power ratio for the two heat exchangers is constant for this case.

6.6 PUMPING POWER PER UNIT HEAT TRANSFER

The ratio of pumping power to the rate of heat transfer P/Q could be used as an evaluation index. Azer et al. [31] used this ratio to evaluate the performance of static-in-line mixers in augmenting the condensation heat transfer inside horizontal tubes. This index P/Q was evaluated for all the tubes at the same mass flow rate of R-113 under the constraints of fixed geometry, same mass flow rate of water and same inlet temperatures of water and R-113. The pumping power was obtained from the product of the pressure drop and the volumetric flow rate of R-113.

P/Q is plotted against Reynolds number of R-113 for all the tubes and sets 0 and 1 in Figs. 6.31 through 6.34 for all the tubes tested. From these figures, the following can be deduced:

1. Pumping power per unit heat transfer increases with increase in Reynolds number of R-113.

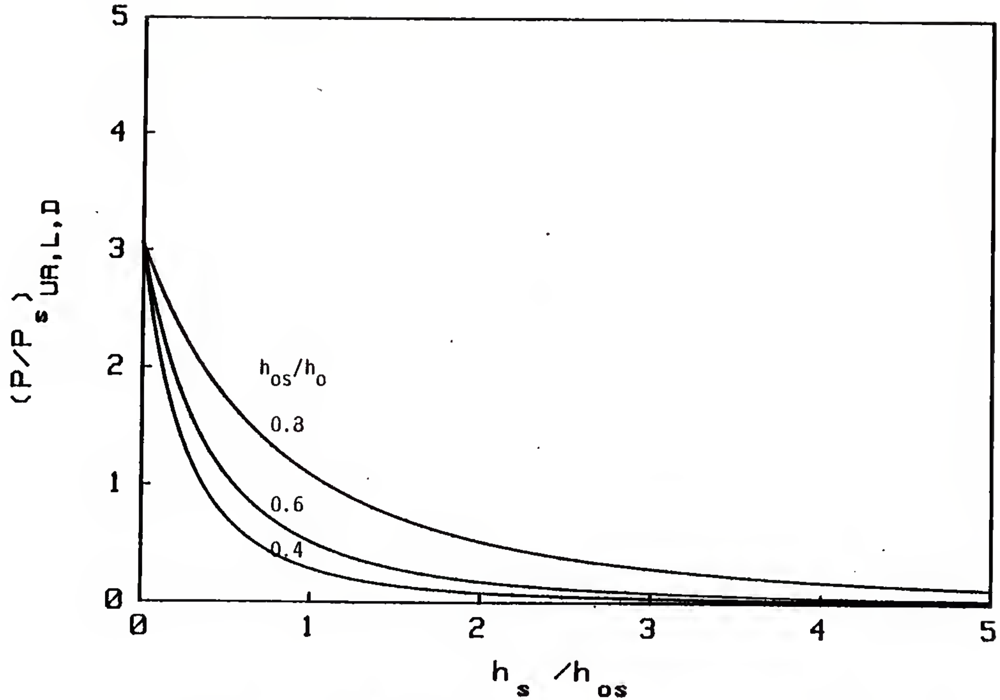

Fig. 6.30 Ratio of pumping power for heat exohanger with augmented inner tubes to that of a heat exchanger with smooth inner tubes, at constant heat duty (UA) and fixed geometry, versus h_s/h_{os} at constant h_{os}/h_o.

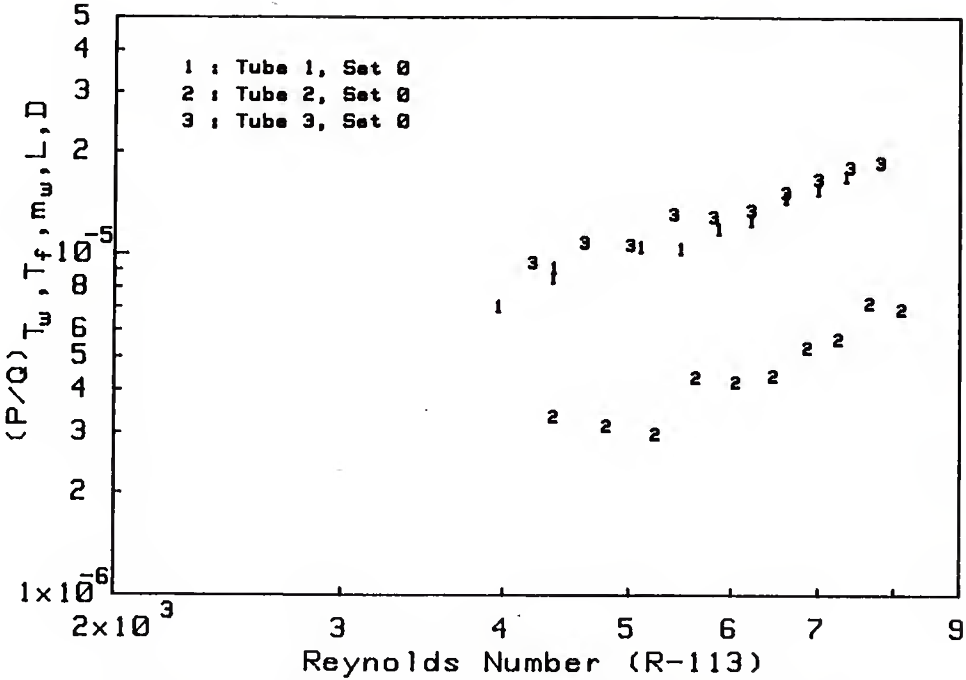

Fig. 6.31 Experimental values of the ratio of pumping power to heat transfer rate versus Reynolds number of R-113 for tubes 1,2 and 3, set 0.

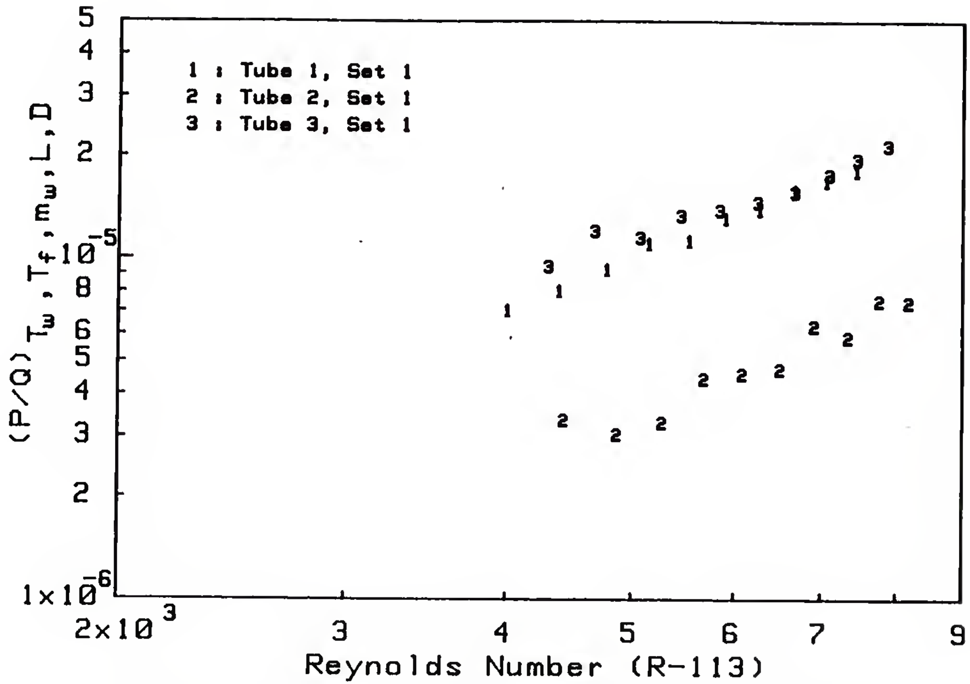

Fig. 6.32 Experimental values of the ratio of pumping power to heat transfer rate versus Reynolds number of R-113 for tubes 1,2 and 3, set 1.

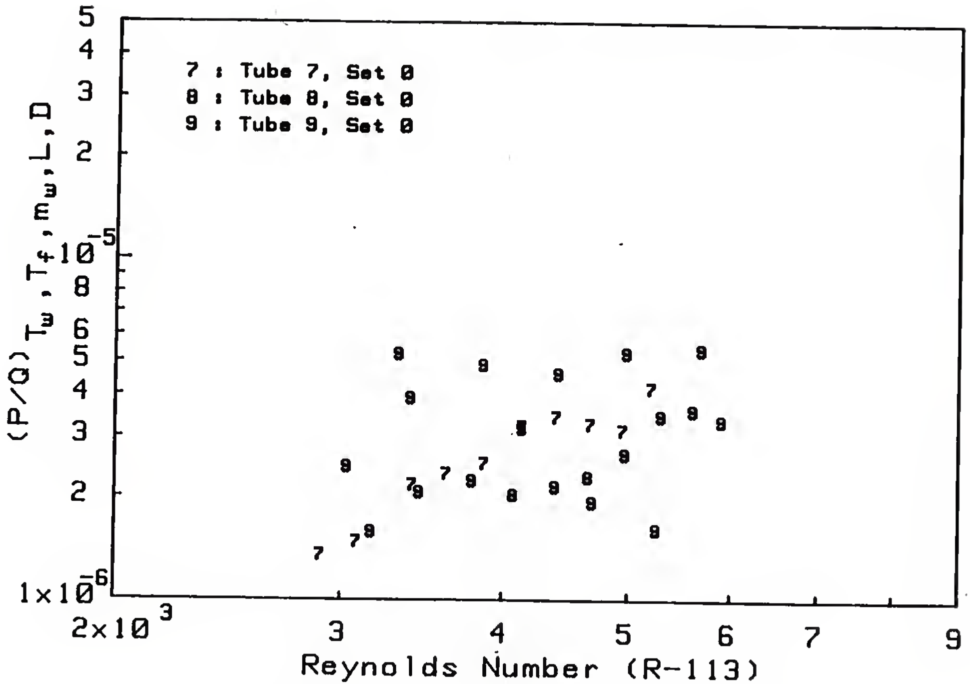

Fig. 6.33 Experimental values of the ratio of pumping power to heat transfer rate versus Reynolds number of R-113 for tubes 7,8 and 9, set 0.

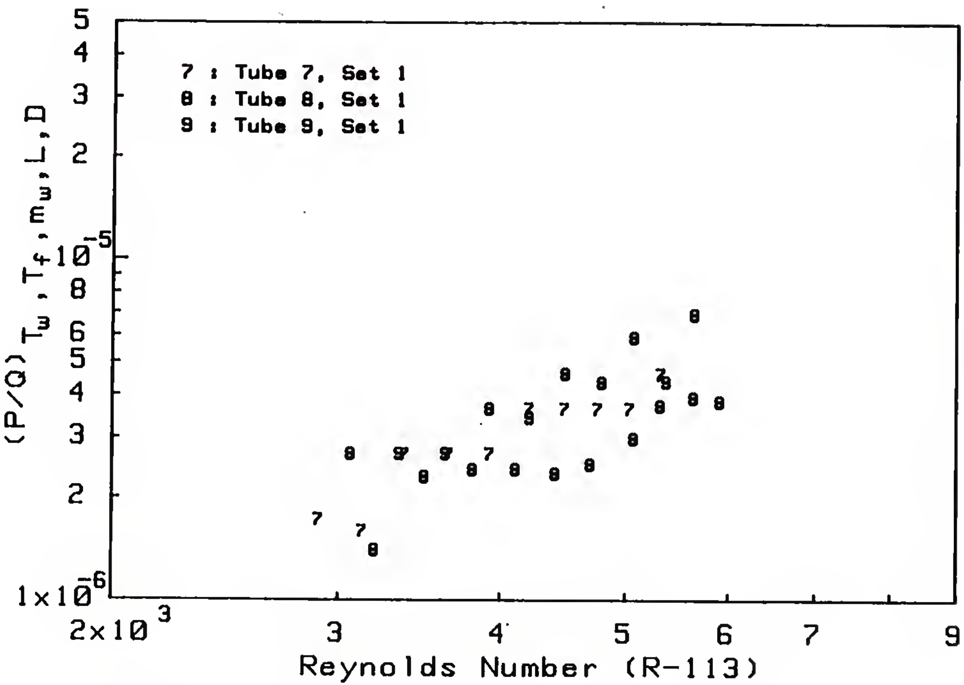

Fig. 6.34 Experimental values of the ratio of pumping power to heat transfer rate versus Reynolds number of R-113 for tubes 7,8 and 9, set 1.

2. Pumping power per unit heat transfer is higher for internally finned tubes than for tubes with smooth inside surfaces.

3. Pumping power per unit heat transfer for tubes 1 and 3 are about the same at the same Reynolds number of R-113. This means that the outside surface does not affect this index (P/Q).

4. There is a lot of scatter in the plot of pumping power per unit heat transfer for tubes 7,8 and 9 (Figs. 6.33 and 6.34). The reason for this is that the pressure drops for these tubes were very low and hence subject to greater error in its measurement.

Chapter VII

SUMMARY, CONCLUSIONS AND RECOMMENDATIONS

Heat transfer and pressure drop data were taken during cooling of liquid R-113 inside 6 different double pipe, counterflow heat exchangers with R-113 flowing inside the inner tube and the cooling water flowing in the annulus. The heat exchangers consisted of two sets with three exchangers in each set. Within each set the inner tube of each exchanger had the same outside diameter. However, in one exchanger the inner tube had internal fins on the inside and a smooth surface on the outside. In the second exchanger the inner tube had knurls on the outside surface and a smooth surface on the inside. In the third exchanger the inner tube had the same fins of the first exchanger on the inside and the same knurls of the second exchanger on the outside. This tube is what is identified as a doubly augmented tube. The two sets of exchangers differed in the outside diameter of the inner tube and the number of fins. The results are summarized in the following:

1. Over the range of Reynolds number tested for R-113 (2700 - 7850), internally finned tubes enhanced the inside heat transfer coefficient by 200% over the smooth tube results on a nominal area basis.
2. The surface rougness created by the knurls enhanced the water side heat transfer coefficient by 25% over the smooth surface over the Reynolds number range encountered (700 - 3500).
3. The overall heat transfer coefficient and consequently the performance of the heat exchanger was controlled by the tubes with fins on the inside. This was due to the fact that the outsside heat transfer coefficient of water was about three times the inside heat transfer coefficient of the inner tube when both surfaces were smooth. Therefore augmenting the side with the lower heat transfer coefficient

controlled the overall heat transfer coefficient.

4. The overall heat transfer coefficient between tubes with augmentation on the inside surface only and tubes which were doubly augmented were not significantly different. The overall heat transfer rate per unit temperature difference UA increased by 190% for doubly augmented tubes compared to smooth inner tube.

5. Heat transfer and pressure drop correlations were developed for cooling of R-113 inside the internally finned tubes. The heat transfer correlation predicted the heat transfer coefficient to within ±10% for 98% of the data points. The pressure drop correlation predicted the pressure drop to within ±15% for 62% of the data points. These correlations are valid for the ranges of Reynolds number and parameters F_1, F_2, F_3 and F_4 covered in the present study.

6. The performance of the heat exchangers with inside doubly augmented tubes was compared with the performance of exchangers with inside singly augmented tubes (finned on the inside and smooth on the outside and/or smooth on the inside and knurled on the outside). The comparisons were based on evaluating the ratio of the overall heat transfer rate per unit temperature difference (UA) of any two exchangers subject to the constraint of the same pumping power of R-113. Also, the performance was evaluated on the basis of the pumping power demand for R-113 for the same overall heat transfer rate. The results showed that the benefits of double augmentation versus single augmentation in improving the overall heat transfer of the heat exchanger depend mainly on the ratio of the inside to the outside heat transfer coefficients of the inner tube before augmenting the inside and outside surfaces. The benefits of having inside augmentation

versus outside augmentation alone, as well as double augmentation of the inner tube of the heat exchanger were also discussed.

Recommendation for Future Studies

The results of the present study have shown that using singly augmented versus doubly augmented tubes in improving the overall heat transfer of a heat exchanger depends mainly on the ratio of the inside to the outside heat transfer coefficients of the inner tube before augmenting both side of the tube. If one side of the inner tube has a heat transfer coefficient which is 2-3 times the heat transfer coefficient on the other side, then double augmentation may not be an answer and single augmentation is recommended for the side which has the lower heat transfer coefficient. These observations were based on single phase flow on both side of the inner tube.

The effect of double augmentation needs also to be investigated for other heat transfer modes such as condensation inside the inner tubes. Refrigerants are the best candidate for such studies. Certain refrigerants have thermophysical and transport properties that result in low condensation heat transfer coefficients that are close to the heat transfer coefficients of the coolant (water). Double augmentation of the inner tubes of such exchangers can be very effective in improving their performance.

Economic studies are also needed to evaluate the benefits of increasing the heat transfer efficiency against the increased cost of manufacturing the doubly augmented tubes.

To predict outside heat transfer coefficients, correlation equations having modifying factors which include outside surface roughness needs to be developed.

The effect of outside augmentation on pressure drop of the fluid flowing in the annulus needs to be studied. Thus performance of heat exchangers can also be investigated in terms of the total pumping power and heat transfer rate.

The equations developed for performance evaluation can be extended to include the effects of fin geometries, such as, pitch, helix angle etc. Each of these parameters could be studied in detail.

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SELECTED BIBLIOGRAPHY

1. Bergles, A.E., "Research Workshop on Augmentation of Convective Heat Transfer", Final Report HTL-8, ISU-ERI-Ames-76026 (1975).

2. Bergles, A.E., Nirmalan, V., Junkhan, G.H., and Webb, R.L., "Bibliography on Augmentation of Convective Heat and Mass Transfer", Bibliographic Report, HTL-31, ISU-ERI-Ames-84211 (1983).

3. Watkinson, A.P., Miletti, D.L., and Tarasoff, P., "Turbulent Heat Transfer and Pressure Drop in Internally Finned Tubes", AIChE Symposium Series, Vol. 69, No. 131, (1973): 94-103.

4. Watkinson, A.P., Miletti, D.L., and Kubanek, G.R., "Heat Transfer and Pressure Drop of Internally Finned Tubes in Turbulent Air Flow", ASHRAE Transactions, Vol. 81, Part 1, (1975): 330-349.

5. Watkinson, A.P., Miletti, D.L., and Kubanek, G.R., "Heat Transfer and Pressure Drop of Internally Finned Tubes in Laminar Oil Flow", ASME Paper 75-HT-41, Presented at AIChE-ASME Heat Transfer Conference, San Francisco, Ca., August (1975).

6. Russell, J.J., and Carnavos, T.C., "An Experimental Study: Cooling Air in Turbulent Flow with Internally Finned Tubes", 16th Natl. Heat Transfer Conf., St. Louis, Mo., AIChE Paper 28, (1976).

7. Carnavos, T.C., "Cooling Air in Turbulent Flow with Internally Finned Tubes", Presented at the 17th Natl. Heat Transfer Conf., Salt Lake City, Utah. AIChE Paper 4, Aug. 14-17 (1977). Also, Heat Transfer Engineering, Vol. 1, No. 2. Oct.-Dec. (1979): 41-46.

8. Carnavos, T.C., "Cooling Air in Turbulent Flow Multi-Passage Internally Finned Tubes", ASME Paper 78-WA/HT-52, Presented at the Winter Annual Meeting, San Francisco, Ca., Dec. 10-15 (1978).

9. Carnavos, T.C., "Heat Transfer Performance of Internally Finned Tubes in Turbulent Flow", Heat Transfer Engineering, Vol. 1, No. 4, April-June (1980): 32-37.

10. Bergles, A.E., Brown, G.S.,Jr., and Snider. W.D., "Heat Transfer Performance of Internally Finned Tubes", ASME Paper 71-HT-31, Presented at ASME-AIChE Heat Transfer Conference, Tulsa, Oklahoma, August (1971).

11. Bergles, A.E., "Laminar Flow Heat Transfer in Horizontal Tubes under Normal and Augmented Conditions", Final Report, HTL-11, ISU-ERI-Ames-77179, December (1975).

12. Hu, M.H., and Chang, Y.P., "Optimization of Finned tubes for Heat Transfer in Laminar Flow", Transactions ASME, Journal of Heat Transfer, Vol. 95, Series C, (1973): 332-338.

13. Nandakumar, K., and Masliyah, J.H., "Fully Developed Viscous Flow in Internally Finned Tubes", The Chemical Eng. J., Vol. 10, (1975): 113-120.

14. Masliyah, J.H., and Nandakumar, K., "Heat Transfer in Internally Finned Tubes", J. of Heat Transfer, Vol. 98, (1976):257-261

15. Soliman, H.M., and Feingold, A., "Analysis of Fully Developed Laminar Flow in Longitudinal Internally Finned Tubes", The Chemical Eng. J., Vol. 14, (1977): 119-128.

16. Soliman. H.M., and Feingold, A., "Analysis of Heat Transfer in Internally Finned Tubes under Laminar Flow Conditions", Heat Transfer, Vol. 2, Hemisphere Publishing Corporation, (1978): 571-576.

17. Rowley. G.J., and Patankar, S.V., "Analysis of Laminar Flow and Heat Transfer in Tubes with Internal Circumferential Fins", Int. J. of Heat Mass Transfer, Vol. 27, No. 4, (1984): 553-560.

18. Bergles, A.E., "heat Transfer Characteristics of Turbotec tubing", Final Report, HTL-24, ISU-ERI-Ames-81018, June (1980).

19. Maeda Iron Works Co. Ltd., Yoshida, Nagana-Ken, Japan, "Turbotec Spiral Tube Used in Heat Transfer - Technical Data", April (1975).

20. Fedynskii. O.S., "Increasing the Heat Transfer Rate During Flow of Water in an Annular Space", Problems of Heat Transfer, Edited by M.A. Mikheev, Atomic Energy Commission Report, AEC-TR-4511, (1959): 44-56.

21. Hudina, M., "Evaluation of Heat Transfer Performance of Rough Surfaces from Experimental Investigation in annular channels", Int. J. of Heat Mass Transfer, Vol. 22, (1979): 1381-1392.

22. Pence, D.T., "An Experimental Study of Steam Condensation On A Single Horizontal Tube", MS Thesis, Naval Postgraduate School, Monterey. Ca., March (1978).

23. Kern, D.Q., and Othmer, D.F., "Effect of Free Convection on Viscous Heat Transfer in Horizontal Tubes", Transactions AIChE, Vol. 39, (1943): 517-555.

24. Bergles, A.E., Blumenkrantz, A.R., and Taborek, J., "Performance Evaluation Criteria for Enhanced Heat Transfer Surfaces", Presented at the 13th National Heat Transfer Conference, Denver, Colorado, AIChE-ASME, Aug. 6-9 (1972).

25. Webb, R.L., and Eckert, E.R.G., "Application of Rough Surfaces to Heat Exchanger Design", Int. J. Heat Mass Transfer, Vol. 15, (1972): 1647-1658.

26. Webb, R.L., and Scott, M.J., "A parametric Analysis of the Performance of Internally Finned Tubes for Heat Exchanger Application:, Transactions of the ASME, Vol. 102. Feb. (1980): 38-43.

27. Webb, R.L., "Performance Evaluation Criteria for use of Enhanced Heat Transfer Surfaces in Heat Exchanger Design", Int. J. Heat Mass Transfer, Vol. 24, No. 4, (1981): 715-726.

28. Marner, W.J., Bergles, A.E., and Chenoweth, J.M., "On the Presentation of Performance Data for Enhanced Tubes Used in Shell and Tube Heat Exchangers", Transactions of the ASME, Vol. 105, May (1983): 358-364.

29. Bejan, A., "General Criterion for Rating Heat Exchanger Performance", Int. J. Heat Mass Transfer, Vol. 21, (1978): 655-658.

30. Bejan, A., and Pfister, P.A.Jr., "Evaluation of Heat Transfer Augmentation Techniques Based on their Impact on Entropy Generation", Letters in Heat and Mass Transfer, Vol. 7, (1980): 97-106.

31. Azer, N.Z., Fan, L.T., and Lin, S.T., "Augmentation of Condensation Heat Transfer with In-line Static Mixers", Proceedings of 1976 HTFMI, (1976): 512-526.

32. ASHRAE Handbook of fundamentals, New York: ASHRAE (1981).

33. ASHRAE Thermodynamic Properties of Refrigerants, New York: ASHRAE (1969).

34. Kline, S.J., and McClintock, F.A., "Describing the Uncertainties in Single-Sample Experiments", Mechanical Engineering 75, January (1953): 3-8.

NOMENCLATURE

Symbol	
a	Coefficient defined in Eq. (6-31)
A	Heat transfer area, m^2 (ft^2)
A_a	Actual heat transfer area, m^2/m (ft^2/ft)
A_{fa}	Actual flow area, m^2 (ft^2)
A_{fc}	Core flow area, m^2 (ft^2)
A_{fn}	Nominal flow area, m^2 (ft^2)
A_n	Nominal heat transfer area, m^2/m (ft^2/ft)
b	Fin height, m (ft), or constant in Eq. (6-32)
B	Constant defined in Eq. (6-3)
Cp	Specific heat of water, W.hr/Kg ^oC (Btu/lbm ^oF)
Cu	Copper
d	Coefficient defined in equation (6-32)
D	Diameter, m (ft)
D_c	Core diameter, m (ft)
D_j	Water jacket diameter, m (ft)
e	Coefficient defined in Eq. (6-31)
f	Friction factor or coefficient defined in Eq. (6-31)
f_d	Darcy's friction factor
f_f	Fanning's friction factor
F_1	Factor defined in equation (5-8)
F_2	Factor defined in equation (5-9)
F_3	Factor defined in equation (5-10)
F_4	Factor defined in equation (5-21)
g	Coefficient defined in Eq. (6-31)
g_c	Gravitational constant
G	Mass flux, Kg/hr m^2 (lbm/hr ft^2)

Gr	Grashof's number = $g_c \beta \Delta T \rho^2 D^3 / \mu^2$
Gz	Graetz number = m Cp / k L
G^*	Mass flux ratio defined in equation (6-21)
h	Enthalpy of liquid R-113, J/Kg (Btu/lbm)
h_i	Inside heat transfer coefficient, W/m^2 ^{o}C (Btu/hr ft^2 ^{o}F)
h_o	Outside heat transfer coefficient, W/m^2 ^{o}C (Btu/hr ft^2 ^{o}F)
k	Thermal conductivity (Copper if no subscript), W/m ^{o}C (Btu/hr ft ^{o}F)
ℓ	Coefficient defined in Eq. (6-32)
L	Length, m (ft)
LMTD	Log-mean temperature difference, ^{o}C (^{o}F)
m	Mass flow rate, kg/hr (lbm/hr), or coefficient defined in Eq. (6-32)
n	Number of fins
Nu	Nusselt number
Δ p	Pressure drop Pa (psi)
P	Pumping power. W (Btu/hr)
$\mathbb{P}$	Pitch of fins (length per turn), m/360^o (ft/360^o)
Pr	Prandtl number
Q	Heat gain/lost, W (Btu/hr), or Volume flow rate, m^3/hr (ft^3/hr)
r	Coefficient defined in Eq. (6-17)
t	Fin thickness, m (ft)
T	Temperature, ^{o}C (^{o}F)
$\bar{T}_w$	Wall temperature ^{o}C (^{o}F)
UA	Overall heat transfer conductance, W/^{o}C (Btu/hr ^{o}F)
V	Velocity. m/s (ft/s)
w	Uncertainty
W	Channel width between internal fins, m (ft)

Greek Letters

α	Spiral tube helix angle
μ	Viscosity, N s/m^2 (lbm/ft hr)
ρ	Density, kg/m^3 (lbm/ft^3)
ε	Height of knurls, m (ft)
β	Coefficient of volume expansion, per ^{o}C (per ^{o}F)

Subscripts

a, aug.	Augmented
h	Based on hydraulic diameter
i	Inside or inlet
o	Outside or outlet
R	Refrigerant R-113
s	Smooth
w	Wall
wa	Water
1-5	Stations 1-5
I-IV	Sections I-IV

APPENDIX A

APPENDIX A

ADDITIONAL INFORMATION ON THE INSTRUMENTATION AND COMPONENTS USED IN THIS STUDY

1. R-113 FLOW CIRCUIT

A. Components

1. Refrigerant R-113 Liquid Circulating Gear Pump:

Dayton Electric Manufacturing Company

Bronze Rotary Gear Pump

Model: 1P777

R.P.M: 1725

Pipe Size: 1/4"

Shaft Diameter: 1/2"

GPM: 3.8 (Free flow)

H.P.: 1/4 (Free flow)

Dripless Mechanical Shaft Seal, Self Lubricated.

2. Refrigerant-113 Liquid Circulating Pump Motor:

Dayton - Electric A.C. Motor

Model No.: 5K991

R.P.M.: 1725

H.P.: 1/2

Hz : 60

3. Refrigerant-113 Filter:

Parker - Liquid Line Filter Dryer

Model No.: 164

Design Pressure: 500 psig

4. Refrigerant-113 Liquid Receiver:

Midland-Ross Refrigerant Type Circular Tank

Serial No.: 2193

Size: 3.5 gallons

Working Pressure: Maximum Allowable Working Pressure

400 psi at 650°F

5. Refrigerant-113 Valves:

Diaphram Packless Line Valves

Superior Brand, Solder to Solder Type

A. Model No.: 214-4S (1/4")

B. Model No.: 216-10S (5/8")

6. Refrigerant-113 After-cooler:

Dunham-Bush Bundle Type Condenser

Model No.: C1C-200-66-L

7. Refrigerant-113 Tube Connectors:

Standard Copper Tube

Sweat Fitting Type

8. Vibration Eliminators:

Anacond Vibration Eliminators Supplied By RECO

Specification: "Has Fatigue-Resistance Corrugated in Bronze Seamless Tubing Core with Bronze Braid Seamless Tubing Core with Bronze Braid Covering. Standard Copper Tube Fittings are Welded on Both Ends."

9. Heater, Locally Constructed:

Material:

Copper Tubing: 5/8" (nominal diameter)

Length 6 ft

Heating element: Ribbon type chromel of 0.204 ohms/ft

Teflon Tape: Temp-r-glas type A2005

Epoxy A-68 and B-68 Types

The Teflon tape was wrapped around the copper tube with one thickness. The heating element was wound uniformly around the tape with 1/4" distance pitch. The epoxy was applied to secure the heating element. Teflon tape was wrapped over the epoxy after it was dry.

10. Thermocouples:

Copper-Constantan Thermocouples of type B&S 24 gage.

B. Instrumentation

1. Refrigerant-113 Flow meters:

A. Fischer-Porter Variable Area Flow Meter

Range: 0 - 0.35 GPM Liquid

Model: 10A3565S

Serial No.: 7207A4733A2

Tube No.: FP-1/2-27-G-10/55

B. Fischer-Porter Variable Area Type Flow Meter

Range: 0 - 0.50 GPM Liquid

Model: 10A3565S

Serial No.: 7207A4733A1

Tube No.: FP-1/2-17-G-10/55

3. Refrigerant-113 Pressure Gauge:

Heise Pressure Gauge of Type H28832

Range: 0 - 200 psig

4. Digital Multimeter:

    Model: 168 Autoranging DMM

    Keithley Instruments Inc.

5. Voltage Regulator:

    Superior Electric Co.

    Powerstat Variable Autotransformer

    Input: 240 V, 60 Hz

    Output: 0 - 280 V, 28 A, 7.8 KW

6. A.C. Ampere Meter:

    Daystrom, Incorporated Weston Instruments Div.

    Weston Instruments, Inc.

    New York, New Jersey

    Model: 433 No. 164330

7. A.C. Volt Meter:

    Daystrom, Incorporated Weston Instruments Div.

    Weston Instruments, Inc.

    New York, New Jersey

    Model: 433 No. 146652

8. Data Acquisition System:

    Esterline Angus an Esterline Company

    Model: PD-2064

    Type: Key Programmable

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

Accuracy:

<u>With Ambient Temperature at 77^{o}F ∓ 9^{o}F</u>

∓ 0.01% of reading, ∓ 0.015% full scale, ∓ 1 count on 4000 mV range;

∓ 0.01% of reading, ∓ 0.030% full scale, ∓ 1 count on 400 mV range;

∓ 0.01% of reading, ∓ 0.040% full scale, ∓ 1 count on 40 mV range;

<u>Over Full Operation Ambient Temperature Range</u>

∓ 0.5 micro-volt per ^{o}C, ∓ 0.01% of reading, ∓.04% full scale, ∓ 1 count on all ranges.

9. Mercury Manometer:

Meriam Instrument Co.

Type: W

Model: 30EC10

Serial No.: B23131

Range: 40"

Manometer with mercury as the indicating fluid

10. Water Manometer:

Meriam Instrument Co.

Model: 22AA25

Range: 50"

Manometer with water as the indicating fluid

11. Differential Pressure Cell:

    Foxboro Type 13A

    Range: -0.034 - 0.184 bar (-0.5 - 2.67 psia)

    D/P cell connections, locally constructed: Copper tubing of 1/4" diameter was used to connect the pressure tap and the D/P cell

    Calibration: The differential pressure cell was calibrated according to the manufacturer's recommended calibration procedure before it was connected to the test condenser. A linear least squares regression correlation was used to obtain a calibration curve. The calibration curve is shown in Fig. A.1.

12. Vaccum Pump:

    Matheson Scientific

    Division of Will Ross, Inc.

    Serial No.: 1173

    Power: 115 V, 60 Hz

    Connections: 3 conductor power cord with 2-prong adaptor. Inlet and outlet of connector to 3/8" diameter hose.

    Function: Portable A.C. powered source of vaccum (to 686mm/27" Hg) or pressure (to 1.7 kg/cm^2, 25 psig).

All thermophysical and transport properties of R-113 were obtained from ASHRAE Handook of Fundamentals [32] and ASHRAE Thermodynamic Properties of Refrigerants.

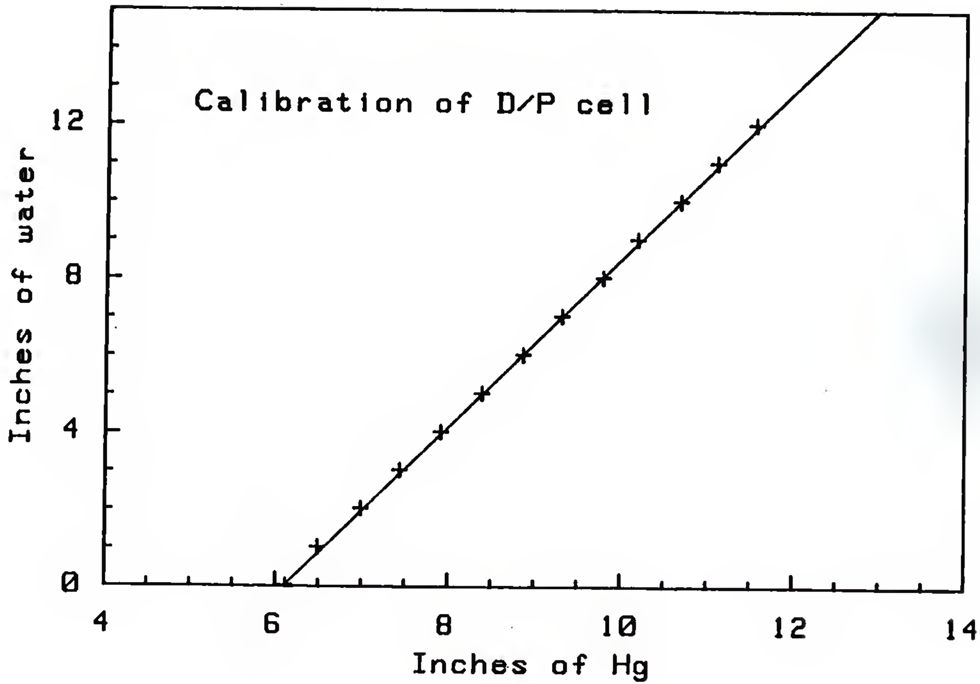

Fig. A.1 D/P Cell calibration curve.

2. WATER FLOW CIRCUIT

1. Cooling Water Flow Meters:

   Brooks Rotameter

   Type: 1110-09H3A1B and 12-1110-10-MF

   Serial No.: 7201-74650/1 and 6402-60732

   Tube No.: R-9M-25-1 BR-3/4-14G10 and R-12M-25-5

   Range: 0 - 3 GPM and 0 - 21 GPM

2. Cooling Water Pump:

   A.O.Smith Co. Pump

   Model No.: C48L2DA11A4

   Serial No.: J69

   H.P.: 1

   R.P.M.: 3450

   Hz: 60

3. Cooling water filter:

   Dayton Electric Manufacturer's

   Model No.: 1P635A

   Replacement Cartridge No.: 1P754

APPENDIX B

APPENDIX B

COMPUTER PROGRAM USED IN DATA REDUCTION

```
100  PI = 3.14159
110 OPEN "O",#1,"scrn:"
120 WIDTH #1,139
130 '**********************************************************************
140 '                              INPUT CONSTANTS
150 'The firet data card contains the outsids and ineide eurface typee.
160 'The optiona available for outside surface are :
170 '                  a) Seooth        and      b) Knurled
180 'The options available for insids surface ars :
190 '                  a) Seooth        and      b) Finned
200 DATA Finned,Knurled
210 READ INSIDE.SURFACEs,OUTSIDE.SURFACEs
220 '**********************************************************************
230 'The eecond data card contains inforeation regarding tube dimeneione.
240 'Ineide diameter, outside dieeater  and noeinal diameter of the tube, water
250 'jacket dieeetar and thermal conductivity of watar ie in thie data statement
260 'All diametars are in inchse. Thereal conductivity is in Btu/hr ft F.
270 DATA 0.494.0.618,0.625,1.500,220.0
280 READ INSIDE.DIAMETER,OUTSIDE.DIAMETER,NOMINAL.DIAMETER,WATER.JACKET.DIAMETER
.THERMAL.CONDUCTIVITY.OF.COPPER
290 '**********************************************************************
300 'Ths third data stateeent contains ths lengths of each section in inchee
310 DATA 27.4375,27.5000,27.3750,27.4375
320 READ LENGTH.OF.SECTION.1,LENGTH.OF.SECTION.2,LENGTH.OF.SECTION.3,LENGTH.OF.S
ECTION.4
330 '**********************************************************************
340 'The fourth data stateeent contains the alope and intercept of the D/P cell
350 'calibration curve.
360 '    Preseure drop("water) = intercept + elope * (D/P call reading in "Hg).
370 DATA 2.223451,-13.724609
380 READ SLOPE.INTERCEPT
390 '**********************************************************************
400 'Ths fifth data card givee ths number of data points.
410 DATA 10
420 READ NUMBER.OF.DATA.POINTS
430 '**********************************************************************
440 '                              INPUT DATA
450 '**********************************************************************
460 'Data for each run is written in the following format:
470 'Thers ars four data cards for each run.
480 'a) The firet data card contains the run number and inlet preesurs (psia).
490 'b) Ths second data card contains freon and water flow rates in gpe.
500 'c) Ths third data card contains temperatures in C. Referance channel
510 '   reading is also included.
520 'd) The fourth data card contains the euperheater ampa and volts.
530 '************************ RUN NUMBER 3001 ******************************
540 DATA 3001,15.486
550 DATA 0.275,2.50
560 DATA 34.2,39.4,15.9,19.8,21.5,19.9,19.2,16.1,15.6,17.4,17.2.16.3,16.8,15.4,1
5.5.16.3.16.0.15.2,15.6,15.2,15.3,15.9,15.8,15.9,15.3,14.9,16.4,19.4
570 DATA 7.6,74.0
580 '************************ RUN NUMBER 3002 ******************************
590 DATA 3002,15.486
600 DATA 0.300,2.50
610 DATA 34.2,39.9,16.0,20.2,21.9,20.2,19.6,16.2,15.6,17.6,17.4,16.5,17.0,15.3,1
5.5,16.5,16.2,15.2,15.7,15.3,15.3,15.9,15.9,15.9,15.3,14.9,16.5,19.1
```

```
620 DATA 8.0,78.0
630 '************************* RUN NUMBER 3003 ******************************
640 DATA 3003,15.486
650 DATA 0.325,2.5
660 DATA 34.2,39.8,16.2,20.4,22.2,20.4,19.8,16.2,15.7,17.9,17.7,16.7,17.1,15.5,1
5.6,16.7,16.3,15.3,15.9,15.3,15.4,16.0,16.0,16.0,15.5,15.0,16.8,18.8
670 DATA 8.3,81.0
680 '************************* RUN NUMBER 3004 ******************************
690 DATA 3004,15.486
700 DATA 0.350,2.5
710 DATA 34.1,40.0,16.1,20.7,22.5,20.6,20.0,16.2,15.6,18.1,17.9,16.8,17.4,15.3,1
5.5,16.7,16.3,15.2,15.9,15.2,15.2,16.1,16.0,15.9,15.4,14.9,16.8,18.5
720 DATA 8.5,88.0
730 '************************* RUN NUMBER 3005 ******************************
740 DATA 3005,15.486
750 DATA 0.375,2.5
760 DATA 34.1,39.9,16.2,20.9,22.8,20.8,20.3,16.3,15.7,18.3,18.1,17.0,17.5,15.4,1
5.6,16.9,16.5,15.3,16.0,15.2,15.3,16.2,16.1,16.0,15.6,14.9,17.1,18.4
770 DATA 9.1,88.0
780 '************************* RUN NUMBER 3006 ******************************
790 DATA 3006,15.486
800 DATA 0.400,2.5
810 DATA 34.1,40.2,16.3,21.2,23.3,21.0,20.6,16.3,15.8,18.6,18.3,17.1,17.7,15.5,1
5.7,17.0,16.6,15.3,16.1,15.3,15.4,16.3,16.2,16.1,15.6,14.9,17.2,18.2
820 DATA 9.3,90.0
830 '************************* RUN NUMBER 3007 ******************************
840 DATA 3007,15.486
850 DATA 0.425,2.5
860 DATA 34.1,40.2,16.5,21.4,23.4,21.0,20.7,16.4,15.9,18.7,18.4,17.2,17.8,15.5,1
5.7,17.1,16.7,15.4,16.2,15.3,15.5,16.5,16.3,16.2,15.7,15.1,17.3,18.2
870 DATA 9.6,93.0
880 '************************* RUN NUMBER 3008 ******************************
890 DATA 3008,15.486
900 DATA 0.450,2.5
910 DATA 34.1,40.0,16.5,21.4,23.5,21.3,20.9,16.4,15.9,18.8,18.5,17.3,17.9,15.6,1
5.7,17.1,16.8.15.4,16.3,15.3,15.4,16.6,16.4,16.2,15.7,15.0,17.5,17.9
920 DATA 9.9,95.0
930 '************************* RUN NUMBER 3009 ******************************
940 DATA 3009,15.486
950 DATA 0.475,2.5
960 DATA 34.1,40.2,16.6,21.8,23.9,21.5,21.2,16.5,16.0,19.0,18.8,17.5,18.2,15.6,1
5.7,17.3,17.0,15.4,16.4,15.4,15.4,16.6,16.5,16.3,15.8,15.0,17.7,17.9
970 DATA 10.2,99.0
980 '************************* RUN NUMBER 3010 ******************************
990 DATA 3010,15.486
1000 DATA 0.500,2.5
1010 DATA 34.0,40.2,16.6,21.9,24.1,21.7,21.4,16.5,16.0,19.2,18.9,17.6,18.3,15.5,
15.7,17.4,16.9,15.4,16.5,15.3,15.3,16.7,16.5,16.3,15.7,14.9,17.8,17.6
1020 DATA 10.4,102.0
1030 DUMMY.RUN.NUMBER = 10000
1040 INPUT "Do you want to store the results in a file (n or y)";STORE.OR.NOT$
1050 INPUT "Do you want data reduction for one data (y or n)";ONE.DATA$
1060 IF ONE.DATA$="n" OR ONE.DATA$="N" THEN 1080
1070 INPUT "WHICH RUN NUMBER";DUMMY.RUN.NUMBER
```

```
1080 '************************************************************************
1090 '                               READ DATA
1100 FOR COUNT= 1 TO NUMBER.OF.DATA.POINTS
1110 READ RUN.NUMBER,INLET.PRESSURE     ' Inlet pressure is in psia.
1120 READ R113.GPM.WATER.GPM             ' In gpm
1130 READ T0,T1,T2,T3,T4,T5,T6,T7,T8,T9,T10,T11,T12,T13,T14,T15,T16,T17,T18,T19,
T20,T21,T22,T23,T24,T25,T26,T27          ' In degrees C
1140 READ SUPERHEAT.AMPS,SUPERHEAT.VOLTS
1150 IF RUN.NUMBER>DUMMY.RUN.NUMBER THEN 2510
1160 PRINT RUN.NUMBER
1170 IF ONE.DATA$="n" OR ONE.DATA$="N" THEN 1200
1180 IF RUN.NUMBER = DUMMY.RUN.NUMBER THEN 1200
1190 GOTO 2500
1200 '************************************************************************
1210 '                         AREA CALCULATIONS (sq ft)
1220 TOTAL.LENGTH.OF.THE.SECTIONS=LENGTH.OF.SECTION.1+LENGTH.OF.SECTION.2+LENGTH
.OF.SECTION.3+LENGTH.OF.SECTION.4
1230 TOTAL.INSIDE.SURFACE.AREA=PI*TOTAL.LENGTH.OF.THE.SECTIONS*INSIDE.DIAMETER/1
44!
1240 TOTAL.OUTSIDE.SURFACE.AREA=PI*TOTAL.LENGTH.OF.THE.SECTIONS*OUTSIDE.DIAMETER
/144!
1250 INSIDE.FLOW.AREA = PI*INSIDE.DIAMETER^2/576!
1260 OUTSIDE.FLOW.AREA = PI*(WATER.JACKET.DIAMETER^2-OUTSIDE.DIAMETER^2)/576!
1270 HYDRAULIC.DIAMETER = WATER.JACKET.DIAMETER-OUTSIDE.DIAMETER
1280 '************************************************************************
1290 '                         CORRECT THE TEMPERATURES (C)
1300 CORRECTION.FACTOR = T0 - 32.9
1310 T1 = T1+.8-CORRECTION.FACTOR
1320 T2 = T2+.9-CORRECTION.FACTOR
1330 T3 = T3+.8-CORRECTION.FACTOR
1340 T4 = T4+.7-CORRECTION.FACTOR
1350 T5 = T5+.4-CORRECTION.FACTOR
1360 T6 = T6+.1-CORRECTION.FACTOR
1370 T7 = T7-.1-CORRECTION.FACTOR
1380 T8 = T8+.7-CORRECTION.FACTOR
1390 T9 = T9+.8-CORRECTION.FACTOR
1400 T10 = T10+.7-CORRECTION.FACTOR
1410 T11 = T11+.8-CORRECTION.FACTOR
1420 T12 = T12+.5-CORRECTION.FACTOR
1430 T13 = T13+.5-CORRECTION.FACTOR
1440 T14 = T14+.3-CORRECTION.FACTOR
1450 T15 = T15+0!-CORRECTION.FACTOR
1460 T16 = T16+.6-CORRECTION.FACTOR
1470 T17 = T17+.7-CORRECTION.FACTOR
1480 T18 = T18+.8-CORRECTION.FACTOR
1490 T19 = T19+.6-CORRECTION.FACTOR
1500 T20 = T20+.6-CORRECTION.FACTOR
1510 T21 = T21+.5-CORRECTION.FACTOR
1520 T22 = T22+.1-CORRECTION.FACTOR
1530 T23 = T23+0!-CORRECTION.FACTOR
1540 T24 = T24+.9-CORRECTION.FACTOR
1550 T25 = T25-1!-CORRECTION.FACTOR
1560 T26 = T26+.7-CORRECTION.FACTOR
1570 T27 = T27+.4-CORRECTION.FACTOR
1580 '************************************************************************
1590 '                         WATER TEMPERATURES (C)
1600 WATER.TEMPERATURE.AT.STATION.1 = T2
1610 WATER.TEMPERATURE.AT.STATION.5 = T25
```

```
1620 '***********************************************************************
1630 '                           HEAT INPUT (Btu/hr)
1640 SUPERHEATER.HEAT = SUPERHEAT.AMPS*SUPERHEAT.VOLTS
1650 TOTAL.HEAT.INPUT = SUPERHEATER.HEAT*3.413
1660 '***********************************************************************
1670 '                     HEAT GAINED BY WATER  (Btu/hr)
1680 MASS.FLOW.RATE.OF.WATER = WATER.GPM*60!*.13368*62.3          ' lbm/hr
1690 WATER.MASS.FLUX = MASS.FLOW.RATE.OF.WATER/OUTSIDE.FLOW.AREA ' lbm/hr ft^2
1700 HEAT.GAIN.IN.SECTION.1=MASS.FLOW.RATE.OF.WATER*(WATER.TEMPERATURE.AT.STATIO
N.1-WATER.TEMPERATURE.AT.STATION.2)*1.8
1710 HEAT.GAIN.IN.SECTION.2=MASS.FLOW.RATE.OF.WATER*(WATER.TEMPERATURE.AT.STATIO
N.2-WATER.TEMPERATURE.AT.STATION.3)*1.8
1720 HEAT.GAIN.IN.SECTION.3=MASS.FLOW.RATE.OF.WATER*(WATER.TEMPERATURE.AT.STATIO
N.3-WATER.TEMPERATURE.AT.STATION.4)*1.8
1730 HEAT.GAIN.IN.SECTION.4=MASS.FLOW.RATE.OF.WATER*(WATER.TEMPERATURE.AT.STATIO
N.4-WATER.TEMPERATURE.AT.STATION.5)*1.8
1740 TOTAL.HEAT.GAIN=HEAT.GAIN.IN.SECTION.1+HEAT.GAIN.IN.SECTION.2+HEAT.GAIN.IN.
SECTION.3+HEAT.GAIN.IN.SECTION.4
1750 HEAT.FLUX = TOTAL.HEAT.GAIN/TOTAL.INSIDE.SURFACE.AREA
1760 '***********************************************************************
1770 '                        HEAT LOST BY R113 (Btu/hr)
1780 TEMPERATURE.IN.DEG.F = T27*1.8+32!
1790 GOSUB 2520                                 ' Find liquid density
1800 MASS.FLOW.RATE.OF.R113 = R113.GPM*LIQUID.DENSITY*60!*.13368  ' lbm/hr
1810 R113.MASS.FLUX = MASS.FLOW.RATE.OF.R113/INSIDE.FLOW.AREA     ' lbm/hr ft^2
1820 TEMPERATURE.IN.DEG.F = T1*1.8+32!
1830 GOSUB 2940                                 ' Find liquid enthalpy
1840 ENTHALPY.AT.STATION.1 = ENTHALPY.OF.LIQUID
1850 TEMPERATURE.IN.DEG.F = T26*1.8+32!
1860 GOSUB 2940                                 ' Find liquid enthalpy
1870 ENTHALPY.AT.STATION.5 = ENTHALPY.OF.LIQUID
1880 TOTAL.HEAT.LOST = MASS.FLOW.RATE.OF.R113*(ENTHALPY.AT.STATION.1-ENTHALPY.AT
.STATION.5)
1890 '***********************************************************************
1900 '                           HEAT BALANCE
1910 HEAT.BALANCE.ERROR = (TOTAL.HEAT.GAIN-TOTAL.HEAT.LOST)/TOTAL.HEAT.GAIN*100!
1920 TEMPERATURE.IN.DEG.F = T27*1.8+32!
1930 GOSUB 2940                                 ' Find liquid enthalpy
1940 HEAT.GAIN.IN.BOILER = MASS.FLOW.RATE.OF.R113*(ENTHALPY.AT.STATION.1-ENTHALP
Y.OF.LIQUID)
1950 HEAT.BALANCE.FOR.THE.BOILER = (HEAT.GAIN.IN.BOILER-HEAT.INPUT)/HEAT.GAIN.IN
.BOILER*100!
1960 '***********************************************************************
1970 '                TEMPERATURES AT STATIONS 1 AND 5 (C)
1980 TEMPERATURE.AT.STATION.1 = T1
1990 TEMPERATURE.AT.STATION.5 = T26
2000 '***********************************************************************
2010 '                 AVERAGE WALL TEMPERATURES  (C)
2020 AVERAGE.WALL.TEMPERATURE.IN.SECTION.1 = (T3+T4+T5+T6)/4!
2030 AVERAGE.WALL.TEMPERATURE.IN.SECTION.2 = (T9+T10+T11+T12)/4!
2040 AVERAGE.WALL.TEMPERATURE.IN.SECTION.3 = (T15+T16+T17+T18)/4!
2050 AVERAGE.WALL.TEMPERATURE.IN.SECTION.4 = (T21+T22+T23+T24)/4!
2060 AVERAGE.WALL.TEMPERATURE.IN.TUBE = (AVERAGE.WALL.TEMPERATURE.IN.SECTION.1+A
VERAGE.WALL.TEMPERATURE.IN.SECTION.2+AVERAGE.WALL.TEMPERATURE.IN.SECTION.3+AVERA
GE.WALL.TEMPERATURE.IN.SECTION.4)/4!
2070 '***********************************************************************
```

```
2080 '                  LMTD -BETWEEN WALL AND WATER  (C)
2090 TEMPERATURE.DIFF.1 = AVERAGE.WALL.TEMPERATURE.IN.TUBE-WATER.TEMPERATURE.AT.
STATION.1
2100 TEMPERATURE.DIFF.2 = AVERAGE.WALL.TEMPERATURE.IN.TUBE-WATER.TEMPERATURE.AT.
STATION.5
2110 WATER.LMTD.TUBE = (TEMPERATURE.DIFF.1-TEMPERATURE.DIFF.2)/LOG(TEMPERATURE.D
IFF.1/TEMPERATURE.DIFF.2)
2120 '**********************************************************************
2130 '                                    LMTD   (C)
2140 TEMPERATURE.DIFF.1=TEMPERATURE.AT.STATION.1-WATER.TEMPERATURE.AT.STATION.1
2150 TEMPERATURE.DIFF.2=TEMPERATURE.AT.STATION.5-WATER.TEMPERATURE.AT.STATION.5
2160 LMTD.TUBE=(TEMPERATURE.DIFF.1-TEMPERATURE.DIFF.2)/LOG(TEMPERATURE.DIFF.1/TE
MPERATURE.DIFF.2)
2170 '**********************************************************************
2180 '          OUTSIDE HEAT TRANSFER COEFFICIENTS  - BASED ON LMTD
2190 OUTSIDE.HTC.TUBE = TOTAL.HEAT.GAIN/(TOTAL.OUTSIDE.SURFACE.AREA*WATER.LMTD.T
UBE*1.8)
2200 '**********************************************************************
2210 '                               U TIMES A
2220 U.TIMES.A.TUBE = TOTAL.HEAT.GAIN/LMTD.TUBE/1.8
2230 '**********************************************************************
2240 '         INSIDE HEAT TRANSFER COEFFICIENT FROM EQUATION
2250 WALL.RESISTANCE = OUTSIDE.DIAMETER*LOG(OUTSIDE.DIAMETER/INSIDE.DIAMETER)/(T
HERMAL.CONDUCTIVITY.OF.COPPER*12!)/2!
2260 INSIDE.HTC.TUBE = (OUTSIDE.DIAMETER/INSIDE.DIAMETER)/(TOTAL.OUTSIDE.SURFACE
.AREA/U.TIMES.A.TUBE - WALL.RESISTANCE - 1!/OUTSIDE.HTC.TUBE)
2270 '**********************************************************************
2280 '                         PROPERTIES OF R113
2290 TEMPERATURE.IN.DEG.F = TEMPERATURE.IN.TUBE*1.8 + 32!
2300 GOSUB 3020
2310 PRANDTL.NUMBER.OF.R113.IN.TUBE = PRANDTL.NUMBER.OF.LIQUID.R113
2320 NUSSELT.NUMBER.OF.R113.IN.TUBE = INSIDE.HTC.TUBE*INSIDE.DIAMETER/THERMAL.CO
NDUCTIVITY.OF.LIQUID.R113/12!
2330 REYNOLDS.NUMBER.OF.R113.IN.TUBE = REYNOLDS.NUMBER.OF.LIQUID.R113
2340 '**********************************************************************
2350 '                         PROPERTIES OF WATER
2360 TEMPERATURE.IN.DEG.C = (WATER.TEMPERATURE.AT.STATION.1+WATER.TEMPERATURE.AT
.STATION.5)/2!
2370 GOSUB 3100                                      ' Properties of water
2380 VISCOSITY.OF.WATER.IN.TUBE = VISCOSITY.OF.WATER
2390 THERMAL.CONDUCTIVITY.OF.WATER.TUBE = THERMAL.CONDUCTIVITY.OF.WATER
2400 PRANDTL.NUMBER.OF.WATER.IN.TUBE = PRANDTL.NUMBER.OF.WATER
2410 '**********************************************************************
2420 '                     DIMENSIONLESS NUMBERS FOR WATER
2430 WATER.REYNOLDS.NUMBER.IN.TUBE = WATER.MASS.FLUX*HYDRAULIC.DIAMETER/12!/VISC
OSITY.OF.WATER.IN.TUBE
2440 WATER.NUSSELT.NUMBER.IN.TUBE = OUTSIDE.HTC.TUBE*HYDRAULIC.DIAMETER/12!/THER
MAL.CONDUCTIVITY.OF.WATER.TUBE
2450 '**********************************************************************
2460 IF STORE.OR.NOT$="y" OR STORE.OR.NOT$="Y" THEN 2480
2470 GOTO 2500
2480 IF COUNT = 1 THEN GOSUB 3160
2490 GOSUB 3340
2500 NEXT COUNT
2510 END
2520 '**********************************************************************
```

```
2530 '                         LIQUID DENSITY
2540 TEMPERATURE.IN.DEG.R = TEMPERATURE.IN.DEG.F+459.6
2550 LIQUID.DENSITY = 122.872 - .0128*TEMPERATURE.IN.DEG.R - .0000636*TEMPERATUR
E.IN.DEG.R^2
2560 RETURN
2570 '**************************************************************************
2580 '                   ENTHALPY OF LIQUID (Btu/lbm)
2590 ENTHALPY.OF.LIQUID  = 7.2706307692#+.217211538462#*TEMPERATURE.IN.DEG.F
2600 RETURN
2610 '**************************************************************************
2620 '              TEMPERATURE CORRESPONDING TO AN ENTHALPY (C)
2630 TEMPERATURE.IN.DEG.C = ((ENTHALPY.OF.LIQUID - 7.2706307692#)/.21721158462#
- 32!)/1.8
2640 RETURN
2650 '**************************************************************************
2660 '                 PHYSICAL PROPERTIES OF LIQUID R-113
2670 THERMAL.CONDUCTIVITY.OF.LIQUID.R113 = .0484566666666667# - 6.575E-05*TEMPERA
TURE.IN.DEG.F
2680 VISCOSITY.OF.LIQUID.R113 = 3.085134927# - .02608903935#*TEMPERATURE.IN.DEG.
F + .0001105465381#*TEMPERATURE.IN.DEG.F^2 - .0000001886574111#*TEMPERATURE.IN.D
EG.F^3
2690 SPECIFIC.HEAT.OF.LIQUID.R113 = .210960316# + .00034037402#*TEMPERATURE.IN.D
EG.F - .0000018262991#*TEMPERATURE.IN.DEG.F^2 + .0000000052609428#*TEMPERATURE.I
N.DEG.F^3
2700 PRANDTL.NUMBER.OF.LIQUID.R113 = VISCOSITY.OF.LIQUID.R113*SPECIFIC.HEAT.OF.L
IQUID.R113/THERMAL.CONDUCTIVITY.OF.LIQUID.R113
2710 REYNOLDS.NUMBER.OF.LIQUID.R113 = R113.MASS.FLUX*INSIDE.DIAMETER/VISCOSITY.O
F.LIQUID.R113/12!
2720 RETURN
2730 '**************************************************************************
2740 '                   PHYSICAL PROPERTIES OF WATER
2750 VISCOSITY.OF.WATER = (605.4727133# - 15.54082397#*TEMPERATURE.IN.DEG.C + .1
935339483#*TEMPERATURE.IN.DEG.C^2-.000925792914#*TEMPERATURE.IN.DEG.C^3)/100!*.6
719
2760 THERMAL.CONDUCTIVITY.OF.WATER = .005778*(55.049708336#+.260982142#*TEMPERAT
URE.IN.DEG.C-.001376726189#*TEMPERATURE.IN.DEG.C^2)
2770 PRANDTL.NUMBER.OF.WATER = 13.5806060632#-.4569841276#*TEMPERATURE.IN.DEG.C+
.00734772793#*TEMPERATURE.IN.DEG.C^2-.00004406565677#*TEMPERATURE.IN.DEG.C^3
2780 RETURN
2790 '**************************************************************************
2800 '                  PRINTING TUBE DATA ONTO A FILE
2810 INPUT "What is the tube number (1 thru 9)";TUBE.NUMBER$
2820 IF (TUBE.NUMBER$="1") OR (TUBE.NUMBER$="2") OR (TUBE.NUMBER$="3") OR (TUBE.
NUMBER$="4") OR (TUBE.NUMBER$="5") OR (TUBE.NUMBER$="6") OR (TUBE.NUMBER$="7") O
R (TUBE.NUMBER$="8") OR (TUBE.NUMBER$="9") THEN 2830  ELSE 2810
2830 INPUT "Which set of readings (0 to 4)";PRESSURE.NUMBER$
2840 IF PRESSURE.NUMBER$="0" OR PRESSURE.NUMBER$="1" OR PRESSURE.NUMBER$="2" OR
PRESSURE.NUMBER$="3" OR PRESSURE.NUMBER$="4" THEN 2850 ELSE 2830
2850 INPUT "Disc where this data is to be stored (A or B)";DISC$
2860 IF DISC$="a" OR DISC$="A" OR DISC$="b" OR DISC$="B" THEN 2870  ELSE 2850
2870 DATA.FILE$=DISC$+":"+"TUBE"+TUBE.NUMBER$+PRESSURE.NUMBER$+"XX"+".spd"
2880 OPEN "O",#2,DATA.FILE$
2890 PRINT #2, INSIDE.SURFACE$
2900 PRINT #2, OUTSIDE.SURFACE$
```

```
2910 PRINT #2, INSIDE.SURFACE.AREA.SECTION.1,INSIDE.SURFACE.AREA.SECTION.2,INSID
E.SURFACE.AREA.SECTION.3,INSIDE.SURFACE.AREA.SECTION.4,TOTAL.INSIDE.SURFACE.AREA
2920 PRINT #2, OUTSIDE.SURFACE.AREA.SECTION.1,OUTSIDE.SURFACE.AREA.SECTION.2,OUT
SIDE.SURFACE.AREA.SECTION.3,OUTSIDE.SURFACE.AREA.SECTION.4,TOTAL.OUTSIDE.SURFACE
.AREA
2930 PRINT #2, LENGTH.OF.SECTION.1,LENGTH.OF.SECTION.2,LENGTH.OF.SECTION.3,LENGT
H.OF.SECTION.4,WATER.JACKET.DIAMETER,THERMAL.CONDUCTIVITY.OF.COPPER
2940 PRINT #2, INSIDE.DIAMETER,OUTSIDE.DIAMETER,INSIDE.FLOW.AREA,OUTSIDE.FLOW.AR
EA
2950 PRINT #2, NUMBER.OF.DATA.POINTS
2960 RETURN
2970 '**************************************************************************
2980 '                    PRINTING REDUCED DATA ONTO THE FILE
2990 PRINT #2, RUN.NUMBER
3000 PRINT #2, T1,T2,T3,T4,T5,T6,T7,T8,T9,T10,T11,T12,T13,T14,T15,T16,T17,T18,T1
9,T20,T21,T22,T23,T24,T25,T26,T27
3010 PRINT #2, WATER.GPM,R113.GPM,MASS.FLOW.RATE.OF.WATER,MASS.FLOW.RATE.OF.R113
,WATER.MASS.FLUX,R113.MASS.FLUX
3020 PRINT #2, INLET,PRESSURE
3030 PRINT #2, SUPERHEAT.AMPS,SUPERHEAT.VOLTS,TOTAL.HEAT.INPUT,TOTAL.HEAT.GAIN,T
OTAL.HEAT.LOST,HEAT.BALANCE.ERROR
3040 PRINT #2, TEMPERATURE.IN.TUBE
3050 PRINT #2, TEMPERATURE.AT.STATION.1,TEMPERATURE.AT.STATION.5
3060 PRINT #2, AVERAGE.WALL.TEMPERATURE.IN.SECTION.1,AVERAGE.WALL.TEMPERATURE.IN
.SECTION.2,AVERAGE.WALL.TEMPERATURE.IN.SECTION.3,AVERAGE.WALL.TEMPERATURE.IN.SEC
TION.4,AVERAGE.WALL.TEMPERATURE.IN.TUBE
3070 PRINT #2, WATER.TEMPERATURE.AT.STATION.1
3080 PRINT #2, LMTD.TUBE
3090 PRINT #2, WATER.LMTD.TUBE
3100 PRINT #2, U.TIMES.A.TUBE
3110 PRINT #2, OUTSIDE.HTC.TUBE
3120 PRINT #2, INSIDE.HTC.TUBE
3130 PRINT #2, ENTHALPY.AT.STATION.1,ENTHALPY.AT.STATION.5
3140 PRINT #2, PRANDTL.NUMBER.OF.WATER.IN.TUBE
3150 PRINT #2, WATER.REYNOLDS.NUMBER.IN.TUBE
3160 PRINT #2, WATER.NUSSELT.NUMBER.IN.TUBE
3170 PRINT #2, PRANDTL.NUMBER.OF.R113.IN.TUBE
3180 PRINT #2, NUSSELT.NUMBER.OF.R113.IN.TUBE
3190 PRINT #2, REYNOLDS.NUMBER.OF.R113.IN.TUBE
3200 RETURN
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APPENDIX C

TABLE C.1 :- REDUCED DATA - FLOW RATES, INLET AND OUTLET TEMPERATURES

Run	Flow rate				Inlet temperature				Exit temperature			
	Water		R-113		Water		R-113		Water		R-113	
#	gpm	cu.m/hr	gpm	cu.m/hr	deg.C	deg.F	deg.C	deg.F	deg.C	deg.F	deg.C	deg.F
1001	2.50	.5678	.275	.0625	14.3	57.7	39.1	102.4	15.2	59.4	15.5	59.9
1002	2.50	.5678	.300	.0681	14.3	57.7	40.3	104.5	15.3	59.5	15.7	60.3
1003	2.50	.5678	.325	.0738	14.3	57.7	40.0	104.0	15.4	59.7	16.0	60.8
1004	2.50	.5678	.350	.0795	14.2	57.6	40.0	104.0	15.4	59.7	16.1	61.0
1005	2.50	.5678	.375	.0852	14.2	57.6	40.1	104.2	15.5	59.9	16.3	61.3
1006	2.50	.5678	.400	.0908	14.2	57.6	40.4	104.7	15.5	59.9	16.4	61.5
1007	2.50	.5678	.425	.0965	14.0	57.2	39.9	103.8	15.4	59.7	16.4	61.5
1008	2.55	.5792	.450	.1022	13.9	57.0	40.3	104.5	15.3	59.5	16.6	61.9
1009	2.50	.5678	.475	.1079	13.9	57.0	40.6	105.1	15.4	59.7	16.8	62.2
1010	2.50	.5678	.500	.1136	13.9	57.0	40.0	104.0	15.5	59.9	17.0	62.6
1101	1.50	.3407	.275	.0625	14.8	58.6	39.6	103.3	16.3	61.3	16.5	61.7
1102	1.50	.3407	.300	.0681	14.8	58.6	39.9	103.8	16.4	61.5	16.8	62.2
1103	1.50	.3407	.325	.0738	14.8	58.6	40.3	104.5	16.6	61.9	17.1	62.8
1104	1.50	.3407	.350	.0795	14.9	58.8	40.1	104.2	16.8	62.2	17.4	63.3
1105	1.50	.3407	.375	.0852	15.0	59.0	40.1	104.2	17.0	62.6	17.8	64.0
1106	1.55	.3520	.400	.0908	15.1	59.2	40.1	104.2	17.0	62.6	17.8	64.0
1107	1.50	.3407	.425	.0965	15.0	59.0	39.8	103.6	17.1	62.8	18.0	64.4
1108	1.50	.3407	.450	.1022	15.0	59.0	40.2	104.4	17.2	63.0	18.3	64.9
1109	1.50	.3407	.475	.1079	15.0	59.0	40.2	104.4	17.3	63.1	18.6	65.5
1110	1.50	.3407	.500	.1136	15.0	59.0	40.2	104.4	17.5	63.5	18.9	66.0
1201	1.50	.3407	.275	.0625	14.6	58.3	36.0	96.8	15.8	60.4	16.1	61.0
1202	1.50	.3407	.300	.0681	14.7	58.5	34.0	93.2	15.9	60.6	16.3	61.3
1203	1.50	.3407	.325	.0738	14.7	58.5	32.6	90.7	15.9	60.6	16.4	61.5
1204	1.50	.3407	.350	.0795	14.6	58.3	31.4	88.5	15.8	60.4	16.4	61.5
1205	1.50	.3407	.375	.0852	14.8	58.6	30.6	87.1	16.0	60.8	16.5	61.7
1206	1.50	.3407	.400	.0908	14.9	58.8	29.7	85.5	16.1	61.0	16.7	62.1
1207	1.50	.3407	.425	.0965	14.8	58.6	29.0	84.2	16.0	60.8	16.7	62.1
1208	1.50	.3407	.450	.1022	15.0	59.0	28.7	83.7	16.2	61.2	17.0	62.6
1209	1.50	.3407	.475	.1079	14.9	58.8	28.1	82.6	16.1	61.0	16.8	62.2
1210	1.50	.3407	.500	.1136	15.0	59.0	27.6	81.7	16.2	61.2	17.0	62.6
1301	1.30	.2953	.500	.1136	14.7	58.5	39.7	103.5	17.4	63.3	18.6	65.5
1302	1.50	.3407	.500	.1136	14.7	58.5	40.0	104.0	17.1	62.8	18.7	65.7
1303	1.60	.3634	.500	.1136	14.7	58.5	40.1	104.2	17.0	62.6	18.5	65.3
1304	1.70	.3861	.500	.1136	14.8	58.6	40.2	104.4	17.0	62.6	18.4	65.1
1305	1.80	.4088	.500	.1136	14.7	58.5	40.5	104.9	16.8	62.2	18.2	64.8
1306	1.90	.4315	.500	.1136	14.5	58.1	40.2	104.4	16.5	61.7	17.9	64.2
1307	2.05	.4656	.500	.1136	14.6	58.3	40.2	104.4	16.4	61.5	17.8	64.0
1308	2.10	.4770	.500	.1136	14.5	58.1	40.3	104.5	16.4	61.5	17.9	64.2
1309	2.20	.4997	.500	.1136	14.5	58.1	40.4	104.7	16.3	61.3	17.9	64.2
1310	2.30	.5224	.500	.1136	14.5	58.1	40.6	105.1	16.3	61.3	17.7	63.9
1401	1.30	.2953	.275	.0625	15.1	59.2	40.6	105.1	16.9	62.4	17.0	62.6
1402	1.40	.3180	.275	.0625	15.1	59.2	40.9	105.6	16.8	62.2	16.9	62.4
1403	1.47	.3339	.275	.0625	14.9	58.8	40.5	104.9	16.5	61.7	16.8	62.2
1404	1.60	.3634	.275	.0625	14.7	58.5	40.5	104.9	16.1	61.0	16.4	61.5
1405	1.70	.3861	.275	.0625	14.6	58.3	40.8	105.4	16.0	60.8	16.3	61.3
1406	1.80	.4088	.275	.0625	14.6	58.3	40.6	105.1	15.9	60.6	16.2	61.2
1407	1.90	.4315	.275	.0625	14.6	58.3	40.7	105.3	15.8	60.4	16.1	61.0
1408	2.00	.4542	.275	.0625	14.6	58.3	40.8	105.4	15.8	60.4	16.1	61.0
1409	2.10	.4770	.275	.0625	14.5	58.1	40.8	105.4	15.6	60.1	[illegible]	[illegible]
1410	2.20	.4997	.275	.0625	14.5	58.1	40.4	104.7	15.5	59.9	16.0	60.8

TABLE C.1 (Contd.)

Run	Flow rate				Inlet temperature				Exit temperature			
	Water		R-113		Water		R-113		Water		R-113	
#	gpm	cu.m/hr	gpm	cu.m/hr	deg.C	deg.F	deg.C	deg.F	deg.C	deg.F	deg.C	deg.F
2001	2.50	.5678	.275	.0625	15.0	59.0	39.4	102.9	15.6	60.1	22.8	73.0
2002	2.50	.5678	.300	.0681	14.8	58.6	40.1	104.2	15.5	59.9	23.0	73.4
2003	2.50	.5678	.325	.0738	14.9	58.8	40.7	105.3	15.7	60.3	23.5	74.3
2004	2.50	.5678	.350	.0795	14.9	58.8	40.0	104.0	15.7	60.3	23.7	74.7
2005	2.45	.5565	.375	.0852	14.8	58.6	40.2	104.4	15.7	60.3	23.8	74.8
2006	2.50	.5678	.400	.0908	14.9	58.8	40.3	104.5	15.8	60.4	24.0	75.2
2007	2.50	.5678	.425	.0965	14.9	58.8	40.3	104.5	15.9	60.6	24.0	75.2
2008	2.50	.5678	.450	.1022	14.9	58.8	39.9	103.8	15.9	60.6	24.0	75.2
2009	2.50	.5678	.475	.1079	14.9	58.8	39.9	103.8	15.9	60.6	24.1	75.4
2010	2.50	.5678	.500	.1136	15.0	59.0	40.4	104.7	16.1	61.0	24.5	76.1
2101	1.50	.3407	.275	.0625	16.0	60.8	40.2	104.4	17.0	62.6	24.1	75.4
2102	1.50	.3407	.300	.0681	16.0	60.8	40.8	105.4	17.2	63.0	24.5	76.1
2103	1.50	.3407	.325	.0738	16.1	61.0	40.8	105.4	17.3	63.1	24.7	76.5
2104	1.40	.3180	.350	.0795	15.9	60.6	40.5	104.9	17.3	63.1	24.9	76.8
2105	1.45	.3293	.375	.0852	15.8	60.4	40.3	104.5	17.2	63.0	24.9	76.8
2106	1.50	.3407	.400	.0908	15.8	60.4	40.4	104.7	17.2	63.0	25.1	77.2
2107	1.55	.3520	.425	.0965	15.6	60.1	40.3	104.5	17.0	62.6	25.0	77.0
2108	1.50	.3407	.450	.1022	15.5	59.9	40.5	104.9	17.1	62.8	25.3	77.5
2109	1.50	.3407	.475	.1079	15.4	59.7	40.5	104.9	17.0	62.6	25.5	77.9
2110	1.50	.3407	.500	.1136	15.5	59.9	40.5	104.9	17.2	63.0	25.6	78.1
2201	1.50	.3407	.275	.0625	16.0	60.8	40.2	104.4	17.0	62.6	24.1	75.4
2202	1.50	.3407	.300	.0681	15.5	59.9	39.9	103.8	16.6	61.9	23.8	74.8
2203	1.50	.3407	.325	.0738	15.4	59.7	37.6	99.7	16.5	61.7	23.6	74.5
2204	1.50	.3407	.350	.0795	15.4	59.7	36.3	97.3	16.4	61.5	23.3	73.9
2205	1.45	.3293	.375	.0852	15.2	59.4	34.9	94.8	16.3	61.3	22.9	73.2
2206	1.55	.3520	.400	.0908	15.2	59.4	34.1	93.4	16.2	61.2	22.5	72.5
2207	1.55	.3520	.425	.0965	15.2	59.4	33.2	91.8	16.2	61.2	22.3	72.1
2208	1.50	.3407	.450	.1022	15.2	59.4	33.3	91.9	16.3	61.3	22.3	72.1
2209	1.45	.3293	.475	.1079	15.2	59.4	32.5	90.5	16.4	61.5	22.2	72.0
2210	1.50	.3407	.500	.1136	15.2	59.4	31.7	89.1	16.3	61.3	21.9	71.4
2301	1.30	.2953	.500	.1136	15.9	60.6	40.6	105.1	17.9	64.2	26.1	79.0
2302	1.40	.3180	.500	.1136	15.8	60.4	40.6	105.1	17.6	63.7	26.0	78.8
2303	1.50	.3407	.500	.1136	15.7	60.3	40.6	105.1	17.4	63.3	25.8	78.4
2304	1.60	.3634	.500	.1136	15.5	59.9	40.6	105.1	17.2	63.0	25.5	77.9
2305	1.70	.3861	.500	.1136	15.4	59.7	40.4	104.7	17.0	62.6	25.3	77.5
2306	1.80	.4088	.500	.1136	15.1	59.2	40.3	104.5	16.6	61.9	25.1	77.2
2307	1.90	.4315	.500	.1136	15.0	59.0	40.4	104.7	16.4	61.5	24.9	76.8
2308	2.00	.4542	.500	.1136	14.9	58.8	40.1	104.2	16.2	61.2	24.6	76.3
2309	2.10	.4770	.500	.1136	14.9	58.8	40.4	104.7	16.2	61.2	24.7	76.5
2310	2.40	.5451	.500	.1136	15.0	59.0	40.3	104.5	16.1	61.0	24.6	76.3
2401	1.30	.2953	.275	.0625	15.6	60.1	40.2	104.4	16.8	62.2	23.9	75.0
2402	1.40	.3180	.275	.0625	15.6	60.1	40.5	104.9	16.7	62.1	23.9	75.0
2403	1.50	.3407	.275	.0625	15.4	59.7	40.5	104.9	16.5	61.7	23.7	74.7
2404	1.60	.3634	.275	.0625	15.4	59.7	40.6	105.1	16.4	61.5	23.5	74.3
2405	1.70	.3861	.275	.0625	15.2	59.4	40.6	105.1	16.2	61.2	23.4	74.1
2406	1.80	.4088	.275	.0625	15.0	59.0	40.5	104.9	15.9	60.6	23.2	73.8
2407	1.90	.4315	.275	.0625	14.9	58.8	40.4	104.7	15.8	60.4	23.1	73.5
2408	2.00	.4542	.275	.0625	14.9	58.8	40.4	104.7	15.7	60.3	23.2	73.8
2409	2.10	.4770	.275	.0625	14.9	58.8	40.4	104.7	15.7	60.3	23.0	73.4
2410	2.30	.5224	.275	.0625	14.8	58.6	40.5	104.9	15.5	59.9	22.9	73.2

TABLE C.1 (Contd.)

Run	Flow rate				Inlet temperature				Exit temperature			
	Water		R-113		Water		R-113		Water		R-113	
#	gpm	cu.m/hr	gpm	cu.m/hr	deg.C	deg.F	deg.C	deg.F	deg.C	deg.F	deg.C	deg.F
3001	2.50	.5678	.275	.0625	14.6	58.3	38.9	102.0	15.5	59.9	15.8	60.4
3002	2.50	.5678	.300	.0681	14.6	58.3	39.4	102.9	15.6	60.1	15.9	60.6
3003	2.50	.5678	.325	.0738	14.7	58.5	39.3	102.7	15.8	60.4	16.2	61.2
3004	2.50	.5678	.350	.0795	14.7	58.5	39.6	103.3	15.8	60.4	16.3	61.3
3005	2.50	.5678	.375	.0852	14.7	58.5	39.5	103.1	15.9	60.6	16.6	61.9
3006	2.50	.5678	.400	.0908	14.7	58.5	39.8	103.6	16.0	60.8	16.7	62.1
3007	2.50	.5678	.425	.0965	14.9	58.8	39.8	103.6	16.2	61.2	16.8	62.2
3008	2.50	.5678	.450	.1022	14.8	58.6	39.6	103.3	16.2	61.2	17.0	62.6
3009	2.50	.5678	.475	.1079	14.8	58.6	39.8	103.6	16.3	61.3	17.2	63.0
3010	2.50	.5678	.500	.1136	14.8	58.6	39.9	103.8	16.4	61.5	17.4	63.3
3101	1.50	.3407	.275	.0625	15.5	59.9	41.0	105.8	17.0	62.6	17.1	62.8
3102	1.50	.3407	.300	.0681	15.5	59.9	40.1	104.2	17.0	62.6	17.2	63.0
3103	1.50	.3407	.325	.0738	15.4	59.7	40.0	104.0	17.1	62.8	17.4	63.3
3104	1.50	.3407	.350	.0795	15.3	59.5	39.6	103.3	17.1	62.8	17.5	63.5
3105	1.55	.3520	.375	.0852	15.2	59.4	39.6	103.3	17.0	62.6	17.4	63.3
3106	1.50	.3407	.400	.0908	15.2	59.4	39.6	103.3	17.2	63.0	17.8	64.0
3107	1.50	.3407	.425	.0965	15.2	59.4	39.9	103.8	17.3	63.1	18.2	64.8
3108	1.50	.3407	.450	.1022	15.2	59.4	40.4	104.7	17.4	63.3	18.3	64.9
3109	1.50	.3407	.475	.1079	15.4	59.7	39.5	103.1	17.7	63.9	18.5	65.3
3110	1.55	.3520	.500	.1136	15.4	59.7	39.8	103.6	17.7	63.9	18.7	65.7
3201	1.50	.3407	.275	.0625	14.3	57.7	36.1	97.0	15.5	59.9	15.8	60.4
3202	1.50	.3407	.300	.0681	14.4	57.9	32.5	90.5	15.6	60.1	15.8	60.4
3203	1.50	.3407	.325	.0738	14.5	58.1	31.3	88.3	15.6	60.1	15.9	60.6
3204	1.50	.3407	.350	.0795	14.3	57.7	30.7	87.3	15.5	59.9	15.9	60.6
3205	1.55	.3520	.375	.0852	14.5	58.1	29.7	85.5	15.6	60.1	16.1	61.0
3206	1.50	.3407	.400	.0908	14.6	58.3	28.8	83.8	15.8	60.4	16.2	61.2
3207	1.55	.3520	.425	.0965	14.5	58.1	28.2	82.8	15.6	60.1	16.2	61.2
3208	1.50	.3407	.450	.1022	14.4	57.9	27.4	81.3	15.6	60.1	16.1	61.0
3209	1.55	.3520	.475	.1079	14.5	58.1	26.9	80.4	15.6	60.1	16.1	61.0
3210	1.50	.3407	.500	.1136	14.4	57.9	26.4	79.5	15.6	60.1	16.2	61.2
3301	1.30	.2953	.500	.1136	15.0	59.0	39.2	102.6	17.7	63.9	18.8	65.8
3302	1.42	.3225	.500	.1136	14.7	58.5	38.8	101.8	17.1	62.8	18.2	64.8
3303	1.50	.3407	.500	.1136	14.9	58.8	39.7	103.5	17.3	63.1	18.4	65.1
3304	1.60	.3634	.500	.1136	14.9	58.8	39.9	103.8	17.3	63.1	18.3	64.9
3305	1.70	.3861	.500	.1136	14.9	58.8	40.0	104.0	17.1	62.8	18.2	64.8
3306	1.80	.4088	.500	.1136	14.9	58.8	39.9	103.8	17.0	62.6	18.1	64.6
3307	1.90	.4315	.500	.1136	14.7	58.5	39.9	103.8	16.8	62.2	17.9	64.2
3308	2.00	.4542	.500	.1136	14.5	58.1	39.8	103.6	16.4	61.5	17.4	63.3
3309	2.10	.4770	.500	.1136	14.5	58.1	39.7	103.5	16.3	61.3	17.4	63.3
3310	2.20	.4997	.500	.1136	14.5	58.1	39.7	103.5	16.3	61.3	17.4	63.3
3401	1.30	.2953	.275	.0625	15.1	59.2	40.0	104.0	16.8	62.2	16.8	62.2
3402	1.35	.3066	.275	.0625	15.1	59.2	40.0	104.0	16.8	62.2	16.8	62.2
3403	1.50	.3407	.275	.0625	14.9	58.8	39.9	103.8	16.4	61.5	16.6	61.9
3404	1.60	.3634	.275	.0625	14.9	58.8	39.9	103.8	16.3	61.3	16.5	61.7
3405	1.70	.3861	.275	.0625	14.8	58.6	39.5	103.1	16.1	61.0	16.3	61.3
3406	1.80	.4088	.275	.0625	14.8	58.6	39.6	103.3	16.0	60.8	16.3	61.3
3407	1.90	.4315	.275	.0625	14.7	58.5	39.4	102.9	15.9	60.6	16.1	61.0
3408	2.00	.4542	.275	.0625	14.7	58.5	39.5	103.1	15.8	60.4	16.0	60.8
3409	2.10	.4770	.275	.0625	14.7	58.5	39.6	103.3	15.8	60.4	16.0	60.8
3410	2.20	.4997	.275	.0625	14.7	58.5	39.6	103.3	15.7	60.3	16.0	60.8

TABLE C.1 (Contd.)

Run	Flow rate				Inlet temperature				Exit temperature			
	Water		R-113		Water		R-113		Water		R-113	
#	gpm	cu.m/hr	gpm	cu.m/hr	deg.C	deg.F	deg.C	deg.F	deg.C	deg.F	deg.C	deg.F
7001	3.40	.7722	.500	.1136	15.4	59.7	40.4	104.7	16.5	61.7	18.7	65.7
7002	3.40	.7722	.475	.1079	15.3	59.5	40.7	105.3	16.4	61.5	18.5	65.3
7003	3.40	.7722	.450	.1022	15.4	59.7	40.3	104.5	16.4	61.5	18.4	65.1
7004	3.40	.7722	.425	.0965	15.4	59.7	40.0	104.0	16.3	61.3	18.2	64.8
7005	3.40	.7722	.400	.0908	15.5	59.9	40.1	104.2	16.4	61.5	18.1	64.6
7006	3.40	.7722	.375	.0852	15.5	59.9	40.0	104.0	16.3	61.3	17.9	64.2
7007	3.40	.7722	.350	.0795	15.6	60.1	40.2	104.4	16.4	61.5	18.0	64.4
7008	3.40	.7722	.330	.0750	15.3	59.5	40.4	104.7	16.1	61.0	17.5	63.5
7009	3.40	.7722	.300	.0681	15.3	59.5	40.1	104.2	16.0	60.8	17.4	63.3
7010	3.40	.7722	.280	.0636	15.4	59.7	40.9	105.6	16.1	61.0	17.4	63.3
7101	1.70	.3861	.275	.0625	18.5	65.3	40.4	104.7	19.6	67.3	20.7	69.3
7102	1.70	.3861	.300	.0681	16.9	62.4	40.1	104.2	18.2	64.8	19.6	67.3
7103	1.70	.3861	.325	.0738	16.8	62.2	39.6	103.3	18.1	64.6	19.3	66.7
7104	1.70	.3861	.350	.0795	16.8	62.2	39.9	103.8	18.2	64.8	19.6	67.3
7105	1.70	.3861	.375	.0852	16.7	62.1	40.2	104.4	18.2	64.8	19.8	67.6
7106	1.70	.3861	.400	.0908	17.1	62.8	40.4	104.7	18.7	65.7	20.4	68.7
7107	1.70	.3861	.425	.0965	17.2	63.0	40.4	104.7	18.9	66.0	20.7	69.3
7108	1.70	.3861	.450	.1022	17.2	63.0	40.5	104.9	19.0	66.2	20.8	69.4
7109	1.70	.3861	.475	.1079	17.3	63.1	40.6	105.1	19.2	66.6	21.1	70.0
7110	1.70	.3861	.500	.1136	17.3	63.1	40.9	105.6	19.3	66.7	21.4	70.5
7201	1.65	.3748	.275	.0625	16.3	61.3	41.5	106.7	17.6	63.7	18.8	65.8
7202	1.65	.3748	.300	.0681	17.5	63.5	40.0	104.0	18.8	65.8	20.1	68.2
7203	1.65	.3748	.325	.0738	17.9	64.2	38.3	100.9	19.1	66.4	20.3	68.5
7204	1.65	.3748	.350	.0795	17.9	64.2	37.7	99.9	19.2	66.6	20.5	68.9
7205	1.65	.3748	.375	.0852	18.0	64.4	36.4	97.5	19.2	66.6	20.5	68.9
7206	1.65	.3748	.400	.0908	18.0	64.4	35.3	95.5	19.3	66.7	20.6	69.1
7207	1.65	.3748	.425	.0965	16.9	62.4	34.9	94.8	18.2	64.8	19.7	67.5
7208	1.65	.3748	.450	.1022	16.5	61.7	33.9	93.0	17.9	64.2	19.4	66.9
7209	1.65	.3748	.475	.1079	16.6	61.9	32.7	90.9	17.9	64.2	19.4	66.9
7210	1.65	.3748	.500	.1136	16.4	61.5	32.0	89.6	17.7	63.9	19.2	66.6
7301	1.60	.3634	.275	.0625	15.4	59.7	37.3	99.1	16.6	61.9	17.8	64.0
7302	1.70	.3861	.275	.0625	15.4	59.7	39.7	103.5	16.6	61.9	17.8	64.0
7303	1.50	.3407	.275	.0625	15.6	60.1	39.8	103.6	16.9	62.4	18.3	64.9
7304	1.40	.3180	.275	.0625	15.7	60.3	39.9	103.8	17.1	62.8	18.5	65.3
7305	1.30	.2953	.275	.0625	15.6	60.1	39.9	103.8	17.2	63.0	18.6	65.5
7306	1.20	.2725	.275	.0625	15.7	60.3	39.9	103.8	17.4	63.3	18.7	65.7
7307	1.10	.2498	.275	.0625	15.9	60.6	39.8	103.6	17.7	63.9	18.9	66.0
7308	1.00	.2271	.275	.0625	16.2	61.2	39.4	102.9	18.1	64.6	19.3	66.7
7309	0.90	.2044	.275	.0625	16.5	61.7	39.5	103.1	18.6	65.5	19.7	67.5
7310	0.80	.1817	.275	.0625	16.6	61.9	39.7	103.5	19.0	66.2	20.0	68.0
7401	0.80	.1817	.500	.1136	16.7	62.1	40.2	104.4	20.5	68.9	22.5	72.5
7402	0.90	.2044	.500	.1136	16.6	61.9	39.2	102.6	20.0	68.0	22.2	72.0
7403	1.00	.2271	.500	.1136	16.5	61.7	39.9	103.8	19.7	67.5	21.9	71.4
7404	1.10	.2498	.500	.1136	16.3	61.3	39.9	103.8	19.3	66.7	21.6	70.9
7405	1.20	.2725	.500	.1136	16.3	61.3	40.0	104.0	19.0	66.2	21.4	70.5
7406	1.30	.2953	.500	.1136	16.0	60.8	39.9	103.8	18.6	65.5	21.0	69.8
7407	1.40	.3180	.500	.1136	16.0	60.8	40.3	104.5	18.3	64.9	20.8	69.4
7408	1.50	.3407	.500	.1136	16.0	60.8	40.4	104.7	18.2	64.8	20.6	69.1
7409	1.60	.3634	.500	.1136	16.0	60.8	40.2	104.4	18.2	64.8	20.5	68.9
7410	1.70	.3861	.500	.1136	15.9	60.6	40.2	104.4	17.9	64.2	20.3	68.5

TABLE C.1 (Contd.)

Run	Flow rate				Inlet temperature				Exit temperature			
	Water		R-113		Water		R-113		Water		R-113	
#	gpm	cu.m/hr	gpm	cu.m/hr	deg.C	deg.F	deg.C	deg.F	deg.C	deg.F	deg.C	deg.F
8001	3.40	.7722	.275	.0625	15.5	59.9	40.7	105.3	15.9	60.6	24.7	76.5
8002	3.40	.7722	.300	.0681	15.4	59.7	40.1	104.2	15.9	60.6	24.8	76.6
8003	3.40	.7722	.325	.0738	15.5	59.9	41.4	106.5	16.0	60.8	25.7	78.3
8004	3.35	.7609	.350	.0795	15.2	59.4	41.3	106.3	15.8	60.4	25.6	78.1
8005	3.40	.7722	.375	.0852	15.2	59.4	41.7	107.1	15.8	60.4	26.0	78.8
8006	3.40	.7722	.400	.0908	15.1	59.2	40.8	105.4	15.7	60.3	25.9	78.6
8007	3.40	.7722	.425	.0965	15.0	59.0	41.3	106.3	15.7	60.3	26.2	79.2
8008	3.40	.7722	.450	.1022	15.0	59.0	41.9	107.4	15.7	60.3	26.6	79.9
8009	3.45	.7836	.475	.1079	15.0	59.0	42.1	107.8	15.7	60.3	26.7	80.1
8010	3.40	.7722	.500	.1136	14.9	58.8	42.0	107.6	15.7	60.3	26.8	80.2
8101	1.70	.3861	.275	.0625	16.3	61.3	41.1	106.0	17.2	63.0	25.9	78.6
8102	1.70	.3861	.300	.0681	16.4	61.5	40.9	105.6	17.3	63.1	26.3	79.3
8103	1.75	.3975	.325	.0738	16.4	61.5	41.4	106.5	17.3	63.1	26.6	79.9
8104	1.70	.3861	.350	.0795	16.2	61.2	41.3	106.3	17.2	63.0	26.9	80.4
8105	1.70	.3861	.375	.0852	16.3	61.3	41.4	106.5	17.4	63.3	27.1	80.8
8106	1.70	.3861	.400	.0908	16.3	61.3	41.0	105.8	17.4	63.3	27.1	80.8
8107	1.64	.3725	.430	.0977	16.2	61.2	41.2	106.2	17.5	63.5	27.5	81.5
8108	1.70	.3861	.450	.1022	16.3	61.3	41.4	106.5	17.6	63.7	27.7	81.9
8109	1.70	.3861	.475	.1079	16.3	61.3	41.8	107.2	17.6	63.7	28.0	82.4
8110	1.70	.3861	.500	.1136	16.3	61.3	41.2	106.2	17.7	63.9	27.9	82.2
8201	1.65	.3748	.275	.0625	16.2	61.2	46.4	115.5	17.3	63.1	27.0	80.6
8202	1.65	.3748	.300	.0681	16.1	61.0	45.6	114.1	17.3	63.1	27.3	81.1
8203	1.65	.3748	.325	.0738	16.2	61.2	44.5	112.1	17.3	63.1	27.6	81.7
8204	1.62	.3679	.350	.0795	16.1	61.0	42.8	109.0	17.3	63.1	27.3	81.1
8205	1.63	.3702	.380	.0863	16.0	60.8	41.3	106.3	17.2	63.0	26.9	80.4
8206	1.65	.3748	.400	.0908	16.0	60.8	39.7	103.5	17.1	62.8	26.4	79.5
8207	1.65	.3748	.425	.0965	15.9	60.6	38.8	101.8	17.0	62.6	26.3	79.3
8208	1.63	.3702	.450	.1022	16.0	60.8	38.5	101.3	17.2	63.0	26.4	79.5
8209	1.67	.3793	.475	.1079	16.1	61.0	37.4	99.3	17.2	63.0	26.1	79.0
8210	1.67	.3793	.500	.1136	16.0	60.8	36.3	97.3	17.1	62.8	25.7	78.3
8301	1.70	.3861	.275	.0625	16.3	61.3	39.8	103.6	17.1	62.8	25.6	78.1
8302	1.56	.3543	.275	.0625	16.4	61.5	39.5	103.1	17.3	63.1	25.3	77.5
8303	1.50	.3407	.275	.0625	16.7	62.1	40.5	104.9	17.6	63.7	26.1	79.0
8304	1.40	.3180	.275	.0625	16.7	62.1	40.6	105.1	17.7	63.9	26.2	79.2
8305	1.17	.2657	.275	.0625	16.9	62.4	40.7	105.3	18.1	64.6	26.4	79.5
8306	1.23	.2794	.275	.0625	17.1	62.8	40.8	105.4	18.2	64.8	26.4	79.5
8307	1.07	.2430	.275	.0625	17.2	63.0	40.9	105.6	18.5	65.3	26.8	80.2
8308	0.97	.2203	.275	.0625	17.4	63.3	41.1	106.0	18.8	65.8	27.2	81.0
8309	0.80	.1817	.275	.0625	17.6	63.7	41.0	105.8	19.2	66.6	27.3	81.1
8310	0.70	.1590	.275	.0625	17.8	64.0	41.2	106.2	19.7	67.5	27.8	82.0
8401	0.70	.1590	.500	.1136	19.6	67.3	39.3	102.7	21.9	71.4	29.6	85.3
8402	0.82	.1862	.500	.1136	19.2	66.6	40.1	104.2	21.3	70.3	29.8	85.6
8403	0.90	.2044	.500	.1136	18.9	66.0	40.2	104.4	20.9	69.6	29.4	84.9
8404	1.02	.2317	.500	.1136	18.5	65.3	40.1	104.2	20.3	68.5	29.1	84.4
8405	1.10	.2498	.500	.1136	18.3	64.9	40.1	104.2	20.1	68.2	29.0	84.2
8406	1.20	.2725	.500	.1136	18.1	64.6	40.2	104.4	19.8	67.6	28.8	83.8
8407	1.30	.2953	.500	.1136	18.0	64.4	40.3	104.5	19.5	67.1	28.7	83.7
8408	1.40	.3180	.500	.1136	17.9	64.2	40.2	104.4	19.3	66.7	28.5	83.3
8409	1.50	.3407	.500	.1136	17.6	63.7	39.9	103.8	18.9	66.0	28.2	82.8
8410	1.60	.3634	.500	.1136	17.4	63.3	40.1	104.2	18.7	65.7	28.1	82.6

TABLE C.1 (Contd.)

Run #	Flow rate Water gpm	Flow rate Water cu.m/hr	Flow rate R-113 gpm	Flow rate R-113 cu.m/hr	Inlet temperature Water deg.C	Inlet temperature Water deg.F	Inlet temperature R-113 deg.C	Inlet temperature R-113 deg.F	Exit temperature Water deg.C	Exit temperature Water deg.F	Exit temperature R-113 deg.C	Exit temperature R-113 deg.F
9001	3.40	.7722	.510	.1158	15.8	60.4	41.3	106.3	17.0	62.6	19.0	66.2
9002	3.40	.7722	.450	.1022	15.7	60.3	40.5	104.9	16.7	62.1	18.6	65.5
9003	3.40	.7722	.400	.0908	15.6	60.1	40.5	104.9	16.5	61.7	18.2	64.8
9004	3.40	.7722	.350	.0795	16.0	60.8	40.5	104.9	16.8	62.2	18.2	64.8
9005	3.40	.7722	.300	.0681	16.2	61.2	41.2	106.2	16.9	62.4	18.1	64.6
9006	3.40	.7722	.325	.0738	16.2	61.2	40.6	105.1	16.9	62.4	18.5	65.3
9007	3.40	.7722	.375	.0852	16.3	61.3	40.1	104.2	17.1	62.8	18.7	65.7
9008	3.40	.7722	.425	.0965	16.2	61.2	40.0	104.0	17.1	62.8	18.9	66.0
9009	3.40	.7722	.475	.1079	16.2	61.2	40.1	104.2	17.2	63.0	19.1	66.4
9010	3.40	.7722	.275	.0625	16.2	61.2	40.7	105.3	16.8	62.2	18.1	64.6
9101	1.70	.3861	.500	.1136	17.9	64.2	40.6	105.1	19.8	67.6	21.9	71.4
9102	1.70	.3861	.450	.1022	18.0	64.4	40.7	105.3	19.8	67.6	21.6	70.9
9103	1.70	.3861	.475	.1079	18.1	64.6	40.9	105.6	20.0	68.0	21.8	71.2
9104	1.70	.3861	.425	.0965	17.3	63.1	41.2	106.2	19.1	66.4	20.9	69.6
9105	1.70	.3861	.400	.0908	17.4	63.3	40.8	105.4	19.0	66.2	20.7	69.3
9106	1.70	.3861	.375	.0852	17.4	63.3	41.1	106.0	19.0	66.2	20.6	69.1
9107	1.70	.3861	.350	.0795	17.5	63.5	41.0	105.8	18.9	66.0	20.4	68.7
9108	1.70	.3861	.325	.0738	17.9	64.2	40.5	104.9	19.2	66.6	20.5	68.9
9109	1.70	.3861	.300	.0681	17.9	64.2	40.4	104.7	19.1	66.4	20.2	68.4
9110	1.70	.3861	.275	.0625	17.9	64.2	40.6	105.1	19.0	66.2	20.1	68.2
9201	1.65	.3748	.500	.1136	17.3	63.1	31.7	89.1	18.5	65.3	19.9	67.8
9202	1.65	.3748	.475	.1079	17.4	63.3	32.3	90.1	18.6	65.5	19.9	67.8
9203	1.65	.3748	.450	.1022	17.3	63.1	33.1	91.6	18.5	65.3	20.0	68.0
9204	1.65	.3748	.425	.0965	17.4	63.3	33.8	92.8	18.6	65.5	19.9	67.8
9205	1.65	.3748	.400	.0908	17.4	63.3	34.7	94.5	18.6	65.5	19.9	67.8
9206	1.65	.3748	.375	.0852	17.2	63.0	35.6	96.1	18.4	65.1	19.7	67.5
9207	1.65	.3748	.350	.0795	17.3	63.1	36.9	98.4	18.6	65.5	19.9	67.8
9208	1.65	.3748	.325	.0738	17.3	63.1	38.0	100.4	18.5	65.3	19.8	67.6
9209	1.65	.3748	.300	.0681	17.4	63.3	39.7	103.5	18.7	65.7	19.8	67.6
9210	1.65	.3748	.275	.0625	17.4	63.3	40.6	105.1	18.6	65.5	19.8	67.6
9301	0.80	.1817	.275	.0625	15.9	60.6	39.3	102.7	18.2	64.8	19.1	66.4
9302	0.90	.2044	.275	.0625	15.9	60.6	39.7	103.5	18.1	64.6	19.2	66.6
9303	1.00	.2271	.275	.0625	15.7	60.3	39.8	103.6	17.8	64.0	18.8	65.8
9304	1.10	.2498	.275	.0625	15.6	60.1	39.9	103.8	17.4	63.3	18.5	65.3
9305	1.20	.2725	.275	.0625	15.5	59.9	40.0	104.0	17.2	63.0	18.5	65.3
9306	1.30	.2953	.275	.0625	15.4	59.7	40.3	104.5	17.0	62.6	18.2	64.8
9307	1.40	.3180	.275	.0625	15.4	59.7	40.4	104.7	16.9	62.4	18.1	64.6
9308	1.45	.3293	.275	.0625	15.4	59.7	40.5	104.9	16.9	62.4	18.1	64.6
9309	1.60	.3634	.275	.0625	15.2	59.4	40.6	105.1	16.6	61.9	17.9	64.2
9310	1.70	.3861	.275	.0625	15.1	59.2	40.6	105.1	16.4	61.5	17.6	63.7
9401	0.80	.1817	.500	.1136	16.1	61.0	39.4	102.9	20.0	68.0	21.9	71.4
9402	0.85	.1931	.500	.1136	15.9	60.6	39.6	103.3	19.7	67.5	21.6	70.9
9403	1.00	.2271	.500	.1136	15.6	60.1	39.9	103.8	18.9	66.0	21.1	70.0
9404	1.10	.2498	.500	.1136	15.4	59.7	39.9	103.8	18.4	65.1	20.7	69.3
9405	1.20	.2725	.500	.1136	15.2	59.4	39.8	103.6	18.0	64.4	20.3	68.5
9406	1.30	.2953	.500	.1136	15.1	59.2	40.0	104.0	17.7	63.9	20.0	68.0
9407	1.40	.3180	.500	.1136	14.9	58.8	39.7	103.5	17.4	63.3	19.6	67.3
9408	1.50	.3407	.500	.1136	14.8	58.6	39.7	103.5	17.2	63.0	19.4	66.9
9409	1.60	.3634	.500	.1136	14.7	58.5	39.7	103.5	16.9	62.4	19.1	66.4
9410	1.70	.3861	.500	.1136	14.7	58.5	39.6	103.3	16.8	62.2	19.0	66.2

TABLE C.2 :- REDUCED DATA - WALL TEMPERATURES, LMTD, HEAT GAIN, HEAT LOST AND HEAT BALANCE ERROR

Run	Twall		LMTD		LMTD(Outside)		Heat gain		Heat lost		%
#	deg.C	deg.F	deg.C	deg.F	deg.C	deg.F	Btu/hr	W	Btu/hr	W	error
1001	16.35	61.43	7.59	13.66	1.56	2.80	2023.8	593.1	2007.37	588.3	0.8
1002	16.67	62.00	8.19	14.74	1.82	3.28	2248.6	659.0	2283.98	669.4	-1.6
1003	16.87	62.36	8.57	15.43	1.97	3.54	2473.5	724.9	2415.01	707.8	2.4
1004	16.96	62.52	8.86	15.96	2.10	3.78	2698.4	790.8	2591.44	759.5	4.0
1005	17.18	62.93	9.14	16.46	2.27	4.09	2923.2	856.7	2765.32	810.4	5.4
1006	17.39	63.30	9.36	16.84	2.48	4.47	2923.2	856.7	2975.33	872.0	-1.8
1007	17.35	63.23	9.51	17.12	2.59	4.66	3148.1	922.6	3097.21	907.7	1.6
1008	17.42	63.37	10.02	18.04	2.77	4.98	3211.0	941.1	3308.74	969.7	-3.0
1009	17.69	63.85	10.31	18.57	2.98	5.37	3372.9	988.5	3507.80	1028.0	-4.0
1010	17.83	64.10	10.35	18.63	3.06	5.51	3597.8	1054.4	3569.33	1046.1	0.8
1101	17.62	63.71	8.25	14.85	1.97	3.55	2023.8	593.1	1965.41	576.0	2.9
1102	17.99	64.39	8.73	15.71	2.30	4.14	2158.7	632.7	2145.02	628.6	0.6
1103	18.35	65.03	9.17	16.51	2.54	4.58	2428.5	711.7	2334.51	684.2	3.9
1104	18.58	65.44	9.32	16.77	2.61	4.70	2563.4	751.3	2460.26	721.0	4.0
1105	18.92	66.07	9.62	17.32	2.81	5.05	2698.4	790.8	2590.29	759.1	4.0
1106	18.95	66.11	9.50	17.11	2.79	5.03	2648.9	776.3	2763.38	809.9	-4.3
1107	19.17	66.52	9.73	17.52	3.00	5.41	2833.3	830.4	2871.09	841.4	-1.3
1108	19.41	66.94	10.15	18.26	3.19	5.74	2968.2	869.9	3054.80	895.3	-2.9
1109	19.77	67.58	10.43	18.78	3.49	6.29	3103.1	909.4	3180.80	932.2	-2.5
1110	19.92	67.87	10.67	19.21	3.53	6.35	3372.9	988.5	3302.18	967.8	2.1
1201	17.05	62.69	7.19	12.95	1.78	3.21	1619.0	474.5	1693.15	496.2	-4.6
1202	17.05	62.69	6.80	12.24	1.68	3.02	1619.0	474.5	1643.35	481.6	-1.5
1203	17.09	62.77	6.57	11.82	1.72	3.10	1619.0	474.5	1630.37	477.8	-0.7
1204	17.00	62.60	6.39	11.50	1.73	3.12	1619.0	474.5	1626.43	476.7	-0.5
1205	17.12	62.81	6.00	10.80	1.65	2.96	1619.0	474.5	1638.04	480.1	-1.2
1206	17.29	63.12	5.83	10.50	1.72	3.09	1619.0	474.5	1611.17	472.2	0.5
1207	17.19	62.95	5.77	10.39	1.72	3.10	1619.0	474.5	1620.63	475.0	-0.1
1208	17.41	63.34	5.73	10.31	1.74	3.14	1619.0	474.5	1632.02	478.3	-0.8
1209	17.24	63.03	5.48	9.86	1.67	3.00	1619.0	474.5	1664.27	487.8	-2.8
1210	17.35	63.23	5.40	9.72	1.68	3.02	1619.0	474.5	1643.34	481.6	-1.5
1301	19.88	67.79	10.75	19.34	3.67	6.60	3157.1	925.3	3242.04	950.2	-2.7
1302	19.68	67.43	10.83	19.50	3.65	6.57	3238.0	949.0	3304.08	968.3	-2.0
1303	19.51	67.12	10.69	19.25	3.54	6.37	3310.0	970.1	3350.14	981.8	-1.2
1304	19.40	66.92	10.52	18.93	3.38	6.09	3364.0	985.9	3380.16	990.6	-0.5
1305	19.19	66.54	10.56	19.01	3.33	5.99	3399.9	996.4	3457.71	1013.4	-1.7
1306	18.86	65.95	10.45	18.82	3.26	5.87	3417.9	1001.7	3458.71	1013.7	-1.2
1307	18.73	65.71	10.27	18.48	3.14	5.65	3319.0	972.7	3474.22	1018.2	-4.7
1308	18.76	65.76	10.51	18.92	3.21	5.78	3588.8	1051.8	3473.72	1018.0	3.2
1309	18.67	65.62	10.57	19.03	3.19	5.74	3561.8	1043.9	3488.72	1022.4	2.1
1310	18.53	65.36	10.41	18.73	3.04	5.48	3723.7	1091.3	3550.74	1040.6	4.6
1401	18.38	65.08	8.64	15.55	2.26	4.06	2104.7	616.8	2007.66	588.4	4.6
1402	18.19	64.75	8.60	15.47	2.13	3.84	2140.7	627.4	2041.10	598.2	4.7
1403	18.03	64.45	8.71	15.68	2.23	4.01	2115.5	620.0	2015.58	590.7	4.7
1404	17.62	63.71	8.52	15.34	2.14	3.86	2014.8	590.5	2050.20	600.9	-1.8
1405	17.54	63.57	8.62	15.51	2.16	3.89	2140.7	627.4	2083.92	610.7	2.7
1406	17.38	63.28	8.44	15.19	2.06	3.70	2104.7	616.8	2076.02	608.4	1.4
1407	17.21	62.98	8.33	14.99	1.95	3.51	2050.8	601.0	2092.73	613.3	-2.0
1408	17.19	62.95	8.35	15.04	1.93	3.48	2158.7	632.7	2100.63	615.6	2.7
1409	17.06	62.71	8.40	15.12	1.96	3.53	2077.7	608.9	2109.44	618.2	-1.5
1410	16.93	62.48	8.33	14.99	1.89	3.40	1978.8	579.9	2075.42	608.2	-4.9

TABLE C.2 (Contd.)

Run #	Twall deg.C	Twall deg.F	LMTD deg.C	LMTD deg.F	LMTD(Outside) deg.C	LMTD(Outside) deg.F	Heat gain Btu/hr	Heat gain W	Heat lost Btu/hr	Heat lost W	% error
2001	16.24	61.24	14.34	25.82	0.91	1.64	1349.2	395.4	1411.35	413.6	-4.6
2002	16.26	61.26	14.93	26.87	1.07	1.92	1574.0	461.3	1587.18	465.2	-0.8
2003	16.48	61.66	15.37	27.66	1.13	2.03	1798.9	527.2	1730.00	507.0	3.8
2004	16.54	61.78	15.26	27.47	1.20	2.16	1798.9	527.2	1766.36	517.7	1.8
2005	16.65	61.97	15.48	27.86	1.35	2.43	1983.3	581.2	1904.97	558.3	3.9
2006	16.72	62.09	15.55	27.99	1.32	2.37	2023.8	593.1	2019.87	592.0	0.2
2007	16.86	62.35	15.51	27.92	1.40	2.53	2248.6	659.0	2146.11	629.0	4.6
2008	16.86	62.35	15.36	27.66	1.40	2.53	2248.6	659.0	2217.23	649.8	1.4
2009	17.08	62.74	15.44	27.78	1.62	2.92	2248.6	659.0	2326.02	681.7	-3.4
2010	17.26	63.07	15.76	28.37	1.65	2.97	2473.5	724.9	2464.30	722.2	0.4
2101	17.86	64.14	14.35	25.83	1.29	2.33	1349.2	395.4	1367.25	400.7	-1.3
2102	18.09	64.57	14.79	26.62	1.41	2.54	1619.0	474.5	1510.95	442.8	6.7
2103	18.24	64.84	14.82	26.68	1.46	2.63	1619.0	474.5	1617.25	474.0	0.1
2104	18.23	64.82	15.00	26.99	1.53	2.75	1762.9	516.7	1688.30	494.8	4.2
2105	18.26	64.87	15.03	27.05	1.67	3.00	1825.9	535.1	1786.74	523.6	2.1
2106	18.29	64.93	15.21	27.37	1.70	3.06	1888.9	553.6	1894.03	555.1	-0.3
2107	18.22	64.79	15.31	27.56	1.83	3.29	1951.8	572.0	2013.57	590.1	-3.2
2108	18.42	65.16	15.63	28.13	2.02	3.64	2158.7	632.7	2118.69	620.9	1.9
2109	18.38	65.08	15.87	28.56	2.07	3.73	2158.7	632.7	2207.61	647.0	-2.3
2110	18.63	65.52	15.79	28.42	2.16	3.90	2293.6	672.2	2308.31	676.5	-0.6
2201	17.89	64.20	14.35	25.83	1.33	2.39	1349.2	395.4	1367.45	400.8	-1.4
2202	17.50	63.50	14.53	26.16	1.38	2.48	1484.1	434.9	1493.93	437.8	-0.7
2203	17.41	63.34	13.65	24.57	1.39	2.50	1484.1	434.9	1408.35	412.7	5.1
2204	17.31	63.16	12.99	23.38	1.35	2.43	1349.2	395.4	1408.75	412.9	-4.4
2205	17.11	62.79	12.36	22.25	1.28	2.30	1434.6	420.4	1393.88	408.5	2.8
2206	17.11	62.80	11.82	21.27	1.35	2.43	1394.2	408.6	1437.87	421.4	-3.1
2207	17.11	62.79	11.34	20.41	1.34	2.42	1394.2	408.6	1435.54	420.7	-3.0
2208	17.14	62.85	11.34	20.41	1.31	2.36	1484.1	434.9	1534.15	449.6	-3.4
2209	17.21	62.98	10.93	19.67	1.32	2.38	1565.0	458.7	1516.33	444.4	3.1
2210	17.08	62.75	10.45	18.82	1.25	2.25	1484.1	434.9	1519.09	445.2	-2.4
2301	19.38	66.89	15.63	28.13	2.34	4.21	2338.6	685.4	2246.02	658.2	4.0
2302	19.11	66.39	15.74	28.34	2.29	4.12	2266.6	664.3	2261.84	662.9	0.2
2303	18.86	65.95	15.75	28.35	2.20	3.97	2293.6	672.2	2292.49	671.9	0.0
2304	18.56	65.41	15.76	28.37	2.10	3.78	2446.5	717.0	2339.29	685.6	4.4
2305	18.32	64.97	15.69	28.25	2.01	3.63	2446.5	717.0	2339.63	685.7	4.4
2306	17.98	64.35	15.88	28.58	2.03	3.66	2428.5	711.7	2355.81	690.4	3.0
2307	17.76	63.97	15.92	28.66	1.98	3.57	2392.5	701.2	2402.30	704.0	-0.4
2308	17.54	63.58	15.75	28.34	1.92	3.46	2338.6	685.4	2402.65	704.1	-2.7
2309	17.47	63.44	15.93	28.67	1.84	3.32	2455.5	719.6	2433.65	713.2	0.9
2310	17.42	63.37	15.79	28.42	1.82	3.28	2374.6	695.9	2433.30	713.1	-2.5
2401	17.67	63.80	14.57	26.22	1.38	2.49	1403.1	411.2	1386.04	406.2	1.2
2402	17.58	63.64	14.71	26.48	1.35	2.43	1385.2	406.0	1411.14	413.6	-1.9
2403	17.35	63.23	14.79	26.62	1.32	2.38	1484.1	434.9	1428.15	418.5	3.8
2404	17.17	62.91	14.71	26.48	1.21	2.17	1439.1	421.8	1453.65	426.0	-1.0
2405	16.99	62.59	14.86	26.74	1.23	2.21	1529.1	448.1	1461.94	428.5	4.4
2406	16.69	62.05	14.93	26.87	1.19	2.14	1457.1	427.0	1470.86	431.1	-0.9
2407	16.54	61.78	14.93	26.87	1.13	2.04	1538.1	450.8	1470.86	431.1	4.4
2408	16.54	61.78	15.04	27.07	1.20	2.16	1439.1	421.8	1462.58	428.6	-1.6
2409	16.42	61.57	14.89	26.80	1.08	1.94	1511.1	442.9	1479.56	433.6	2.1
2410	16.29	61.32	15.00	26.99	1.10	1.98	1448.1	424.4	1496.59	438.6	-3.3

TABLE C.2 (Contd.)

Run #	Twall deg.C	Twall deg.F	LMTD deg.C	LMTD deg.F	LMTD(Outside) deg.C	LMTD(Outside) deg.F	Heat gain Btu/hr	Heat gain W	Heat lost Btu/hr	Heat lost W	% error
3001	16.36	61.44	7.47	13.45	1.25	2.26	2023.8	593.1	1964.84	575.8	2.9
3002	16.54	61.77	7.74	13.93	1.38	2.48	2248.6	659.0	2181.53	639.3	3.0
3003	16.72	62.09	8.00	14.39	1.40	2.52	2473.5	724.9	2324.11	681.1	6.0
3004	16.92	62.45	8.22	14.80	1.61	2.89	2473.5	724.9	2525.29	740.1	-2.1
3005	17.09	62.77	8.61	15.50	1.72	3.10	2698.4	790.8	2659.60	779.5	1.4
3006	17.26	63.07	8.80	15.84	1.84	3.31	2923.2	856.7	2862.51	838.9	2.1
3007	17.37	63.26	8.61	15.50	1.74	3.13	2923.2	856.7	3028.25	887.5	-3.6
3008	17.46	63.42	8.97	16.14	1.87	3.37	3148.1	922.6	3151.99	923.8	-0.1
3009	17.65	63.77	9.25	16.65	2.01	3.61	3372.9	988.5	3327.10	975.1	1.4
3010	17.83	64.10	9.49	17.09	2.13	3.84	3597.8	1054.4	3487.72	1022.2	3.1
3101	18.04	64.48	8.27	14.89	1.68	3.03	2023.8	593.1	2032.30	595.6	-0.4
3102	18.09	64.57	8.20	14.76	1.74	3.13	2023.8	593.1	2124.91	622.8	-5.0
3103	18.32	64.97	8.57	15.43	1.95	3.50	2293.6	672.2	2272.81	666.1	0.9
3104	18.26	64.87	8.73	15.72	1.92	3.46	2428.5	711.7	2394.54	701.8	1.4
3105	18.26	64.86	8.76	15.76	2.02	3.64	2509.5	735.5	2578.30	755.6	-2.7
3106	18.53	65.36	9.19	16.55	2.18	3.92	2698.4	790.8	2701.03	791.6	-0.1
3107	18.81	65.85	9.71	17.47	2.41	4.33	2833.3	830.4	2857.50	837.5	-0.9
3108	18.98	66.16	9.93	17.87	2.52	4.53	2968.2	869.9	3081.81	903.2	-3.8
3109	19.10	66.38	9.59	17.26	2.37	4.26	3103.1	909.4	3091.55	906.0	0.4
3110	19.37	66.86	9.89	17.80	2.65	4.78	3206.5	939.7	3270.23	958.4	-2.0
3201	16.63	61.94	7.29	13.12	1.66	2.99	1619.0	474.5	1728.18	506.5	-6.7
3202	16.28	61.31	6.22	11.20	1.18	2.13	1619.0	474.5	1551.85	454.8	4.1
3203	16.38	61.49	5.92	10.65	1.25	2.25	1484.1	434.9	1550.53	454.4	-4.5
3204	16.34	61.42	6.04	10.87	1.36	2.44	1619.0	474.5	1605.44	470.5	0.8
3205	16.39	61.51	5.74	10.34	1.27	2.28	1533.6	449.4	1581.10	463.4	-3.1
3206	16.51	61.71	5.44	9.80	1.21	2.18	1619.0	474.5	1562.50	457.9	3.5
3207	16.34	61.42	5.44	9.79	1.21	2.18	1533.6	449.4	1582.01	463.6	-3.2
3208	16.29	61.33	5.21	9.38	1.19	2.15	1619.0	474.5	1577.58	462.3	2.6
3209	16.29	61.32	4.96	8.93	1.15	2.07	1533.6	449.4	1592.00	466.6	-3.8
3210	16.28	61.30	5.02	9.04	1.17	2.11	1619.0	474.5	1582.69	463.8	2.2
3301	19.42	66.96	10.21	18.38	2.87	5.16	3157.1	925.3	3163.11	927.0	-0.2
3302	18.85	65.93	9.98	17.96	2.78	5.00	3065.3	898.4	3195.96	936.6	-4.3
3303	19.05	66.29	10.18	18.33	2.78	5.00	3238.0	949.0	3302.66	967.9	-2.0
3304	18.92	66.07	10.14	18.25	2.65	4.76	3453.9	1012.2	3349.65	981.7	3.0
3305	18.76	65.77	10.12	18.21	2.61	4.70	3364.0	985.9	3379.70	990.5	-0.5
3306	18.66	65.58	10.01	18.02	2.56	4.62	3399.9	996.4	3379.21	990.4	0.6
3307	18.44	65.19	10.07	18.12	2.54	4.58	3588.8	1051.8	3411.69	999.9	4.9
3308	18.07	64.52	9.82	17.67	2.50	4.50	3417.9	1001.7	3473.72	1018.0	-1.6
3309	17.94	64.29	9.82	17.67	2.43	4.37	3399.9	996.4	3458.21	1013.5	-1.7
3310	17.92	64.27	9.82	17.67	2.41	4.35	3561.8	1043.9	3458.21	1013.5	2.9
3401	17.91	64.24	8.23	14.81	1.83	3.30	1987.8	582.6	1973.64	578.4	0.7
3402	17.76	63.96	8.23	14.81	1.66	3.00	2064.2	605.0	1973.35	578.3	4.4
3403	17.42	63.35	8.30	14.94	1.66	2.98	2023.8	593.1	1981.85	580.8	2.1
3404	17.34	63.22	8.17	14.71	1.65	2.96	2014.8	590.5	1990.07	583.2	1.2
3405	17.11	62.80	7.97	14.35	1.57	2.83	1987.8	582.6	1973.06	578.2	0.7
3406	17.06	62.70	8.02	14.44	1.58	2.85	1942.8	569.4	1981.28	580.7	-2.0
3407	16.89	62.40	7.84	14.10	1.51	2.72	2050.8	601.0	1981.57	580.7	3.4
3408	16.78	62.21	7.72	13.89	1.46	2.63	1978.8	579.9	1998.59	585.7	-1.0
3409	16.69	62.04	7.74	13.93	1.36	2.46	2077.7	608.9	2007.06	588.2	3.4
3410	16.65	61.97	7.76	13.97	1.39	2.50	1978.8	579.9	2005.79	588.1	-1.4

TABLE C.2 (Contd.)

Run #	Twall deg.C	Twall deg.F	LMTD deg.C	LMTD deg.F	LMTD(Outside) deg.C	LMTD(Outside) deg.F	Heat gain Btu/hr	Heat gain W	Heat lost Btu/hr	Heat lost W	% error
7001	18.23	64.82	10.40	18.73	2.24	4.03	3364.0	985.9	3362.25	985.4	0.1
7002	18.06	64.51	10.41	18.73	2.17	3.90	3364.0	985.9	3266.79	957.4	2.9
7003	18.01	64.42	10.07	18.13	2.07	3.73	3058.1	896.3	3052.15	894.5	0.2
7004	17.82	64.07	9.79	17.61	1.93	3.48	2752.3	806.6	2869.43	840.9	-4.3
7005	17.82	64.07	9.55	17.19	1.83	3.30	2752.3	806.6	2723.83	798.3	1.0
7006	17.74	63.94	9.30	16.74	1.81	3.27	2446.5	717.0	2564.46	751.6	-4.8
7007	17.67	63.80	9.33	16.79	1.64	2.95	2446.5	717.0	2403.63	704.4	1.8
7008	17.31	63.16	9.20	16.56	1.58	2.84	2446.5	717.0	2337.74	685.1	4.4
7009	17.19	62.95	9.02	16.23	1.52	2.73	2140.7	627.4	2105.74	617.1	1.6
7010	17.20	62.96	9.06	16.30	1.42	2.56	2140.7	627.4	2033.43	595.9	5.0
7101	20.79	69.42	8.28	14.90	1.68	3.02	1682.0	492.9	1673.94	490.6	0.5
7102	19.64	67.35	9.17	16.51	2.02	3.63	1987.8	582.6	1901.10	557.2	4.4
7103	19.43	66.98	8.83	15.89	1.91	3.43	1987.8	582.6	2039.73	597.8	-2.6
7104	19.63	67.34	9.23	16.61	2.05	3.69	2140.7	627.4	2197.91	644.1	-2.7
7105	19.81	67.65	9.64	17.36	2.27	4.09	2293.6	672.2	2367.54	693.9	-3.2
7106	20.33	68.59	9.77	17.59	2.33	4.20	2446.5	717.0	2476.57	725.8	-1.2
7107	20.59	69.07	9.92	17.85	2.45	4.40	2599.4	761.8	2592.26	759.7	0.3
7108	20.73	69.32	10.02	18.03	2.53	4.55	2752.3	806.6	2745.94	804.8	0.2
7109	20.93	69.68	10.18	18.33	2.57	4.62	2905.2	851.4	2869.07	840.8	1.2
7110	21.14	70.05	10.53	18.96	2.72	4.89	3058.1	896.3	3020.51	885.2	1.2
7201	18.86	65.94	9.48	17.06	1.83	3.29	1929.3	565.4	1930.54	565.8	-0.1
7202	20.06	68.11	8.86	15.95	1.84	3.31	1929.3	565.4	1847.07	541.3	4.3
7203	20.33	68.60	8.08	14.54	1.76	3.17	1780.9	521.9	1811.25	530.8	-1.7
7204	20.26	68.47	8.10	14.59	1.63	2.93	1929.3	565.4	1864.43	546.4	3.4
7205	20.42	68.77	7.62	13.72	1.76	3.16	1780.9	521.9	1847.42	541.4	-3.7
7206	20.44	68.79	7.37	13.27	1.71	3.07	1929.3	565.4	1822.65	534.2	5.5
7207	19.64	67.35	7.78	14.01	2.02	3.63	1929.3	565.4	2002.15	586.8	-3.8
7208	19.17	66.52	7.67	13.81	1.89	3.40	2077.7	608.9	2023.17	592.9	2.6
7209	19.20	66.56	7.21	12.97	1.88	3.38	1929.3	565.4	1960.24	574.5	-1.6
7210	18.93	66.08	7.05	12.69	1.80	3.25	1929.3	565.4	1986.41	582.2	-3.0
7301	17.80	64.04	8.49	15.29	1.73	3.12	1726.9	506.1	1658.63	486.1	4.0
7302	17.91	64.24	9.14	16.45	1.85	3.33	1834.9	537.8	1862.50	545.8	-1.5
7303	18.36	65.05	9.45	17.01	2.04	3.68	1753.9	514.0	1828.48	535.9	-4.3
7304	18.76	65.76	9.54	17.17	2.29	4.11	1762.9	516.7	1819.98	533.4	-3.2
7305	18.81	65.86	9.73	17.52	2.32	4.18	1870.9	548.3	1811.21	530.8	3.2
7306	18.94	66.09	9.68	17.42	2.28	4.11	1834.9	537.8	1802.97	528.4	1.7
7307	19.24	66.63	9.56	17.22	2.32	4.18	1780.9	521.9	1777.97	521.1	0.2
7308	19.64	67.35	9.44	17.00	2.36	4.25	1709.0	500.8	1709.42	501.0	-0.0
7309	20.09	68.17	9.43	16.98	2.39	4.31	1700.0	498.2	1683.66	493.4	1.0
7310	20.42	68.75	9.58	17.24	2.42	4.36	1726.9	506.1	1675.16	490.9	3.0
7401	22.51	72.52	11.37	20.46	3.58	6.45	2734.3	801.4	2742.48	803.7	-0.3
7402	22.06	71.70	11.04	19.87	3.48	6.27	2752.3	806.6	2634.02	772.0	4.3
7403	21.78	71.20	11.22	20.19	3.43	6.17	2878.2	843.5	2788.97	817.4	3.1
7404	21.43	70.58	11.27	20.29	3.41	6.15	2968.2	869.9	2835.45	831.0	4.5
7405	21.24	70.23	11.23	20.22	3.41	6.14	2914.2	854.1	2881.93	844.6	1.1
7406	20.82	69.47	11.25	20.24	3.35	6.03	3040.2	891.0	2928.41	858.2	3.7
7407	20.66	69.18	11.30	20.34	3.38	6.08	2896.2	848.8	3020.51	885.2	-4.3
7408	20.38	68.68	11.18	20.13	3.15	5.67	2968.2	869.9	3066.98	898.8	-3.3
7409	20.26	68.46	11.03	19.85	3.02	5.44	3166.1	927.9	3051.93	894.4	3.6
7410	20.03	68.06	11.03	19.85	3.02	5.44	3058.1	896.3	3082.47	903.4	-0.3

TABLE C.2 (Contd.)

Run #	Twall deg.C	Twall deg.F	LMTD deg.C	LMTD deg.F	LMTD(Outside) deg.C	LMTD(Outside) deg.F	Heat gain Btu/hr	Heat gain W	Heat lost Btu/hr	Heat lost W	% error
8001	16.59	61.87	15.73	28.32	0.88	1.58	1223.3	358.5	1359.94	398.6	-11.2
8002	16.56	61.81	15.65	28.17	0.89	1.60	1529.1	448.1	1419.29	416.0	7.2
8003	16.75	62.15	16.66	29.99	0.98	1.76	1529.1	448.1	1578.44	462.6	-3.2
8004	16.62	61.91	16.84	30.31	1.09	1.96	1807.9	529.8	1700.60	498.4	5.9
8005	16.70	62.06	17.26	31.07	1.17	2.11	1834.9	537.8	1822.87	534.2	0.7
8006	16.66	61.99	16.96	30.52	1.24	2.23	1834.9	537.8	1845.85	541.0	-0.6
8007	16.70	62.06	17.42	31.35	1.32	2.37	2140.7	627.4	1987.83	582.6	7.1
8008	16.83	62.30	17.92	32.26	1.45	2.62	2140.7	627.4	2133.25	625.2	0.3
8009	16.85	62.33	18.06	32.52	1.47	2.65	2172.2	636.6	2267.14	664.4	-4.4
8010	16.98	62.56	18.16	32.68	1.64	2.96	2446.5	717.0	2355.81	690.4	3.7
8101	18.06	64.50	15.68	28.22	1.25	2.26	1376.2	403.3	1291.76	378.6	6.1
8102	18.16	64.69	15.77	28.39	1.26	2.27	1376.2	403.3	1353.76	396.7	1.6
8103	18.32	64.97	16.17	29.10	1.42	2.56	1416.6	415.2	1487.31	435.9	-5.0
8104	18.26	64.87	16.50	29.71	1.51	2.71	1529.1	448.1	1559.34	457.0	-2.0
8105	18.42	65.16	16.53	29.76	1.51	2.72	1682.0	492.9	1659.12	486.2	-1.4
8106	18.51	65.31	16.37	29.47	1.59	2.87	1682.0	492.9	1720.72	504.3	-2.3
8107	18.58	65.44	16.74	30.13	1.64	2.95	1917.6	562.0	1823.95	534.5	4.9
8108	18.77	65.78	16.85	30.32	1.74	3.13	1987.8	582.6	1908.51	559.3	4.0
8109	19.03	66.24	17.20	30.96	2.01	3.61	1987.8	582.6	2029.24	594.7	-2.1
8110	19.05	66.29	16.86	30.34	1.97	3.54	2140.7	627.4	2059.25	603.5	3.8
8201	18.43	65.18	18.46	33.23	1.62	2.92	1632.5	478.4	1648.21	483.0	-1.0
8202	18.39	65.11	18.45	33.21	1.62	2.92	1780.9	521.9	1696.84	497.3	4.7
8203	18.41	65.14	18.17	32.70	1.60	2.88	1632.5	478.4	1698.10	497.7	-4.0
8204	18.39	65.10	17.38	31.28	1.61	2.90	1748.5	512.4	1677.72	491.7	4.0
8205	18.25	64.85	16.64	29.95	1.57	2.83	1759.3	515.6	1693.48	496.3	3.7
8206	18.12	64.61	15.72	28.29	1.50	2.70	1632.5	478.4	1647.16	482.7	-0.9
8207	18.14	64.65	15.40	27.73	1.63	2.93	1632.5	478.4	1644.36	481.9	-0.7
8208	18.28	64.91	15.20	27.37	1.61	2.89	1759.3	515.6	1685.86	494.1	4.2
8209	18.28	64.89	14.51	26.11	1.56	2.81	1652.3	484.2	1662.11	487.1	-0.6
8210	18.17	64.70	13.91	25.04	1.55	2.80	1652.3	484.2	1641.92	481.2	0.6
8301	17.94	64.29	15.02	27.03	1.19	2.15	1223.3	358.5	1206.24	353.5	1.4
8302	18.05	64.49	14.55	26.19	1.14	2.05	1262.8	370.1	1206.42	353.6	4.5
8303	18.49	65.28	15.16	27.29	1.29	2.31	1214.3	355.9	1223.06	358.4	-0.7
8304	18.59	65.47	15.23	27.41	1.33	2.40	1259.2	369.0	1223.41	358.5	2.8
8305	18.92	66.05	15.12	27.21	1.33	2.39	1262.8	370.1	1214.56	356.0	3.8
8306	19.13	66.44	14.98	26.96	1.41	2.54	1217.0	356.7	1223.06	358.4	-0.5
8307	19.44	66.99	15.11	27.19	1.49	2.69	1251.1	366.7	1197.58	351.0	4.3
8308	19.73	67.51	15.20	27.37	1.52	2.73	1221.5	358.0	1180.59	346.0	3.3
8309	20.16	68.28	14.94	26.90	1.63	2.93	1151.3	337.4	1163.43	341.0	-1.1
8310	20.58	69.05	15.02	27.04	1.65	2.98	1196.3	350.6	1137.96	333.5	4.9
8401	23.10	73.58	13.36	24.05	2.15	3.87	1448.1	424.4	1499.25	439.4	-3.5
8402	22.63	72.73	14.31	25.76	2.21	3.98	1548.9	453.9	1592.21	466.6	-2.8
8403	22.18	71.91	14.46	26.02	2.12	3.82	1619.0	474.5	1669.26	489.2	-3.1
8404	21.71	71.07	14.72	26.50	2.18	3.93	1651.4	484.0	1700.17	498.3	-3.0
8405	21.31	70.35	14.87	26.76	1.97	3.55	1780.9	521.9	1715.88	502.9	3.7
8406	20.99	69.79	15.03	27.06	1.92	3.46	1834.9	537.8	1762.26	516.5	4.0
8407	20.81	69.46	15.19	27.35	1.97	3.54	1753.9	514.0	1792.91	525.5	-2.2
8408	20.53	68.95	15.17	27.31	1.84	3.31	1762.9	516.7	1808.37	530.0	-2.6
8409	20.14	68.26	15.21	27.38	1.82	3.27	1753.9	514.0	1809.42	530.3	-3.2
8410	19.91	67.84	15.44	27.79	1.78	3.21	1870.9	548.3	1855.27	543.7	0.8

TABLE C.2 (Contd.)

Run #	Twall deg.C	Twall deg.F	LMTD deg.C	LMTD deg.F	LMTD(Outside) deg.C	LMTD(Outside) deg.F	Heat gain Btu/hr	Heat gain W	Heat lost Btu/hr	Heat lost W	% error
9001	18.58	65.44	10.41	18.73	2.12	3.81	3669.8	1075.5	3519.74	1031.5	4.1
9002	18.09	64.56	9.93	17.87	1.84	3.32	3058.1	896.3	3049.94	893.9	0.3
9003	17.71	63.88	9.63	17.33	1.62	2.92	2752.3	806.6	2759.38	808.7	-0.3
9004	17.73	63.92	9.04	16.28	1.29	2.32	2446.5	717.0	2412.35	707.0	1.4
9005	17.74	63.93	8.79	15.82	1.15	2.07	2140.7	627.4	2139.72	627.1	0.0
9006	17.88	64.18	9.17	16.51	1.29	2.33	2140.7	627.4	2218.66	650.2	-3.6
9007	18.12	64.61	9.11	16.41	1.38	2.48	2446.5	717.0	2479.99	726.8	-1.4
9008	18.25	64.85	9.45	17.01	1.56	2.80	2752.3	806.6	2772.86	812.6	-0.7
9009	18.49	65.29	9.68	17.42	1.75	3.14	3058.1	896.3	3085.74	904.3	-0.9
9010	17.70	63.86	8.69	15.64	1.17	2.11	1834.9	537.8	1917.84	562.1	-4.5
9101	21.50	70.70	10.19	18.34	2.53	4.56	2905.2	851.4	2888.19	846.4	0.6
9102	21.17	70.10	9.84	17.71	2.14	3.86	2752.3	806.6	2654.20	777.9	3.6
9103	21.45	70.61	9.93	17.88	2.27	4.08	2905.2	851.4	2802.06	821.2	3.6
9104	20.47	68.84	10.19	18.35	2.14	3.86	2752.3	806.6	2665.79	781.3	3.1
9105	20.25	68.45	9.80	17.64	1.94	3.49	2446.5	717.0	2483.54	727.9	-1.5
9106	20.18	68.33	9.78	17.60	1.87	3.36	2446.5	717.0	2373.96	695.7	3.0
9107	20.10	68.18	9.45	17.02	1.81	3.26	2140.7	627.4	2225.53	652.2	-4.0
9108	20.24	68.43	8.89	16.00	1.60	2.88	1987.8	582.6	2004.31	587.4	-0.8
9109	20.06	68.11	8.54	15.37	1.48	2.67	1834.9	537.8	1868.64	547.6	-1.8
9110	19.99	67.98	8.49	15.29	1.47	2.65	1682.0	492.9	1737.34	509.2	-3.3
9201	19.46	67.03	6.52	11.74	1.48	2.67	1780.9	521.9	1824.09	534.6	-2.4
9202	19.49	67.08	6.58	11.85	1.40	2.53	1780.9	521.9	1820.73	533.6	-2.2
9203	19.49	67.08	7.05	12.69	1.51	2.72	1780.9	521.9	1822.01	534.0	-2.3
9204	19.56	67.20	7.04	12.66	1.48	2.66	1780.9	521.9	1825.34	535.0	-2.5
9205	19.56	67.21	7.30	13.14	1.48	2.67	1780.9	521.9	1828.67	535.9	-2.7
9206	19.43	66.98	7.62	13.72	1.55	2.80	1780.9	521.9	1841.26	539.6	-3.4
9207	19.59	67.27	8.05	14.48	1.55	2.80	1929.3	565.4	1836.87	538.3	4.8
9208	19.52	67.13	8.28	14.90	1.54	2.78	1780.9	521.9	1825.53	535.0	-2.5
9209	19.64	67.36	8.58	15.44	1.50	2.70	1929.3	565.4	1841.43	539.7	4.5
9210	19.53	67.14	8.85	15.92	1.44	2.60	1780.9	521.9	1764.06	517.0	0.9
9301	19.29	66.72	9.49	17.08	2.02	3.64	1655.0	485.0	1719.67	504.0	-3.9
9302	19.33	66.80	9.74	17.53	2.15	3.86	1780.9	521.9	1744.70	511.3	2.0
9303	18.91	66.04	9.64	17.36	1.98	3.56	1888.9	553.6	1787.26	523.8	5.4
9304	18.59	65.47	9.57	17.22	1.96	3.52	1780.9	521.9	1821.04	533.7	-2.3
9305	18.50	65.30	9.76	17.57	2.03	3.66	1834.9	537.8	1829.28	536.1	0.3
9306	18.15	64.67	9.68	17.42	1.84	3.30	1870.9	548.3	1880.33	551.1	-0.5
9307	18.07	64.52	9.61	17.30	1.82	3.27	1888.8	553.6	1897.07	556.0	-0.4
9308	18.06	64.50	9.64	17.35	1.80	3.25	1956.3	573.3	1905.58	558.5	2.5
9309	17.71	63.88	9.75	17.55	1.72	3.09	2014.8	590.5	1930.82	565.9	4.2
9310	17.46	63.42	9.56	17.21	1.62	2.92	1987.8	582.6	1956.62	573.4	1.6
9401	21.73	71.10	11.26	20.27	3.30	5.94	2806.3	822.4	2714.23	795.5	3.3
9402	21.43	70.57	11.36	20.44	3.26	5.88	2905.2	851.4	2791.38	818.1	3.9
9403	20.75	69.35	11.57	20.82	3.22	5.80	2968.2	869.9	2916.70	854.8	1.7
9404	20.29	68.52	11.57	20.82	3.15	5.68	2968.2	869.9	2978.76	873.0	-0.4
9405	19.89	67.81	11.50	20.69	3.08	5.55	3022.2	885.7	3025.30	886.6	-0.1
9406	19.60	67.28	11.48	20.67	3.02	5.43	3040.2	891.0	3102.87	909.4	-2.1
9407	19.23	66.62	11.30	20.35	2.90	5.23	3148.1	922.6	3118.39	913.9	0.9
9408	18.99	66.18	11.28	20.30	2.82	5.07	3238.0	949.0	3149.42	923.0	2.7
9409	18.70	65.66	11.18	20.13	2.76	4.96	3166.1	927.9	3195.42	936.8	-1.0
9410	18.49	65.28	11.09	19.96	2.60	4.68	3211.0	941.1	3196.42	936.8	0.5

TABLE C.3 :- REDUCED DATA - INSIDE AND OUTSIDE HEAT TRANSFER COEFFICIENTS AND HEAT DUTY (UA)

Run	Inside HTC		Outside HTC		UA	
#	Btu/hr sqft F	W/sq.m C	Btu/hr sqft F	W/sq.m C	Btu/hr F	W/ C
1001	147.34	836.61	478.21	2715.42	148.17	78.16
1002	155.16	881.02	453.71	2576.31	152.58	80.49
1003	164.55	934.35	462.44	2625.84	160.35	84.59
1004	175.23	995.00	472.85	2684.95	169.12	89.22
1005	186.86	1061.02	473.85	2690.65	177.62	93.70
1006	186.84	1060.93	433.46	2461.32	173.59	91.57
1007	199.77	1134.35	447.65	2541.87	183.85	96.99
1008	194.54	1104.63	427.05	2424.90	178.04	93.92
1009	202.16	1147.93	416.24	2363.54	181.68	95.84
1010	216.96	1231.97	432.28	2454.60	193.09	101.86
1101	141.56	803.83	377.03	2140.88	136.26	71.88
1102	147.54	837.76	345.01	1959.08	137.43	72.50
1103	160.88	913.51	351.08	1993.54	147.06	77.58
1104	167.86	953.18	361.22	2051.09	152.83	80.62
1105	173.99	987.97	353.62	2007.94	155.83	82.21
1106	173.40	984.63	348.89	1981.10	154.85	81.69
1107	184.95	1050.19	347.02	1970.49	161.70	85.30
1108	187.40	1064.09	342.64	1945.59	162.52	85.73
1109	196.56	1116.10	326.78	1855.55	165.27	87.18
1110	207.50	1178.24	351.66	1996.81	175.56	92.62
1201	131.40	746.13	334.01	1896.62	125.07	65.98
1202	138.76	787.90	354.72	2014.18	132.24	69.76
1203	146.86	833.92	345.34	1960.93	137.00	72.27
1204	152.59	866.45	344.04	1953.56	140.75	74.25
1205	163.38	927.69	361.75	2054.09	149.94	79.10
1206	172.75	980.95	346.65	1968.35	154.15	81.32
1207	175.73	997.86	345.34	1960.93	155.83	82.21
1208	178.47	1013.39	341.48	1938.99	156.98	82.81
1209	186.52	1059.09	357.49	2029.95	164.13	86.58
1210	191.15	1085.42	354.72	2014.17	166.54	87.85
1301	195.98	1112.81	316.73	1798.45	163.21	86.10
1302	198.16	1125.19	326.30	1852.83	166.07	87.61
1303	203.33	1154.55	344.10	1953.90	171.96	90.72
1304	207.15	1176.23	365.97	2078.09	177.66	93.72
1305	206.60	1173.14	375.86	2134.25	178.85	94.35
1306	208.84	1185.83	385.60	2189.55	181.62	95.81
1307	204.68	1162.22	388.92	2208.42	179.60	94.75
1308	216.14	1227.32	410.90	2333.19	189.66	100.05
1309	212.19	1204.85	410.66	2331.86	187.21	98.76
1310	222.29	1262.24	450.18	2556.25	198.77	104.86
1401	144.80	822.23	343.13	1948.37	135.36	71.41
1402	145.42	825.74	369.40	2097.52	138.36	72.99
1403	143.25	813.44	348.98	1981.59	134.88	71.15
1404	138.68	787.47	345.86	1963.91	131.36	69.30
1405	145.58	826.66	364.18	2067.92	137.99	72.79
1406	144.76	822.01	376.43	2137.45	138.53	73.08
1407	141.17	801.60	386.62	2195.34	136.79	72.16
1408	147.62	838.23	411.05	2334.03	143.58	75.74
1409	141.67	804.46	389.72	2212.92	137.41	72.49
1410	134.85	765.70	385.72	2190.25	131.99	69.63

TABLE C.3 (Contd.)

Run #	Inside HTC Btu/hr sqft F	Inside HTC W/sq.m C	Outside HTC Btu/hr sqft F	Outside HTC W/sq.m C	UA Btu/hr F	UA W/ C
2001	46.58	264.49	549.71	3121.40	52.26	27.57
2002	52.67	299.08	546.93	3105.63	58.58	30.90
2003	58.59	332.70	591.92	3361.09	65.03	34.30
2004	59.34	336.97	556.64	3160.76	65.49	34.55
2005	65.13	369.80	545.18	3095.69	71.19	37.55
2006	65.97	374.59	570.01	3236.65	72.31	38.14
2007	73.95	419.90	594.67	3376.70	80.53	42.48
2008	74.73	424.36	594.67	3376.71	81.31	42.89
2009	75.54	428.96	513.97	2918.47	80.93	42.70
2010	81.37	462.03	555.82	3156.10	87.20	46.00
2101	47.92	272.08	387.51	2200.37	52.23	27.56
2102	56.13	318.74	426.34	2420.86	60.83	32.09
2103	56.21	319.16	410.90	2333.17	60.68	32.01
2104	60.71	344.71	428.93	2435.57	65.31	34.45
2105	63.38	359.90	406.93	2310.66	67.50	35.61
2106	64.87	368.36	412.76	2343.77	69.01	36.41
2107	67.16	381.35	395.83	2247.62	70.81	37.36
2108	73.62	418.01	396.59	2251.93	76.75	40.49
2109	72.60	412.26	386.53	2194.80	75.58	39.87
2110	78.11	443.50	393.27	2233.11	80.69	42.57
2201	48.04	272.76	377.92	2145.91	52.23	27.56
2202	52.32	297.10	399.91	2270.78	56.74	29.93
2203	56.15	318.85	396.12	2249.27	60.41	31.87
2204	53.77	305.31	370.59	2104.33	57.71	30.44
2205	60.05	341.00	416.59	2365.50	64.49	34.02
2206	61.79	350.83	382.95	2174.48	65.54	34.57
2207	64.71	367.46	384.81	2185.05	68.31	36.03
2208	68.66	389.90	420.05	2385.17	72.72	38.36
2209	75.62	429.41	439.11	2493.37	79.58	41.98
2210	74.83	424.93	440.12	2499.10	78.87	41.61
2301	81.69	463.85	370.89	2106.02	83.15	43.86
2302	78.18	443.94	367.49	2086.70	79.99	42.20
2303	78.56	446.06	386.24	2193.16	80.89	42.67
2304	83.10	471.85	432.66	2456.75	86.23	45.49
2305	82.99	471.25	450.94	2560.53	86.60	45.69
2306	81.41	462.26	443.28	2517.08	84.98	44.83
2307	79.63	452.16	448.38	2546.03	83.48	44.04
2308	78.49	445.66	451.89	2565.97	82.50	43.52
2309	80.89	459.31	494.58	2808.35	85.64	45.18
2310	78.87	447.83	484.33	2750.13	83.54	44.07
2401	49.35	280.22	376.59	2138.40	53.51	28.23
2402	48.07	272.95	380.53	2160.78	52.30	27.59
2403	51.13	290.32	415.85	2361.32	55.76	29.42
2404	49.42	280.64	442.69	2513.72	54.35	28.67
2405	52.03	295.43	462.76	2627.65	57.18	30.16
2406	49.18	279.24	455.50	2586.47	54.23	28.61
2407	51.71	293.65	503.07	2856.59	57.24	30.20
2408	48.22	273.83	445.31	2528.61	53.17	28.05
2409	50.73	288.08	521.36	2960.39	56.38	29.74
2410	48.33	274.43	488.39	2773.20	53.65	28.30

TABLE C.3 (Contd.)

Run #	Inside HTC Btu/hr sqft F	Inside HTC W/sq.m C	Outside HTC Btu/hr sqft F	Outside HTC W/sq.m C	UA Btu/hr F	UA W/ C
3001	153.29	870.41	606.48	3443.75	150.44	79.36
3002	166.60	946.00	612.86	3480.00	161.42	85.15
3003	176.73	1003.49	664.60	3773.79	171.87	90.66
3004	176.23	1000.68	578.08	3282.48	167.11	88.15
3005	184.70	1048.77	587.39	3335.35	174.05	91.81
3006	197.91	1123.80	597.63	3393.49	184.49	97.33
3007	200.56	1138.82	631.30	3584.71	188.55	99.47
3008	209.25	1188.20	632.15	3589.53	195.05	102.89
3009	219.80	1248.10	630.83	3582.01	202.62	106.89
3010	230.66	1309.76	633.54	3597.41	210.54	111.07
3101	144.73	821.79	451.24	2562.26	135.92	71.70
3102	147.49	837.50	437.40	2483.65	137.08	72.32
3103	163.14	926.34	442.38	2511.95	148.64	78.41
3104	168.17	954.89	473.85	2690.65	154.53	81.52
3105	175.71	997.74	465.36	2642.43	159.20	83.98
3106	181.39	1029.98	464.62	2638.22	163.05	86.01
3107	182.97	1038.98	442.24	2511.13	162.17	85.55
3108	188.81	1072.11	442.79	2514.29	166.07	87.61
3109	202.71	1151.03	492.29	2795.35	179.82	94.86
3110	209.18	1187.79	453.49	2575.01	180.19	95.06
3201	135.43	768.99	366.27	2079.79	123.37	65.08
3202	151.31	859.21	514.53	2921.62	144.54	76.25
3203	149.92	851.27	445.15	2527.67	139.37	73.52
3204	162.88	924.88	448.13	2544.62	148.89	78.55
3205	161.36	916.24	455.14	2584.43	148.32	78.25
3206	180.32	1023.89	502.96	2855.94	165.29	87.20
3207	170.89	970.37	475.19	2698.28	156.57	82.59
3208	190.01	1078.92	508.67	2888.37	172.54	91.02
3209	189.76	1077.51	500.14	2839.94	171.70	90.57
3210	198.42	1126.66	517.51	2938.57	179.07	94.46
3301	202.68	1150.89	413.56	2348.31	171.73	90.59
3302	200.93	1140.92	414.07	2351.19	170.72	90.06
3303	206.35	1171.71	437.40	2483.65	176.68	93.21
3304	217.57	1235.41	490.07	2782.77	189.30	99.86
3305	211.38	1200.27	483.95	2747.98	184.71	97.44
3306	215.44	1223.33	497.75	2826.35	188.69	99.54
3307	225.13	1278.34	529.50	3006.62	198.04	104.47
3308	220.37	1251.34	513.39	2915.18	193.41	102.03
3309	217.05	1232.49	525.90	2986.17	192.39	101.49
3310	227.04	1289.17	553.92	3145.33	201.55	106.32
3401	146.48	831.74	407.16	2311.97	134.24	70.82
3402	148.20	841.52	465.76	2644.70	139.40	73.54
3403	143.51	814.90	458.52	2603.58	135.45	71.46
3404	145.38	825.52	459.65	2610.02	136.93	72.23
3405	146.39	831.22	474.14	2692.29	138.53	73.08
3406	142.15	807.18	461.35	2619.65	134.59	71.00
3407	152.73	867.27	510.31	2897.66	145.40	75.71
3408	149.10	846.64	507.82	2883.56	142.48	75.16
3409	153.58	872.07	571.74	3246.51	149.15	78.68
3410	146.31	830.81	534.26	3033.66	141.62	74.71

TABLE C.3 (Contd.)

Run #	Inside HTC Btu/hr sqft F	Inside HTC W/sq.m C	Outside HTC Btu/hr sqft F	Outside HTC W/sq.m C	UA Btu/hr F	UA W/ C
7001	121.26	688.55	378.61	2149.85	179.62	94.76
7002	120.17	682.37	390.88	2219.51	179.56	94.73
7003	112.55	639.08	371.41	2108.97	168.70	88.99
7004	103.17	585.84	358.20	2033.97	156.26	82.43
7005	104.99	596.17	378.13	2147.13	160.15	84.48
7006	96.16	546.02	339.38	1927.08	146.13	77.09
7007	93.59	531.46	376.33	2136.90	145.71	76.87
7008	94.45	536.33	390.02	2214.62	147.73	77.93
7009	83.99	476.90	355.20	2016.91	131.92	69.59
7010	82.49	468.38	379.07	2152.48	131.33	69.28
7101	74.94	425.52	252.32	1432.74	112.86	59.54
7102	81.74	464.12	247.91	1407.68	120.40	63.51
7103	84.48	479.72	262.22	1488.98	125.07	65.98
7104	87.75	498.25	262.54	1490.79	128.85	67.97
7105	91.57	519.94	253.82	1441.28	132.12	69.70
7106	96.83	549.81	263.80	1497.90	139.12	73.39
7107	102.42	581.55	267.48	1518.80	145.64	76.83
7108	108.15	614.10	274.33	1557.74	152.56	80.53
7109	112.26	637.47	285.08	1618.77	158.50	83.62
7110	115.19	654.09	283.42	1609.33	161.32	85.10
7201	74.19	421.25	265.37	1506.84	113.07	59.65
7202	80.77	458.61	264.43	1501.49	120.93	63.79
7203	82.96	471.06	254.15	1443.13	122.46	64.60
7204	87.65	497.70	298.50	1694.95	132.28	69.78
7205	89.34	507.32	255.09	1448.47	129.81	68.48
7206	100.16	568.74	284.69	1616.56	145.35	76.67
7207	98.48	559.18	240.62	1366.28	137.70	72.64
7208	105.78	600.67	276.80	1571.73	150.49	79.39
7209	106.51	604.78	258.92	1470.23	148.72	78.45
7210	108.20	614.37	269.21	1528.63	151.98	80.18
7301	75.12	426.54	251.08	1425.67	112.96	59.59
7302	73.99	420.16	249.91	1419.05	111.51	58.82
7303	69.66	395.57	215.97	1226.35	103.13	54.40
7304	71.50	406.00	194.17	1102.56	102.70	54.18
7305	74.23	421.49	202.86	1151.87	106.77	56.33
7306	72.98	414.39	202.30	1148.70	105.33	55.57
7307	72.33	410.68	193.01	1095.99	103.44	54.57
7308	70.97	403.00	182.15	1034.30	100.54	53.04
7309	71.02	403.27	178.88	1015.70	100.13	52.82
7310	71.00	403.17	179.33	1018.27	100.18	52.85
7401	103.37	586.98	192.09	1090.76	133.63	70.49
7402	107.25	608.98	198.84	1129.05	138.53	73.08
7403	108.78	617.66	211.23	1199.40	142.54	75.19
7404	111.22	631.55	218.81	1242.45	146.32	77.19
7405	109.65	622.61	215.03	1220.97	144.11	76.02
7406	113.36	643.69	228.26	1296.10	150.17	79.22
7407	107.62	611.11	215.89	1225.85	142.42	75.13
7408	108.75	617.53	237.33	1347.62	147.48	77.80
7409	116.46	661.30	263.52	1496.34	159.51	84.14
7410	112.43	638.38	254.73	1446.43	154.04	81.25

TABLE C.3 (Contd.)

Run #	Inside HTC Btu/hr sqft F	Inside HTC W/sq.m C	Outside HTC Btu/hr sqft F	Outside HTC W/sq.m C	UA Btu/hr F	UA W/ C
8001	26.58	150.91	381.38	2165.57	43.20	22.79
8002	33.43	189.83	471.06	2674.80	54.28	28.63
8003	31.47	178.69	427.93	2429.90	50.99	26.90
8004	37.06	210.44	453.77	2576.62	59.66	31.47
8005	36.81	209.02	427.93	2429.91	59.05	31.15
8006	37.68	213.95	405.89	2304.72	60.12	31.71
8007	42.92	243.72	444.52	2524.10	68.27	36.02
8008	41.97	238.29	403.51	2291.24	66.37	35.01
8009	42.26	239.97	404.13	2294.76	66.81	35.24
8010	47.82	271.55	407.98	2316.64	74.85	39.49
8101	30.79	174.82	300.90	1708.59	48.77	25.73
8102	30.60	173.78	299.34	1699.74	48.48	25.57
8103	31.01	176.06	272.98	1550.05	48.68	25.68
8104	32.91	186.86	277.83	1577.58	51.47	27.15
8105	36.14	205.20	305.38	1734.05	56.53	29.82
8106	36.73	208.55	289.15	1641.86	57.07	30.10
8107	40.99	232.75	320.30	1818.77	63.64	33.57
8108	42.47	241.17	313.22	1778.52	65.55	34.58
8109	42.23	239.79	271.55	1541.92	64.21	33.87
8110	46.42	263.58	298.02	1692.23	70.56	37.22
8201	31.28	177.61	276.14	1568.00	49.12	25.91
8202	34.16	193.96	301.08	1709.59	53.63	28.29
8203	31.80	180.55	279.50	1587.10	49.92	26.33
8204	35.79	203.25	296.80	1685.31	55.89	29.48
8205	37.70	214.08	306.09	1738.04	58.75	30.99
8206	37.06	210.46	297.69	1690.37	57.70	30.44
8207	38.25	217.17	275.04	1561.73	58.88	31.06
8208	41.77	237.16	299.85	1702.62	64.28	33.91
8209	41.20	233.93	289.97	1646.51	63.27	33.38
8210	43.16	245.05	291.18	1653.41	65.97	34.80
8301	28.56	162.15	280.85	1594.73	45.26	23.87
8302	30.39	172.57	303.06	1720.84	48.22	25.44
8303	28.24	160.35	258.77	1469.36	44.49	23.47
8304	29.24	166.02	259.02	1470.76	45.93	24.23
8305	29.56	167.86	260.16	1477.23	46.41	24.48
8306	28.94	164.35	236.35	1342.05	45.14	23.81
8307	29.66	168.42	229.34	1302.24	46.01	24.27
8308	28.81	163.56	220.28	1250.81	44.63	23.55
8309	27.90	158.44	193.82	1100.54	42.81	22.58
8310	28.87	163.95	198.23	1125.57	44.24	23.34
8401	41.69	236.75	184.62	1048.33	60.22	31.77
8402	41.32	234.63	191.87	1089.52	60.13	31.72
8403	42.36	240.56	209.19	1187.85	62.22	32.82
8404	42.51	241.38	207.13	1176.14	62.31	32.87
8405	44.58	253.11	247.50	1405.35	66.54	35.10
8406	45.18	256.52	261.80	1486.57	67.81	35.77
8407	42.81	243.07	244.12	1386.17	64.13	33.83
8408	42.68	242.33	262.90	1492.80	64.55	34.05
8409	42.27	240.00	264.44	1501.56	64.05	33.79
8410	44.24	251.19	287.23	1630.96	67.33	35.52

TABLE C.3 (Contd.)

Run #	Inside HTC Btu/hr sqft F	Inside HTC W/sq.m C	Outside HTC Btu/hr sqft F	Outside HTC W/sq.m C	UA Btu/hr F	UA W/ C
9001	142.59	809.64	472.37	2682.26	195.89	103.34
9002	121.75	691.30	452.65	2570.26	171.11	90.27
9003	110.62	628.15	463.03	2629.19	158.80	83.77
9004	101.52	576.44	517.14	2936.46	150.27	79.27
9005	90.18	512.05	506.65	2876.89	135.31	71.38
9006	87.38	496.16	451.30	2562.62	129.63	68.38
9007	101.78	577.94	483.36	2744.67	149.11	78.66
9008	112.25	637.39	482.12	2737.59	161.83	85.37
9009	124.12	704.77	477.58	2711.85	175.54	92.60
9010	78.54	445.95	426.03	2419.11	117.32	61.89
9101	122.13	693.50	312.88	1776.60	158.39	83.56
9102	115.17	653.99	350.05	1987.66	155.45	82.01
9103	122.02	692.83	349.20	1982.82	162.47	85.71
9104	110.03	624.78	350.05	1987.66	149.98	79.12
9105	100.19	568.89	343.68	1951.52	138.71	73.17
9106	99.49	564.96	357.09	2027.65	138.97	73.31
9107	90.10	511.60	322.42	1830.79	125.79	66.36
9108	87.71	498.01	338.72	1923.33	124.20	65.52
9109	83.67	475.10	337.56	1916.74	119.42	63.00
9110	77.02	437.33	312.14	1772.44	110.02	58.04
9201	113.69	645.57	327.63	1860.36	151.65	80.00
9202	110.63	628.19	346.18	1965.68	150.27	79.27
9203	103.41	587.18	321.90	1827.85	140.33	74.03
9204	103.07	585.25	329.09	1868.68	140.62	74.18
9205	98.47	559.12	327.63	1860.36	135.50	71.48
9206	94.44	536.24	312.37	1773.72	129.81	68.48
9207	95.62	542.98	338.54	1922.35	133.22	70.28
9208	85.06	483.02	315.03	1788.84	119.55	63.07
9209	87.73	498.17	350.52	1990.34	124.99	65.94
9210	77.36	439.28	336.63	1911.49	111.84	59.00
9301	71.28	404.75	222.96	1266.05	96.88	51.11
9302	75.42	428.26	226.26	1284.77	101.58	53.58
9303	79.26	450.08	260.12	1477.03	108.80	57.40
9304	75.27	427.42	248.08	1408.68	103.42	54.56
9305	76.34	433.49	246.15	1397.72	104.42	55.08
9306	76.75	435.78	278.01	1578.64	107.43	56.67
9307	77.92	442.45	283.55	1610.05	109.16	57.59
9308	80.29	455.90	295.83	1679.79	112.74	59.47
9309	80.69	458.20	319.73	1815.53	114.81	60.57
9310	80.53	457.28	334.58	1899.83	115.52	60.94
9401	113.41	643.99	231.95	1317.07	138.41	73.02
9402	115.54	656.09	242.71	1378.15	142.11	74.97
9403	114.47	650.01	251.14	1426.01	142.53	75.19
9404	113.52	644.62	256.72	1457.71	142.54	75.19
9405	115.65	656.70	267.18	1517.10	146.05	77.04
9406	115.57	656.25	274.95	1561.23	147.09	77.60
9407	120.65	685.09	295.63	1678.66	154.72	81.62
9408	123.27	699.94	313.22	1778.57	159.54	84.16
9409	120.91	686.58	313.39	1779.52	157.27	82.96
9410	121.72	691.15	337.13	1914.30	160.86	84.86

TABLE C.4 :- REDUCED DATA - MASS FLUX AND REYNOLDS NUMBER

Run	Mass flux				Reynolds number	
	Water		R-113		Water	R-113
#	lbm/sqft hr	kg/sq.m hr	lbm/sqft hr	kg/sq.m hr		
1001	123183.3	601433.6	144715.4	706562.6	3218.31	3968.7
1002	123183.3	601433.6	157963.0	771243.0	3222.36	4372.8
1003	123183.3	601433.6	171201.0	835876.6	3226.40	4739.2
1004	123183.3	601433.6	184477.0	900695.7	3222.36	5110.2
1005	123183.3	601433.6	197682.5	965170.6	3226.40	5486.9
1006	123183.3	601433.6	210922.2	1029813.0	3226.41	5870.0
1007	123183.3	601433.6	224234.2	1094807.0	3214.27	6219.7
1008	125646.9	613462.0	237527.0	1159709.0	3270.33	6614.8
1009	123183.3	601433.6	250759.0	1224313.0	3210.23	7006.6
1010	123183.3	601433.6	264032.7	1289121.0	3214.27	7357.9
1101	73910.0	360860.1	144757.4	706767.6	1970.12	4009.9
1102	73910.0	360860.1	157985.9	771354.8	1972.59	4393.9
1103	73910.0	360860.1	171201.0	835876.6	1977.53	4783.6
1104	73910.0	360860.1	184397.0	900305.1	1984.97	5155.7
1105	73910.0	360860.1	197625.3	964891.3	1992.42	5540.3
1106	76373.6	372888.7	210830.8	1029366.0	2061.41	5910.5
1107	73910.0	360860.1	224072.5	1094018.0	1994.91	6277.6
1108	73910.0	360860.1	237321.7	1158706.0	1997.41	6679.6
1109	73910.0	360860.1	250542.4	1223255.0	1999.90	7065.8
1110	73910.0	360860.1	263766.9	1287823.0	2004.90	7453.5
1201	73910.0	360860.1	144757.4	706767.6	1952.92	3903.7
1202	73910.0	360860.1	157963.0	771243.0	1957.82	4208.2
1203	73910.0	360860.1	171225.7	835997.2	1957.82	4521.4
1204	73910.0	360860.1	184477.0	900695.7	1952.92	4831.6
1205	73910.0	360860.1	197653.9	965031.0	1962.74	5152.0
1206	73910.0	360860.1	210861.3	1029515.0	1967.66	5469.9
1207	73910.0	360860.1	224169.5	1094492.0	1962.74	5787.2
1208	73910.0	360860.1	237321.7	1158706.0	1972.59	6126.8
1209	73910.0	360860.1	250578.5	1223432.0	1967.66	6433.5
1210	73910.0	360860.1	263766.9	1287823.0	1972.59	6758.1
1301	64055.3	312745.4	263918.9	1288565.0	1728.92	7428.2
1302	73910.0	360860.1	263918.9	1288565.0	1987.45	7438.1
1303	78837.3	384917.4	263880.9	1288380.0	2117.30	7432.1
1304	83764.6	408974.7	263804.9	1288009.0	2252.44	7429.9
1305	88692.0	433032.2	263804.9	1288009.0	2376.01	7434.9
1306	93619.3	457089.4	263880.9	1288380.0	2492.37	7407.5
1307	101010.3	493175.5	263880.9	1288380.0	2689.13	7402.6
1308	103473.9	505203.9	263842.9	1288194.0	2751.27	7411.4
1309	108401.3	529261.6	263804.9	1288009.0	2878.68	7415.2
1310	113328.6	553318.8	263804.9	1288009.0	3009.53	7415.2
1401	64055.3	312745.4	144736.4	706665.1	1726.77	4049.5
1402	68982.6	336802.8	144694.4	706460.1	1857.27	4053.7
1403	72431.8	353643.0	144694.4	706460.1	1937.98	4040.3
1404	78837.3	384917.4	144736.4	706665.1	2093.58	4030.7
1405	83764.6	408974.7	144715.4	706562.6	2218.87	4035.5
1406	88692.0	433032.2	144757.4	706767.6	2346.44	4028.6
1407	93619.3	457089.4	144736.4	706665.1	2473.70	4028.0
1408	98546.6	481146.7	144694.4	706460.1	2603.89	4029.6
1409	103473.9	505203.9	144715.4	706562.6	2723.82	4017.5
1410	108401.3	529261.6	144715.4	706562.6	2849.95	4015.7

TABLE C.4 (Contd.)

Run	Mass flux				Reynolds number	
	Water		R-113		Water	R-113
#	lbm/sqft hr	kg/sq.m hr	lbm/sqft hr	kg/sq.m hr		
2001	123100.6	601029.9	159479.2	778645.8	3264.57	4380.3
2002	123100.6	601029.9	174103.7	850048.8	3252.32	4810.1
2003	123100.6	601029.9	188667.0	921153.1	3264.58	5249.9
2004	123100.6	601029.9	203268.2	992442.4	3264.58	5637.8
2005	120638.6	589009.3	217881.9	1063793.0	3195.28	6055.0
2006	123100.6	601029.9	232441.0	1134877.0	3268.67	6472.1
2007	123100.6	601029.9	246968.5	1205806.0	3272.77	6876.7
2008	123100.6	601029.9	261571.7	1277105.0	3272.77	7264.4
2009	123100.6	601029.9	276143.2	1348249.0	3272.77	7674.0
2010	123100.6	601029.9	290719.1	1419415.0	3285.08	8126.4
2101	73860.4	360617.9	159293.5	777739.1	2018.39	4435.4
2102	73860.4	360617.9	173876.0	848937.1	2023.43	4872.9
2103	73860.4	360617.9	188420.6	919950.0	2028.47	5287.4
2104	68936.3	336576.6	203003.0	991147.6	1888.53	5692.9
2105	71398.3	348597.3	217629.6	1062561.0	1951.11	6095.2
2106	73860.4	360617.9	232205.6	1133727.0	2018.39	6516.0
2107	76322.4	372638.5	246861.4	1205283.0	2075.29	6918.3
2108	73860.4	360617.9	261458.3	1276551.0	2008.35	7351.1
2109	73860.4	360617.9	276063.5	1347860.0	2003.34	7771.8
2110	73860.4	360617.9	290593.2	1418800.0	2010.86	8186.1
2201	73860.4	360617.9	159316.7	777852.4	2018.39	4436.1
2202	73860.4	360617.9	174053.1	849801.8	1995.85	4827.5
2203	73860.4	360617.9	188694.4	921286.9	1990.87	5148.9
2204	73860.4	360617.9	203268.2	992442.4	1988.39	5488.5
2205	71398.3	348597.3	217881.9	1063793.0	1914.91	5813.4
2206	76322.4	372638.5	232508.1	1135204.0	2044.41	6154.4
2207	76322.4	372638.5	247039.9	1206155.0	2044.41	6491.3
2208	73860.4	360617.9	261609.4	1277289.0	1980.94	6878.7
2209	71398.3	348597.3	276143.2	1348249.0	1917.31	7217.3
2210	73860.4	360617.9	290761.0	1419620.0	1980.94	7543.4
2301	64012.3	312535.5	290551.2	1418595.0	1756.78	8216.6
2302	68936.3	336576.6	290593.2	1418800.0	1893.24	8212.5
2303	73860.4	360617.9	290551.2	1418595.0	2020.91	8200.8
2304	78784.4	384659.1	290593.2	1418800.0	2144.91	8186.1
2305	83708.4	408700.3	290635.2	1419005.0	2270.45	8166.2
2306	88632.4	432741.5	290719.1	1419415.0	2383.08	8152.7
2307	93556.4	456782.6	290719.1	1419415.0	2506.06	8147.5
2308	98480.5	480823.8	290761.0	1419620.0	2628.07	8117.1
2309	103404.5	504865.1	290761.0	1419620.0	2759.47	8138.1
2310	118176.6	576988.8	290719.1	1419415.0	3153.68	8126.4
2401	64012.3	312535.5	159502.4	778759.1	1736.23	4435.5
2402	68936.3	336576.6	159456.0	778532.5	1867.45	4442.8
2403	73860.4	360617.9	159456.0	778532.5	1990.87	4437.1
2404	78784.4	384659.1	159456.0	778532.5	2120.94	4434.2
2405	83708.4	408700.3	159432.8	778419.2	2242.26	4430.5
2406	88632.4	432741.5	159479.2	778645.8	2359.34	4423.2
2407	93556.4	456782.6	159479.2	778645.8	2484.19	4417.6
2408	98480.5	480823.8	159502.4	778759.1	2611.66	4421.1
2409	103404.5	504865.1	159502.4	778759.1	2742.24	4415.3
2410	113252.5	552947.2	159502.4	778759.1	2992.13	4415.3

TABLE C.4 (Contd.)

Run	Mass flux				Reynolds number	
	Water		R-113		Water	R-113
#	lbm/sqft hr	kg/sq.m hr	lbm/sqft hr	kg/sq.m hr		
3001	122609.5	598632.1	163447.9	798022.6	3253.36	4220.6
3002	122609.5	598632.1	178384.5	870949.5	3257.44	4624.8
3003	122609.5	598632.1	193333.9	943938.9	3269.72	5019.1
3004	122609.5	598632.1	208266.0	1016844.0	3269.72	5421.2
3005	122609.5	598632.1	223174.5	1089634.0	3273.82	5817.1
3006	122609.5	598632.1	238121.6	1162612.0	3277.93	6223.2
3007	122609.5	598632.1	253004.2	1235275.0	3294.39	6616.6
3008	122609.5	598632.1	268002.9	1308505.0	3290.27	7008.8
3009	122609.5	598632.1	282891.9	1381199.0	3294.39	7417.9
3010	122609.5	598632.1	297867.0	1454314.0	3298.52	7826.2
3101	73565.7	359179.3	163400.5	797791.2	2011.53	4316.0
3102	73565.7	359179.3	178306.9	870570.6	2011.53	4684.9
3103	73565.7	359179.3	193249.9	943528.8	2011.53	5080.8
3104	73565.7	359179.3	208205.7	1016549.0	2009.02	5463.2
3105	76017.9	371151.9	223174.5	1089634.0	2070.81	5852.0
3106	73565.7	359179.3	238087.2	1162444.0	2009.02	6259.7
3107	73565.7	359179.3	253040.7	1235453.0	2011.53	6683.8
3108	73565.7	359179.3	267964.2	1308316.0	2014.04	7106.2
3109	73565.7	359179.3	282891.9	1381199.0	2026.63	7467.3
3110	76017.9	371151.9	297824.0	1454104.0	2094.19	7887.6
3201	73565.7	359179.3	163590.2	798717.5	1944.68	4145.5
3202	73565.7	359179.3	178565.4	871832.8	1949.57	4415.7
3203	73565.7	359179.3	193473.8	944621.9	1952.02	4748.5
3204	73565.7	359179.3	208446.7	1017726.0	1944.68	5095.0
3205	76017.9	371151.9	223400.2	1090736.0	2017.08	5430.6
3206	73565.7	359179.3	238293.6	1163451.0	1959.38	5760.8
3207	76017.9	371151.9	253332.9	1236880.0	2017.08	6099.1
3208	73565.7	359179.3	268273.4	1309826.0	1949.57	6418.8
3209	76017.9	371151.9	283258.9	1382991.0	2017.08	6753.9
3210	73565.7	359179.3	298167.3	1455781.0	1949.57	7089.7
3301	63756.9	311288.7	297952.9	1454734.0	1747.68	7864.9
3302	69642.2	340023.1	298124.5	1455572.0	1887.67	7817.4
3303	73565.7	359179.3	297952.9	1454734.0	2004.01	7870.1
3304	78470.1	383124.6	297995.7	1454943.0	2137.61	7876.5
3305	83374.5	407069.8	297909.9	1454524.0	2265.54	7874.2
3306	88278.8	431015.2	297867.0	1454314.0	2395.81	7862.6
3307	93183.2	454960.4	297995.7	1454943.0	2516.30	7855.6
3308	98087.6	478905.7	297995.7	1454943.0	2628.92	7824.4
3309	102992.0	502851.1	297995.7	1454943.0	2756.91	7819.2
3310	107896.4	526796.5	297995.7	1454943.0	2888.19	7819.2
3401	63756.9	311288.7	163471.7	798138.9	1730.31	4280.8
3402	66209.1	323261.4	163447.9	798022.6	1796.86	4280.2
3403	73565.7	359179.3	163447.9	798022.6	1981.59	4271.7
3404	78470.1	383124.6	163424.2	797907.0	2111.05	4258.2
3405	83374.5	407069.8	163424.2	797907.0	2234.58	4251.1
3406	88278.8	431015.2	163400.5	797791.2	2363.07	4253.4
3407	93183.2	454960.4	163424.2	797907.0	2488.11	4242.6
3408	98087.6	478905.7	163424.2	797907.0	2615.78	4242.6
3409	102992.0	502851.1	163424.2	797907.0	2746.57	4245.5
3410	107896.4	526796.5	163400.5	797791.2	2873.75	4244.9

TABLE C.4 (Contd.)

Run	Mass flux				Reynolds number	
	Water		R-113		Water	R-113
#	lbm/sqft hr	kg/sq.m hr	lbm/sqft hr	kg/sq.m hr		
7001	96309.1	470222.2	129171.2	630669.1	3333.86	5214.4
7002	96309.1	470222.2	122677.2	598962.6	3325.54	4955.5
7003	96309.1	470222.2	116186.9	567274.2	3329.70	4677.9
7004	96309.1	470222.2	109732.1	535759.1	3325.54	4403.4
7005	96309.1	470222.2	103217.4	503951.6	3333.86	4142.0
7006	96309.1	470222.2	96738.3	472317.8	3329.70	3874.3
7007	96309.1	470222.2	90262.9	440702.1	3338.03	3622.1
7008	96309.1	470222.2	85105.0	415519.1	3313.09	3408.4
7009	96309.1	470222.2	77334.5	377580.0	3308.94	3089.0
7010	96309.1	470222.2	72136.9	352203.0	3317.23	2896.7
7101	48154.6	235111.2	70838.4	345863.4	1800.11	2897.5
7102	48154.6	235111.2	77312.0	377470.3	1734.67	3133.3
7103	48154.6	235111.2	83766.8	408985.6	1730.37	3377.1
7104	48154.6	235111.2	90262.9	440702.1	1732.52	3653.4
7105	48154.6	235111.2	96752.3	472386.3	1730.37	3929.0
7106	48154.6	235111.2	103232.4	504024.8	1749.77	4214.2
7107	48154.6	235111.2	109700.3	535603.8	1756.28	4487.1
7108	48154.6	235111.2	116203.7	567356.3	1758.45	4759.3
7109	48154.6	235111.2	122659.4	598875.7	1764.98	5036.9
7110	48154.6	235111.2	129133.9	630487.0	1767.16	5323.7
7201	46738.2	228196.1	70900.2	346165.4	1658.75	2884.9
7202	46738.2	228196.1	77379.4	377799.4	1708.84	3144.3
7203	46738.2	228196.1	83888.5	409579.4	1723.67	3375.3
7204	46738.2	228196.1	90367.6	441213.2	1725.79	3626.3
7205	46738.2	228196.1	96864.4	472933.5	1727.92	3853.6
7206	46738.2	228196.1	103366.8	504681.0	1730.05	4085.0
7207	46738.2	228196.1	109811.4	536146.3	1683.65	4302.2
7208	46738.2	228196.1	116321.2	567930.0	1669.09	4517.6
7209	46738.2	228196.1	122871.9	599913.2	1671.16	4733.6
7210	46738.2	228196.1	129376.0	631669.0	1662.88	4953.9
7301	45321.9	221281.1	70910.5	346215.6	1570.84	2787.4
7302	48154.6	235111.2	70900.2	346165.4	1669.02	2832.0
7303	42489.3	207451.0	70900.2	346165.4	1481.88	2843.3
7304	39656.7	193620.9	70900.2	346165.4	1388.28	2848.9
7305	36824.1	179790.8	70890.0	346115.1	1289.11	2850.4
7306	33991.4	165960.8	70900.2	346165.4	1194.41	2852.7
7307	31158.8	152130.7	70920.8	346266.0	1101.71	2855.4
7308	28326.2	138300.6	70900.2	346165.4	1010.31	2854.6
7309	25493.6	124470.6	70890.0	346115.1	918.35	2863.6
7310	22661.0	110640.5	70890.0	346115.1	821.39	2873.1
7401	22661.0	110640.5	129171.2	630669.1	837.78	5339.1
7402	25493.6	124470.6	129171.2	630669.1	935.55	5293.9
7403	28326.2	138300.6	129171.2	630669.1	1034.38	5307.8
7404	31158.8	152130.7	129171.2	630669.1	1129.41	5297.4
7405	33991.4	165960.8	129171.2	630669.1	1227.51	5293.9
7406	36824.1	179790.8	129171.2	630669.1	1318.31	5276.6
7407	39656.7	193620.9	129133.9	630487.0	1414.44	5282.0
7408	42489.3	207451.0	129133.9	630487.0	1513.59	5278.5
7409	45321.9	221281.1	129152.5	630577.8	1614.49	5263.3
7410	48154.6	235111.2	129133.9	630487.0	1706.89	5261.2

TABLE C.4 (Contd.)

Run	Mass flux				Reynolds number	
	Water		R-113		Water	R-113
#	lbm/sqft hr	kg/sq.m hr	lbm/sqft hr	kg/sq.m hr		
8001	94077.4	459326.2	80193.6	391539.2	3365.77	3170.9
8002	94077.4	459326.2	87522.1	427320.1	3361.56	3449.5
8003	94077.4	459326.2	94856.9	463131.9	3369.98	3792.1
8004	92693.9	452571.4	102198.0	498974.4	3299.71	4080.3
8005	94077.4	459326.2	109545.5	534848.0	3348.96	4396.2
8006	94077.4	459326.2	116882.3	570669.5	3340.58	4660.6
8007	94077.4	459326.2	124205.4	606424.0	3336.40	4978.1
8008	94077.4	459326.2	131549.6	642281.5	3336.40	5306.3
8009	95460.9	466081.0	138898.1	678160.0	3385.46	5613.5
8010	94077.4	459326.2	146229.6	713955.5	3332.22	5909.8
8101	47038.7	229663.1	80181.9	391482.3	1727.57	3203.4
8102	47038.7	229663.1	87483.9	427133.8	1731.88	3499.6
8103	48422.2	236417.9	94815.5	462930.1	1782.81	3812.4
8104	47038.7	229663.1	102168.4	498829.9	1725.42	4113.3
8105	47038.7	229663.1	109466.1	534460.4	1731.88	4415.6
8106	47038.7	229663.1	116797.7	570256.4	1731.88	4699.2
8107	45378.5	221557.3	125612.1	613292.1	1670.75	5073.3
8108	47038.7	229663.1	131435.5	641724.4	1736.19	5322.1
8109	47038.7	229663.1	138737.5	677375.9	1736.19	5642.9
8110	47038.7	229663.1	146081.8	713233.9	1738.35	5915.1
8201	45655.2	222908.3	80158.5	391368.3	1676.76	3335.3
8202	45655.2	222908.3	87483.9	427133.8	1674.67	3628.7
8203	45655.2	222908.3	94801.8	462862.9	1676.76	3912.5
8204	44825.1	218855.5	102123.9	498612.6	1644.22	4161.6
8205	45101.8	220206.4	110957.9	541744.0	1650.26	4467.1
8206	45655.2	222908.3	116848.5	570504.4	1668.43	4641.3
8207	45655.2	222908.3	124115.6	605985.5	1664.27	4898.2
8208	45101.8	220206.4	131454.6	641817.6	1650.26	5181.1
8209	46208.6	225610.2	138777.7	677572.2	1692.86	5420.2
8210	46208.6	225610.2	146145.2	713543.5	1688.65	5552.4
8301	47038.7	229663.1	80146.8	391311.2	1725.42	3169.1
8302	43164.9	210749.7	80158.5	391368.3	1589.25	3157.3
8303	41504.7	202643.9	80135.2	391254.1	1539.57	3192.3
8304	38737.8	189134.4	80158.5	391368.3	1438.72	3198.3
8305	32373.7	158062.3	80135.2	391254.1	1211.34	3203.5
8306	34033.9	166168.0	80135.2	391254.1	1278.20	3205.6
8307	29606.7	144552.7	80135.2	391254.1	1117.46	3215.9
8308	26839.7	131043.1	80135.2	391254.1	1019.30	3228.3
8309	22135.9	108076.7	80123.5	391197.1	846.91	3227.8
8310	19368.9	94567.2	80123.5	391197.1	747.47	3242.3
8401	19368.9	94567.2	145827.7	711993.3	784.97	5897.3
8402	22689.3	110778.7	145848.9	712096.8	908.40	5935.9
8403	24902.9	121586.4	145827.7	711993.3	988.54	5923.7
8404	28223.2	137797.9	145827.7	711993.3	1106.71	5908.6
8405	30436.8	148605.6	145848.9	712096.8	1187.67	5905.7
8406	33203.8	162115.1	145848.9	712096.8	1287.70	5902.0
8407	35970.8	175624.7	145827.7	711993.3	1388.16	5901.1
8408	38737.8	189134.4	145827.7	711993.3	1489.43	5889.8
8409	41504.7	202643.9	145912.5	712407.3	1582.10	5870.7
8410	44271.7	216153.6	145870.1	712200.3	1679.25	5872.7

TABLE C.4 (Contd.)

Run	Mass flux				Reynolds number	
	Water		R-113		Water	R-113
#	lbm/sqft hr	kg/sq.m hr	lbm/sqft hr	kg/sq.m hr		
9001	94077.4	459326.2	148917.1	727077.0	3425.14	5695.7
9002	94077.4	459326.2	131397.4	641538.4	3408.09	4986.0
9003	94077.4	459326.2	116746.9	570008.4	3395.35	4418.4
9004	94077.4	459326.2	102064.5	498322.6	3425.14	3862.7
9005	94077.4	459326.2	87394.7	426698.2	3437.97	3320.7
9006	94077.4	459326.2	94719.0	462458.7	3437.97	3594.2
9007	94077.4	459326.2	109338.9	533839.3	3450.84	4140.8
9008	94077.4	459326.2	123989.6	605370.3	3446.55	4698.7
9009	94077.4	459326.2	138637.0	676885.2	3450.84	5264.2
9010	94077.4	459326.2	80065.0	390911.5	3433.69	3032.1
9101	47038.7	229663.1	145721.5	711474.8	1819.76	5654.4
9102	47038.7	229663.1	131111.1	640140.5	1822.00	5080.9
9103	47038.7	229663.1	138415.3	675802.8	1828.74	5377.9
9104	47038.7	229663.1	123899.4	604929.9	1790.83	4795.1
9105	47038.7	229663.1	116577.2	569179.8	1790.83	4494.1
9106	47038.7	229663.1	109259.3	533450.7	1790.83	4217.5
9107	47038.7	229663.1	101930.7	497669.4	1790.83	3926.9
9108	47038.7	229663.1	94553.0	461648.3	1806.37	3633.1
9109	47038.7	229663.1	87279.7	426136.9	1804.14	3344.8
9110	47038.7	229663.1	79959.5	390396.4	1801.92	3066.3
9201	45655.2	222908.3	145848.9	712096.8	1725.31	5263.8
9202	45655.2	222908.3	138536.3	676393.5	1729.59	5020.2
9203	45655.2	222908.3	131225.8	640700.5	1725.31	4784.2
9204	45655.2	222908.3	123899.4	604929.9	1729.59	4535.4
9205	45655.2	222908.3	116577.2	569179.8	1729.59	4293.2
9206	45655.2	222908.3	109259.3	533450.7	1721.05	4042.5
9207	45655.2	222908.3	101945.6	497742.1	1727.45	3809.9
9208	45655.2	222908.3	94636.1	462053.8	1725.31	3550.3
9209	45655.2	222908.3	87305.3	426261.8	1731.73	3321.7
9210	45655.2	222908.3	80018.1	390682.7	1729.59	3062.5
9301	22135.9	108076.7	80321.7	392165.0	819.06	3033.3
9302	24902.9	121586.4	80298.5	392051.5	920.30	3043.0
9303	27669.8	135096.0	80298.5	392051.5	1016.22	3037.0
9304	30436.8	148605.6	80286.8	391994.5	1110.90	3032.5
9305	33203.8	162115.1	80275.2	391937.7	1207.37	3034.1
9306	35970.8	175624.7	80275.2	391937.7	1303.09	3034.1
9307	38737.8	189134.4	80263.5	391880.8	1401.58	3033.6
9308	40121.3	195889.1	80263.5	391880.8	1451.64	3035.6
9309	44271.7	216153.6	80251.9	391824.0	1591.83	3033.2
9310	47038.7	229663.1	80263.5	391880.8	1684.99	3027.6
9401	22135.9	108076.7	146335.0	714470.1	839.62	5633.8
9402	23519.4	114831.5	146313.9	714367.1	886.60	5629.3
9403	27669.8	135096.0	146377.1	714675.6	1028.93	5624.4
9404	30436.8	148605.6	146377.1	714675.6	1122.02	5609.6
9405	33203.8	162115.1	146377.1	714675.6	1214.91	5591.2
9406	35970.8	175624.7	146377.1	714675.6	1309.61	5577.5
9407	38737.8	189134.4	146377.1	714675.6	1401.58	5561.2
9408	41504.7	202643.9	146377.1	714675.6	1496.08	5554.5
9409	44271.7	216153.6	146398.2	714778.7	1587.66	5544.3
9410	47038.7	229663.1	146398.2	714778.7	1684.99	5536.9

TABLE C.5 :- REDUCED DATA - PRANDTL AND NUSSELT NUMBER

Run	Prandtl number		Nusselt number	
#	Water	R-113	Water	R-113
1001	8.297	8.485	102.99	149.48
1002	8.284	8.429	97.69	157.72
1003	8.270	8.429	99.55	167.27
1004	8.284	8.425	101.81	178.15
1005	8.270	8.413	102.01	190.05
1006	8.270	8.398	93.31	190.14
1007	8.311	8.417	96.42	203.15
1008	8.338	8.394	92.02	198.00
1009	8.324	8.374	89.67	205.90
1010	8.311	8.390	93.11	220.85
1101	8.086	8.425	80.95	143.92
1102	8.073	8.402	74.06	150.12
1103	8.047	8.374	75.34	163.85
1104	8.008	8.370	77.47	170.99
1105	7.969	8.354	75.80	177.33
1106	7.957	8.354	74.77	176.73
1107	7.957	8.358	74.37	188.47
1108	7.944	8.331	73.42	191.15
1109	7.931	8.319	70.01	200.58
1110	7.906	8.308	75.31	211.83
1201	8.177	8.587	71.81	132.86
1202	8.151	8.662	76.23	139.95
1203	8.151	8.716	74.22	147.86
1204	8.177	8.767	73.97	153.38
1205	8.125	8.797	77.71	164.07
1206	8.099	8.827	74.44	173.32
1207	8.125	8.858	74.19	176.14
1208	8.073	8.858	73.31	178.88
1209	8.099	8.893	76.77	186.75
1210	8.073	8.906	76.15	191.31
1301	7.957	8.331	67.88	199.90
1302	7.995	8.323	69.97	202.18
1303	8.008	8.327	73.80	207.43
1304	7.995	8.327	78.48	211.32
1305	8.034	8.323	80.64	210.80
1306	8.099	8.347	82.81	212.90
1307	8.099	8.350	83.52	208.64
1308	8.112	8.343	88.26	220.38
1309	8.125	8.339	88.22	216.38
1310	8.125	8.339	96.71	226.68
1401	7.969	8.366	73.55	147.52
1402	7.982	8.358	79.20	148.19
1403	8.047	8.378	74.89	145.88
1404	8.125	8.394	74.30	141.15
1405	8.151	8.386	78.27	148.21
1406	8.164	8.398	80.91	147.32
1407	8.177	8.398	83.12	143.66
1408	8.177	8.394	88.37	150.25
1409	8.217	8.398	83.83	144.17
1410	8.230	8.413	82.99	137.15

TABLE C.5 (Contd.)

Run #	Prandtl number Water	Prandtl number R-113	Nusselt number Water	Nusselt number R-113
2001	8.151	8.189	118.27	45.48
2002	8.191	8.155	117.74	51.49
2003	8.151	8.115	127.36	57.37
2004	8.151	8.133	119.77	58.07
2005	8.164	8.122	117.32	63.75
2006	8.138	8.111	122.62	64.60
2007	8.125	8.111	127.90	72.42
2008	8.125	8.126	127.90	73.15
2009	8.125	8.122	110.54	73.95
2010	8.086	8.089	119.48	79.75
2101	7.843	8.111	83.01	46.92
2102	7.818	8.074	91.29	55.05
2103	7.793	8.067	87.95	55.13
2104	7.818	8.070	91.85	59.54
2105	7.843	8.078	87.17	62.15
2106	7.843	8.067	88.42	63.63
2107	7.893	8.074	84.85	65.86
2108	7.893	8.056	85.01	72.24
2109	7.918	8.049	82.89	71.27
2110	7.881	8.045	84.29	76.68
2201	7.843	8.111	80.95	47.04
2202	7.957	8.133	85.80	51.20
2203	7.982	8.227	85.02	54.75
2204	7.995	8.288	79.56	52.31
2205	8.034	8.358	89.48	58.28
2206	8.047	8.406	82.27	59.86
2207	8.047	8.449	82.67	62.61
2208	8.034	8.445	90.23	66.44
2209	8.021	8.481	94.30	73.08
2210	8.034	8.526	94.54	72.21
2301	7.743	8.023	79.33	80.26
2302	7.793	8.027	78.66	76.81
2303	7.830	8.034	82.72	77.15
2304	7.881	8.045	92.73	81.58
2305	7.918	8.060	96.70	81.43
2306	8.008	8.070	95.18	79.84
2307	8.047	8.074	96.33	78.09
2308	8.086	8.096	97.14	76.90
2309	8.086	8.081	106.31	79.30
2310	8.086	8.089	104.11	77.30
2401	7.918	8.118	80.76	48.31
2402	7.931	8.107	81.62	47.08
2403	7.982	8.115	89.26	50.06
2404	7.995	8.118	95.04	48.39
2405	8.047	8.122	99.42	50.93
2406	8.112	8.133	97.95	48.12
2407	8.138	8.141	108.22	50.59
2408	8.151	8.137	95.81	47.18
2409	8.151	8.144	112.17	49.62
2410	8.191	8.144	105.14	47.27

TABLE C.5 (Contd.)

Run #	Prandtl number Water	Prandtl number R-113	Nusselt number Water	Nusselt number R-113
3001	8.217	8.481	131.50	146.36
3002	8.204	8.457	132.86	159.20
3003	8.164	8.449	144.00	168.92
3004	8.164	8.433	125.25	168.54
3005	8.151	8.425	127.25	176.69
3006	8.138	8.409	129.44	189.43
3007	8.086	8.406	136.63	191.99
3008	8.099	8.406	136.84	200.32
3009	8.086	8.390	136.53	210.53
3010	8.073	8.378	137.09	221.02
3101	7.906	8.347	97.41	138.83
3102	7.906	8.378	94.42	141.33
3103	7.906	8.374	95.50	156.34
3104	7.918	8.386	102.31	161.09
3105	7.944	8.390	100.51	168.30
3106	7.918	8.374	100.32	173.83
3107	7.906	8.347	95.47	175.52
3108	7.893	8.323	95.57	181.27
3109	7.830	8.350	106.15	194.42
3110	7.830	8.331	97.79	200.77
3201	8.257	8.595	79.46	128.81
3202	8.230	8.746	111.59	143.22
3203	8.217	8.793	96.52	141.68
3204	8.257	8.819	97.22	153.81
3205	8.217	8.853	98.69	152.21
3206	8.177	8.888	109.00	169.90
3207	8.217	8.914	103.04	160.89
3208	8.230	8.954	110.32	178.67
3209	8.217	8.976	108.45	178.32
3210	8.230	8.994	112.23	186.35
3301	7.881	8.350	89.24	194.40
3302	7.995	8.390	89.50	192.45
3303	7.944	8.347	94.47	197.95
3304	7.944	8.343	105.85	208.73
3305	7.969	8.343	104.57	202.80
3306	7.982	8.350	107.57	206.64
3307	8.034	8.358	114.51	215.87
3308	8.112	8.382	111.15	211.13
3309	8.125	8.386	113.88	207.93
3310	8.125	8.386	119.95	217.49
3401	7.982	8.398	87.99	140.26
3402	7.982	8.398	100.66	141.91
3403	8.060	8.409	99.20	137.36
3404	8.073	8.413	99.46	139.13
3405	8.112	8.437	102.66	139.98
3406	8.125	8.433	99.90	135.95
3407	8.151	8.449	110.55	145.99
3408	8.164	8.449	110.03	142.52
3409	8.164	8.445	123.88	146.82
3410	8.177	8.445	115.78	139.87

TABLE C.5 (Contd.)

Run #	Prandtl number Water	R-113	Nusselt number Water	R-113
7001	7.982	8.308	104.36	176.84
7002	8.008	8.304	107.79	175.28
7003	7.995	8.323	102.40	164.05
7004	8.008	8.343	98.78	150.28
7005	7.982	8.343	104.23	152.93
7006	7.995	8.354	93.57	140.01
7007	7.969	8.343	103.72	136.33
7008	8.047	8.354	107.61	137.52
7009	8.060	8.370	98.02	122.22
7010	8.034	8.339	104.57	120.17
7101	7.237	8.231	68.78	109.59
7102	7.585	8.284	67.94	119.30
7103	7.609	8.315	71.89	123.18
7104	7.597	8.292	71.96	128.04
7105	7.609	8.273	69.59	133.71
7106	7.502	8.242	72.20	141.54
7107	7.467	8.231	73.17	149.78
7108	7.455	8.223	75.03	158.20
7109	7.420	8.208	77.93	164.32
7110	7.408	8.185	77.46	168.74
7201	7.731	8.261	72.88	108.37
7202	7.443	8.269	72.31	117.95
7203	7.362	8.327	69.41	120.90
7204	7.351	8.343	81.51	127.67
7205	7.339	8.394	69.65	129.90
7206	7.328	8.433	77.72	145.43
7207	7.585	8.485	65.94	142.73
7208	7.670	8.538	75.95	153.05
7209	7.658	8.587	71.03	153.84
7210	7.707	8.624	73.91	156.09
7301	7.969	8.465	69.20	108.95
7302	7.969	8.370	68.87	107.67
7303	7.906	8.347	59.47	101.46
7304	7.868	8.335	53.44	104.18
7305	7.868	8.331	55.82	108.17
7306	7.830	8.327	55.64	106.36
7307	7.768	8.323	53.04	105.42
7308	7.682	8.323	49.99	103.45
7309	7.585	8.304	49.02	103.59
7310	7.526	8.284	49.10	103.63
7401	7.339	8.170	52.45	151.51
7402	7.408	8.219	54.35	156.91
7403	7.455	8.204	57.77	159.23
7404	7.526	8.216	59.91	162.75
7405	7.562	8.219	58.91	160.42
7406	7.646	8.238	62.61	165.74
7407	7.682	8.231	59.25	157.39
7408	7.694	8.235	65.15	159.02
7409	7.694	8.246	72.33	170.22
7410	7.743	8.254	69.97	164.28

TABLE C.5 (Contd.)

Run #	Prandtl number Water	Prandtl number R-113	Nusselt number Water	Nusselt number R-113
8001	8.047	8.070	109.43	36.75
8002	8.060	8.089	135.19	46.20
8003	8.034	8.009	122.77	43.62
8004	8.099	8.016	130.30	51.36
8005	8.099	7.987	122.88	51.07
8006	8.125	8.023	116.59	52.20
8007	8.138	7.994	127.72	59.53
8008	8.138	7.959	115.93	58.29
8009	8.138	7.948	116.11	58.72
8010	8.151	7.948	117.24	66.45
8101	7.781	8.012	86.01	42.67
8102	7.756	8.005	85.53	42.43
8103	7.756	7.977	78.00	43.04
8104	7.793	7.969	79.43	45.69
8105	7.756	7.959	87.26	50.19
8106	7.756	7.973	82.62	50.98
8107	7.756	7.952	91.52	56.95
8108	7.731	7.938	89.46	59.04
8109	7.731	7.913	77.56	58.76
8110	7.719	7.938	85.11	64.53
8201	7.781	7.789	78.93	43.74
8202	7.793	7.806	86.07	47.74
8203	7.781	7.833	79.89	44.39
8204	7.793	7.902	84.85	49.83
8205	7.818	7.969	87.54	52.34
8206	7.830	8.045	85.15	51.31
8207	7.855	8.081	78.70	52.87
8208	7.818	8.089	85.75	57.72
8209	7.805	8.141	82.91	56.82
8210	7.830	8.197	83.29	59.40
8301	7.793	8.070	80.29	39.49
8302	7.756	8.092	86.59	41.99
8303	7.682	8.027	73.86	39.12
8304	7.670	8.020	73.92	40.51
8305	7.597	8.009	74.16	40.98
8306	7.562	8.005	67.34	40.13
8307	7.514	7.987	65.29	41.15
8308	7.455	7.966	62.66	40.00
8309	7.385	7.966	55.07	38.74
8310	7.305	7.941	56.26	40.13
8401	6.868	7.945	52.03	57.94
8402	6.974	7.909	54.17	57.50
8403	7.049	7.920	59.13	58.93
8404	7.159	7.934	58.65	59.10
8405	7.203	7.938	70.13	61.96
8406	7.259	7.941	74.25	62.79
8407	7.305	7.941	69.28	59.50
8408	7.339	7.952	74.65	59.29
8409	7.420	7.973	75.18	58.67
8410	7.467	7.969	81.72	61.41

TABLE C.5 (Contd.)

Run #	Prandtl number Water	Prandtl number R-113	Nusselt number Water	Nusselt number R-113
9001	7.868	8.261	135.19	195.79
9002	7.918	8.308	129.64	166.90
9003	7.957	8.323	132.69	151.57
9004	7.868	8.323	148.01	139.09
9005	7.830	8.300	144.92	123.66
9006	7.830	8.308	129.09	119.79
9007	7.793	8.319	138.19	139.47
9008	7.805	8.315	137.86	153.84
9009	7.793	8.304	136.54	170.17
9010	7.843	8.319	121.89	107.62
9101	7.282	8.178	88.76	168.22
9102	7.271	8.185	99.29	158.59
9103	7.237	8.170	99.00	168.10
9104	7.432	8.193	99.54	151.47
9105	7.432	8.216	97.73	137.80
9106	7.432	8.208	101.54	136.89
9107	7.432	8.219	91.68	123.91
9108	7.351	8.235	96.19	120.55
9109	7.362	8.250	95.88	114.94
9110	7.374	8.246	88.68	105.82
9201	7.502	8.608	93.26	154.26
9202	7.479	8.583	98.51	150.23
9203	7.502	8.546	91.63	140.60
9204	7.479	8.522	93.64	140.25
9205	7.479	8.485	93.23	134.15
9206	7.526	8.457	88.95	128.79
9207	7.490	8.398	96.35	130.68
9208	7.502	8.358	89.68	116.41
9209	7.467	8.292	99.72	120.34
9210	7.479	8.258	95.79	106.24
9301	7.707	8.335	63.66	97.62
9302	7.719	8.315	64.62	103.37
9303	7.781	8.327	74.35	108.59
9304	7.843	8.335	70.98	103.09
9305	7.881	8.331	70.46	104.57
9306	7.918	8.331	79.63	105.12
9307	7.931	8.331	81.23	106.73
9308	7.931	8.327	84.74	109.99
9309	7.995	8.331	91.68	110.53
9310	8.034	8.343	95.99	110.26
9401	7.467	8.223	65.99	155.95
9402	7.526	8.227	69.11	158.86
9403	7.658	8.235	71.65	157.34
9404	7.743	8.250	73.34	155.95
9405	7.818	8.269	76.41	158.76
9406	7.868	8.273	78.69	158.63
9407	7.931	8.300	84.69	165.44
9408	7.969	8.308	89.78	168.99
9409	8.021	8.319	89.89	165.69
9410	8.034	8.327	96.72	166.75

TABLE C.6 :- REDUCED DATA - PRESSURE DROP AND FRICTION COEFFICIENT

Run #	R-113 Flow Rate gpm	R-113 Flow Rate cu.m/hr	Reynolds #	Pressure drop psi	Pressure drop Pa	Friction coeff.
101	0.500	0.1136	6845.2	0.0822	566.5	0.0553
102	0.475	0.1079	6502.9	0.0743	512.0	0.0553
103	0.450	0.1022	6160.6	0.0703	484.8	0.0584
104	0.425	0.0965	5818.4	0.0624	430.3	0.0581
105	0.400	0.0908	5476.1	0.0585	403.0	0.0614
106	0.375	0.0852	5133.9	0.0545	375.8	0.0652
107	0.350	0.0795	4791.6	0.0545	375.8	0.0748
108	0.325	0.0738	4449.4	0.0466	321.3	0.0742
109	0.300	0.0681	4107.1	0.0387	266.8	0.0723
110	0.275	0.0625	3764.8	0.0347	239.6	0.0773
201	0.500	0.1136	7187.4	0.0229	157.9	0.0121
202	0.475	0.1079	6828.0	0.0229	157.9	0.0134
203	0.450	0.1022	6468.7	0.0189	130.6	0.0124
204	0.425	0.0965	6109.3	0.0189	130.6	0.0139
205	0.400	0.0908	5749.9	0.0150	103.4	0.0124
206	0.375	0.0852	5390.6	0.0150	103.4	0.0141
207	0.350	0.0795	5031.2	0.0150	103.4	0.0162
208	0.325	0.0738	4671.8	0.0110	76.1	0.0139
209	0.300	0.0681	4312.4	0.0110	76.1	0.0163
210	0.275	0.0625	3953.1	0.0110	76.1	0.0194
301	0.500	0.1136	7274.7	0.0901	621.0	0.0451
302	0.475	0.1079	6911.0	0.0861	593.7	0.0478
303	0.450	0.1022	6547.2	0.0782	539.2	0.0484
304	0.425	0.0965	6183.5	0.0703	484.8	0.0487
305	0.400	0.0908	5819.8	0.0664	457.5	0.0519
306	0.375	0.0852	5456.0	0.0624	430.3	0.0556
307	0.350	0.0795	5092.3	0.0624	430.3	0.0638
308	0.325	0.0738	4728.6	0.0545	375.8	0.0646
309	0.300	0.0681	4364.8	0.0545	375.8	0.0758
310	0.275	0.0625	4001.1	0.0466	321.3	0.0772

TABLE C.6 (Contd.)

Run #	R-113 Flow Rate gpm	cu.m/hr	Reynolds #	Pressure drop psi	Pa	Friction coeff.
701	0.500	0.1136	4791.6	0.0189	130.6	0.0732
702	0.475	0.1079	4552.0	0.0150	103.4	0.0642
703	0.450	0.1022	4312.4	0.0150	103.4	0.0715
704	0.425	0.0965	4072.9	0.0150	103.4	0.0801
705	0.400	0.0908	3833.3	0.0150	103.4	0.0905
706	0.375	0.0852	3593.7	0.0110	76.1	0.0758
707	0.350	0.0795	3354.1	0.0110	76.1	0.0870
708	0.325	0.0738	3114.5	0.0110	76.1	0.1009
709	0.300	0.0681	2875.0	0.0071	48.9	0.0761
710	0.275	0.0625	2635.4	0.0071	48.9	0.0905
801	0.500	0.1136	5097.5	0.0110	76.1	0.0321
802	0.475	0.1079	4842.6	0.0110	76.1	0.0356
803	0.450	0.1022	4587.7	0.0110	76.1	0.0397
804	0.425	0.0965	4332.8	0.0091	62.5	0.0365
805	0.400	0.0908	4078.0	0.0071	48.9	0.0322
806	0.375	0.0852	3823.1	0.0071	48.9	0.0367
807	0.350	0.0795	3568.2	0.0071	48.9	0.0421
808	0.325	0.0738	3313.3	0.0071	48.9	0.0488
809	0.300	0.0681	3058.5	0.0071	48.9	0.0573
810	0.275	0.0625	2803.6	0.0047	32.6	0.0454
901	0.500	0.1136	5097.5	0.0268	185.1	0.0778
902	0.475	0.1079	4842.6	0.0229	157.9	0.0735
903	0.450	0.1022	4587.7	0.0189	130.6	0.0678
904	0.425	0.0965	4332.8	0.0189	130.6	0.0760
905	0.400	0.0908	4078.0	0.0189	130.6	0.0858
906	0.375	0.0852	3823.1	0.0150	103.4	0.0773
907	0.350	0.0795	3568.2	0.0150	103.4	0.0887
908	0.325	0.0738	3313.3	0.0110	76.1	0.0758
909	0.300	0.0681	3058.5	0.0110	76.1	0.0889
910	0.275	0.0625	2803.6	0.0110	76.1	0.1058

APPENDIX D

APPENDIX D

SAMPLE OF DATA REDUCTION AND CALCULATION PROCEDURE OF HEAT TRANSFER COEFFICIENTS

The raw data of all experimental runs were fed into a computer program, for a Zenith microcomputer Model Z-150, which was written to reduce the data into useful form. The computer listing is given in Appendix B. As was pointed earlier, the experimental runs in the reduced data are coded in four digit numbers. The first digit represents the tube number, the second digit represents the set number and the remaining two digits represent the run number. A sample of the calculation procedure for run number 3001 is given below. Since all the experimental measurements like pressures, temperatures, flow rates etc. were in British units, the calculations for data reduction were done in British units and quantities of interest were converted into SI system of units. During the calculation procedure reference is made to the thermocouple locations and stations 1 through 5. These locations are shown in Figure D.1. Properties of R-113 were taken from ASHRAE Tables [32,33].

B.1 HEAT TRANSFER CALCULATIONS

Recorded Experimental Data for Run number 3001

Reference (ambient) temperature	= 74°F
Atmospheric pressure	= 14.12 psia
Tube I.D., D_i	= 0.494 in.
Tube O.D., D_o	= 0.618 in.
Water jacket inside diameter, D_j	= 1.500 in.
Length of subsection I, L_I	= 27.4375 in.
Length of subsection II, L_{II}	= 27.5000 in.
Length of subsection III, L_{III}	= 27.3750 in.

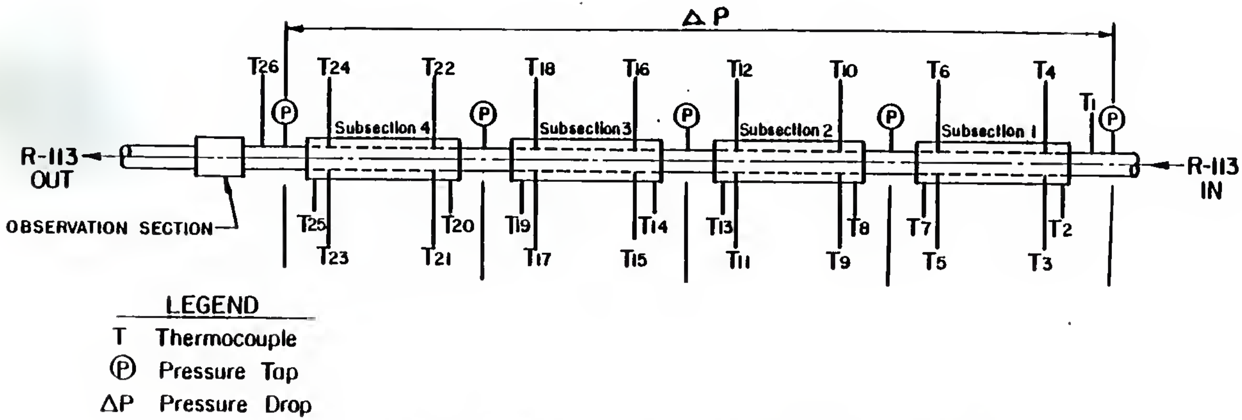

Fig. D.1 Thermocouple locations and locations of stations.

Length of subsection IV, L_{IV}	= 27.4375 in.
Tube thermal conductivity, k	= 220 Btu/hr ft^2 ^oF
Test fluid flow rate, GPM_R	= 0.275 gal/min.
Cooling water flow rate, GPM_{wa}	= 2.50 gal/min.
Test section inlet pressure, P_{in}	= 15.486 psia

<u>Coolant Temperatures</u>

Coolant inlet temperature at subsection IV, T_{25}	= 58.28 ^oF
Coolant outlet temperature at subsection IV, T_{20}	= 58.28 ^oF
Coolant inlet temperature at subsection III, T_{19}	= 58.10 ^oF
Coolant outlet temperature at subsection III, T_{14}	= 58.10 ^oF
Coolant inlet temperature at subsection II, T_{13}	= 58.28 ^oF
Coolant outlet temperature at subsection II, T_8	= 59.00 ^oF
Coolant inlet temperature at subsection I, T_7	= 58.46 ^oF
Coolant outlet temperature at subsection I, T_2	= 59.90 ^oF

<u>R-113 Temperatures</u>

Test fluid inlet temperature, T_1	= 102.02 ^oF
Test fluid outlet temperature, T_{26}	= 60.44 ^oF
Test fluid temperature at flowmeter, T_{27}	= 65.30 ^oF

<u>Wall Temperatures in ^oF</u>

Subsection I	Subsection II	Subsection III	Subsection IV
T_3 = 66.74	T_9 = 66.02	T_{15} = 59.00	T_{21} = 59.18
T_4 = 69.62	T_{10} = 61.88	T_{16} = 59.54	T_{22} = 58.28
T_5 = 66.20	T_{11} = 60.44	T_{17} = 58.28	T_{23} = 58.28
T_6 = 64.40	T_{12} = 60.80	T_{18} = 59.18	T_{24} = 58.82

Calculation Procedure

Sectional inside heat transfer area, A_{il}:

$$A_{il} = D_i \, L_l \quad \text{where } l = I, II, III \text{ or } IV$$

$$A_{iI} = (0.494)(27.4375)/144 = 0.29570 \text{ ft}^2$$

$$A_{iII} = (0.494)(27.5000)/144 = 0.29638 \text{ ft}^2$$

$$A_{iIII} = (0.494)(27.3750)/144 = 0.29503 \text{ ft}^2$$

$$A_{iIV} = (0.494)(27.4375)/144 = 0.29570 \text{ ft}^2$$

Total inside heat transfer area, A_{it}:

$$A_{it} = A_{iI} + A_{iII} + A_{iIII} + A_{iIV} = 1.18282 \text{ ft}^2$$

Sectional outside heat transfer area, A_{ol}:

$$A_{ol} = D_o \, L_l \quad \text{where } l = I, II, III \text{ or } IV$$

$$A_{oI} = (0.618)(27.4375)/144 = 0.36993 \text{ ft}^2$$

$$Ao_{II} = (0.618)(27.5000)/144 = 0.37077 \text{ ft}^2$$

$$A_{oIII} = (0.618)(27.3750)/144 = 0.36909 \text{ ft}^2$$

$$A_{oIV} = (0.618)(27.4375)/144 = 0.36993 \text{ ft}^2$$

Total inside heat transfer area, A_{ot}:

$$A_{ot} = A_{oI} + A_{oII} + A_{oIII} + A_{oIV} = 1.47972 \text{ ft}^2$$

Cross sectional flow area, A_{fn}:

$$A_{fn} = \frac{\pi D_i^2}{4} = \frac{\pi (0.494)^2}{(4)(144)} = 0.001331 \text{ ft}^2$$

Annular flow area for water, A_o:

$$A_o = \frac{\pi (D_j^2 - D_o^2)}{4} = \frac{\pi (1.5^2 - 0.618^2)}{(4)(144)} = 0.0101888 \text{ ft}^2$$

Coolant mass flow rate, m_{wa}:

$$m_{wa} = 2.50 \frac{\text{gallon}}{\text{minute}} \times 60 \frac{\text{minute}}{\text{hour}} \times 0.13368 \frac{\text{ft}^3}{\text{gallon}} \times 62.30 \frac{\text{lbm}}{\text{ft}^3}$$

$= 1249.24$ lbm/hr

R-113 mass flow rate, m_R:

$$m_R = 0.275 \frac{\text{gallon}}{\text{minute}} \times 60 \frac{\text{minute}}{\text{hour}} \times 0.13368 \frac{\text{ft}^3}{\text{gallon}} \times 98.63 \frac{\text{lbm}}{\text{ft}^3}$$

$= 217.55$ lbm/hr

Mass flux of water, G_{wa}:

$$G_{wa} = m_{wa}/A_o = \frac{1249.24}{0.010188} = 122609.50 \text{ lbm/hr.ft}^2$$

Mass flux of R-113, G_R:

$$G_R = m_R/A_{fn} = \frac{217.55}{0.001331} = 163447.90 \text{ lbm/hr.ft}^2$$

Total heat gain by water, Q_{wa}:

$$Q_{wa} = m_{wa}\, Cp_{wa}\, (T_2 - T_{25}) = (1249.24)(1)(59.90-58.28)$$

$= 2023.8$ Btu/hr

Enthalpy of R-113 at station 1 (inlet to heat exchanger), h_1:

h_1 is the specific enthalpy of liquid R-113 at $T_1 = 102.02\ ^oF$.

$h_1 = 29.43$ Btu/lbm

Enthalpy of R-113 at station 5 (outlet of heat exchanger), h_5:

h_5 is the specific enthalpy of liquid R-113 at $T_{26} = 60.44\ ^oF$.

$h_5 = 20.40$ Btu/lbm

Total heat lost by R-113, Q_R:

$$Q_R = m_R\,(h_1 - h_5)$$

$= 217.55\ (29.43 - 20.40) = 1964.84$ Btu/hr

Heat Balance error in percentage, Error %:

$$\text{Error \%} = 100 \times (Q_{wa} - Q_R)/Q_{wa}$$
$$= 100 \times (2023.77 - 1964.84)/2023.77$$
$$= 2.91\ \%$$

Inlet and outlet temperatures of R-113:

Inlet temperature of R-113 = T_{Ri} = 102.02 ^{o}F.

Outlet temperature of R-113 = T_{Ro} = 60.44 ^{o}F.

Inlet and outlet temperatures of cooling water:

Inlet temperature of cooling water = T_{wai} = 58.28 ^{o}F.

Outlet temperature of cooling water = T_{wao} = 59.90 ^{o}F.

Average outside wall temperature for each section, T_{oi}, i=I,II,III or IV:

Average outside wall temperature for subsection I = T_{oI}

$$= (T_3+T_4+T_5+T_6)/4$$
$$= 66.74\ ^{o}F$$

Average outside wall temperature for subsection II = T_{oII}

$$= (T_9+T_{10}+T_{11}+T_{12})/4$$
$$= 61.34\ ^{o}F$$

Average outside wall temperature for subsection III = T_{oIII}

$$= (T_{15}+T_{16}+T_{17}+T_{18})/4$$
$$= 59.00\ ^{o}F$$

Average outside wall temperature for subsection IV = T_{oIV}

$$= (T_{21}+T_{22}+T_{23}+T_{24})/4$$
$$= 58.64\ ^{o}F$$

Average outside wall temperature of the tube, $\bar{T}_w$:

$$\bar{T}_w = (T_{oI} + T_{oII} + T_{oIII} + T_{oIV})/4 = 61.45\ ^{o}F$$

Overall LMTD:

$$LMTD = \frac{(T_{Ri} - T_{wao}) - (T_{Ro} - T_{wai})}{\ln[(T_{Ri} - T_{wao})/(T_{Ro} - T_{wai})]} = 13.45\ {}^{o}F$$

LMTD between the wall and water, $LMTD_o$:

$$LMTD_o = \frac{T_{wao} - T_{wai}}{\ln[(\bar{T}_w - T_{wai})/(\bar{T}_w - T_{wao})]}$$

$$= 2.26\ {}^{o}F$$

Heat duty, UA:

$$UA = Q_{wa}/LMTD$$

$$= 150.44\ Btu/hr\ {}^{o}F$$

Outside (water side) heat transfer coefficient, h_o:

$$h_o = Q_{wa}/((A_{oI}+A_{oII}+A_{oIII}+A_{oIV}) \times LMTD_o)$$

$$= 606.48\ Btu/hr\ ft^2\ {}^{o}F$$

Inside (R-113 side) heat transfer coefficient, h_i:

$$h_i = \frac{D_o/D_i}{(A_o/UA) - (1/h_o) - (D_o/2k)\ln(D_o/D_i)}$$

$$= 153.29\ Btu/hr\ ft^2\ {}^{o}F$$

Nusselt number of water, Nu_{wa}:

$$Nu_{wa} = h_o \times (D_j - D_o) / k_{wa}$$

$$= 131.51$$

Reynolds number of water, Re_{wa}:

$$Re_{wa} = G_{wa}(D_j - D_o)/\mu$$

$$= 3253.36$$

Prandtl number of water, Pr_{wa}:

$$Pr_{wa} = \mu \, Cp / k_{wa}$$
$$= 8.217$$

Nusselt number of R-113, Nu_R:

$$Nu_R = h_i \, D_i \, / \, k_R$$
$$= 146.36$$

Reynolds number of R-113, Re_R:

$$Re_R = G_R \, D_i \, / \, \mu$$
$$= 4220.61$$

Prandtl number of R-113, Pr_R:

$$Pr_R = \mu \, Cp_R \, / \, k_R$$
$$= 8.481$$

B.2 PRESSURE DROP CALCULATIONS

Recorded Experimental Data for Run number 301

Temperature of R-113, T_R	= 75.2 °F
GPM of R-113, GPM_R	= 0.500 GPM
Length of the tube, L	= 133.75 in.
Inside diameter of the tube, D_i	= 0.494 in.
Pressure drop, Δp	= 0.0901 psi

Calculation Procedure

R-113 mass flow rate, m_R:

$$m_R = 0.500 \, \frac{\text{gallon}}{\text{minute}} \times 60 \, \frac{\text{minute}}{\text{hour}} \times 0.13368 \, \frac{\text{ft}^3}{\text{gallon}} \times 97.84 \, \frac{\text{lbm}}{\text{ft}^3}$$

$$= 392.36 \text{ lbm/hr}$$

Mass flux of R-113, G_R:

$$G_R = m_R/A_{fn} = 392.36/0.001331 = 294787.78 \text{ lbm/hr.ft}^2$$

Reynolds number of R-113, Re_{R113}:

$$Re_R = G_R\, D_i \;/\; \mu_R$$
$$= (294787.78)(0.494/12)/(1.668) \qquad = 7274.71$$

Volume flow rate, Q_R:

$$Q_R = 0.5\ \frac{\text{gallon}}{\text{minute}} \times 60\ \frac{\text{minute}}{\text{hour}} \times 0.13368\ \frac{\text{ft}^3}{\text{gallon}} = 4.0104\ \frac{\text{ft}^3}{\text{hr}}$$

$$= 4.010 \text{ ft}^3/\text{hr}$$

Pumping power, P:

$$P = 0.0901\ \frac{\text{lbf}}{\text{in}^2} \times 144\ \frac{\text{in}^2}{\text{ft}^2} \times 4.010\ \frac{\text{ft}^3}{\text{hr}} \times \frac{1}{778}\ \frac{\text{Btu}}{\text{lbf ft}}$$

$$= 0.06684 \text{ Btu/hr}$$

Friction factor, f:

$$f = 2\, \Delta p\, D_i \;/\; L\, \rho\, V^2$$

$$= \frac{0.0901\ \frac{\text{lbf}}{\text{in}^2} \times 144\ \frac{\text{in}^2}{\text{ft}^2} \times 32.2\ \frac{\text{lbm ft}}{\text{s}^2\text{ lbf}} \times 3600^2\ \frac{\text{s}^2}{\text{hr}^2}}{\frac{13.75 \text{ in.}}{0.94 \text{ in.}} \times 0.5 \times 97.84\ \frac{\text{lbm}}{\text{ft}^3} \times \left(\frac{294787.78 \text{ lbm ft}^3}{97.836\ \text{ft}^2 \text{ hr lbm}}\right)^2}$$

$$= 0.04512$$

APPENDIX E

APPENDIX E

UNCERTAINTY ANALYSIS IN EXPERIMENTAL MEASUREMENTS OF OVERALL HEAT TRANSFER COEFFICIENT

The parameter of interest in the error analysis is the uncertainty in the experimental measurements of the overall heat transfer coefficients. The method outlined by Kline and McClintock [34] was used in estimating the uncertainty associated with the overall heat transfer conductance UA for the internally finned tube, run number 7310.

The overall heat transfer coefficient was calculated by:

$$UA = \frac{Q}{\dfrac{(T_{Ri} - T_{wao}) - (T_{Ro} - T_{wai})}{\ln[(T_{Ri}-T_{wao})/(T_{Ro}-T_{wai})]}} \qquad \text{(E-1)}$$

The experimental uncertainty for the overall average heat transfer coefficient is given by:

$$w_{UA}=\left[\left(\frac{\partial UA}{\partial Q} w_Q\right)^2+\left(\frac{\partial UA}{\partial T_{Ri}} w_{TRi}\right)^2+\left(\frac{\partial UA}{\partial T_{Ro}} w_{TRo}\right)^2+\left(\frac{\partial UA}{\partial T_{wai}} w_{Twai}\right)^2+\left(\frac{\partial UA}{\partial T_{wao}} w_{Twao}\right)^2\right]^{1/2} \qquad \text{(E-2)}$$

From equation (E-1):

$$\frac{\partial UA}{\partial Q} = \frac{1}{LMTD}$$

$$\frac{\partial UA}{\partial T_{Ri}} = \frac{Q\,[LMTD/(T_{Ri}-T_{wao}) - 1]}{LMTD^2\{\ln[(T_{Ri}-T_{wao})/(T_{Ro}-T_{wai})]\}} \qquad \text{(E-3)}$$

$$\frac{\partial UA}{\partial T_{Ro}} = \frac{Q\,[1 - LMTD/(T_{Ro}-T_{wai})]}{LMTD^2\{\ln[(T_{Ri}-T_{wao})/(T_{Ro}-T_{wai})]\}} \qquad \text{(E-4)}$$

$$\frac{\partial UA}{\partial T_{wai}} = \frac{Q\,[LMTD/(T_{Ro}-T_{wai}) - 1]}{LMTD^2\{\ln[(T_{Ri}-T_{wao})/(T_{Ro}-T_{wai})]\}} \qquad (E-5)$$

$$\frac{\partial UA}{\partial T_{wao}} = \frac{Q\,[1 - LMTD/(T_{Ri}-T_{wao})]}{LMTD^2\{\ln[(T_{Ri}-T_{wao})/(T_{Ro}-T_{wai})]\}} \qquad (E-6)$$

For this run:

$Q = 1675.16$ Btu/hr

$T_{iR113} = 103.5\ ^{o}F$

$T_{oR113} = 68.0\ ^{o}F$

$T_{iwa} = 61.9\ ^{o}F$

$T_{owa} = 66.2\ ^{o}F$

$LMTD = 17.24\ ^{o}F$

Substituting these values in equations (E-3) - (E-6):

$$\frac{\partial UA}{\partial Q} = 0.058$$

$$\frac{\partial UA}{\partial T_{Ri}} = -1.67$$

$$\frac{\partial UA}{\partial T_{Ro}} = -5.69$$

$$\frac{\partial UA}{\partial T_{wai}} = 5.69$$

$$\frac{\partial UA}{\partial T_{wao}} = 1.67$$

The estimation of the uncertainty for the energy transfer, w_Q:

The energy transfer is given by:

$$Q = m_{wa}\, Cp_{wa}\, (T_{wao} - T_{wai})$$

The uncertainty in Q is given by:

$$w_Q=[(\frac{\partial Q}{\partial m_{wa}} w_{mwa})^2+(\frac{\partial Q}{\partial Cp_{wa}} w_{Cpwa})^2+(\frac{\partial Q}{\partial T_{wao}} w_{Twao})^2+(\frac{\partial Q}{\partial T_{wai}} w_{Twai})^2]^{1/2}+\dot{Q}_e \quad (E-7)$$

where $\dot{Q}_e$ = heat exchange with the environment.

For this run:

m_{wa} = 399.75 lbm/hr

Cp_{wa} = 1.0 Btu/lbm ^oF

Therefore:

$$\frac{\partial Q}{\partial m_{wa}} = Cp_{wa}\ (T_{wai}-T_{wao}) = 4.3 \text{ Btu/lbm}$$

$$\frac{\partial Q}{\partial Cp_{wa}} = m_{wa}\ (T_{wao}-T_{wai}) = 1718.96 \text{ lbm } ^oF/hr$$

$$\frac{\partial Q}{\partial T_{wao}} = m_{wa}\ Cp_{wa} = 399.75 \text{ Btu/hr } ^oF$$

$$\frac{\partial Q}{\partial T_{wai}} = -m_{wa}\ Cp_{wa} = -399.75 \text{ Btu/hr } ^oF$$

The uncertainty associated with the water flow was assumed to be within ± 1% of the flow rate. Therefore:

w_{mwa} = 0.01 x 399.76 = 0.4 lbm/hr

and the uncertainty of the specific heat w_{Cpwa} was assumed to be:

w_{Cpwa} = 0.004 Btu/lbm ^oF

The uncertainties in T_{wao}, T_{wai}, T_{iR113} and T_{oR113} were determined from the uncertainties of thermocouples and data acquisition system readings as follows:

Uncertainties due to thermocouple wire inaccuracies: ±0.5 ^oF

Uncertainties due to data acquisition system: ±0.01% of the reading.

Therefore:

$$w_{Twao} = [\ (0.50)^2 + (66.2 \times 0.0001)^2\ ]^{1/2} = 0.50\ ^oF$$

$$w_{Twai} = [\ (0.50)^2 + (61.9 \times 0.0001)^2\ ]^{1/2} = 0.50\ ^oF$$

$$w_{TRi} = [\ (0.50)^2 + (103.5 \times 0.0001)^2\ ]^{1/2} = 0.50\ ^oF$$

$$w_{TRo} = [\ (0.50)^2 + (68.0 \times 0.0001)^2\ ]^{1/2} = 0.50\ ^oF$$

Since the test condenser was completely insulated, the heat exchange with the environment was considered to be small and was estimated to be 1% of the heat transferred. Thus from equation (E-7):

$$w_Q = 300.47 \text{ Btu/hr}$$

Hence the uncertainty from equation (E-2) is:

$$w_{UA} = \pm\ 17.92 \text{ Btu/hr } ^oF.$$

Thus, for this run:

$$UA = 100.18 \pm 17.92 \text{ Btu/hr } ^oF.$$

Therefore, the uncertainty for the experimental overall average heat transfer coefficient of this run is about ±17.9%.

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SINGLE PHASE HEAT TRANSFER ENHANCEMENT BY DOUBLY AUGMENTED TUBES

ABSTRACT

The present investigation reports the results of an experimental study in which heat transfer and pressure drop data were taken during cooling of liquid R-113 inside 6 different double pipe, counter flow heat exchangers with R-113 flowing inside the inner tube and the cooling water flowing in the annulus. The heat exchangers consisted of two sets with three exchangers in each set. Within each set the inner tube of each exchanger had the same outside diameter. However, in one exchanger the inner tube had internal fins on the inside and a smooth surface on the outside. In the second exchanger the inner tube had knurls on the outside surface and a smooth surface on the inside. In the third exchanger the inner tube had the same fins of the first exchanger on the inside and the same knurls of the second exchanger on the outside. This tube is identified as a doubly augmented tube. The two sets of exchangers differed in the outside diameter of the inner tube and the number of fins. The results are summarized in the following:

1. Over the range of Reynolds number tested for R-113 (2700-7850), internally finned tubes enhanced the inside heat transfer coefficient by 200% over the smooth tube results on a nominal area basis.
2. The surface roughness by the knurls enhanced the water side heat transfer coefficient by 25% over the smooth surface over the Reynolds number range encountered (700-3500).
3. Heat transfer and pressure drop correlations were developed for cooling of R-113 inside the internally finned tubes. The heat transfer correlation predicted the heat transfer coefficient to within ±10% for 98% of the data points. The pressure drop correlation

predicted the pressure drop to within $\pm15\%$ for 62% of the data points.

4. The performance of the heat exchangers with inside doubly augmented tubes was compared with the performance of exchangers with inside singly augmented tubes (finned on the inside and smooth on the outside and/or smooth on the inside and knurled on the outside). The comparisons were based on evaluating the ratio of the overall heat conductance (UA) of any two exchangers subject to the constraint of same inlet conditions of R-113 and water (flow rates and temperatures). Also, comparisons were made by evaluating the ratio of the overall heat transfer conductance (UA) of any two exchangers subject to the constraint of the same pumping power of R-113.
5. The heat exchanger performance was also evaluated on the basis of the pumping power demand for R-113 for the same overall heat transfer rate and geometry.
6. The ratio of pumping power to the heat transfer rate for the same inlet conditions of R-113 and water was also used to compare the performance of heat exchangers with inner tubes that has internal fins or are smooth on the inside. The results showed that the power demand per unit heat transfer rate is higher for internally finned tubes.
7. The results also showed that the benefits of double augmentation versus single augmentation in improving the overall heat transfer of the heat exchanger depended mainly on the ratio of the inside to the outside heat transfer coefficients of the inner tube before augmenting the inside and outside surfaces. The benefits of having inside augmentation versus outside augmentation alone, as well as double augmentation of the inner tube of the heat exchanger are also discussed.