

**Development of an Experimental Facility
to Evaluate the Performance of Air-Cooled
Automotive and Household Refrigerator
Condensers Utilizing Ozone-Safe Refrigerants**

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ACRC TR-08

October 1991

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*Prepared as part of ACRC Project 03
Condenser Performance
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The Air Conditioning and Refrigeration Center was founded in 1988 with a grant from the estate of Richard W. Kritzer, the founder of Peerless of America Inc. A State of Illinois Technology Challenge Grant helped build the laboratory facilities. The ACRC receives continuing support from the Richard W. Kritzer Endowment and the National Science Foundation. The following organizations have also become sponsors of the Center.

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DEVELOPMENT OF AN EXPERIMENTAL FACILITY TO EVALUATE THE PERFORMANCE
OF AIR-COOLED AUTOMOTIVE AND HOUSEHOLD REFRIGERATOR CONDENSERS
UTILIZING OZONE-SAFE REFRIGERANTS

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ABSTRACT

Chlorofluorocarbons or CFC's have long been the refrigerants of choice in vapor compression refrigeration systems, including mobile air conditioning and household refrigerator/freezer applications. Due to environmental concerns regarding ozone layer depletion and global warming, international agreements have been legislated that will gradually reduce and eventually eliminate these refrigerants. Although alternative, ozone-safe refrigerants have been developed, their effect on the performance of air-cooled condensers must still be evaluated so that industry may more effectively design air conditioning and refrigeration system components in the future. A full condenser test apparatus has been designed and constructed to allow experimental determination of the heat transfer and friction characteristics of air-cooled condensers utilizing alternative refrigerants. Baseline testing has recently been conducted using CFC-12 and simulating a wide range of air-side and refrigerant-side conditions at the condenser inlet. Separate experiments were included either single phase or two phase refrigerant flow through the condenser. Modified Wilson Plot procedures have been implemented to evaluate the air-side heat transfer resistances and to develop a Colburn j-factor correlation for an automotive condenser with two rows of staggered, wavy finned tubes. The correlation agrees fairly well with other empirical correlations appearing in the literature for the same type of condenser geometry. Additional expected trends in the experimental data were observed; however, existing uncertainties in temperature and pressure measurements require that refinements be made to the system in the near future.

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NOMENCLATURE

a	Reynolds number exponent	-	-
A	Total heat transfer surface area on one side of heat exchanger	ft ²	m ²
A _c	Intube flow cross sectional area	ft ²	m ²
A _{ct}	Minimum free flow area based on a bare tube bank	ft ²	m ²
A _f	Fin or secondary surface area	ft ²	m ²
A _{fr}	Frontal or face area of the exchanger	ft ²	m ²
A _i	Tube inside heat transfer surface area	ft ²	m ²
A _{min}	Minimum free flow area	ft ²	m ²
A _o	Outside heat transfer surface area (A _p + A _f)	ft ²	m ²
A _p	Primary surface area	ft ²	m ²
A _t	Outside surface area of bare tubes alone	ft ²	m ²
b	Prandtl number exponent or Reduced fin height	- ft	- m
C _{air}	Air heat capacity rate	Btu/h °F	W/°C
C _{max}	Maximum heat capacity rate	Btu/h °F	W/°C
C _{min}	Minimum heat capacity rate	Btu/h °F	W/°C
C _{ref}	Refrigerant heat capacity rate	Btu/h °F	W/°C
C _r	Heat capacity rate ratio	-	-
c _p	Specific heat of fluid at constant pressure	Btu/lb _m °F	J/kg °C
D	Nominal tube diameter	ft	m
D _c	Fin collar outside diameter	ft	m
D _h	Hydraulic diameter	ft	m
d _i	Tube inside diameter (I.D.)	in	mm

d_o	Tube outside diameter (O.D.)	i n	m m
f	Fanning friction factor	-	-
f'	Modified friction factor per tube row	-	-
f_F	Friction factor due to fins alone	-	-
G	Mass air velocity based on A_{min}	lb _m /h ft ²	kg/m ² s
Gz	Graetz number	-	-
h	Convective heat transfer coefficient or Enthalpy	Btu/h ft ² °F	W/m ² °C
h_f	Enthalpy of saturated liquid	Btu/lb _m	J/kg
h_g	Enthalpy of saturated vapor	Btu/lb _m	J/kg
h_L	Single-phase heat transfer coefficient	Btu/h ft ² °F	W/m ² °C
h_{TP}	Two-phase heat transfer coefficient	Btu/h ft ² °F	W/m ² °C
j	Colburn modulus or j-factor	-	-
j_N	j-factor corresponding to Nth row	-	-
k	Thermal conductivity	Btu/h ft °F	W/m °C
k_f	Thermal conductivity of fin material	Btu/h ft °F	W/m °C
k_w	Thermal conductivity of tube wall material	Btu/h ft °F	W/m °C
K_{min}	Minimum of $2K_{s1}$ and $2K_{s2}$	ft	m
K_{s1}	Condenser core dimension	ft	m
K_{s2}	Condenser core dimension	ft	m
l	Fin height	ft	m
L_1	Condenser core depth	ft	m
L_2	Condenser core width	ft	m
L_3	Condenser core height	ft	m
m	Fin efficiency parameter or Property ratio exponent	ft ⁻¹	m ⁻¹
$\dot{m}$	Mass flow rate	lb _m /h r	lb _m /h r

n_f	Number of fins per unit length	ft^{-1}	m^{-1}
N, N_R	Number of tube rows in air flow direction	-	-
N_f	Total number of fins in heat exchanger	-	-
N_t	Total number of tubes in heat exchanger	-	-
NTU	Number of heat transfer units	-	-
Nu	Nusselt number	-	-
n	Property ratio exponent		
P_b	Barometric pressure	lb_f/ft^2	Pa
P_s	Static pressure	lb_f/ft^2	Pa
ΔP	Pressure drop	lb_f/ft^2	Pa
Pr	Prandtl number	-	-
Q, q	Heat transfer rate	Btu/h	J
q_b	Heat transfer rate at fin base	Btu/h	W
q_e	Heat transfer rate at fin tip	Btu/h	W
q_{cond}	Condensing region heat transfer	Btu/h	J
q_{desuper}	Desuperheating region heat transfer	Btu/h	J
q_{sub}	Subcooling region heat transfer	Btu/h	J
R_{air}	Air-side thermal resistance	$h \text{ } ^\circ\text{F}/\text{Btu}$	$^\circ\text{C}/\text{W}$
$R_{f,i}$	Fouling resistance of inside surface	$h \text{ } ^\circ\text{F}/\text{Btu}$	$^\circ\text{C}/\text{W}$
$R_{f,o}$	Fouling resistance of outside surface	$h \text{ } ^\circ\text{F}/\text{Btu}$	$^\circ\text{C}/\text{W}$
R_{ref}	Refrigerant-side thermal resistance	$h \text{ } ^\circ\text{F}/\text{Btu}$	$^\circ\text{C}/\text{W}$
R_w	Wall thermal resistance	$h \text{ } ^\circ\text{F}/\text{Btu}$	$^\circ\text{C}/\text{W}$
Re, Re_D	Reynolds number based on D_h	-	-
Re_d	Reynolds number based on d_o	-	-
Re_{eq}	Reynolds number based on equivalent mass flux	-	-
Re_g	Vapor phase Reynolds number	-	-

Re_l	Liquid phase Reynolds number	-	-
Re_L	Reynolds number based on S_l	-	-
r_d	Hydraulic radius	ft	m
r_i	Tube inner radius	ft	m
r_o	Tube outer radius	ft	m
s	Spacing between adjacent fins ($S_t - d_o$)	in	mm
S_d	Peak to valley wave depth for wavy fins or Diagonal between adjacent tube rows	in	mm
S_p	One-half the wave length for wavy fins	in	mm
St	Stanton number	-	-
T	Temperature	$^{\circ}F$	$^{\circ}C$
T_a	Ambient fluid temperature	$^{\circ}F$	$^{\circ}C$
T_b	Fin base temperature or Bulk mean fluid temperature	$^{\circ}F$	$^{\circ}C$
T_{db}	Air dry-bulb temperature	$^{\circ}F$	$^{\circ}C$
T_{wb}	Air wet-bulb temperature	$^{\circ}F$	$^{\circ}C$
T_w	Temperature evaluated at tube wall	$^{\circ}F$	$^{\circ}C$
ΔT	Temperature difference	$^{\circ}F$	$^{\circ}C$
U	Overall heat transfer coefficient	Btu/h ft ² $^{\circ}F$	W/m ² $^{\circ}C$
UA	Overall thermal conductance	hr $^{\circ}F$ /Btu	$^{\circ}C$ /W
V	Velocity	ft/s	m/s
v	Specific volume	ft ³ /lb _m	m ³ /kg
W	Mass flow rate	lb _m /s	kg/s
S_t	Transverse tube pitch	in	mm
S_l	Longitudinal tube pitch	in	mm

Greek Symbols

α	Ratio of outside heat transfer surface area to total volume (A_o/Volume)	ft^2/ft^3	m^2/m^3
δ	Fin thickness	in.	mm
ε	Effectiveness	-	-
η_f	Fin efficiency	-	-
η_o	Overall surface efficiency	-	-
μ	Absolute or dynamic viscosity	$\text{lb}_m/\text{ft h}$	m Pa/sec
μ_m	Viscosity at mean fluid temperature	$\text{lb}_m/\text{ft h}$	m Pa/sec
μ_w	Viscosity at tube wall temperature	$\text{lb}_m/\text{ft h}$	m Pa/sec
π	$\text{pi} = 3.14159$	-	-
ρ	Density	lb_m/ft^3	kg/m^3
σ	Ratio of minimum free flow area to frontal area ($A_{\min}/A_{fr}$)	-	-
ω	Air humidity ratio	-	-

CHAPTER 1

INTRODUCTION

1.1. Background

In June of 1974, Molina and Rowland [74] first postulated that certain halogenated chlorofluorocarbons, in particular CFC-11 and CFC-12, may pose a threat to the ozone layer surrounding the earth, i.e., the stratospheric layer which protects the earth from harmful ultraviolet radiation from the sun. However, it was not until the mid-1980's that significant public and scientific attention focused on the possible environmental and health consequences of ozone layer depletion and climate modification. This dramatic shift was due to further studies that confirmed partial depletion of the stratospheric ozone layer over Antarctica and that revealed an additional link between halogenated chlorofluorocarbons and the so-called "greenhouse effect", i.e., the potential for global warming and climate change as a result of chlorinated and brominated compound emissions. A comprehensive survey of the research related to ozone layer depletion and greenhouse warming is summarized by Embler [40].

In September 1987, in response to the increasing public and scientific community concern of ozone layer depletion and the "greenhouse effect", the United States along with twenty-four other nations and the European Economic Community reached a landmark global agreement and signed the Montreal Protocol on Substances that Deplete the Ozone Layer [120]. The Protocol mandates international action to limit the production of certain chlorofluorocarbons (CFC's) that have been linked to the destruction of the world's ozone layer in addition to providing for continual reassessment of the scientific data connecting CFC's and stratospheric ozone levels. Specifically, the original Protocol currently limits the manufacture of the refrigerants CFC-11, CFC-12, CFC-113, CFC-114 and CFC-115 to 1986 production levels in terms of their respective ozone depleting potentials. This production freeze shall be followed by further major curtailments in mid-1993 and mid-1998 of 20 percent and 50 percent, respectively. During recent meetings in May 1989 and June 1990, the contracting parties of the United Nations Environment Programme (UNEP) ratified additional declarations to the Montreal Protocol [121]. These provisions include a complete phase-out of all CFC's by the year 2000 for developed countries and by the year 2010 for underdeveloped

countries, in addition to regular review of the utilization of transitional substances, namely the HCFC's, with a view of replacing them by no later than 2040 and if possible, by 2020.

In April 1991, the Environmental Protection Agency announced that ozone layer loss at the latitudes of the U.S. is proceeding at a rate of more than twice that previously predicted by researchers. [108].

As indicated, one of the fully halogenated chlorofluorocarbons that is scheduled for eventual elimination is dichlorodifluoromethane (CCl_2F_2), or as more commonly referred to as refrigerant CFC-12 or R-12. CFC-12 has long been the refrigerant of choice in mobile air conditioning and household refrigerator/freezer applications. In all, the refrigeration, air conditioning and heat pump sector represents 25 percent of the global consumption of chlorofluorocarbons [66]. In addition, the global refrigerant R-12 consumption for charging and servicing automobile air conditioning systems accounts for 50 percent of the total CFC usage [122]. Inevitably, the eventual ban of CFC-12 near the turn of the next century will force the air conditioning and refrigeration industry to find a replacement refrigerant.

The primary alternative refrigerant that is currently being evaluated as a replacement for CFC-12 is the hydrofluorocarbon HFC-134a. Refrigerant HFC-134a, chemical name of tetrafluoroethane ($\text{CF}_3\text{CH}_2\text{F}$), contains no chlorine or bromine molecules and has zero ozone depleting potential. Conclusions regarding the global warming potential of HFC-134a are less clear, making long-term usage of this alternative refrigerant questionable. However, one item for certain is that based on equal system efficiencies, the global warming potential of HFC-134a is significantly less than that of CFC-12 [81]. Further, since no other competitive alternatives appear to be available and in order to abide with the phase-out regulations for R-12, the adoption of HFC-134a as an environmentally acceptable, working fluid seems to be inevitable.

One of the additional important problems with HFC-134a is that it possesses different thermodynamic and transport properties than CFC-12. To date, only a limited amount of experimental data has been published on the thermodynamic and transport characteristics of HFC-134a [69,83,132]. This subsequently means that extensive research must still be conducted in order to ensure the viability of HFC-134a as an alternative refrigerant in particular its miscibility with lubricants and compatibility with materials. The air conditioning and refrigeration industry requires a better understanding of how HFC-134a will affect the performance of their products in order to be able to design future systems more effectively.

1.2. Purpose of ACRC and Project

In response to the Montreal Protocol mandate, improved appliance energy efficiency standards [123] enacted by the National Appliance Energy Conservation Act (NAECA), and the industry need for research and development of alternative refrigerants for use with mobile air conditioning and household refrigerator/freezer systems, a National Science Foundation (NSF) Industry/University Cooperative Research Center for Air Conditioning and Refrigeration Technologies was established at the University of Illinois at Urbana-Champaign in November 1988. The Air Conditioning and Refrigeration Center (ACRC) has two fundamental purposes [1]. "The first is to contribute to the technology base for a new generation of equipment that must eliminate reliance on chlorofluorocarbons refrigerants (CFC's) that are linked to the problem of stratospheric ozone depletion. The second is to provide an opportunity for manufacturers to coordinate research and share results, as the international marketplace grows increasingly competitive in response to the energy efficiency standards being adopted by many countries."

The test apparatus described in this document has been specifically designed and constructed to conduct some of the research needed with mobile air conditioning and household refrigerator/freezer systems and alternative refrigerants. This ACRC SP03 project, officially titled Condenser Performance, was initiated in January 1990. It is a component-oriented project that fits between the fundamental single tube, condensation heat transfer experiments and the full system experiments which are additional parts of the ACRC. Some of the

information gained from this condenser research shall be utilized in developing mechanistic simulation models for heat exchanger design and optimization.

In order to establish a reference base for current and future experiments with HFC-134a, the operating characteristics of the full condenser system have been recently evaluated with CFC-12, and the results are documented herein. Operating conditions typically found in condensers of mobile air conditioning and household refrigerator/freezer systems have been simulated. Short term goals include the investigation of steady-state condenser performance at practical operating conditions and the evaluation of air-side heat transfer characteristics. Long term goals include transient performance tests, the formulating of comprehensive heat exchanger simulation models, analysis of the effects of airflow maldistributions, and the development of improved heat exchanger surfaces including optimization of refrigerant circuiting and of condenser weight and volume.

1.3. Summary of Contents

This thesis document is divided into a total of nine chapters. For chapters containing figures or tables or both, these items may be found organized at the end of each chapter.

Chapter 2 provides a comprehensive review of the published literature related to the determination of the air-side heat transfer and friction (pressure drop) characteristics of extended surface, fin-and-tube heat exchangers. This Chapter includes documentation of several air-side convective heat transfer coefficient correlations published to date and an introduction of the effects of changes in surface geometry (e.g., tube arrangement, number of tube rows and fin density) and of flow maldistribution and variable fluid properties. In addition, a brief review of refrigerant-side condensation and available single phase and two phase flow correlations is presented.

Chapter 3 describes the characteristics inherent in the design and construction of the full condenser experimental facility. Components of the air-side and refrigerant-side loops, including instrumentation and the data acquisition system, are presented in full detail. In addition, start-up, operating and maintenance procedures are described.

Chapter 4 details one of the alternative methods commonly employed to evaluate the outside heat transfer coefficients in heat exchangers, particularly evaporators and condensers. It is sometimes called the Wilson Plot technique; however, other researchers have formulated their own "modified" approaches to overcome the limitations of the original methodology. The adaptation of modified Wilson Plot techniques to finned tube, air-cooled condensers is also presented.

Chapter 5 examines the data reduction techniques employed in order to properly analyze experimental data. Calculation procedures, including formulas, are documented for the heat transfer rate, the overall thermal resistance, and individual heat transfer resistances and coefficients.

Chapter 6 describes the experimental testing performed in order to quantify the air-side heat transfer resistance. Tabulated summaries indicate the testing conditions utilized for single phase and two phase refrigerant flow experiments. In addition, this Chapter contains the proposed testing envelope for evaluating the steady-state performance of condensers in mobile air conditioning and household refrigerator/freezer applications.

Chapter 7 furnishes the results of analyzing the experimental data, including several modified Wilson Plots utilized to determine the air-side heat transfer resistance over a wide range of flow conditions and correlation of the Colburn j -factors. A comparison of single phase and two phase refrigerant flow data is made in order to validate the results and the correlation.

Chapter 8 identifies the uncertainty analysis techniques employed and quantifies the individual and overall uncertainties existing in the single phase and two phase experimental results.

Finally, Chapter 9 summarizes the project achievements and recommends improvements that may be made to the experimental facility to reduce the uncertainty in the final results. In addition, the relationship between some of the future work necessary with alternative refrigerants and the full condenser experimental apparatus is examined.

CHAPTER 2

LITERATURE SEARCH

The literature that encompasses heat exchanger research forms an extensive database. Studies include both the experimental and analytical evaluation of local and average heat transfer and friction (pressure drop) characteristics. A comprehensive search and review of the literature related to the air-side and refrigerant-side heat transfer and friction behavior of forced air-cooled condensers was performed. Some of the results are presented within this Chapter.

2.1. General Condenser Design

The typical condensers used in mobile air conditioning and household refrigerator/freezer applications may be classified as compact heat exchangers. Fundamentally, a compact heat exchanger may be defined as a gas-to-gas, gas-to-liquid or gas-to-condensing fluid or evaporating fluid heat exchanger incorporating a heat transfer surface with a higher area density than conventional shell-and-tube heat exchangers. Although somewhat arbitrary, a compact heat exchanger typically possesses an area density of approximately 200 ft²/ft³ (656 m²/m³) or better. This higher area density or surface area to volume ratio is accomplished by the use of secondary surfaces or fins on one or more of the fluid sides and also by the use of small diameter tubes.

In most two fluid, compact heat exchanger applications involving a gas on one fluid side, the heat transfer surface on the gas-side is the one enhanced or increased by the use of extended surfaces or fins. This is primarily because the convective heat transfer coefficient for gases is typically one or two orders of magnitude less than that for water, oil, refrigerant and other liquids. By adding the heat transfer surface area to the gas-side, 'hA' products on both sides of the heat exchanger are made more equivalent. If the heat transfer surface area on the gas-side was not enhanced, then much larger tubes than would have to be employed, making the heat exchanger considerably more bulky and massive.

In general, there are six (6) important design and operating considerations related to extended surface, compact heat exchangers that may be summarized as follows [51]:

- 1) Usually at least one of the fluids is a gas.
- 2) Fluids must be relatively free of suspended particles and non-corrosive. If suspended particles are present, as typically discovered in automotive and industrial radiators, it must be accounted for in the design by oversizing the exchanger or by employing larger spacing between the fins to permit elimination of deposits.
- 3) The mechanical pumping power required to move fluids through the heat exchanger and to overcome fluid friction (pressure drop) may often be as important of a design constraint as the heat transfer rate.
- 4) Operating pressures and temperatures are more limited compared to conventional shell-and-tube heat exchangers due to construction features such as brazing, mechanical expansion, extrusion, etc.
- 5) Since the resultant shape of a compact heat exchanger is typically one having a large frontal area and a short flow length, the header design is important for achieving uniform flow distribution;
- 6) Each new design must be carefully evaluated separately for its heat transfer and pressure drop characteristics and operating limits on temperature and pressure. The market potential must be large enough to offset the sizable manufacturing, research and tooling costs.

Compact heat exchangers may be classified according to the types of compact elements they employ. The compact elements generally fall into one of the following four categories: (1) Tubular surfaces, (2) Finned tube surfaces, (3) Plate-fin surfaces, and (4) Matrix surfaces. Shah [97] and Kern and Kraus [60] furnish comprehensive pictorial and verbal descriptions of these different surface types.

The two most common types of extended surface heat exchangers are the plate-fin type, where fins are sandwiched between parallel plates, and the finned tube type, where fluid flow occurs normal to finned banks of smooth tubes. The characteristics of the finned tube construction are presented in more detail herein since they are more applicable to mobile air conditioning and household refrigerator/freezer applications.

In a finned tube heat exchanger, round and rectangular tubes are most commonly employed, although oval tubes are also being used. Depending on the specific application, fins may be implemented either on the outside, on the inside, or on both the outside and the inside of the tubes. External fins may be categorized as (1) continuous fins on an array of tubes, including the plain or flat, wavy, louver and interrupted fin types; (2) normal fins on individual tubes, referred to simply as individually finned tubes; and (3) longitudinal fins on individual tubes. An exchanger having continuous fins on an array of tubes is also often referred to as a plate-fin and tube heat exchanger. Since longitudinal fins on individual tubes are normally not classified as compact heat exchangers, they will not be considered in the discussion of compact heat exchangers in this Chapter. In Figures 2.1 and 2.2, two different types of finned tube surfaces are presented.

Fins are typically attached to tubes by soldering, brazing, welding, gluing, tension winding, or mechanical extrusion and expansion processes. Internal fins are either integral or attached fins. The typical fin density associated with a finned tube heat exchanger surface varies from 5 to 20 fins/inch (197 to 788 fins/meter). In addition, fin thickness ranges from 0.0035 to 0.010 inches (0.09 to 0.25 mm); and fin flow length ranges from 1 to 10 inches (25 to 250 mm).

For most refrigeration and air conditioning applications utilizing extended surface compact heat exchangers, crossflow is the common flow arrangement. This flow arrangement is advantageous because it simplifies the header design at the entrance and the exit for each fluid [97]. The header design is more complex for a counterflow heat exchanger since it is necessary to separate the fluids at each end. A parallel-flow arrangement is seldom ever used as a compact heat exchanger since it produces the lowest exchanger effectiveness for a given NTU, hence making the required size to achieve the equivalent performance of a crossflow unit too excessive.

2.2. The Condensing Process

Condensers have the primary function of removing the latent heat from vapors and changing them from the vapor state to the liquid state. In the condensing process, heat is transferred from the refrigerant to the air in three main phases: (1) desuperheating, (2) condensing, and (3) subcooling. Figure 2.3 shows the three changes of state of a typical refrigerant along with the corresponding changes in the cooling air temperature as it passes through the condenser.

As shown in the figure, refrigerant enters the condenser as superheated vapor and cools, or desuperheats, by single phase convection to the saturation temperature, after which condensation begins. If the tube wall is below the saturation temperature of the refrigerant, condensation may actually occur while the condenser core is still superheated. Eventually, all the vapor condenses to liquid, and in most instances, the condensed liquid is subcooled prior to leaving the condenser in a single phase convection process.

In a properly designed system, condensing occurs in approximately 85 percent of the condenser area at nearly constant temperature. The slight decrease in temperature indicated in the region of two phase flow is a result of pressure losses due to friction, momentum, acceleration, and gravity through the condenser. The desuperheating, condensing, and subcooling regions shown are merely for reference purposes. In reality, these zones may vary as much as 5 to 10 percent from that shown. The deviation is primarily dependent on the amount of refrigerant superheat at the entrance of the core and the air flow rate across the condenser. An article by Rubin [89] examines the three processes in greater detail, including information on the relative amount of heat release and the distribution of temperature in each region.

2.3. Refrigerant-Side Heat Transfer

The purpose of this section is to summarize some of the available single-phase and two-phase flow heat transfer coefficient correlations applicable to the forced convection condensation of refrigerant flowing inside of tubes. For a more detailed explanation of actual condensation theory, literature by Collier [30] and McAdams [68] are recommended for further reading.

2.3.1. Single Phase Flow Correlations

Many empirical and theoretical correlations exist in the literature for determining refrigerant-side heat transfer coefficients for single phase, fully developed turbulent flow in smooth pipes. Most of these correlations are expressed in terms of the Nusselt number, Reynolds number and Prandtl number. Chapter 3 of the 1989 ASHRAE Handbook of Fundamentals provides a generalized list of heat transfer correlations for single phase flow. These correlations are in the form of the original Nusselt correlation, namely:

$$Nu = C Re^a Pr^b \quad (2.3.1)$$

For fully developed turbulent flow in smooth tubes, the following relation is recommended by Dittus and Boelter [36]:

$$Nu = 0.023 Re^{0.8} Pr^n \quad (2.3.2)$$

where the Prandtl number exponent 'n' has the following values:

$$n = \begin{cases} 0.4 & \text{for heating of the fluid} \\ 0.3 & \text{for cooling of the fluid} \end{cases}$$

This classical correlation was developed for the ranges of $0.7 \leq Pr \leq 120$ and $2500 \leq Re \leq 1.24 \times 10^5$. The objective of providing different Prandtl number exponents for heating and cooling was to account for the variation of fluid properties with temperature.

A critical evaluation of additional analytical solutions and Reynolds analogy based correlations for heat transfer with constant properties and turbulent flow is presented by Webb [125]. Correlations that appear in Webb's evaluation include the analytical solutions of Deissler (1955), Sparrow, Hallman and Seigel (1957), and Petukhov and Popov (1955) and the Reynolds analogy based correlation of Friend and Metzner (1958), since these were the most popular correlations existing at the time. The dates in parentheses indicate the year in which the correlation was developed. The Petukhov-Popov correlation was determined to be the "most superior" in correlating existing heat and mass transfer data. In addition, the recommended equation was shown to represent the data more accurately than the classical empirical equations of Dittus-Boelter and Colburn.

The general form of the Petukhov-Popov correlation is as follows [82]:

$$Nu = \frac{(f/2) Re Pr}{C + 12.7 (f/2)^{1/2} (Pr^{2/3} - 1)} \left(\frac{\mu_b}{\mu_w} \right)^n \quad (2.3.3)$$

In publications containing simplified versions of Equation (2.2.3), the constant C is equal to 1.07. However, the parameter may actually be determined by the equation below.

$$C = 1.07 + \frac{900}{Re} - \frac{0.63}{1 + 10Pr} \quad (2.3.4)$$

In addition, in the Petukhov-Popov correlating equation, the Fanning friction factor, f, may be obtained from the following relationship for smooth tubes:

$$f = (1.58 \log_{10} Re - 3.28)^{-2} \quad (2.3.5)$$

or by using any one of the other Fanning friction factor correlations given by Equations (2.3.10) and (2.3.11).

The Petukhov-Popov correlation, Equation (2.3.3) is applicable for the following ranges:

$$\begin{array}{rclcl} 0.5 & \leq & Pr & \leq & 10^6 \\ 4000 & \leq & Re & \leq & 5 \times 10^6 \\ 0.08 & \leq & (\mu_b/\mu_w) & \leq & 40 \end{array}$$

Two publications containing a more complete survey of the available empirical and theoretical single phase heat transfer correlations for fully developed turbulent flow in smooth pipes have been published by Bhatti and Shah [18] and Shah and Johnson [100]. In each of these publications, the Gnielinski correlation [46] is suggested for determination of refrigerant-side heat transfer coefficients based on constant property, fully developed turbulent flow through a smooth circular tube. Comparisons with other correlations indicate that it yields more reliable results than the Dittus-Boelter [36] and the Petukhov-Popov [82] correlations. An additional primary advantage of the Gnielinski correlation is that it accurately covers a broader range than most other correlations.

The general expression of the Gnielinski correlation is as follows [46]:

$$Nu = \frac{(f/2) (Re - 1000) Pr}{1 + 12.7 (f/2)^{1/2} (Pr^{2/3} - 1)} \left(\frac{\mu_b}{\mu_w} \right)^n \quad (2.3.6)$$

It is applicable for the following ranges:

$$\begin{array}{rclcl} 0.6 & \leq & Pr & \leq & 10^5 \\ 2300 & \leq & Re & \leq & 5 \times 10^6 \\ 0.08 & \leq & (\mu_b/\mu_w) & \leq & 40 \end{array}$$

According to Bhatti and Shah [18], the Petukhov-Popov correlation agrees with the most reliable experimental data on heat and mass transfer within an accuracy of ± 5.0 percent and is often used as a basis in formulating other heat transfer coefficient correlations. However, since the Gnielinski correlation covers a lower range of Reynolds number than most other correlations, including Petukhov-Popov, and it agrees with the Petukhov-Popov correlation within $+7.8/-2.0$ percent, it is also widely accepted as a reference.

The Gnielinski correlation is actually a modification of the Petukhov-Popov correlation in order to represent the heat transfer behavior near the transition region between laminar and turbulent flow. Finally, in the same work, Shah indicates the traditional Dittus-Boelter relationship yields results that may be up to 94 percent higher than the Gnielinski equation.

Sleicher and Rouse [103] formulated a simpler correlation in power-law form that agrees with the Gnielinski correlation within 10 percent for $Re > 10^4$. It is as follows:

$$Nu = 5 + 0.015 Re^a Pr^b \quad (2.3.7)$$

where:

$$a = 0.88 - \frac{0.24}{4 + Pr} \quad (2.3.8)$$

$$b = 0.333 + 0.5 \exp(-0.6 Pr) \quad (2.3.9)$$

The Fanning friction factor appears as a variable in the Gnielinski and Petukhov-Popov correlations and Equation (2.3.5) is one relationship, derived by Filonenko and appearing in Shah [97], to determine its value for fully developed turbulent flow through a smooth circular tube. The classical and standard correlation for Fanning friction factor the Karman-Nikuradse correlation, is oftenly expressed in two different forms as shown by Equations (2.3.10) and (2.3.11):

$$\frac{1}{\sqrt{f}} = 4.0 \log_{10} (Re \sqrt{f}) - 0.4 \quad (2.3.10)$$

$$\frac{1}{\sqrt{4f}} = 2.0 \log_{10} (Re \sqrt{4f}) - 0.8 \quad (2.3.11)$$

These theoretically based correlations agree with experimental data within two percent for the Reynolds number range $4000 \leq Re \leq 3 \times 10^6$. Since these equations are theoretically derived, they may be extrapolated to arbitrarily large Reynolds numbers. The explicit Filonenko correlation, Equation (2.3.5), agrees with the implicit Karman-Nikuradse correlations within 0.5 percent for $3 \times 10^4 \leq Re \leq 10^7$ and within 1.8 percent at a Reynolds number of 10,000. Several other explicit correlations are summarized in Shah and Johnson [100].

Finally, the absolute viscosity ratio (μ_b/μ_w) is incorporated in the Petukhov-Popov and Gnielinski equations in order to account for the effects of temperature dependent fluid properties. On the basis of analytical and experimental solutions, different values for the exponent are given by Kays and London [59], depending on the flow situation. For refrigerant applications with relatively small temperature differences, the viscosity ratio is normally assumed to be unity, and subsequently, omitted from the computations. Fluid properties are then evaluated at the bulk mean temperature.

2.3.2. Two Phase Flow Correlations

In two phase refrigerant flow, the dependence of the heat transfer mechanism upon the local flow characteristics (i.e., the local flow regime, vapor velocity, and resulting shear at the liquid-vapor interface) complicates an accurate prediction of the inside convective heat transfer coefficient. Several experimental studies have successfully identified and investigated the two phase flow patterns that exist with condensing refrigerant R-12 inside horizontal tubes. Works by Soliman and Azer [105,106], Traviss and Rohsenow [118] and Tandon, Varma and Gupta [115] are good examples. Figure 2.4 depicts schematically the sequence of flow patterns reported in these studies for conditions of high and low mass flux. The major difference between the two parts of Figure 2.4 is the existence of mist flow in the entry part of the condenser at high mass fluxes. This flow pattern data base also represents one of the main sources in developing empirical flow regime maps for predicting the heat transfer coefficient for condensation of pure components in a horizontal tube, e.g., Breber et. al. [22] and Tandon et. al. [115]. A newly defined theoretical approach to the design of horizontal condenser tubes that emphasizes the relationship between heat transfer and the existing flow situation is documented by Nitheandan [79].

Two phase flow heat transfer correlations that were specifically derived for a certain flow pattern could only be located for the wavy flow regime and annular flow regime. Some of the more well-recognized correlations are presented in the following paragraphs.

Based on their experiments with refrigerant R-12, Akers and Rosson [15] developed an equation for the wavy flow region. This correlation, recommended by the 1985 ASHRAE Fundamentals Handbook, is shown below.

$$Nu = 13.8 Pr_l^{1/3} \left[Re_g \left(\frac{\mu_g}{\mu_l} \right) \left(\frac{\rho_l}{\rho_g} \right)^{0.5} \right]^{0.2} \left(\frac{c_{pl} \Delta T}{h_{fg}} \right)^{-1/6} \quad (2.3.12)$$

In Equation (2.3.12), ΔT is equal to the difference between the tube wall temperature and the fluid saturation temperature.

The greatest number of published two phase flow correlations exists for the annular flow regime, since this pattern tends to occupy a significant portion of the condenser during high velocity condensation. ASHRAE recommends the correlation by Akers, Deans and Crosser [14] for annular flow; although other researchers have concluded that this correlation predicts too low. The correlation is expressed in two parts as follows:

$$Nu = 5.03 Pr_l^{1/3} Re_{eq}^{1/3} \quad \text{for } Re_{eq} < 5 \times 10^4 \quad (2.3.13)$$

$$Nu = 0.0265 Pr_l^{1/3} Re_{eq}^{0.8} \quad \text{for } Re_{eq} > 5 \times 10^4 \quad (2.3.14)$$

where Re_{eq} is defined as the Reynolds number based on the equivalent mass flux. Mathematically, it is expressed as a function of the liquid phase Reynolds number (Re_l) and the vapor phase Reynolds number (Re_g); namely:

$$Re_{eq} = Re_l + \left(\frac{\mu_g}{\mu_l} \right) \left(\frac{\rho_l}{\rho_g} \right)^{0.5} Re_g \quad (2.3.15)$$

A review of the work by Akers and Rosson [15] and Akers, et. al [14] as well as other available predictive and experimental studies related to heat transfer during forced convection condensation in tubes and annuli is presented by M.M. Shah [96]. In his report, Shah also discusses the individual effects of vapor superheat, noncondensable gases, oil and return bends on the refrigerant-side heat transfer coefficient.

An empirical correlation for in-tube forced convective condensation that is the most verified predictive technique and has the widest data base in terms of fluid properties, flow rates, tube diameter and orientation is the one by M.M. Shah [95] as follows:

$$\Psi = \frac{h_{TP}}{h_l} = 1 + \frac{3.8}{Z^{0.95}} \quad (2.3.16)$$

The variables in Equation (2.3.16) are defined as follows:

$$Z = \left(\frac{1}{x} - 1 \right)^{0.8} P_r^{0.4} \quad (2.3.17)$$

$$h_l = h_l (1 - x)^{0.8} \quad (2.3.18)$$

$$h_l = 0.023 \left(\frac{GD}{\mu} \right)^{0.8} Pr^{0.4} \left(\frac{k_l}{D} \right) \quad (2.3.19)$$

In Shah's formulation, P_r is equal to the reduced pressure and h_l is equal to the single phase heat transfer coefficient calculated from the Dittus-Boelter correlation, i.e., Equation (2.3.2), assuming that all the mass to be flowing as liquid. Shah recommends the above correlation for general use provided that all the following conditions are satisfied.

$$V_g \geq 5.905 \text{ ft/min (3 m/sec)}$$

$$Re_l \geq 350$$

$$Re_g \geq 35,000$$

where V_g is equivalent to the fluid velocity assuming all the mass to be flowing as vapor.

Another correlation, well-verified for the halocarbon refrigerants R-11, R-12, R-21, R-22, R-113 and R-114, was furnished by Cavallini and Zecchin [27] and identified by Equation (2.3.20).

$$h_{TP} = 0.05 \operatorname{Re}_{eq}^{0.8} \operatorname{Pr}_l^{0.33} \left(\frac{k_l}{D} \right) \quad (2.3.20)$$

where Re_{eq} is given by Equation (2.3.15). During experimental testing, the ratio (ρ_l/ρ_g) varied from 11 to 314, and Re_l varied from 7000 to 53,000. The standard deviation of the data sets ranged from 8 to 47 percent.

2.4. Condenser Performance

The results of previous experimental and theoretical studies related to the determination of air-side heat transfer and flow friction characteristics of finned tube heat exchanger surfaces are summarized in this Section. Available heat transfer correlations are presented, as well as the performance characteristics of different fin types and surface geometries, including the influences of fin spacing, number of tube rows, and tube arrangement. A more in-depth review of the performance of several different heat transfer surfaces is the primary subject of Shah [97] and Webb [126].

2.4.1. Presentation of Heat Transfer and Friction Data

Standardly in industry, the terminology "surface performance data" implies a dimensional plot of heat transfer rate and pressure drop as a function of the fluid flow rate through the heat exchanger. However, the dimensionless heat transfer and fluid flow friction (pressure drop) characteristics of a heat transfer surface are usually referred to simply as the "surface basic characteristics", or "surface basic data".

In most cases, the dimensionless experimental heat transfer characteristics of a finned tube surface are presented in terms of the Colburn modulus, j , versus Reynold's number, Re . The correlating parameter, j , was introduced by Colburn in 1933 [29] based on available data at the time for fully developed turbulent flow. The Colburn modulus or j -factor is defined as follows:

$$j = St Pr^{2/3} \quad (2.4.1)$$

where St is the Stanton number, a dimensionless parameter based on the mass flux through the minimum flow cross sectional area of the heat exchanger. The Stanton number may be expressed in terms of either dimensional or nondimensional quantities as follows:

$$St = \frac{h}{Gc_p} = \frac{Nu}{Pr Re} \quad (2.4.2)$$

Based on Equations (2.4.1) and (2.4.2), the Colburn j-factor may be redefined as

$$j = \frac{Nu}{Re Pr^{1/3}} \quad (2.4.3)$$

In contrast, the theoretical characteristics of a heat transfer surface are presented in terms of the Nusselt number, Nu , versus Re . This relationship is generally expressed as Equation (2.3.1) which is repeated here:

$$Nu = C_o Re^a Pr^b \quad (2.4.4)$$

where the terms C_o , a , and b are empirical constants.

The dimensionless pressure drop or fluid friction characteristics of a heat transfer surface are generally presented in terms of the Fanning friction factor f versus Re or f' versus Re , where f' represents the friction factor on an individual tube row basis.

In general, for any finned tube surface, the air-side heat transfer coefficient and fluid friction factor are each a function of several different parameters, including surface geometry and fluid properties. Specifically, as presented theoretically by Mirkovic [73], the Nusselt number and friction factor dependence may be expressed in terms of a number of different dimensionless parameters as follows.

$$Nu = Nu(Re, Pr, l/d_o, \delta/d_o, n_f/d, S_l/d, S_t/d, N_t, T_w/T_b, \mu_w/\mu_b) \quad (2.4.5)$$

$$f = f(Re, l/d_o, \delta/d_o, n_f/d_o, S_l/d_o, S_t/d, N_t, T_w/T_b, \mu_w/\mu_b) \quad (2.4.6)$$

where: Nu = Nusselt number
 Re = Reynolds number, based on hydraulic diameter
 Pr = Prandtl number
 d_o = outside diameter of bare tubes
 l = fin height
 δ = fin thickness
 n_f = fin pitch or number of fins per unit length

S_t = transverse pitch of tube bundle
 S_l = longitudinal pitch of tube bundle
 N_t = total number of finned tube rows
 T_w = tube wall temperature
 T_b = bulk mean fluid temperature
 μ_w = absolute fluid viscosity evaluated at tube wall temperature
 μ_b = absolute fluid viscosity evaluated at bulk mean temperature

The ratio of absolute viscosity $(\mu_w/\mu_b)^n$ and of temperature $(T_w/T_b)^n$ are incorporated as correction factors to account for the variation of fluid properties with temperature. Since the exponent is typically low, the value is generally close to unity and omitted from many correlations with no explanation.

The above factors make successful analytical prediction of the heat transfer and pressure drop characteristics of finned tube surfaces a complex art. At the present time, few numerical predictions appear in the literature. Thus, the majority of basic data for compact heat exchanger surfaces is obtained empirically, and the dimensionless heat transfer and pressure drop characteristics of these surfaces are presented in terms of j versus Re , and f versus Re , where the Reynolds number used is most often based on the hydraulic diameter, D_h . In some instances, either the Reynolds number based on tube spacing (Re_L), tube outside diameter (Re_d), or the Graetz number Gz has been employed as the independent flow variable. However, whether Re_D , Re_L , Re_d , or Gz is utilized is arbitrary. Since j , f , and Re are dimensionless parameters, then the test data gathered by the stated approach is applicable to other surfaces of any hydraulic diameter provided that complete geometric similarity is maintained. For example, if the fin density or tube pitch is modified on a finned tube surface for which j and f versus Re data already exists, then the original j and f versus Re information is no longer applicable.

In detail, R.K. Shah [97] emphasizes the limitations of the j versus Re plots based on presently available experimental data; particularly, that the j factor is clearly dependent on the Prandtl number for the fully developed turbulent flow and fully developed laminar flow regions, but independent of the Prandtl number for developing laminar flows.

In a compact heat exchanger, the air-side overall pressure drop is equivalent to a combination of the entrance and exit, flow acceleration, and core friction losses. Generally, the core frictional pressure drop is the dominating factor accounting for approximately 90 percent or more of the total pressure drop, particularly for highly interrupted fin surfaces since the flow is more fully mixed. Entrance and exit losses tend to gain in importance heat exchanger cores with short flow lengths and at high values of Reynolds number. Shah [97] thoroughly describes the specific pressure drop relationships for plate-fin, tube-fin and individually finned heat exchanger geometries.

2.4.2. Air-Side Correlations

Significant progress has been achieved by industry in the formulation of finned tube heat transfer surfaces and the development of empirical correlations for the air-side heat transfer coefficient and the fluid friction factor. Unfortunately, only a portion of this information has been reported in open literature, particularly that related to interesting new and innovative surface geometries that may increase the air-side heat transfer coefficients considerably compared to conventional finned tube surfaces. This is primarily because most information related to these types of surface geometries is highly proprietary. In the subsections that follow, an attempt has been made to give an accurate overview of many of the different heat transfer coefficient and Fanning friction factor correlations available currently. In addition, the results of experimental and theoretical studies related to the evaluation of local and average heat transfer and friction behavior is presented. The information furnished is classified according to the type of surface geometry.

2.4.2.1. Continuous Plain Fins on an Array of Round Tubes

It appears that more published data and correlations exist for this type of finned tube surface geometry than any other geometry. This simple design is one of the most commonly used for air conditioning and refrigeration applications, such as exchangers or coils in which high pressure must be maintained on the refrigerant-side. Although its performance is limited compared to many other enhanced surfaces, the plain finned tube design has been "an automotive industry standard" due to its competitive cost, high reliability, and ease of

manufacturing [32]. In addition, for the same total heat transfer surface area, continuous plain finned tubes exhibit the lowest pressure drop compared to other finned tube surfaces. Continuous plain fins are also commonly referred to as flat fins, plate fins or straight fins and are characterized by their longer, uninterrupted flow passages compared to other fin types (e.g., augmented geometries, such as louver fins, wavy fins, etc.).

For single row heat exchangers with plain finned tubes, some of the first and most extensive experimental analysis is attributable to Sheperd [102] and Gebhart [45]. In Sheperd's results, it is concluded that the selection of optimum tube pitch is highly dependent on the fin efficiency. In an extension of Sheperd's work, Gebhart indicated that the use of perforations, slots, holes or tabs as augmentation schemes on plain fins may waste more heat transfer surface than can be effectively replaced by a higher heat transfer coefficient.

Saboya and Sparrow [90,91,92] employed a mass transfer technique, called naphthalene sublimation, to establish local and average mass transfer coefficients of one, two and three row coils each having plain finned tubes and a staggered tube layout. The results of Saboya and Sparrow, expressed in terms of the dimensionless Sherwood number, may be interpreted in terms of heat transfer (i.e., Nusselt number) through the heat and mass transfer analogy. In general, the advantages of using a mass transfer system to simulate a heat transfer system include the potential for improved accuracy of measurement and more precise control of boundary conditions.

The importance of Saboya and Sparrow's work concerns the identification of different transfer mechanisms that operate in different portions of the fin and how each contributes to the high transfer rates. In particular, for the first row, the boundary layer on the forward part or leading edge of the fin is the dominant contributor to mass or heat transfer with the vortex-induced transfer mechanism on the fin portion in front and at the sides of the tube gaining in importance at higher Reynolds numbers (approximately 1000 to 1200). On the other hand, for the downstream rows, lower transfer coefficients exist and there is no region of boundary layer development, i.e., only the vortex system is responsible for the heat transfer. As the Reynold's number increases, the stronger vortices act as a natural augmentation mechanism and the contribution of the downstream rows to

the heat transfer increases. At a Reynolds number of approximately 1100, the contributions of each tube row are roughly equivalent. In a later work, Seshimo and Fujii [94] actually rearranged the local mass transfer coefficients obtained by Saboya and Sparrow and graphically represented the data in the form of an equivalent heat transfer distribution.

McQuiston [70] was able to formulate an empirical heat transfer correlation based on his studies of four row, plate-fin and tube coils with a staggered tube arrangement. His correlation is based not only on his own experimental data [71,72], but also on that of Rich [87] and Kays and London [59]. The correlation is expressed in terms of the Colburn parameter, j , and the Reynolds number, Re_d , based on the tube outside diameter, as follows:

$$j = 0.0014 + 0.2618 Re_d^{-0.4} \left(\frac{A}{A_t} \right)^{-0.15} \quad (2.4.7)$$

where (A/A_t) is the ratio of the total air-side surface area, including the fins, to the bare tube surface area. For staggered banks of plain fins, this ratio may be equally expressed in terms of individual geometric variables.

$$\frac{A}{A_t} = \frac{4}{\pi} \frac{S_l}{D_h} \frac{S_t}{d_o} \sigma \quad (2.4.8)$$

For ninety percent of the experimental data, McQuiston's j -factor correlation agrees within ± 10 percent.

McQuiston [70] also was able to develop a relationship for the Fanning friction factor associated with plate fins:

$$f = 0.004904 + 1.382 Re_d^{-0.5} \left(\frac{r_o}{r^*} \right)^{0.5} \left[\frac{(S_t - d_o)n_f}{4(1 - \delta n_f)} \right]^{-0.8} \left[\frac{S_t}{2r^*} - 1 \right]^1 \quad (2.4.9)$$

where the pseudo hydraulic radius r^* is defined by the function:

$$\frac{r^*}{r_o} = \frac{A/A_t}{(S_t - d_o)n_f + 1} \quad (2.4.10)$$

However, the friction factor correlation, Equation (2.4.9) was only able to accurately represent the data of the same 17 coil geometries within ± 35 percent.

Because the McQuiston plate-fin and tube correlations are empirically-based, it is important to define the range of parameters employed in developing the correlations. As presented by McQuiston in dimensional form, the ranges are:

$$\begin{aligned}
 (9.53 \text{ mm}) \ 3/8 \text{ in.} &\leq D \leq 5/8 \text{ in.} \ (15.9 \text{ mm}) \\
 (25.4 \text{ mm}) \ 1 \text{ in.} &\leq S_t \leq 2 \text{ in.} \ (50.8 \text{ mm}) \\
 (25.4 \text{ mm}) \ 1 \text{ in.} &\leq S_l \leq 2 \text{ in.} \ (50.8 \text{ mm}) \\
 (0.157 \text{ fins/mm}) \ 4 \text{ fins/inch} &\leq n_f \leq 14 \text{ fins/inch} \ (0.551 \text{ fins/mm}) \\
 (0.152 \text{ mm}) \ 0.006 \text{ in.} &\leq \delta \leq 0.010 \text{ in.} \ (0.254 \text{ mm}) \\
 (1 \text{ m/s}) \ 200 \text{ ft/min} &\leq V_b \leq 800 \text{ ft/min} \ (4 \text{ m/s})
 \end{aligned}$$

A recently published correlation by Gray and Webb [49] provides nominally the same accuracy as McQuiston for the heat transfer coefficient. The correlation was developed based on their experiments with 16 different flat finned, circular tube heat exchangers each having a staggered tube layout. In addition, the correlation shown below is valid for heat exchangers having four or more rows and assumes that the heat transfer coefficient does not change after four rows. The rms error of the resulting correlation is 7.3 percent compared to the rms error of 9.5 percent associated with the McQuiston correlation, Equation (2.4.7).

$$j = 0.14 \text{ Re}_d^{-0.328} \left(\frac{S_t}{S_l} \right)^{-0.502} \left(\frac{s}{d_o} \right)^{0.0312} \quad (2.4.11)$$

In the same study, Gray and Webb [49] also developed a Fanning friction factor correlation that is considerably more accurate than the McQuiston correlation. Their correlation is based on the superposition model initially proposed by Rich [86] in which the total pressure drop is comprised of two components; namely, the pressure drop due to the drag force on the bare tubes ΔP_t and the pressure drop due to friction on the fins ΔP_f . The resulting friction factor correlation, which correlates 95 percent of the data within ± 13 percent, is expressed by Equation (2.4.12), shown at the top of the next page.

$$f = f_f \left(\frac{A_f}{A} \right) + f_t \left(1 - \frac{A_f}{A} \right) \left(1 - \frac{\delta}{n_f} \right) \quad (2.4.12)$$

where f_f and f_t represent the individual friction factors due to the bare tubes and fins, respectively. These two friction factors, correlated separately, are defined by Equations (2.4.13) and (2.4.14).

$$f_f = 0.508 \operatorname{Re}_d^{-0.521} \left(\frac{S_t}{d_o} \right)^{1.318} \quad (2.4.13)$$

$$f_{tz} NX = f_t \left(\frac{A_t}{A_{ct}} \right) = f_t \left(\frac{\pi d_o}{S_t - d_o} \right) \quad (2.4.14)$$

where f_{tz} is equal to the well-documented Zukauskas [136] friction factor correlation derived for bare tube banks and X is an empirical function of the Reynolds number and the tube spacing ratio S_t/d_o .

The range of validity for the Gray and Webb correlations, identified in terms of dimensionless parameters is as follows:

$$\begin{aligned} 400 &\leq \operatorname{Re}_d \leq 24700 \\ 1.97 &\leq S_t/d_o \leq 2.55 \\ 1.70 &\leq S_l/d_o \leq 2.58 \\ 0.08 &\leq s/d_o \leq 0.64 \\ 4 &\leq N_R \leq 8 \end{aligned}$$

Most recently, Webb [129] developed heat transfer correlations for plain finned and wavy finned tube surfaces with staggered layouts, expressed in terms of the Nusselt number and Graetz number and based on the extensive air-side heat transfer data of Beecher and Fagan [17] based on their experiments with 7 flat fin and 20 wavy finned tube surfaces. Webb's flat fin correlations provide better accuracy than McQuiston, predicting ninety-eight percent of the experimental data within ± 5 percent.

The flat fin correlations, as developed by Webb, are as follows:

For $Gz \leq 25$

$$Nu_a = 0.40 Gz^{0.73} \left(\frac{s}{D_c} \right)^{-0.23} N_R^{0.23} \quad (2.4.15)$$

For $Gz > 25$

$$Nu_a = 0.53 Gz^{0.62} \left(\frac{s}{D_c} \right)^{-0.23} N_R^{0.32} \quad (2.4.16)$$

The limitations of Webb's flat fin correlations are as follows:

$$\begin{aligned} D_c &= 0.5 \text{ in.} \quad (12.7 \text{ mm}) \\ \delta &= 0.005 \text{ in.} \quad (0.127 \text{ mm}) \end{aligned}$$

$$\begin{aligned} 5 &\leq Gz \leq 180 \\ 2 &\leq N_R \leq 6 \\ 0.112 &\leq s/D_c \leq 0.154 \\ 1.15 &\leq S_j/S_1 \leq 1.67 \end{aligned}$$

In Equation (2.4.15) and (2.4.16), the Graetz number Gz is defined as follows:

$$Gz = \frac{Re \ Pr \ D_h}{L} \quad (2.4.17)$$

Because Beecher and Fagan arbitrarily selected the Graetz number as the independent flow variable for their heat transfer data, Webb also used the Graetz number in his Nusselt number correlation. Equations (2.4.15) and (2.4.16) may be similarly represented in terms of the j -factor by substitution into Equation (2.4.2).

In graphical form, Kays and London [59] provide linearized j versus Re and f versus Re data obtained from the Trane Company for two multi-row, continuous plate-fin surfaces having staggered tube arrangements and nominal tube diameters of 3/8-inch (9.53 mm) and 5/8-inch (15.9 mm), respectively. The fin spacing and hydraulic diameter are nearly identical for each surface, approximately 8 fins/inch (315/meter) and 0.0115 ft (0.0035 meters), respectively. However, the 3/8-inch (9.53 mm) nominal tube diameter geometry possesses a surface area density roughly six percent greater than that of the other geometry.

For the same range of air-side Reynolds number from 400 to 10000, the smaller tube diameter, higher area density geometry yields j-factors that are about 15 percent larger and friction factors that are nearly 35 percent larger than the larger tube diameter, smaller area density geometry. For the 3/8-inch (15.9 mm) nominal tube diameter surface, j and f may be represented by the following two correlating equations:

$$j = 0.188 \text{ Re}_D^{-0.415} \quad (2.4.18)$$

$$f = 0.118 \text{ Re}_D^{-0.200} \quad (2.4.19)$$

For the 5/8-inch (15.9 mm) nominal tube diameter surface, j and f may be represented by:

$$j = 0.102 \text{ Re}_D^{-0.357} \quad (2.4.20)$$

$$f = 0.090 \text{ Re}_D^{-0.217} \quad (2.4.21)$$

2.4.2.2. Continuous Wavy Fins on an Array of Round Tubes

This continuous fin geometry, sometimes referred to as the corrugated or herringbone fin geometry, is one of the most popularly used designs for air conditioning condensers, because of its associated ruggedness and higher heat transfer coefficients compared to plain fins. It is also commonly used in mobile air conditioning condensers.

In conjunction with the naphthalene sublimation technique, Goldstein and Sparrow [47] simulated a one-row wavy fin design and employed the heat and mass transfer analogy to determine local and average air-side heat transfer coefficients. Their local heat transfer coefficient results revealed the presence of several vortex systems that are activated and strengthened with increasing Reynolds number and concluded that the windward facets of the wavy fins are primarily responsible for the enhancement of the heat transfer coefficients; whereas, the leeward facets have lower heat transfer coefficients and are strongly affected by the flow separation. A comparison of average heat transfer coefficients of the wavy fin with those of a similar plain fin yielded that the

augmentation due to the wavy fin surface increased with the Reynolds number. At a Reynolds number of approximately 1000, the average mass transfer coefficients for the corrugated fin system were 45 percent higher than those for the plain fin design of similar geometry.

As mentioned in Section 2.4.1.1., Webb [129] also developed a general air-side heat transfer correlation for the wavy fin data gathered by Beecher and Fagan. The mathematical expressions, which correlate 80% of the data within $\pm 5\%$ and 96% of the data within $\pm 10\%$, are as follows:

For $Gz \leq 25$

$$Nu = 0.50 Gz^{0.86} \left(\frac{S_t}{D_c}\right)^{0.11} \left(\frac{s}{D_c}\right)^{-0.09} \left(\frac{S_d}{S_l}\right)^{0.12} \left(\frac{2S_p}{S_l}\right)^{-0.34} \quad (2.4.22)$$

For $Gz > 25$

$$Nu = 0.83 Gz^{0.76} \left(\frac{S_t}{D_c}\right)^{0.13} \left(\frac{s}{D_c}\right)^{-0.16} \left(\frac{S_d}{S_l}\right)^{0.25} \left(\frac{2S_p}{S_l}\right)^{-0.43} \quad (2.4.23)$$

where S_d and S_p are defined as the peak-to-valley wave depth and one-half the wave length, respectively.

The limitations of Webb's wavy fin correlations are as follows:

$$\begin{aligned} \delta &= 0.005 \text{ in. (0.127 mm)} \\ (9.53 \text{ mm}) \quad 0.375 \text{ in.} &\leq D_c \leq 0.5 \text{ in. (12.7 mm)} \\ 5 &\leq Gz \leq 180 \\ 2 &\leq N_R \leq 6 \\ 0.112 &\leq s/D_c \leq 0.154 \\ 1.15 &\leq S_t/S_l \leq 1.67 \end{aligned}$$

For Webb's wavy fin correlations, the Graetz number is also defined by Equation (2.4.17).

2.4.2.3. Louver Fins on a Circular Tube Array

Similar to wavy fins, the louver fin design also represents an augmentation scheme employed in finned tube heat exchangers, particularly in automotive condensers, in order to obtain higher air-side heat transfer coefficients and improved performance. In this design, arrays of small metal strips are raised from base plate fins. Further, these metal strips are bent into the flow stream at frequent intervals. This angle of incidence with the flow stream is referred to as the "louver pitch". Although the bent fins increase the form drag, the induced flow separation and secondary flows associated with louvered fin surfaces are primarily responsible for the improvement in heat transfer performance. Unfortunately, most surface basic data and correlations related to louver fin geometries is proprietary and only a limited amount of information is published in the literature.

Hosoda, Uzuhashi and Kobayashi [54] demonstrated the advantages of louvered fin surfaces in air conditioning evaporator and condenser cores with circular tubes and staggered tube arrangements. Their results were presented in dimensional form. They indicate a 60 percent improvement in the air-side heat transfer coefficient, a 30 percent increase in the overall heat transfer coefficient and a slightly higher pressure drop attributable to the louver fins compared to the wavy fin. In addition, Hosoda et. al. [54] employed flow visualization techniques to analyze the air flow patterns over their louver fin and wavy fin coils.

Nakayama and Xu [77] established heat transfer and friction information for louvered fins on the basis of heat exchanger modeling and experimental data. Heat transfer coefficients for the louvered fin were 20 to 200 percent higher than for plain fins at the same Reynolds number. Davenport [35] generated Stanton number versus Reynolds number plots and friction factor versus Reynolds number plots for eight different louvered fin surfaces and found that both Stanton number and friction factor increased with decreasing louver pitch, i.e., the size of the louver or its uninterrupted flow length in the air flow direction.

Suga and Aoki [113] conducted experimental and numerical studies on multi-louvered fins typically used in automotive condensers and concluded that the control of the thermal wakes formed downstream of the louvers is a key factor in determining the optimum geometry. In particular, Suga and Aoki noted the existence of an optimum ratio of fin pitch to louver pitch for each louver angle that best maximizes heat transfer and minimizes air pressure drop.

2.4.2.4. Continuous Fins on an Array of Flat Tubes

The interest in flat or oval tubes (i.e., rectangular tubes with sharp or rounded corners) is high due to the associated larger air-side heat transfer coefficients and reduced pressure drop compared to circular tubes. The performance advantages achieved by using flat or oval tubes rather than circular tubes is primarily due to the lower drag force and the smaller wake region on the fin behind the tube. In addition, the heat transfer coefficients for flow inside flat tubes is higher than those for flow inside circular tubes, especially at lower Reynolds numbers.

Formerly, the use of flat tubes was mainly limited to low-pressure applications, such as vehicular radiators. However, according to Webb [126], higher design pressures are possible by using flattened aluminum tubes made by extrusion processes. These tubes can be made with internal full height ribs, which strengthen the tube and allow a higher tube-side design pressure. Subsequently, the use of finned oval or flat tubes is now being adapted more often to mobile air conditioning systems.

For individually finned tube surfaces, an analysis by Brauer [21] was one of the first to indicate additional significant performance advantages offered by the use of flat or oval tubes. Based on his results, oval tubes may provide up to 15 percent more heat transfer and up to 25 percent less pressure drop than circular tubes. Similarly, for continuously plain finned tube surfaces, results by Rich [86] indicate that the form drag on the tubes accounts for 40 to 60 percent of the total pressure loss, depending on fin density. Rich's results are described in more detail in Section 2.4.5.

On a logarithmic scale, Kays and London [59] provide j-factor and friction factor data over a range of air-side Reynolds number from approximately 400 to 10,000 for five different heat exchanger geometries employing continuously finned, flat tubes and staggered and in-line tube arrangements. In general, for the same flat tube arrangement, the wavy fin geometry provides heat transfer coefficients that are 30 percent higher than those for the plain fin geometry.

Goodremote, Guntly and Costello [48] of Modine Manufacturing Company published the results of one full condenser study related to the patented parallel flow type or PF™ condenser, employed in the automotive market since 1987. The brazed aluminum condenser utilizes oval refrigerant tubes with louvered air fins, a serpentine configuration, and multi-flow refrigerant passages. Their experimental results indicate that the oval finned tube condenser not only possessed higher heat transfer and lower air and refrigerant flow resistances, but also exhibits lower refrigerant charge, reduced physical size, and lighter weight compared to conventional plate-fin and tube surfaces with circular tubes and a serpentine arrangement. In a later study, Struss, Henkes, El-Bourini and Eigenberger [112] performed additional tests on the PF™ automotive condenser. The condenser was installed in several vehicles and both wind tunnel and road performance tests were conducted and the results compared with a typical tube and fin, serpentine condenser. The experiments confirmed that the PF™ condenser equaled the thermal performance of the serpentine condenser while providing a 25 percent reduction in refrigerant charge. Another publication by Sugihara and Lukas [114] of Nikkei Heat Exchanger Company similarly focuses on the performance, design and manufacturing advantages of the PF™ condenser.

2.4.2.5. Circular Fins on an Array of Round Tubes

Considerably more information has been published on this particular individually finned tube geometry compared to the plain and other continuously finned tube geometries. This fact is probably due to it being the most simple and widely used geometry. Circular fins are also called helical or disk fins. Most experimental analysis and reported data have involved the staggered tube arrangement, 6 to 8 rows deep. Webb [126] provides a complete survey of the published data and correlations.

Some of the most extensive data for individually finned circular tube banks with staggered layouts was gathered by Jameson [57]. Kays and London [59] graphically present j and f versus Re data for the eight different circular finned tube surface geometries originally studied by Jameson.

For the air-side convective heat transfer coefficient in individually finned circular tube heat exchangers, the most often recommended correlation is that formulated by Briggs and Young [23] based on their experiments with 14 different equilateral triangular pitch tube banks with smooth, integral, helically finned tubes in crossflow. Although originally documented in terms of the Nusselt number, the Briggs and Young correlation may also be stated as follows:

$$j = 0.134 Re_d^{-0.319} \left(\frac{s}{l}\right)^{0.20} \left(\frac{s}{\delta}\right)^{0.1134} \quad (2.4.24)$$

The validity limits of this correlation are defined as:

$$\begin{aligned} N_R &= 6 \\ 1.54 &\leq S/D \leq 8.23 \\ 0.011 &\leq \delta/D \leq 0.15 \\ 1.01 &\leq s/\delta \leq 6.62 \\ 1100 &\leq Re_d \leq 18000 \end{aligned}$$

In Equation (2.4.24), the Reynolds number is based on the tube outside diameter and l is equal to the circular fin height. In addition, although the equation is strictly applicable to equilateral triangular configurations, it may be adapted to finned tube banks other than six rows by employing correlation charts from Ward and Young [124].

Briggs and Young also attempted to correlate the Fanning friction factor in the same study [23]; however, they were unsuccessful.

For isothermal pressure drops, Robinson and Briggs [88] developed one of the most recommended equations:

$$f' = \frac{\Delta P \rho g_c}{N_R G^2} = 18.93 \text{ Re}_d^{-0.316} \text{ Pr}^{1/3} \left(\frac{S_t}{d_o} \right)^{-0.927} \left(\frac{S_t}{S_d} \right)^{0.515} \quad (2.4.25)$$

where f' represents the modified fanning Friction factor per tube row, and in this case, S_d is equal to the shortest diagonal distance between a tube and an adjacent tube on the closest neighboring row. It may be defined by Equation (2.4.26) below.

$$S_d = \left[(S/2)^2 + S_l^2 \right]^{1/2} \quad (2.4.26)$$

The validity limits of the Robinson and Briggs correlation are given as:

$$\begin{aligned} N_R &= 6 \\ 1.8 &\leq S_l/d_o \leq 4.6 \\ 0.011 &\leq \delta/d_o \leq 0.025 \\ 3.75 &\leq s/\delta \leq 6.03 \\ 2000 &\leq \text{Re}_d \leq 50000 \end{aligned}$$

Briggs and Young [23], who contributed to the formulation of Equation (2.4.25), claim that the pressure drop per row is nearly constant so that it may be applied to finned tube banks of other than six rows. In addition, based on the previous two relationships for the j -factor and the pressure drop, it is evident that for the same tube diameter, the transverse tube pitch S_t has no effect on the heat transfer coefficient. However, the pressure drop is strongly affected by the transverse tube pitch and actually decreases with an increase in S_t .

2.4.2.6. Other Individually Finned Tube Surfaces

Although exhibiting more ruggedness, individually finned tube geometries are typically less compact than continuous fin geometries as they possess a smaller surface area to volume ratio. Other than the circular plain fins presented in the previous section, several other geometries fall into the classification of individually finned tubes including the segmented or serrated fins, spine fins, studded fins, slotted fins, and wire-loop fins. However, these other geometries provide enhanced heat transfer and increased heat transfer coefficients compared to the circular plain fins through the periodic development of thin boundary layers on the small diameter spines, flat strips or wires, and subsequent dissipation in the wake region between the elements. Shah [97] and Webb [126] furnishes a complete discussion and reference list pertaining to these augmented surface geometries.

2.4.2.7. Skyve/Fin™ Geometry

The Skyve/fin™ is a patented design process for producing integral fin, serpentine tube heat exchanger cores from a single and relatively long aluminum extrusion. The fins are formed from the thick-walled extrusion utilizing a modified high speed punch press without the creation of any scrap material. Several investigations are documented which discuss the Skyve/fin™ surface geometry and allude to the excellent performance potential of this type of finned tube design. It has been employed extensively in automobile air conditioning condenser applications since the mid-1970's when the first patents were issued. O'Connor and Pasternak [80] were one of the first to obtain a patent. In addition, Peerless of America Corporation has produced patented Skyve/fin™ coils for many years.

A summary of some of the research related to Skyve/fin™ surface geometries furnished by Webb [128]. Cox [31] describes the use of the Skyve/fin™ technique to fabricate finned tubes. Although the direct interest of this study is not finned tubes, the performance potential of the Skyve/fin™ surface geometry with circular tubes was demonstrated. Cox and Jallouk [33] experimentally addressed the heat transfer behavior of a Skyve/fin™ surface with rectangular shaped refrigerant passages. Their performance results for the Skyve/fin™ were compared with those of the more standard straight and offset rectangular fins

employed in parallel plate fin heat exchangers in terms of several criteria, including heat transfer rate per per unit heat exchanger surface area and per unit heat exchanger volume. The resulting degree of improvement recorded with the Skyve/fin™ design varied depending on the selection criteria; however, no attempt was made to further improve or optimize the Skyve/fin™ heat exchanger design.

Scott and Goldschmidt [93] considered the Skyve/fin™ by developing simplified analytical relationships for surface efficiency. Their approach divides the composite fin into two sections: a rectangular base and a trapezoidal tip. The fin heat transfer is then evaluated by assuming a known base temperature and neglecting the heat transfer through the tip. Additional boundary conditions were also incorporated, such as continuity in temperature and heat flow at the interface between the two fin sections, and a uniform convective heat transfer coefficient between the fin and the airstream. This calculation technique is used extensively by Burr [25] in his experimental and analytical evaluations of the Skyve/fin™ in order to optimize fin size and weight and to quantify individual convective heat transfer coefficients.

2.4.2.8. Wire and Tube Surfaces

Wire and tube heat exchanger surfaces are typically employed as condensers and evaporators in domestic refrigeration circuits. In this design, copper or steel wires (typically, 16 to 22 gage thickness) are soldered to a serpentine tubing arrangement. This configuration is similar to continuous plain fins on a circular tube array; however, in this case, the wires act as the fins. In addition, the enhancement characteristics of small diameter wires is important, particularly at low flows when the enhancement of other interrupted fins diminishes.

Witzell and Fontaine [133] were one of the first to investigate the effects of wire and tube geometry on heat exchanger performance. Four geometrical parameters that influence the design and total heat transfer were identified. These parameters are the wire diameter, total wire length, tube diameter and total tube length.

2.4.3. Staggered Versus Inline Tube Arrangements

In a multi-row coil, tube rows are normally arranged either in a staggered or an aligned (in-line) layout in the direction of the fluid flow. The actual configuration may be characterized by the tube diameter, and by the transverse and longitudinal tube pitches measured between tube centers. For a given finned tube geometry, staggered and in-line tube banks differ substantially in performance. This is basically a result of the different flow phenomena that exist in each arrangement, particularly the boundary layer separation and wake interaction effects. Although an in-line tube bank normally exhibits a lower air pressure drop, the favorable air flow patterns and downstream mixing associated with staggered tube arrangements make them superior to in-line tube arrangements. These two factors contribute to increased heat transfer coefficients and more overall heat transfer per unit friction pressure loss.

The main disadvantage of in-line tube banks is generally referred to as the "tube bypass effect", in which some parts of the fluid stream in the flow direction fail to come into direct contact with the heat transfer surface. In particular, these bypass zones are located at straight through-flow channels between the tube rows of width $(S_t - d_o)$ and offer the path of least resistance. In addition to the bypass effect in an in-line tube arrangement, a much greater fraction of the fin surface area in the second and subsequent rows downstream is in the low-velocity wake region. These two factors significantly degrade the performance of an in-line tube arrangement and are not as noticeable in staggered tube layouts. Fukui and Sakamoto [43] were able to experimentally verify the low-velocity wake and bypass streams associated with in-line tube banks through flow visualization. Brauer [21] found that the differences between the tube arrangements diminishes with reduced fin height and fin tip clearance, i.e., the distance between fins in adjacent tube rows.

Due to their performance differences, multi-row finned tube heat exchanger studies are more widely related to staggered tube orientations rather than in-line tube layouts. However, recent experiments and analysis by Cox, Patel and Study [32] of Ford Motor Company indicate achievement of lower air pressure drop and equivalent or slightly greater heat transfer in a newly designed automotive air-cooled condenser coil consisting of two finned rows of small diameter aluminum tubes arranged in-line and equipped with multi-flow refrigerant

passages. Heat transfer capacity is maintained by increasing the number of tubes per unit volume; whereas, refrigerant flow resistance is reduced by incorporating a new refrigerant distribution manifold in which multiple flow passages are served concurrently. In addition, this condenser while maintaining heat rejection performance and reducing air flow resistance, has less weight, package volume, and manufacturing complexity than conventional tube and fin designs by 25 to 49 percent. The new design also results in nearly a 15 percent reduction in the refrigerant charge. These aforementioned characteristics of the newly designed condenser are considered important, particularly since package space and front end air flow are at a premium in today's compact automobiles.

2.4.4. Effects of Fin Density

One of the most extensive empirical analysis related to the determination of the effects of fin density or fin spacing on surface basic data is attributable to Rich [86]. Rich measured and correlated the heat transfer and friction characteristics of nine multi-row, continuous plate-fin heat exchanger coils with the same geometry, except for fin spacing, which systematically varied from 0 to 21 fins/inch (0 to 830 fins/meter). Figure 2.5 shows the experimental Colburn j -factor and friction factor as a function of Reynolds number for the eight fin spacings examined. In this case, the Reynolds number is based on the hydraulic diameter, D_h . Figure 2.6 displays the resulting j -factor and friction factor correlations formulated by Rich to account for the effects of fin spacing. In addition, Figure 2.6 includes a tabulated list of the actual fin spacings tested by Rich. The Reynolds number in these correlations and in Figure 2.6 is based on the longitudinal spacing between the tube rows, given by Equation (2.4.26) below.

$$Re_L = \frac{G S_l}{\mu} = Re_D \left(\frac{S_l}{D_h} \right) \quad (2.4.26)$$

The use of a length based Reynolds number in lieu of a hydraulic diameter based Reynolds number suggests that the flow may be characterized by alternating growth and wake destruction of the boundary layer as the air flow passes each tube row.

Figure 2.6 indicates that the j factor and the heat transfer coefficient are independent of fin spacing and are merely a function of the mass velocity through the minimum free flow area, G , for fin spacings of 3 to 20 fins/inch (118 to 787 fins/meter). In addition, at the same mass velocity, the bare tube bank exhibits a heat transfer coefficient that is 40 percent larger than that for a finned tube bank.

Figure 2.6 also indicates that the friction factor is independent of the fin spacing for the range of 3 to 14 fins/inch (118 to 551 fins/meter) and an increase in fin density tends to increase the friction factor over the same range of Reynolds numbers. In addition, as indicated by the correlated data, the j/f factor ratio generally increased as the fin spacing was reduced. This is because the fractional parasitic drag associated with the tube is lessened. The friction correlation is based on the "superposition model", in which it is assumed that the overall pressure drop is the sum of the drag force on a bare tube bank and the drag force on the fins. The friction component due to the fins alone may be defined as:

$$f_F = (\Delta P - \Delta P_T) \frac{2A_c \rho}{G_c^2 A_f} \quad (2.4.27)$$

where ΔP_T is the pressure drop for a tube bundle having no fins.

According to Rich, the heat transfer and friction data shown in Figure 2.6 may be approximated by straight lines given by Equations (2.4.28) and (2.4.29) below:

$$j = 0.195 \text{ Re}_L^{-0.35} \quad (2.4.28)$$

$$f_f = 1.70 \text{ Re}_L^{-0.50} \quad (2.4.29)$$

where the j factor correlation is most accurate in the fin density range of 3 to 20 fins/inch (118 to 787 fins/meter) and the friction factor correlation is most accurate in the fin density range of 3 to 14 fins/inch (118 to 551 fins/meter).

2.4.5. Effects of Number of Tube Rows

In general, most published correlations for heat transfer and friction are for tube banks of four rows or more and do not account for row effects. However, some limited information is available. The heat transfer and friction associated with a particular row of a multi-row, finned tube heat exchanger is highly dependent on its position in the tube bank. In-line tube banks are more sensitive to row effects than staggered tube banks. For an in-line tube bank, the convective heat transfer coefficient will decrease as the number of tube rows increases due to the flow bypass effect (See Section 2.4.3). On the other hand, the convective heat transfer coefficient will increase with number of tube rows in a staggered bank due to the spanwise mixing and turbulent eddies propagated from the upstream rows to the downstream rows.

In a later study, Rich [87] performed systematic studies in order to determine the effects of the number of tube rows in staggered tube banks having continuous plain fins. Rich found that the row effect is most noticeable at low Reynolds numbers and becomes negligible at a Reynolds number greater than 15,000. In this analysis, the Reynolds number is also defined in terms of the longitudinal tube spacing (S_l) rather than the hydraulic diameter.

Based on Rich's data, Gray and Webb [49] formulated an expression for the j-factor to account for the row effect in finned tube heat exchangers having 1 to 4 tube rows. This expression is given by Equation (2.4.30).

$$\frac{j_N}{j_4} = 0.991 \left[2.24 \text{Re}_d^{-0.092} \left(\frac{N_R}{4} \right)^{-0.031} \right]^{0.607(4-N)} \quad (2.4.30)$$

where j_N is equal to the j-factor of the Nth row and j_4 is given by Equation (2.4.11). The above equation assumes that the row effect is negligible for $N_R > 4$. Equation (2.4.30) correlates Rich's data within +8/-4 percent.

McQuiston [70] also utilized Rich's data to develop an empirical correlation to account for the row effect. His correlation is given by Equation (2.4.31) and accurately correlates the experimental data within ± 7 percent for the Reynolds number range of 3000 to 15,000. It also verifies that the j -factor or heat transfer coefficient increases with the number of tube rows.

$$\frac{j_N}{j_1} = 1 - 1280 N_R Re_L^{-1.2} \quad (2.4.31)$$

Based on McQuiston's correlation for surfaces with four rows of tubes, the row effect correlation may also be expressed by Equation (2.4.32) which agrees with the data for 1 to 4 tube rows within $+16/-9$ percent.

$$\frac{j_N}{j_4} = \frac{1 - 1280 N_R Re_L^{-1.2}}{1 - 5120 Re_L^{-1.2}} \quad (2.4.32)$$

where j_N is equal to the j -factor of the N th row and j_4 is given by Equation (2.4.7).

The results of several research studies indicate that the row-by-row average heat transfer coefficient tends to approach an asymptotic value after roughly 4 to 6 tube rows in staggered arrangements. Comparatively speaking, Mirkovic [73] noticed that the asymptotic value for the heat transfer coefficient was approximately 30 percent higher for the sixth row than the heat transfer coefficient for the first row in his experiments with staggered tube layouts. However, there is no good agreement on the exact point where the asymptotic value is attained, other than that the tube row effect is a complicated concept and is influenced by the Reynolds number, longitudinal and transverse tube pitches (although Neal and Hitchcock [78] found that the row-by-row performance is more sensitive to transverse tube pitch), and basic fin geometry (circular or continuous fin). Webb [126] and Shah and Webb [101] summarize the published works available.

2.4.6. Effects of Airflow Maldistribution

One of the key idealizations built into heat exchanger performance relations and analysis is that uniform flow distribution exists throughout the flow cross section. However, according to London [67], two different types of flow maldistributions may be anticipated in compact heat exchangers. On a macroscopic and microscopic basis, respectively, these are as described below.

1. Gross nonuniformities as a result of the dynamic effects of high velocity incoming flow impinging on the face area of the core, and on the flow constrictions imposed by the header or ducting for the exit flow stream.
2. Passage-to-passage nonuniformities arising from apparent small differences in passage geometry. Differently sized and shaped flow passages as a result of manufacturing tolerances exhibit nonidentical flow resistances.

These two types of flow maldistributions act independently of one another and are highly undesirable due to their detrimental impact on overall heat transfer and pressure drop.

A comprehensive review of over 100 publications related to the identification and analysis of various types of flow maldistribution in heat exchangers and the associated detrimental effects, thermally and mechanically, is presented by Mueller and Chiou [75]. For air-cooled heat exchangers, it is surmised that the most important factor affecting the heat exchanger performance is the maldistribution's effect on the average effective temperature difference. The effect of the maldistribution on the average heat transfer coefficient represents a minor factor. Additional analytical techniques for predicting the degree of deterioration in heat exchanger performance due to air flow nonuniformities are presented by Chiou [28] and Fagan [41]. The results of these studies indicate that exchanger capacity and effectiveness reduction may be more significant due to air flow maldistribution than to nonuniform temperature distributions.

2.4.7. Effects of Variable Properties

The absolute viscosity ratio was introduced in Section 2.3 as a means to account for the effect of temperature dependent, fluid properties on the heat transfer coefficient of liquids. However, for gases, not only the viscosity, but also the thermal conductivity and density vary with temperature to a considerable degree. In order to account for temperature dependent, fluid properties in gases, the most common adjustment made in analytical and experimental heat exchanger analysis is the "property ratio method". In this method, the Nusselt number and friction factor for the variable fluid property case are related to the constant property values as follows:

$$\frac{Nu}{Nu_{cp}} = \left(\frac{T_w}{T_m} \right)^m \quad (2.4.33)$$

$$\frac{f}{f_{cp}} = \left(\frac{T_w}{T_m} \right)^n \quad (2.4.34)$$

where the "cp" subscript refers to the constant property variable. In the above equations, all the properties in the dimensionless Nusselt number and friction factor are meant to be evaluated at the bulk mean fluid temperature. The two exponents assume various magnitudes of property variations and are dependent on the flow geometry and flow situation.

Values of 'n' and 'm' are given for various flow geometries and flow conditions in Kays and London [59]. On the basis of heat transfer experiments for gas flow normal to banks of circular finned tubes, the value of each exponent is zero. This is primarily due to the fact that the predominant heat transfer mechanism in this situation is through a laminar boundary layer, and the temperature dependent property effect is negligible. It is therefore recommended by Kays and London that fluid properties should be evaluated at the mixed mean fluid temperature with no additional correction for property variations.

2.4.8. Fin Efficiency Considerations

Fin efficiency is an important consideration in the analysis of heat transfer characteristics of extended surfaces. Because of longitudinal heat conduction, a fin decreases in temperature from its base (prime surface) to its tip, as represented in Figure 2.7 for a plain fin with base temperature T_b . Since the heat transfer to or from a fin is directly proportional to the temperature difference between the fin and surrounding medium, as the temperature drops, this reduces the temperature potential between the fin and heat transfer fluid so that the convective heat transfer is also reduced. In order to account for this reduction, a fin temperature effectiveness or fin efficiency term, η_f , is introduced and is defined as the ratio of the actual heat transfer rate, q_b , through the base to the maximum possible heat transfer rate, q_{max} . The maximum possible heat transfer would be attained by a "perfect fin", which would possess infinite thermal conductivity and the same temperature from base to tip.

One normally computes the fin efficiency by utilizing the appropriate formula for the fin geometry under consideration and by incorporating the following assumptions outlined by Gardner [44] in his mathematical and thermal analysis of finned surfaces.

1. The heat transfer is steady-state.
2. The fin material is homogeneous and isotropic, i.e., thermal conductivity of the fin does not vary with temperature or direction.
3. The convective heat transfer coefficient between the fin and the surroundings is constant and uniform throughout the entire surface area.
4. The temperature of the fluid surrounding the fin is constant and uniform.
5. The fin thickness is negligibly small. Hence, the temperature gradients normal to the fin surface are negligible compared to those between the fin and the surrounding fluid.
6. The fin base temperature is uniform and constant.

7. The contact thermal resistance where the fin base mates with the prime surface is negligible.
8. There are no heat sources or heat sinks within the fin.
9. The heat flow to or from the finned surface is directly proportional to the temperature difference between the surface and the surroundings.
10. The fin tip and sides are adiabatic, i.e., the heat transfer that occurs through the edges of the fin is negligible compared to that dissipated through its lateral surfaces.

Significant contributions to the analytical calculation of the heat conduction of a fin were made by Gardner [44] who, in addition to incorporating the above constraints, presented generalized equations for the temperature distribution over the fin and gave correlation equations for determining the effectiveness of various fin types, including straight fins, annular fins, and spines. Kraus [63] presents a brief discussion of 75 sources that have systematically supported or attacked Gardner's ten assumptions; however, Gardner's formulations are generally accepted as a basis for most other studies. For example, for a plain or wavy fin of uniform rectangular cross section with fin thickness ' δ ' and fin height ' l ' and based on the foregoing assumptions, the fin efficiency as given by Gardner is

$$\eta_f = \frac{1}{m b} \tanh (mb) \quad (2.4.35)$$

where the parameters ' m ' and ' b ' are defined as

$$m = \sqrt{\frac{2 h}{k \delta}} \quad (2.4.36)$$

$$b = \frac{l}{2} \quad (2.4.37)$$

Shah [97] contains fin efficiency relations for several different continuously finned tube and individually finned tube surfaces that actually include the effects of heat transfer through the outermost edges of the fin. As an example, for plain, wavy or offset strip fins, the expression for fin efficiency is equivalent to Equation (2.4.35); however, the parameters 'm' and 'b' must be redefined as follow:

$$m = \sqrt{\frac{2h}{k\delta} \left(1 + \frac{\delta}{b} \right)} \quad (2.4.38)$$

$$b = \frac{1}{2} - \delta \quad (2.4.39)$$

Once the fin efficiency has been established, the overall surface efficiency of the composite prime ($\eta_f = 1$) and secondary (finned, $\eta_f < 1$) surface is given by the Equation (2.4.36):

$$\eta_o = 1 - (1 - \eta_f) \frac{A_f}{A} \quad (2.4.40)$$

and the corresponding thermal resistance for heat transfer from the finned surface to the surrounding medium is $1/\eta_o h A$.

Since fin efficiency decreases with increasing fin height, the minimum acceptable fin efficiency is strongly influenced by the economic analysis of the heat exchanger design, e.g., the cost of the prime and secondary surface. It is customary to design for fin efficiencies in excess of 60 percent. The high thermal conductivity of fins made of copper or aluminum allows the utilization of smaller thickness fins than would be required for low conductivity materials, such as steel. Finally, for given fin dimensions and material, the fin efficiency for a low heat transfer coefficient fluid, such as a gas, is larger than that for high heat transfer coefficient liquids. This is one reason that extended surfaces are not always employed on the liquid-side of a two fluid heat exchanger.

2.5. Testing with Alternative Refrigerants

Compared to most other performance investigations, the amount of experimental and analytical data regarding the performance of refrigeration and air conditioning systems using alternative refrigerants, particularly HFC-134a, is limited at this time, although it is common knowledge that studies are being undertaken due to the impact of the Montreal Protocol and other government regulations. The practical performance of HFC-134a and other potential substitute refrigerants, especially in terms of cooling capacity and energy efficiency, is of considerable interest to the air conditioning and refrigeration industry, so that future systems may be designed more effectively. Some of the critical performance characteristics of air conditioning or refrigeration systems that need to be examined include the capacity at varying operating conditions and the resulting performance during pull-down and idling periods. The effect of alternative refrigerants on compressor power requirements and the resulting fuel consumption must also be investigated.

This section describes some of the published works that have analyzed the performance of commercial automotive air conditioning systems or components with alternative refrigerants. Note that although the experimental testing referenced in this Section required utilizing refrigerant/oil mixtures, thorough investigation of oil effects or its compatibility with HFC-134a was outside the scope of the particular study.

Bateman [16] experimentally studied and compared the performance of CFC-12 and HFC-134a in an existing automotive air conditioning system. In-vehicle performance tests were performed within a wind tunnel and with radiative heat loads. The test vehicle was a U.S. manufactured pickup truck. From the results, he concluded that roughly 10 percent more energy was required to compress the HFC-134a gas to higher condenser inlet pressures so as to achieve the same cooling performance as that generated with CFC-12.

A more extensive performance comparison of CFC-12 and HFC-134a in automotive air conditioning systems, including steady-state and transient testing, is reported by Struss, Henkes, and Gabby [111] of Modine Manufacturing Company. On the basis of vehicular wind tunnel tests, their results indicated significantly higher head pressures generated with HFC-134a than CFC-12 (i.e., approximately 20 to 30

percent, depending on the conditions) in order to sustain the same passenger compartment temperatures. In addition, the compression ratio increased with HFC-134a by roughly 15 to 25 percent. Although there were no distinct differences in performance exhibited, two different condensers were employed in the analysis. The two condenser coils consisted of flat oval refrigerant tubes and louvered air fins. One was a parallel flow condenser with "jumped" or manifolded refrigerant passages, similar to the PF™. The other was a typical serpentine coil.

Petterson and Rekstad [81] conducted laboratory tests with a European motorcar air conditioning system, operating alternatively with refrigerant CFC-12 and HFC-134a. The automotive air conditioning system was tested outside the vehicle in an independently constructed test rig. Experimental data reveals that for the same simulated test conditions, the cooling capacity is increased 5 to 10 percent with HFC-134a while the theoretical coefficient of performance (COP) is slightly lower compared to CFC-12. Improved heat transfer coefficients and higher compressor efficiency were also observed with HFC-134a. These two factors may offset the loss in COP in practical automotive systems.

A numerical analysis of CFC-12 and HFC-134a in an existing finned flat tube and a round tube automotive condenser was recently published by Yan, Haji-Shiekh, and Lou [134]. Their results showed that a system utilizing CFC-12 has a higher coefficient of performance (COP) than one using HFC-134a and 12 to 13 percent higher condenser inlet pressures and lower refrigerant mass flow rates are required with HFC-134a in order to obtain equivalent system performance. However, they suggest more effective design and operation of the compressor and condenser, particularly the air-side heat transfer surface of the condenser, may improve the system performance with HFC-134a.

Eckels and Pate [37,38] have published two studies that have included evaluations of the refrigerant-side convective heat transfer coefficients and pressure drop of oil-free HFC-134a and CFC-12 during evaporation, condensation and single-phase flow inside smooth tubes. The first analysis is based strictly on implementing existing single phase and two phase flow correlations to predict the convective heat transfer coefficients. Results indicate that the in-tube heat transfer coefficients of HFC-134a are 27 to 45 percent higher than those of CFC-12;

however, the degree of difference is highly dependent on refrigerant temperature. The condensation heat transfer coefficient relationships employed by Eckels and Pate for this analytical comparison include the single phase flow correlations by Dittus-Boelter [36] and Petukhov-Popov [82] and the two phase flow correlations by M.M. Shah [95], Cavallini-Zecchin [27], and Traviss, Rohsenow and Baron [119].

The second study by Eckels and Pate, an empirical analysis, included results for both the average heat transfer coefficients and pressure drop. On the basis of equivalent mass fluxes, the evaporation and condensation heat transfer coefficients for HFC-134a were 40 and 25 percent larger, respectively, compared to CFC-12. Pressure drops were also greater for HFC-134a. For the same heating or cooling capacity, the in-tube heat transfer coefficients for HFC-134a during both evaporation and condensation were approximately 10 percent higher. In contrast, the pressure drop of HFC-134a was less than that of CFC-12. Koyama, Gao, and Fujii [62] also conducted experiments on the condensation of CFC-12 and HFC-134a inside a horizontal smooth tube and found the local heat transfer coefficients with HFC-134a to be 10 to 25 percent higher than those of CFC-12 based on the same mass velocity, saturation temperature, and heat flux-vapor quality relation.

Another empirical comparison by Eckels and Pate [39] includes consideration of refrigerant-lubricant mixtures. Specifically, an HFC-134a and 165 SUS PAG oil mixture and a CFC-12 and 150 SUS naphthenic oil mixture were tested with lubricant concentrations varying from 0 to 5.4 percent. Generally, results of condensation experiments showed that the in-tube heat transfer coefficients for CFC-12 decreased with the addition of lubricant; whereas, for HFC-134a, the lubricant did not have a significant effect. In addition, condensation pressure drops for HFC-134a increased by roughly 20 to 40 percent; whereas, for CFC-12 the pressure drops were not significantly influenced by the addition of oil.

In contrast to the analyses by Bateman [16], Struss, et. al. [111], Pettersen and Rekstad [81], and Yan, et. al. [134], which are component or system oriented studies, the works by Eckels and Pate [37,38,39] and Koyama, et. al. [62] were based on refrigerant flow through a single smooth tube.

2.6. Standardized Condenser Testing Procedures

Air-cooled condenser manufacturers are required to publish their equipment ratings in accordance with American Refrigeration Institute (ARI) Standard 460-80 and American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE) Standard 20-70, entitled Methods of Testing for Rating Remote Mechanical Draft Air-Cooled Refrigerant Condensers [4]. Standard test procedures for determining the performance of household refrigerator and freezer systems are specified by American National Standards Institute/Association of Home Appliance Manufacturers (ANSI/AHAM) Standard HRF-1-1986 [2]. These two documents were carefully reviewed when the test envelope for the full condenser studies was being established. In addition, solicitations were mailed to several of the Air Conditioning and Refrigeration Center (ACRC) industrial advisory members in order to gain a better knowledge of testing procedures and acceptable ranges. A more detailed review of mobile air conditioning and household refrigerator/freezer testing, including the results of the solicitations, is furnished in Chapter 6.

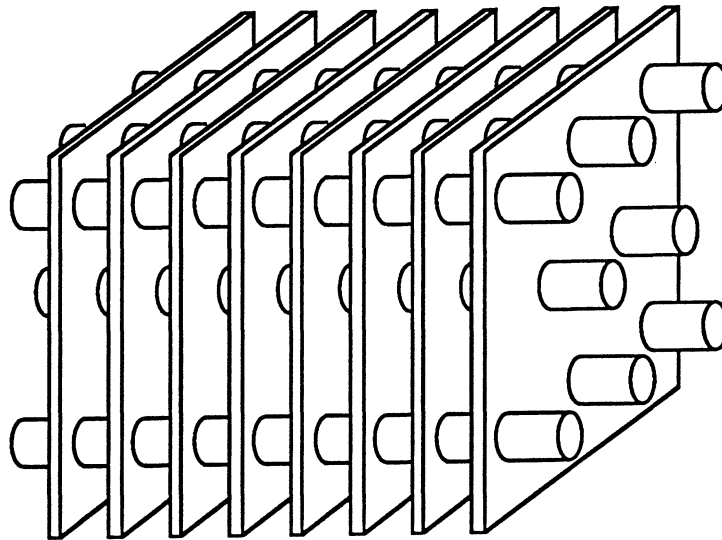


FIGURE 2.1
PLAIN OR FLAT FINS ON A ROUND TUBE ARRAY

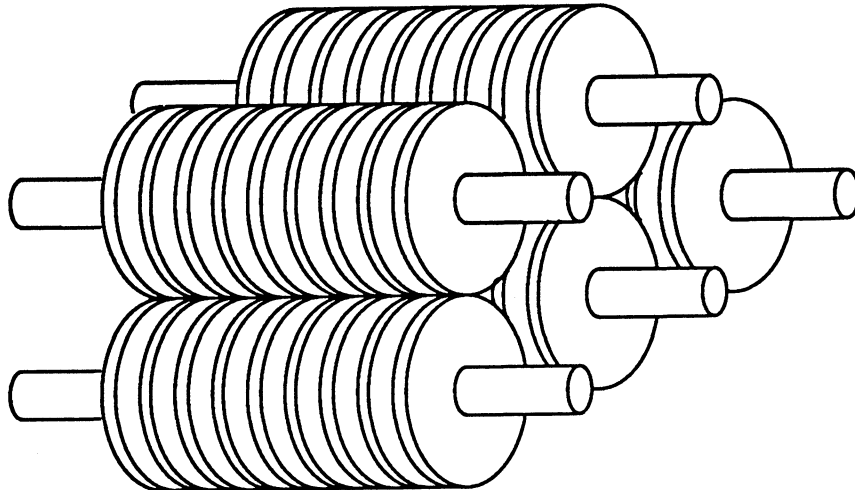


FIGURE 2.2
CIRCULAR FINNED TUBES ON A ROUND TUBE ARRAY

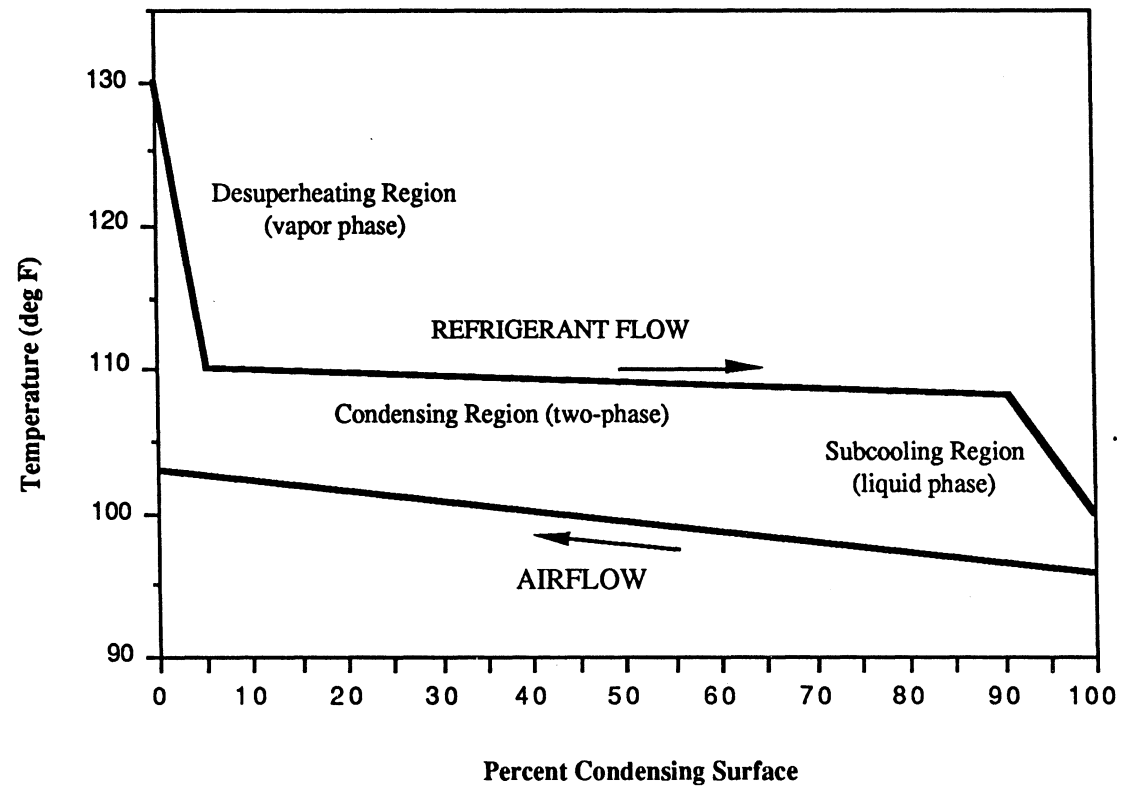


FIGURE 2.3
TYPICAL PHASE CHANGES IN AN AIR-COOLED CONDENSER

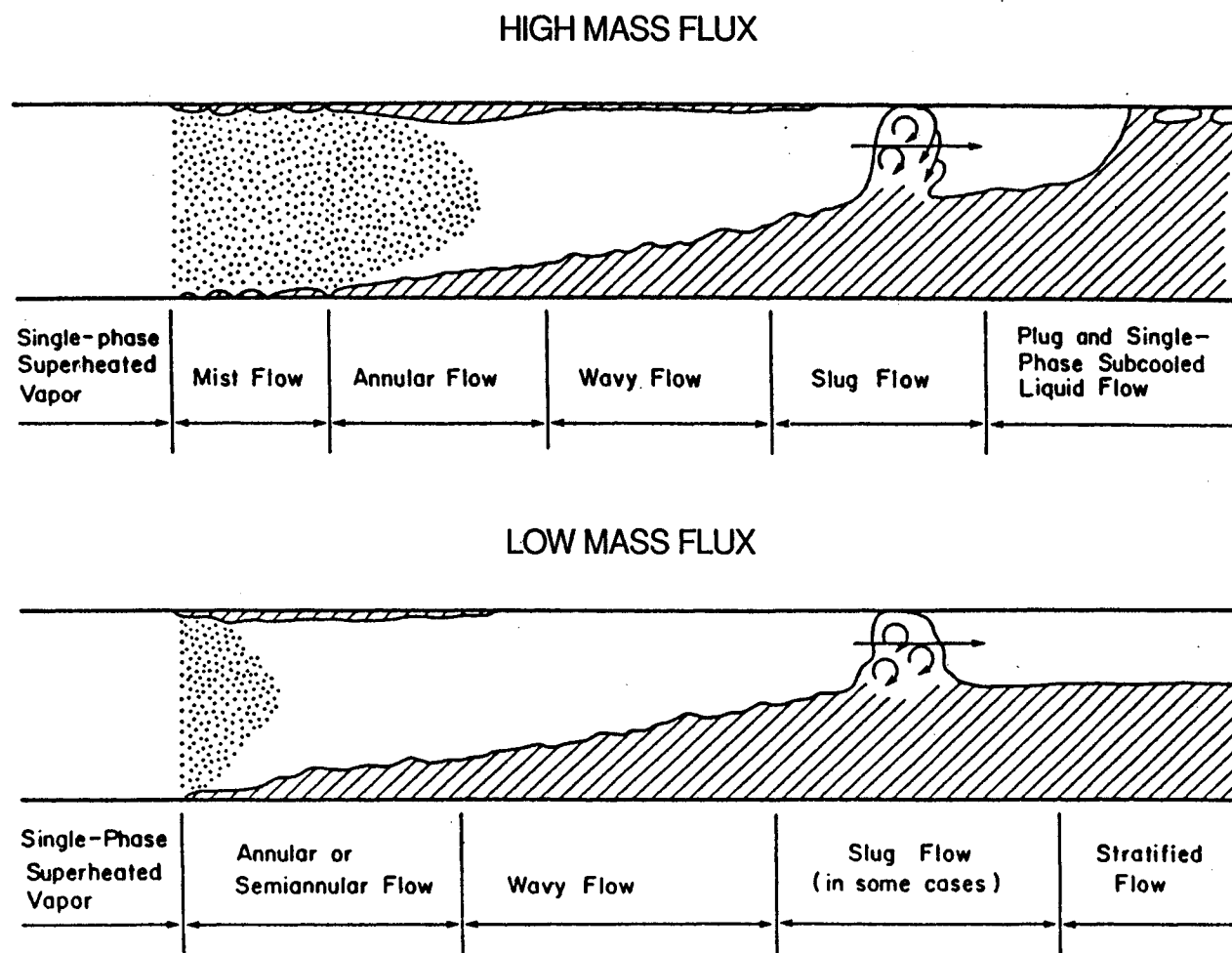


FIGURE 2.4
FLOW PATTERN DEVELOPMENT DURING HORIZONTAL IN-TUBE CONDENSATION (REF. 104)

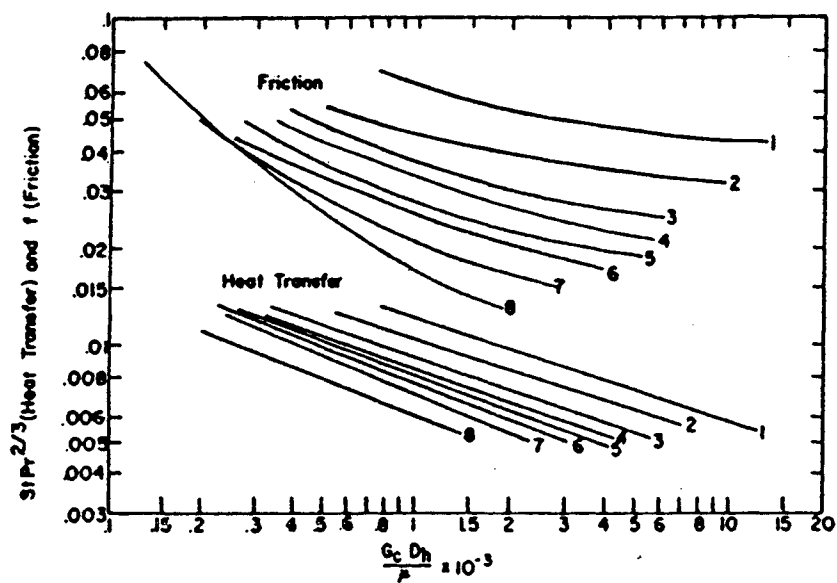


FIGURE 2.5
HEAT TRANSFER AND FRICTION DATA AS MEASURED BY RICH
FOR DIFFERENT FIN SPACINGS (REF. 86)

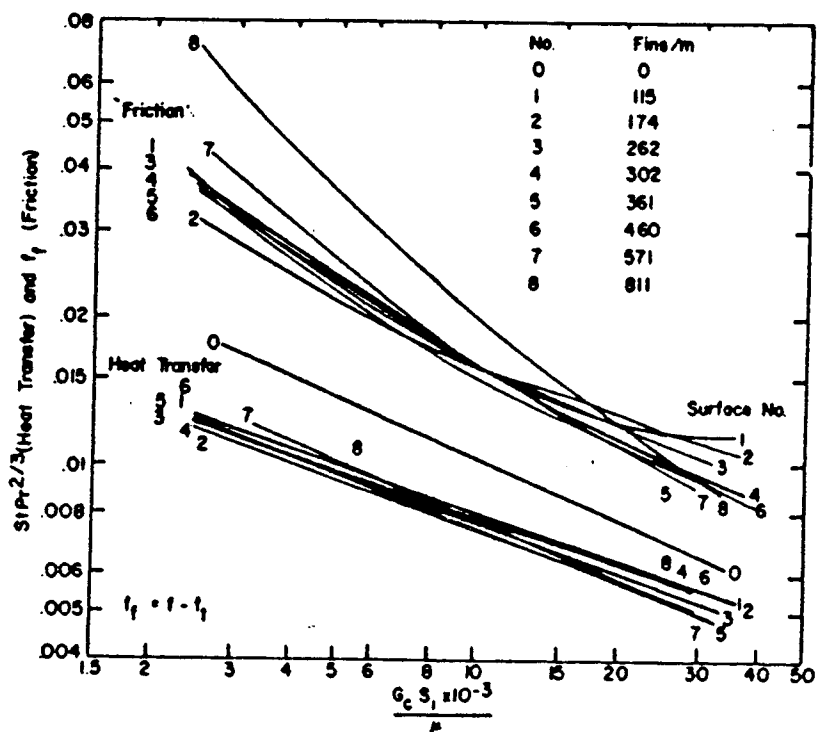


FIGURE 2.6
HEAT TRANSFER AND FRICTION DATA AS CORRELATED BY RICH
TO ACCOUNT FOR EFFECTS OF FIN SPACING (REF. 86)

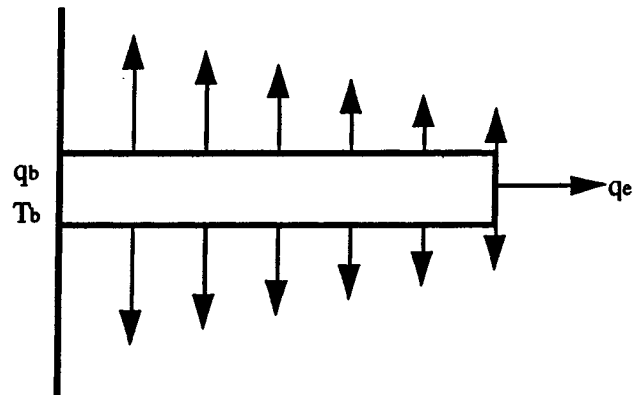
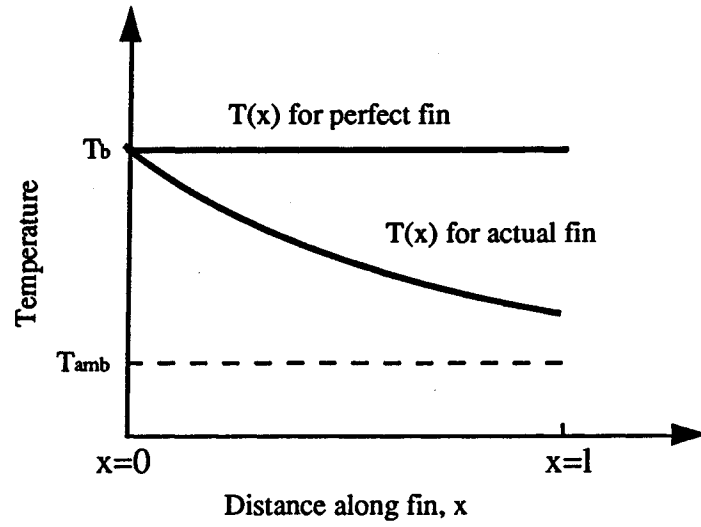


FIGURE 2.7
TEMPERATURE DISTRIBUTION FOR ACTUAL AND PERFECT FINS
WITH CONVECTION

CHAPTER 3

DESIGN AND CONSTRUCTION OF EXPERIMENTAL APPARATUS

This chapter provides a complete description of the full condenser test apparatus designed and fabricated for the Air Conditioning and Refrigeration Center (ACRC) at the University of Illinois at Urbana-Champaign. An overall schematic of the full condenser experimental apparatus, including air-side and refrigerant-side loops, is presented in Figure 3.1 at the end of this Chapter. In order to better clarify its presentation, the apparatus description provided herein is subdivided into five sections as follows: refrigerant-side components, air-side components, condenser test section, instrumentation and control methods, and the data acquisition system.

The test apparatus is designed to examine and research the heat transfer and friction characteristics of air-cooled, fin-and-tube condensers typically found in mobile air conditioning and household refrigerator/freezer applications. The requirements for the experimental facility were based on the need for flexibility reproducibility with simple operating control. Initially, in order to establish a foundation from which the test apparatus, operating procedures and instrumentation could be developed and from which the aforementioned requirements could be accomplished, ASHRAE Standard 20-70, entitled "Methods of Testing for Rating Remote Mechanical-Draft Air-Cooled Refrigerant Condensers" [4], was consulted for standardized air-side and refrigerant-side specifications. The following sections in this chapter further describe the experimental apparatus that meets the outlined objectives.

3.1. Refrigerant-Side Components

As depicted in Figure 3.1, the refrigerant is circulated counterclockwise through a copper piping loop. Liquid refrigerant is drawn from the receiver and flows through a filter-drier and sight flow indicator prior to passing through one of two circulation pumps. The volume flow rate of refrigerant is measured by a positive displacement flowmeter downstream of the pumps. A flowmeter bypass loop is installed to circulate refrigerant during startup and other transient periods. After passing through the flowmeter, the liquid refrigerant enters the

refrigerant heater where the fluid exit temperature, pressure and amount of superheat may be varied by controlling the heat addition rate. Essentially, the refrigerant heater exit represents the exit of the compressor in an actual air conditioning or refrigeration circuit. A sight flow indicator downstream of the heater permits observing the exit phase and flow pattern of the refrigerant. The refrigerant then enters the test condenser. Temperature and pressure are measured at the inlet and exit of the condenser for purposes of calculating the fluid thermodynamic and transport properties and the heat transfer rate. Finally, an additional sight flow indicator downstream of the condenser exit allows visual inspection of the phase and flow pattern of the refrigerant prior to returning the receiver.

One of the main features inherent in the design of the refrigerant-side assembly is its ability to accommodate many different refrigerants other than CFC-12. In order to accomplish this feature, the apparatus has been designed and tested to handle working pressures as high as 500 psia (3447 kPa). A more detailed description of some of the individual refrigerant-side components follows.

3.1.1. Receiver

Under normal operating conditions, the receiver acts as a storage reservoir for liquid refrigerant and is sized to contain a refrigerant liquid charge equivalent to 100 percent of the system volume. During testing with two phase refrigerant flow, the receiver may also act as an aftercondenser to further cool refrigerant exiting the condenser.

The vertically-mounted, cylindrical receiver was donated by Refrigeration Research, Inc. of Michigan and is, nominally, 6 inches in diameter and 48 inches in height. It is capable of holding a refrigerant volume of roughly 0.55 ft^3 (0.0156 m^3) and is also designed for working pressures up to 500 psig (4137 kPa). The volumetric capacity is equivalent to a pump-down capacity of approximately 44.1 lb_m (20.0 kg) based on refrigerant CFC-12 at a temperature of 90°F (32.2°C). Similarly, at the same temperature, the receiver possesses a storage capacity of approximately 40.5 lb_m (18.4 kg) based on refrigerant HFC-134a, or roughly a mass 9% less than that of CFC-12.

A single reflex, liquid level gauge is connected and mounted parallel to the receiver to indicate the amount of liquid in the receiver. In addition, the level indicator serves as a means of checking refrigerant inventory in the system. The level indicator is a Type RL gauge, armored and rated for 600 psia (4137 kPa), as manufactured by Papailias Company of New York.

3.1.2. Refrigerant Pumps

Self-priming pumps circulate the refrigerant through the loop, thereby eliminating the need for a compressor and expansion device. The use of these circulation pumps also allows for testing in a pure refrigerant environment without the presence of oil from a compressor.

Two (2) circulation pumps, mounted in parallel, are present in the refrigerant loop. Each of these pumps is the positive displacement, two-gear type with magnetic drive as manufactured by Micropump Corporation of California. The pump motors are 1/4 hp, DC powered and manufactured by Pacific Scientific, Motor & Control Division, in Illinois. Each pump is controllable via a separate, manually adjustable, variable speed drive controller with a 20-to-1 turndown ratio. These speed controllers are the Series 253D, as manufactured by Dart Controls in Indiana.

An Option 55B control board has been purchased from Dart Controls, but it still requires field installation. This control board permits either a grounded or ungrounded 0 to 5 volt DC voltage speed input command from a remote source, such as a data logger or computer. The output signal of this option board, derived from the 120 volts AC supply power required, supplies a linear signal to the speed controller.

The primary reason for installing two circulation pumps is to accommodate the broad range of refrigerant flow rates encompassed by mobile air conditioning and household refrigerator/freezer systems. The first pump, labeled "Low Range Pump" in Figure 3.1, has a port size of 1/4 inch (6.35 mm) NPT and is capable of producing volumetric flow rates from 0.0 to 0.774 gpm at 150 psi (0.0 to 0.0488 liters/sec at 1034.2 kPa). This pump will handle the lower refrigerant flow rates typical in household refrigerator/freezer systems. The second pump, labeled

"High Range Pump" in Figure 3.1, has a port size of 3/8 inch (9.53 mm) NPT and is capable of producing volumetric flow rates from 0.0 to 1.602 gpm (0.0 to 0.101 liters/second). This pump provides the higher refrigerant flow rates typical in mobile air conditioning systems.

Shutoff valves are installed immediately upstream of each circulation pump to permit only one pump to be operated at a time and to avoid short circuiting of the refrigerant. For best operating performance, Micropump technical representatives recommend that the plastic pump gears be replaced for every 2000 hours of usage. O-ring seals made of ethylene-propylene terpolymer, an elastomer abbreviated EPDM, are used in the pump body.

3.1.3. Flowmeter

The refrigerant flowmeter provides flexibility in two ways. First, it is insensitive to fluid viscosity over a wide range of temperature. Thus, the volume flow rates of several different working fluids may be measured without the need for recalibration. Second, the flowmeter maintains its accuracy over a complete range of flow rates. The flowmeter characteristics are further described herein.

The refrigerant flow measuring device is a Model 214-411 volumetric flowmeter, as manufactured by Max Machinery of California. The flowmeter is a positive displacement, radial piston type device. In the flowmeter operation, fluid pressure causes the oscillating motion of four radial pistons which are connected to and rotate a vertical crankshaft. Each revolution of the crankshaft is proportional to the volume of fluid displaced. A Max Machinery, Series 272, two-wire electronic analog transmitter is magnetically coupled to the crankshaft and produces a 4 to 20 mA output signal proportional to the flow rate.

The refrigerant flowmeter is capable of measuring volumetric flow rates from 0.0 to 2.0 gpm (0.0 to 0.126 liters/second) with an accuracy of approximately 0.5 percent of the full scale reading or roughly within 0.01 gpm (0.0006 liters/second) of the actual flow rate. Factory calibration data relating the flowmeter current output, in milliamps, to the corresponding volumetric flow rate, in gallons per minute, is shown in Table 3.1. Additional calibration data for the flowmeter, including the volumetric flow rate dependence on the K-factor

and the associated linearity error is provided in Table 3.2. The K-factor, in pulses per cubic centimeter (pulses/cc), is a correlating parameter that relates the flowmeter output signal to the volumetric flow rate.

Since the flowmeter operates on a current output basis (4 to 20 mA) and the data acquisition system is only compatible with a voltage input signal, a 250-ohm precision resistor is wired in series with the flowmeter current loop. Figure 3.2 displays a schematic diagram of the flowmeter wiring with the data acquisition system. The voltage drop produced by the current flowing through the resistor is directly indicative of the volume flow rate of refrigerant. Based on the Ohm's Law relationship shown below:

$$\text{Volts (V)} = \text{Current (I)} * \text{Resistance (R)}$$

the 4 to 20 mA output signal from the flowmeter transmitter corresponds to a 1 to 5 volts DC output signal as a result of the voltage drop across the resistor.

Figure 3.3 represents the Max Machinery flowmeter calibration data, converted proportionally from the 4 to 20 milliamps to the 1 to 5 volts DC in accordance with the Ohm's Law relationship. Equation (3.1.1) is a curve fit, fifth order polynomial derived in order to correlate the volumetric refrigerant flow rate in GPM as a function of the voltage output signal from the flowmeter current loop, in volts DC. It accurately represents the flowmeter factory calibration within approximately +0.7/-0.5 percent over the entire flow range.

$$\begin{aligned} \text{GPM} = & -0.48918 + 0.48860\text{VDC} - 0.0042043\text{VDC}^2 \\ & + 0.0070551\text{VDC}^3 - 0.0021324\text{VDC}^4 + 0.00019276\text{VDC}^5 \end{aligned} \quad (3.1.1)$$

Several static tests with no refrigerant flow were performed to determine an appropriate flowmeter output voltage offset. As a result of these experiments in which the flowmeter output voltage fluctuated about a mean value of 0.992 volts DC, a voltage offset of 0.008 volts DC or 8 millivolts was concluded. Figure 3.4 indicates the typical results obtained from over 200 minutes of data recording in which the flowmeter output was sampled once per minute. Two similar tests performed for over 900 minutes produced the same average value for the voltage offset. This voltage offset is incorporated into the data acquisition system.

Bypass plumbing is installed around the flowmeter as a safety precaution during system startup and shutdown. During these transient periods, the bypass valve should be opened and the flowmeter inlet valve closed to avoid any possible damage to the flowmeter. In particular, the flowmeter is highly sensitive to any air or vapor present in the system that may cause overspeeding and subsequent damage to the internal mechanical components and electronic circuitry. Closing the valve is also important during system shutdown because the refrigerant pressure tends to decrease more rapidly than the temperature, thereby causing the refrigerant to prematurely boil as its temperature exceeds the refrigerant saturation temperature corresponding to system pressure.

3.1.4. Refrigerant Heater

The primary responsibility of the refrigerant heater is to allow generation of a large range of condenser inlet conditions (i.e., temperature, pressure, amount of superheat, etc.) that resemble those typically found at the exit of a compressor in a true vapor compression system in mobile and household refrigerator/freezer applications. This is accomplished by the input of heat to the refrigerant. Two sketches that represent the unique refrigerant heater design and construction are Figures 3.5 and 3.6.

The refrigerant heater is comprised of approximately 85 feet of Type K copper tubing that is arranged in a 16-pass, serpentine configuration and wrapped longitudinally with enough Minco Thermofoil electric resistance heaters to provide the equivalent of 18 kW of power. The serpentine arrangement was constructed by Bohn Heat Transfer of Illinois, and is field mounted in an internally insulated, Unistrut-supported, wooden enclosure of dimensions 78" (L) x 64" (H) x 16" (W). Piping is capable of withstanding a maximum working pressure to 600 psia (4137 kPa) and is also insulated with two inches of fiberglass wrap insulation, rated for service temperatures up to 850 °F (454 °C).

The refrigerant heater is designed in two portions; namely, an evaporating section and superheating section. The evaporating section precedes the superheating section. In the evaporator, 12 kW of heating capacity exists in order to at least elevate the liquid refrigerant to the saturated vapor phase, and slightly superheat the refrigerant, if desired. In the superheater, an additional 6 kW of

capacity exists in order to allow achievement and more precise control of variable degrees of refrigerant vapor superheat.

In order to provide heating in both sections, Minco Products Model HR5443 silicone rubber insulated, thermfoil heaters are mounted on 15 of the 16 tube rows. For the 12 kW evaporator section, electric heaters each having a watt density of 32.45 W/in^2 and a nominal active resistance of 90 ohms are employed on approximately 15 feet of straight tubing or roughly four rows of the heater. For the 6 kW superheater section, electric heaters each having a watt density of 5.76 W/in^2 and a nominal active resistance of 500 ohms are employed on approximately 40 feet of straight tubing or roughly 11 rows.

The Minco thermfoil heaters are flexible and specifically designed for mounting on cylindrical surfaces. They are secured to the copper tube surfaces by using Minco Type #20 Heater Installation Tape, which is a self-adhering, glass-supported silicone rubber tape of 0.020" (0.508 mm) nominal thickness and with a service temperature range of -60 F to 392 °F (-51 C to 200 °C). A two-way stretch characteristic produces the normal holding force required for mounting the heaters.

The original design of the refrigerant heater was based on a standard cannister-type circulation heater with dual immersion heating elements, similar to the Chromalox Model CHTV and the Watlow Type CF. However, these manufacturer's standard models are designed for normal working pressures up to only 300 psia (2068 kPa), and the high refrigerant temperatures associated with condensation testing required a heater be designed and constructed to handle pressures up to 600 psia (4137 kPa). Bids were solicited and received from several manufacturers for a high pressure refrigerant circulation heater, including a variable heating control system. Based on the results of bidding, the original heater design was eliminated due to its projected cost, weight, delivery time, and safety. The tube heater previously described replaced this original design approach.

In order to achieve a wide range of refrigerant exit conditions, control of the refrigerant heater is provided in two ways. First, an on/off switch is provided for each row of electric resistance heaters on the refrigerant heater. These switches, 15 in all, are mounted on the rear of the enclosure exterior. Second, the amount

of heat delivered to the refrigerant in the active superheating and evaporating sections of the heater is modulated by individual SCR (silicon-controlled rectifier) controllers. A separate, remote-mounted operating station is equipped with two potentiometer dials to manually regulate the SCR controllers. These SCR controllers may also be regulated automatically via a 0 to 5 volts DC input signal from the data logger or computer. A more complete description of SCR controllers is provided in Section 3.4.1.

Precision watt-hour transducers are wired to measure the amount of power input to the refrigerant heater. During steady state operation of the system, this measurement may serve as an additional check of the energy balance. Indirectly, the accuracy of the measurement recorded by the watt-hour transducers is indicative of the reliability of the pressure and temperature instrumentation at the condenser inlet and exit. These watt-hour transducers are described in further detail in Section 3.4.2.

3.1.5. Expansion Tank

For single-phase experiments with saturated liquid refrigerant, a removable expansion tank must be attached to the system at the Schrader valve nearest the condenser inlet and at the highest point in the refrigerant loop. The purpose of the expansion tank is to allow for the natural expansion of refrigerant caused by the input of heat from the refrigerant heater. If the tank were not attached and the system was completely full of liquid during testing, the pressure increase as a result of heat input would lead to sudden overpressurization of the refrigerant-side and potential serious damage of equipment.

The expansion tank is actually a Robin Air, refillable refrigerant cannister with a fluid holding capacity of 30 lb_m (13.6 kg) and a maximum pressure rating of 450 psig (3103 kPa). Separate cannisters exist for dedicated testing with either refrigerant R-12 or R-134a. The cannister may be connected to the refrigerant loop via the standard high pressure refrigeration tubing normally used for charging purposes.

Prior to connection to the refrigerant loop, the cannister should be at least 50 percent full of liquid refrigerant. Single phase liquid flow is sustained in the loop during testing by maintaining the expansion tank pressure higher than the saturation pressure corresponding to the temperature at the condenser inlet. This may be accomplished by wrapping the cannister with two Minco 90-ohm electric resistance heaters, described in Section 3.1.4 for the refrigerant heater evaporating section, and manually cycling the amount of heat input. The measurement recorded by the pressure transducer at the inlet to the condenser is suitable for monitoring the tank pressure. The presence of single-phase flow may be detected by observing the sight flow indicator located between the refrigerant heater exit and the condenser test section.

3.1.6. Sight Flow Indicators

Sight flow indicators are installed at four refrigerant loop locations in order to detect positive refrigerant flow and associated phase condition at the inlet or exit of particular equipment. These devices are 500 psig (3447 kPa) rated, Type AMI Moisture/Liquid Indicators as manufactured by Alco Controls. One sight flow indicator is installed immediately upstream of both circulation pumps and another one is located upstream of the flowmeter and the bypass loop. These two indicators are important since they allow visual verification of any two phase refrigerant flow. The two remaining sight flow indicators are located upstream and downstream of the condenser test section in order to detect entering and exiting flow conditions.

The original sight flow indicators installed in the refrigerant loop were 600 psig (4137 kPa) rated, flapper type sight glasses with teflon-coated neoprene seals and of multi-piece construction, as manufactured by Papailias Company of New York. However, these Papailias sight flow indicators were prone to refrigerant leakage through the threaded end connections and between the removable glass and carbon steel body, and subsequently were replaced by the Alco indicators with unibody construction, fused glass and soldered end connections.

3.1.7. Piping, Valves and Miscellaneous Specialties

Refrigerant piping is designed and sized in order to accommodate mobile air conditioning and household refrigerator/freezer operating conditions. An ASHRAE publication related to refrigerant piping, ASHRAE RP185 [9], was used to size refrigerant lines. In addition, piping is constructed and installed in accordance with ASME Standard B31.5, entitled "Refrigeration Piping".

Primarily, the refrigerant piping consists of 3/8-inch (9.53 mm), 1/2-inch (12.7 mm), and 3/4-inch (19.1 mm) nominal diameter, Type ACR copper tubing. Type ACR is similar to Type K copper tubing. However, Type ACR piping is strictly suited for air conditioning and refrigeration systems, hence the name "ACR". Relative pipe dimensions, including actual inside and outside diameters, and wall thicknesses are presented below:

<u>Nom. Dia.</u>	<u>O.D.</u>	<u>I.D.</u>	<u>thickness</u>
3/8 in. (9.53 mm)	0.500 in. (12.7 mm)	0.402 in. (10.2 mm)	0.049 in. (1.25 mm)
1/2 in. (12.7 mm)	0.625 in. (15.9 mm)	0.527 in. (13.4 mm)	0.049 in. (1.25 mm)
3/4 in. (19.1 mm)	0.875 in. (22.2 mm)	0.745 in. (18.9 mm)	0.065 in. (1.65 mm)

The largest pipe size is used between the refrigerant heater exit and the condenser inlet. This is in order to reduce the vapor velocity of superheated refrigerant leaving the heater and subsequently, the pressure drop in the section. The remaining lines are 1/2-inch (12.7 mm) nominal O.D., except for short lengths for the upper circulation pump and at the inlet to the refrigerant heater. These short runs are 3/8-inch (9.53 mm) nominal O.D. copper.

The ability to simulate mobile air conditioning operating conditions served as the main constraint in determining piping requirements. Since industry information was unavailable at the time, computations were based on an ideal vapor compression cycle utilizing refrigerant CFC-12 and arbitrarily, a refrigerating effect of roughly 12,000 to 24,000 Btuh (3.52 to 7.03 kW), condensing temperatures from 60 to 170 °F (15.6 C to 76.7 °C), and evaporating temperatures

from -40 to 40 °F (-40 to 4.4 °C). Results of the computations showed that piping be sized to accommodate refrigerant mass flow rates from 200 to 750 lb_m/hr (0.0252 to 0.0945 kg/sec), corresponding to condenser heat rejection capacities of about 15,000 to 32,000 Btuh (4.40 to 9.38 kW). Based on these results, all copper piping including liquid lines and hot gas lines were sized in accordance with ASHRAE RP185, entitled "Refrigerant Line Sizing" [9].

Connections between copper tubing and equipment consist mainly of brazed and threaded fittings, although some compression fittings have been used for temperature instrumentation. Threaded fittings are wrapped with teflon or coated with Leak-Lok™ for tighter sealing.

Bi-directional, refrigeration ball valves, as manufactured by Mueller Brass Company, are installed throughout the refrigerant loop. These 1/4 turn valves are consistently located at or near the inlet and exit sides of major equipment to permit isolation for routine maintenance and repairs. Originally, Nupro plug valves with threaded end connections, obtained from the Peoria Valve Company, were installed in the refrigerant loop. These shutoff valves were prone to refrigerant leakage through their ends and stems, and subsequently, were replaced by the Mueller Brass shutoff valves with soldered end connections. The Mueller valves are rated for working pressures up to 500 psig (3447 kPa) and are normally stocked at most refrigeration supply warehouses locally.

Pressure relief valves with a manually adjustable setpoint range from 350 to 750 psia (2400 to 5150 kPa) are located between shutoff valves to protect from overpressurization in accordance with ASME codes. The pressure relief valves, abbreviated "PRV" on the schematic, are Model R3A valves as manufactured by Nupro Valve Company.

Two liquid line filter-driers are also installed in the refrigerant loop to keep the system clean and free of any possible foreign contaminants (e.g., solids such as metal filings, flux, dust and dirt; or solubles such as acid, water, wax and resins) that may potentially damage system components. One filter-drier, an Alco Controls Model EK165S, is installed immediately upstream of the refrigerant circulation pumps. The other filter-drier, a Sporlan Type C305S, is installed immediately upstream of the flowmeter.

Each filter-drier consists of a 10 micron, fiberglass mesh and has a safe working pressure of 500 psig (3447 kPa). According to the manufacturer's technical representatives, based on the results of factory testing to date, the filter-driers are also compatible with pure refrigerant R-134a and R-134a combined with PAG (polyoxyalkylene glycol ether) lubricants.

A total of four (4) service or Schrader valves are installed in the refrigerant loop. One is located at the highest point in the system near the inlet to the condenser to permit evacuation and reclamation of the refrigerant from the condenser test section. In addition, this port serves as the connection point for the refrigerant expansion tank during single phase experiments (see Section 3.1.5). There are two Schrader valves located near the low points of the system. Specifically, one is situated at the inlet to the refrigerant heater and the other is located at the exit of the receiver. These valves are installed to permit charging of the system, total evacuation of the system or, only a partial evacuation of refrigerant between unopened shutoff valves (e.g., for servicing equipment). The final Schrader valve location is at the inlet to the receiver.

3.1.8. Pressure Measurement

The pressure of the refrigerant is measured at five (5) locations throughout the loop. These locations are:

- Receiver exit or pump inlet pressure
- Refrigerant heater inlet pressure
- Refrigerant heater exit pressure
- Condenser inlet pressure
- Condenser exit pressure

Refrigerant pressures are each measured by a Setra Systems, Model 280E electronic pressure transducer with a sensing range of 0 to 500 psia (0 to 3447 kPa). Each transducer is diaphragm-operated and provides a 0 to 5 volts DC output signal proportional to the measured absolute pressure. The catalogued accuracy of the Setra transducers is 0.11 percent of the full scale or equivalently within 0.55 psia (3.79 kPa) of the actual measured pressure.

In an attempt to duplicate the manufacturer's calibration data, each pressure transducer was also tested on the deadweight piston tester in the Mechanical Engineering Lab Calibration Room. The deadweight piston tester may be classified as a primary standard. Primary standards, such as the deadweight tester and the mercury manometer, are considered to be the most accurate type of laboratory calibration devices and are commonly employed to calibrate secondary and line type standards, such as pressure gages with digital display indicators. In addition, primary standards are certified by laboratories with respect to traceability of calibration to the National Institute of Standards and Technology (N.I.S.T.). The general principle of the deadweight tester is that by knowing the amount of mass applied to a frictionless piston in a close-fitting cylinder of known cross sectional area, the resulting pressure is equivalent to the balancing force between the piston-weight mass and the hydraulic fluid confined within the cylinder. This calibrating pressure may readily be computed based on the following relationship:

$$P_{dw} = F_c/A_p$$

where F_c is the force of the piston-weight mass, A_p is the piston area, and P_{dw} is the calibrating pressure. When used properly, the deadweight piston tester serves as an accurate calibrating instrument for a pressure range of 0.01 to 10,000 psig (69 kPa to 68.9 MPa) in steps as small as 0.01 percent of range within a calibration accuracy of 0.01 to 0.5 percent of the reading [7].

A comparison of the results from the deadweight piston tester with the Setra factory calibration data has been prepared for each pressure transducer and is shown in Figures 3.7 thru 3.11. Calibration test data points obtained from the deadweight tester correspond to pressures of approximately 90 psia (620.5 kPa), 240 psia (1655 kPa), and 390 psia (2689 kPa) on each graph. For each pressure transducer, a total of thirteen (13) data points exist related to the manufacturer's calibration. The maximum percent error between the data exists at the lowest measured pressure of 90 psia (620.5 kPa), being in the range of approximately 1.0 to 2.2 percent for the five sensors. On the other hand, at the two higher pressures measured, the error averages less than 0.6 and 0.3 percent, respectively, for the pressure transducers. Since the results from the laboratory calibration closely approximated the Setra data, it was determined that there was no further need to

question the Setra calibration. Therefore, based on the Setra calibration test data, a fourth order curve-fit polynomial relating the absolute pressure, in psia, to the 0 to 5 volts DC output signal was developed for each of the five pressure transducers. These functions are shown below.

$$\begin{aligned}
 P_{pocr} &= -3.1891 + 99.537 \text{ VDC} + 0.41847 \text{ VDC}^2 \\
 &\quad - 0.080283 \text{ VDC}^3 + 0.0032337 \text{ VDC}^4 \\
 P_{pop} &= -3.2062 + 99.300 \text{ VDC} + 0.57790 \text{ VDC}^2 \\
 &\quad - 0.099489 \text{ VDC}^3 + 0.0021678 \text{ VDC}^4 \\
 P_{picr} &= -3.0897 + 99.217 \text{ VDC} + 0.53612 \text{ VDC}^2 \\
 &\quad - 0.094909 \text{ VDC}^3 + 0.0041525 \text{ VDC}^4 \\
 P_{por} &= -3.1437 + 99.849 \text{ VDC} + 0.17453 \text{ VDC}^2 \\
 &\quad - 0.025879 \text{ VDC}^3 - 0.000802158 \text{ VDC}^4 \\
 P_{pos} &= -3.0949 + 99.879 \text{ VDC} + 0.22359 \text{ VDC}^2 \\
 &\quad - 0.048260 \text{ VDC}^3 + 0.0015741 \text{ VDC}^4
 \end{aligned}$$

where:

- P_{pocr} = condenser exit pressure
- P_{pop} = heater inlet or pump exit pressure
- P_{picr} = condenser inlet pressure
- P_{por} = receiver exit or pump inlet pressure
- P_{pos} = refrigerant heater exit pressure

By utilizing a fourth order polynomial, the five previous correlating expressions consistently represent the manufacturer's calibration data within an accuracy of ± 0.1 percent for the full pressure sensing range of 0 to 500 psia (0 to 3447 kPa).

3.1.9. Temperature Measurement

The temperature of the refrigerant is measured by RTD immersion probes at four (4) different locations around the loop. These locations are:

- Receiver exit or pump inlet temperature
- Refrigerant heater inlet temperature
- Condenser inlet temperature
- Condenser exit temperature

Refrigerant temperatures are each measured by an Omega PR-11 platinum resistance temperature detector probe. Where possible, the measuring elements are installed at 90 degree pipe elbows to insure turbulence and adequate mixing of the refrigerant stream and subsequently, a more representative temperature measurement. A resistance temperature detector or RTD operates on the principle that a change in electrical resistance in a wire is a representation of the measured temperature. As the temperature increases, the resistance increases. The converse is also true.

The sensing element in each RTD probe is made of 99.99% pure platinum wire with a calibrated resistance of 100 ohms at 32 °F (0 °C). Platinum-based RTD's or PRT's were selected for refrigerant-side temperature measurements since they best meet the needs for precision thermometry (e.g., a high degree of stability and repeatability) and inherently possess negligible drift and error with age. In addition, platinum is more resistant to contamination, is highly refined and is more mechanically and electrically stable compared to other materials, such as nickel, copper, or nickel-iron. The calibration of each RTD element satisfies the 0.1 percent DIN standard tolerance and conforms to the DIN 43760 Standard based on a value of 0.00385 ohms/ohms/°C (0.00214 ohms/ohms/°F).

Calibration of the individual RTD probes employed on the apparatus was verified by monitoring the temperature recorded by the same type of RTD in an liquid/crushed ice mixture in a Dewar flask. Results proved favorable, as the measured temperature fluctuated about a mean value of 32 °F (0 °C) by less than ± 0.25 °F (0.14 °C).

3.2. Air-Side Components

The air-side of the apparatus is designed to allow for the controlled flow of air over a wide variety of environmental test conditions. However, in contrast to the refrigerant-side closed loop, the apparatus air-side components are constructed in an open loop arrangement. A schematic of the air-side arrangement is also given in Figure 3.1. The air-side portion of the apparatus, mounted approximately 10 feet above the finished floor level on the top of the control room, makes use of an existing air flow facility designed and constructed for the ASHRAE-sponsored research project RP529. The existing air flow facility consists of a supply air fan, water coil, and air flow measurement chamber interconnected by a fiberglass insulated, plywood/sheet rock constructed air duct. A new duct section, constructed from sheet metal, was added to connect the air loop of the ACRC project with the existing air flow system. Manual shutoff dampers were also installed to allow isolation of each air system. A complete discussion of the design and capacity of the existing ventilation system is contained in work by Haas [52].

As represented in the schematic, unconditioned air is induced from the Mechanical Engineering Lab through a filter bank by the supply fan equipped with a new motor and variable speed drive. The water coil, located immediately downstream of the fan, is piped to a heat pump and is used for preheating or precooling the air, if necessary. After passing over the water coil, the air travels through a settling means and to the air flow measurement chamber, where recording of the static pressure and differential pressure across a nozzle bank permits determination of the air flow rate. The air then passes through an SCR-controlled, electric duct heater, where its temperature may be increased to the desired condenser inlet condition. The air then flows toward the condenser test section, where its temperature is measured both upstream and downstream of the test section. In addition to air temperatures, the pressure drop across the test section is measured via static pressure taps. Provisions have also been made to measure relative humidity at the condenser inlet and exit by installing duct sensors. After temperature measurement, the conditioned air is then discharged into the Mechanical Engineering Lab. A more detailed description of some of the individual air-side components follows.

3.2.1. Supply Fan

The existing supply air fan is a belt driven, forward curved utility fan with a wheel diameter of 16.5 inches. Since the fan manufacturer and model number were never recorded previously and no nameplate information was located, its actual flow characteristics are not accurately known. However, the fan is geometrically similar to the 16FC Belt Driven Utility Fan, as manufactured by the Trane Company in Wisconsin. The Trane fan is rated for an approximate air flow capacity of 4800 CFM (2.266 m³/sec) at a pressure drop of two inches of water (0.498 kPa).

The entire supply fan assembly is housed in a 50" (L) x 52" (W) x 70" (H) wooden enclosure and consists of the fan, a new 5 HP motor, and a metal support stand. The supply fan induces unconditioned room air from the Mechanical Engineering Lab through a 47" x 20" intake opening in the side of the enclosure. In order to reduce the amount of dust and debris circulated through the air loop, the air intake opening is occupied by a filter bank consisting of three (3) Airguard Industries, Type DP4-40 extended surface air filters with nominal dimensions of 16" x 20" x 4". Removal of the filters from the intake opening permits easy access to the supply fan, motor and drive assembly for normal maintenance, such as aligning and balancing of the sheaves, oiling of the bearings, and changing or tightening of the V-belt and pulleys.

Fan speed, and therefore, the air flow rate, is adjustable over a wide range by means of a variable speed drive controller. This controller is described in greater detail in Section 3.4.3.

3.2.2. Air Flow Measurement Chamber

The existing air flow measurement system incorporated for this research project is based upon the recommended system and measurement procedures described in the following two recognized standards [5,6].

ANSI/ASHRAE Standard 51-1985
"Laboratory Methods of Testing Fans for Rating"
ANSI/ASHRAE Standard 41.2-1987
"Standard Methods for Laboratory Air-Flow Measurement"

The existing system, previously developed as part of ASHRAE research project RP529, employs an array of two interchangeable, elliptical flow nozzles selected from several different throat diameters available in a 136" (L) x 36" (W) x 72" (H) fiberglass insulated chamber. In different combinations, it has been verified that the nozzles allow accurate measurement of the air flow rate over a wide range, including roughly 10 to 4500 CFM (0.0047 to 2.124 m³/sec) based on an air dry bulb temperature of 80 °F (26.7 °C).

A flow nozzle may be categorized as an obstruction flow metering device, similar to a venturi or an orifice plate, in which the airstream is constricted into a smaller cross sectional area. The air pressure drop measured across this constriction permits an accurate determination of the air flow rate. In order to properly calculate the air flow rate, additional variables other than the nozzle air pressure drop must also be known. These include the nozzle geometry characteristics (e.g., throat diameter, L/D ratio), the nozzle inlet dry bulb and wet bulb temperatures, the nozzle inlet static pressure, and barometric pressure.

In total, the flow cross sectional area of the air measurement chamber is approximately 18 square feet (i.e., 36 inches wide by 72 inches high), so that for the previously indicated range of air flow rates, the corresponding air velocity range is 0.55 to 250 ft/min (0.0028 to 1.45 m/sec). Further, the velocity pressure contribution to the total pressure in this velocity range is negligible, thus it may be assumed that the nozzle inlet total pressure for the air flow rate computation is simply equal to the measured inlet static pressure. The air flow rate computation is presented more descriptively in the data reduction techniques of Chapter 5.

Six (6) different nozzles are available for use in the air flow measurement chamber. The throat diameters corresponding to these six nozzles are 1-inch (25.4 mm), 2-inch (50.8 mm), 3-inch (76.2 mm), 4-inch (101.6 mm), 7-inch (177.8 mm), and 10-inch (254 mm), respectively. The discharge coefficient C_{noz} for these nozzles is based upon a measured value of $L/D=0.6$ [52]. Any combination of two nozzles may be accommodated with the existing mounting configuration, including a single nozzle arrangement. A 32" x 20" access door is provided on the side of the chamber to permit easy removal and installation of the nozzles.

In accordance with the ANSI/ASHRAE standards, a wire cloth mesh settling means is installed upstream and downstream of the nozzles. The primary function of the settling means is to provide a more uniform velocity distribution in the flow chamber. With respect to the nozzles, the upstream settling means evenly distributes the air flow pattern so that a relatively uniform air flow passes through the nozzles; whereas, the downstream settling means absorbs the kinetic energy of the expanding stream emerging from the nozzles.

Immediately downstream of the first settling means and approximately 40 inches upstream of the nozzles is an Omega RTD probe to measure the chamber air temperature. This temperature serves as the nozzle inlet air temperature. This temperature measurement is made approximately 24 inches from the cooling coil discharge and 45 inches from the supply fan discharge. This Omega RTD is the same type of device that is employed for the refrigerant temperature measurements, described in Section 3.1.9.

In order to measure the nozzle inlet static and differential pressures, six static pressure taps are each located around the inner perimeter of the flow chamber at the nozzle inlet and exit. The location of these pressure taps is more clearly shown in Figure 3.12. Each pressure tap is constructed of a flat brass plate welded to a hollow brass rod of 0.0625 inches (1.588 mm) inside diameter and is mounted flush with the flow chamber surface to avoid creating any flow disturbances. Since the static pressure taps on each side of the nozzles are only sensitive to the static pressure distribution in their immediate vicinity, they are manifolded into a piezometer ring constructed of 1/2 inch (12.7 mm) nominal diameter plastic tubing. The static air pressure within each piezometer ring is equal to the average of the six individual measurements. From each piezometer ring, plastic tubing of 1/4 inch (6.35 mm) nominal diameter is routed to the data acquisition room, where the electronic pressure measurement equipment is located.

The nozzle inlet static pressure and differential pressure are measured by two Setra Systems, Model 264 electronic pressure transducers. In addition, an inclined manometer is connected in parallel with the transducers and serves as an independent, manual check of the electronic pressure measurements.

The Setra pressure sensors have a sensing range of 0 to 2.5 inches w.g. (0 to 0.622 kPa) and a calibrated full scale accuracy of less than ± 1.0 percent. In addition, each sensor operates on a supply voltage of 24 volts DC and provides a 0 to 5 volts DC output signal proportional to the pressure measurement. The inclined manometer is an Ellison draft gage with a pressure range of 0 to 2.0 inches w.g. (0 to 0.498 kPa) and calibrated in 0.01 inch w.g. (0.069 kPa) scale divisions.

Separate air flow rate calibration tests with five individual nozzles (1, 3, 4, 7 and 10 inches) and one combination of nozzles (10 and 4 inches) and over a wide range of flow rates were performed in order to verify that the Setra pressure transducers and the inclined manometer produced the same measurements. The results of this calibration are tabulated in the Tables 3.3 thru 3.8. In summary, the measurements recorded with the inclined manometer and the Setra transducers compare favorably and fail to differ consistently by more than two percent.

The nozzle test data shown in Tables 3.3 thru 3.8 may also be considered as a selection guide for determining the most appropriate nozzle throat diameter(s) to be installed for a particular range of test air flow rates. The following tabulated summary consists of the nozzles tested and a recommended applicability range based on the accuracy presented in the previous tables. Although the tabulated test data was developed for nozzle inlet temperatures from 72 to 80 °F (22.2 to 26.7 °C), the following data may still be used as a rough guide.

Nozzle Diameter(s) (inches)	Recommended Airflow Range (cfm)	Maximum Fan Speedpot Setting
1	6 - 30	4.5
3	60 - 270	5.0
4	200 - 490	5.5
7	400 - 1460	7.0
10	800 - 2400	8.5
10 & 4	1250 - 4250	8.5

In the previous table, the column labeled "Maximum Fan Speedpot Setting" refers to the fan speed controller and the maximum air flow rate setting attainable before the nozzle(s) must be interchanged. Above this maximum speedpot setting, either the nozzle inlet static pressure induced is beyond the range of that measureable by the inclined manometer or the fan is too undersized to overcome the pressure losses in the duct system and in all likelihood, will blow its circuit if operated any longer.

3.2.3. Electric Duct Heater

The primary purpose of the duct heater is to increase and control the temperature of the airstream entering the condenser. The duct heater is an SCR controlled, electric process air heating coil, supplied and manufactured by the Indeeco Company of St. Louis. It possesses a total heating capacity of 24 kW or approximately 82,000 Btuh and operates on a supply power of 240 volts, 3 phase.

The kilowatt rating of the duct heater is designed based on a maximum air flow rate of 2500 cfm (1.180 m³/sec) and a desired air temperature rise of 30 °F (16.7 °C) through the heater with 70 °F (21.1 °C) air at the inlet. Similarly, at the same inlet temperature condition with an air flow rate of 1000 cfm, a maximum air temperature rise of about 75 °F (41.7 °C) is achievable. The following formula is convenient for calculating the kW requirements at the previously mentioned air flow conditions and other full capacity or partial capacity conditions:

$$kW = \frac{60 \dot{m} c_p \Delta T}{3413} = \frac{60 \rho \dot{V} c_p \Delta T}{3413}$$

The electric duct heater construction consists of spirally finned tubular sheathed elements made of 80% nickel, 20% chromium resistance wire and having a watt density of 33 watts per square inch. The heater is confined to a heavy duty, galvanized steel frame for slip-in style duct mounting, i.e., the entire frame slides through a rectangular opening cut in the side of the air duct.

A factory-built, NEMA 12 control panel is mounted integrally with the heater and outside the airstream. It includes a disconnect switch interlocked with the panel door handle, pilot lights to indicate heater on/off operation, an airflow switch, primary and secondary thermal cutouts, magnetic contactors, fuses and a control transformer in addition to the two (2) SCR controllers, one for each heating stage. In addition to the interlocked disconnect switch, the airflow switch and thermal cutouts are built-in safety features. The airflow switch is an air velocity sensitive, diaphragm-operated differential pressure switch, interlocked to prevent heater operation if there is less than 0.07 inches w.g. (0.0174 kPa) pressure difference between the inside and outside of the duct. The thermal cutouts are interlocked to protect the duct heater against overheating as a result of low airflow. The primary thermal cutout is an adjustable high limit switch with a setpoint range from 60 to 250 °F (15.6 C to 121.1 °C). This cutout automatically resets when the temperature decreases to safe levels. The secondary thermal cutout is provided in case of failure of the primary cutout and requires manual reset when the temperature rises 50 °F (27.8 °C) above the primary setpoint.

The finned tubular construction was selected over the open coil construction since the finned tubular style is more mechanically stable, more moisture and contamination resistant, and more tolerant to nonuniform airflow conditions. In particular, due to their larger mass, the finned tubular elements possess a greater thermal lag (about two minutes) compared to open coil elements (about 30 seconds) so that the potential for temperature stratification downstream of the heater is reduced.

In order to control the exit air temperature, the duct heater output is regulated by two (2) SCR controllers. Operating on a 4 second time base and 60 Hertz half-cycles, the SCR controllers permit fractional heat inputs of 1/480th the total heater capacity (50 watts). An algorithm has been implemented in the data acquisition system to automatically control the duct heater capacity based on the user-specified exit air temperature setpoint and several other parameters. A more complete description of the SCR controllers and the control algorithm appears in Section 3.4.1 and Section 3.5, respectively.

3.2.4. Water Loop

A schematic representation of the water loop is presented in Figure 3.13. The major equipment comprising the water loop are a heat pump, a storage tank, a water coil, two circulation pumps and the interconnected piping network. As mentioned previously, the water coil is an existing part of the air ventilation system constructed for the ASHRAE-sponsored research project. In addition, a previously installed water source heat pump and copper piping network was removed in favor of the larger capacity water system described in this Section. The primary purpose of the water loop is to precool or preheat the airstream by circulating a chilled or heated water/ethylene glycol mixture through the existing coil.

The water loop is actually composed of two subsystems that operate continuously and independently of one another. A separate water circulation pump is dedicated to each of these systems. The first system circulates a constant flow of water between the storage tank and the heat pump. During its operation in either heating or cooling mode, the heat pump cycles on and off so as to heat or cool the water and maintain a constant tank water temperature. A Penn thermostat or aquastat with a manually adjustable setpoint is installed to sense and control the tank water temperature. The second subsystem circulates hot or chilled water between the storage tank and the coil. Forced air in crossflow over the water coil is subsequently heated or cooled prior to entering the air flow measurement chamber.

3.2.4.1. Heat Pump

The heat pump is a WS Series, nominal 20-ton, compact water-cooled chilling and heating unit, as manufactured by Thermal Energy Transfer Corporation (TETCO) of Ohio, and operates on an AC supply power of 240 volts, 3 phase. It is equipped with a two-speed hermetic compressor with vibration eliminators, a manual switchover refrigerant reversing valve, and manually cleanable, water-to-refrigerant, AISI 316 stainless steel plate frame heat exchangers. The unit operates on a refrigerant charge of approximately 18 lb_m of R-22, and is self contained within a 29" (L) x 29" (W) x 25" (H) acoustically-insulated, metal enclosure.

A complete description of the heat pump system, including operating and maintenance instructions, heat exchanger performance data and unit wiring diagrams, is contained in the TETCO manual entitled, "W Series Installation Guide" on file.

The catalogued chilling and heating capacity ratings of the heat pump and the corresponding refrigerant evaporating and condensing temperatures are listed below:

	<u>Cooling Mode</u>	<u>Heating Mode</u>
Total Capacity		
MBtuh	288.3	263.2
(kW)	(84.5)	(77.1)
Evaporating Temperature		
°F	45.0	30.0
(°C)	(7.2)	(-1.1)
Condensing Temperature		
°F	100.0	110.0
(°C)	(37.8)	(43.3)

The above capacity ratings are based on an entering water temperature of 50 °F (10 °C) and a water flow rate of 2 gpm/ton (0.0358 mL/J), where one ton is equal to 12,000 Btu/hr (3.52 kW). According to the manufacturer, by utilizing a 75 percent water, 25 percent ethylene glycol mixture, it is possible to obtain an evaporator water discharge temperature as low as 25 °F (3.9 °C) in cooling mode at high compressor speed. At this condition, the total chilling capacity is derated to 11.25 tons or 135 MBtuh (39.6 kW). Likewise, in heating mode at low compressor speed, the heating capacity is derated to 11.5 tons or 138 MBtuh (40.4 kW) and a discharge water temperature of 100 °F (37.8 °C) is possible.

A NEMA 1, front access and insulated, hinged door control cabinet is mounted on the face of the heat pump and contains a factory wired control system that includes a 24 VAC, 100 VA control transformer, a manually resettable refrigerant high pressure safety control, two manually resettable refrigerant low pressure safety controls, mechanically and electrically interlocked compressor contactors, high speed and low speed electronic time delays for compressor restart, connections to the compressor overload protection control module, and a terminal

board. In addition, diagnostic lights are provided on the cabinet exterior to indicate conditions of positive power supply, reversing valve position signifying cooling or heating mode (light is energized in cooling mode), refrigerant high and low pressure cutouts and no water flow. Heat pump internal fusing is located below the main control cabinet in a separate metal enclosure with screw access, removable front cover.

A field wired, switch-operated control box includes a compressor on/off switch, a refrigerant reversing valve switch, and a compressor high/low speed switch. It is important that the heat pump power be disconnected prior to changing the refrigerant reversing valve position.

Water flow on the evaporator side is accomplished by means of a steady flow, centrifugal pump, described in Section 3.2.4.3. Water flow on the condenser side is accomplished by means of city water pressure, which typically fluctuates from 40 to 65 psia (275.6 to 447.9 kPa). Hand-operated, ball valves are installed in the condenser inlet and outlet lines to throttle the condenser water flow. For best results during heat pump operation, the inlet valve should be fully open and the exit valve adjusted until the compressor suction and discharge pressures remain constant. Individual pressure gauges with a range of 0 to 100 psia are installed upstream and downstream of the condenser. By recording the pressure drop across the condenser, the actual water flow rate may be determined by consulting the manufacturer's pressure drop curves. These heat exchanger pressure drop curves are available in the "W Series Installation Guide", Appendix A.

A McDonnell & Miller, Series FS4-3 flow switch is installed in the evaporator loop water piping. This paddle-actuated flow switch is wired to de-energize the heat pump (i.e., the compressor) when the evaporator-side water flow is 7.0 gpm or less. Heat pump operation at such reduced evaporator-side water flows may cause freeze-up of the heat exchanger during cooling duty.

Standard refrigerant pressure gauges with a range of 0 to 500 psia (0 to 3445 kPa) are connected to the compressor suction and discharge ports to monitor refrigerant pressures during start-up and operation.

As previously identified, the TETCO heat pump is also protected by both high and low refrigerant pressure limit controls. These controls are factory set to de-energize the compressor whenever the refrigerant pressures at the compressor discharge and suction ports reach an abnormally high or low level. As examples, these undesirable pressure levels may be a result of insufficient water flow or circulation pump failure, lack of refrigerant charge, or heat exchanger fouling. The low pressure cutout is activated when the compressor suction pressure falls below 50 psia (344.5 kPa); whereas, the high pressure cutout is activated when the discharge pressure rises above 350 psia (2412 kPa). Proper throttling of the condenser water flow typically prevents these cutouts from activating. Manual resets are provided on the face of the control cabinet to restart the compressor after a lockout has occurred. However, repeated resetting of these safety controls (i.e., more than twice for each 30 minutes) should be avoided since this action may eventually damage or freeze-up the system.

In addition to the pressure limit controls, the compressor is also furnished with automatic reset, internal overload protection. Should the compressor amperage drawn exceed factory specified limits, the thermal overload will interrupt the power supply and shut off the compressor. The overload protector will then automatically reset once the compressor has adequately cooled. After tripping several times, this overload may require as long as an hour or longer to reset.

For heat pump operation with a 75 percent water, 25 percent ethylene glycol solution, the manufacturer has verbally indicated that the low pressure setting may be reduced to 35 psia (241.3 kPa) by manually adjusting the screw setpoint device. The heat pump manufacturer also recommends that a water/glycol solution be employed when producing chilled water temperatures of less than 40 °F (4.4 °C), in order to prevent freeze-up and possible damage to the heat exchanger.

3.2.4.2. Water Coil

The existing water coil is a Bohn Heat Transfer, Model 5WH6A-15T24 and is housed in a 32" (L) x 36" (W) x 30" (H) insulated, wooden enclosure. A summary of the coil geometry is listed in Table 3.9. The coil is installed directly downstream of the fan discharge and is equipped with a field-mounted galvanized steel sheet metal drain pan for removal of condensate. A "Water Coil Rating Report" obtained from Bohn indicates the rated cooling performance of the coil based on the following input parameters:

- five (5) different air flow rates from 800 to 1600 CFM
- entering air temperatures of 70 °F dry bulb and 63 °F wet bulb
- an entering water/glycol temperature of 15 °F
- a 10 °F water/glycol temperature rise through the coil
- two (2) different water/glycol solutions of 25% and 30%

A complete listing of the results from the Bohn simulation may be found in Table 3.10. In summary, the coil total cooling capacity ranges from approximately 69,000 to 125,000 Btu/hr (2.02 to 3.66 kW), and the discharge air temperature ranges from approximately 24 to 31 °F (-4.4 to -0.55 °C) based on the aforementioned conditions. In addition, the required water/glycol flow rate ranges from about 15 gpm to 27 gpm (0.946 to 1.704 liters/sec). Finally, in general terms, an increase in the amount of glycol from 25% to 30% tends to decrease the total cooling capacity by around 1.2%.

Based on the results of the Bohn simulation, it was determined that the existing water coil was suitable for meeting the low temperature, discharge air requirements of the ACRC full condenser project. Additional published rating information related to the water coil, in accordance with ARI Standard 410-87 [3], may be found in Bohn Heat Transfer, Bulletin 3112.

Since the fan discharges such a short distance to the coil, manually adjustable turning vanes are installed between the fan and the inlet face of the coil to assure a more uniform distribution of air flow across the entire coil face area. The vanes adjustment may be accessed through the supply fan housing.

Chemical cleaning agents, namely Drewsperser 739 and Drewsperser 794, are available from the University of Illinois, Operations and Maintenance Department for flushing the water loop, including piping and system equipment, of debris. These liquid concentrate antifoulants permit effective removal of silt, oil, iron and copper oxide deposits, and other suspended matter from the water piping system, when utilized properly in accordance with specifications found in UIUC Building Standards, Section 15457, and official Material Safety Data Sheets generated by Ashland Oil Company, Drew Chemical Division of New Jersey. Prior to installation of the new water piping and heat pump system, the Drewsperser chemicals were circulated through the existing water coil to remove copper oxide deposits.

3.2.4.3. Water Circulation Pumps

As previously mentioned, two (2) water circulation pumps are installed in the hydronic system. These pumps are rigidly mounted on a wood platform directly above the heat pump. The first pump is a Bell & Gossett Model 1522-3S centrifugal pump with a 1/2 hp, 1750 rpm, direct drive motor. It is used to circulate a constant flow of water between the storage tank and the heat pump. According to performance characteristic curves obtained from Bell & Gossett, the pump is rated for a water flow rate of 30 gpm (1.89 liters/sec) at a total head of approximately 28 feet (8.53 m). The corresponding pump efficiency at this rating point is around 51 percent. In addition, this pump is not self-priming and therefore, requires routine bearing lubrication in accordance with its nameplate instructions. Bearing lubrication is particularly important when the pump has been idle for extended periods of time.

The second pump is a Cole Parmer Model 7112-40, self-priming centrifugal pump with a 1-1/2 hp, 3450 rpm, direct drive motor. This pump is equipped with a manually adjustable, variable speed drive and is employed to circulate water between the coil and the storage tank. It is capable of producing a range of water flow rates from 0 to 30 gpm (0 to 1.89 liters/sec) and sized based on conservatively calculated piping pressure losses of roughly 52 ft (15.8 m) or 23 psi (160 kPa) at a 75% water, 25% ethylene glycol flow rate of 30 gpm (1.89 liters/sec). A full description of the variable speed drive may be found in Section 3.4.3.

3.2.4.4. Water Storage Tank

The water storage tank is an AB Series cylindrical unit with a holding capacity of 200 gallons (758 liters), as manufactured by Chemtainer Industries of New York. It is mounted approximately 40 inches above the heat pump and the circulation pumps, on a Unistrut platform. The tank, approximately 36 inches in diameter and 48 inches in height, is constructed of 5/16 inch thick polypropylene and is provided with a removable top cover to limit the amount of dust and debris from the surroundings that may contaminate the water. Polypropylene was chosen as the tank material since it is rated for continuous operating temperatures up to 230 °F (110 °C); whereas, polyethylene is merely rated for continuous operating temperatures up to 160 °F (71.1 °C). Both polypropylene and polyethylene possess a Class "A" compatibility rating with ethylene glycol.

3.2.4.5. Water Piping and Miscellaneous Components

Most of the water circulatory system is constructed of 1-1/2 inch nominal diameter, polyvinyl chloride (PVC), Schedule 80 pipe. Information on PVC pipe and fittings dimensions, ratings and availability may be found in "Plastic Piping Systems, Encyclopedia for the '80s" [84]. Only a small portion of the system, the city water supply and return lines for the heat pump, is constructed of copper pipe of 2 inches nominal diameter.

Schedule 80 thermoplastic pipe was incorporated in lieu of copper pipe primarily due to its equivalent or slightly better rank in terms of economy, versatility, impact resistance, chemical resistance, ability to withstand long-term pressures, and ease of installation. In addition, Schedule 80 was selected rather than Schedule 40 due to its slightly higher pressure ratings at elevated water temperatures. For example, in accordance with Plastic Piping Systems, Inc. [84], at a water temperature of 70 °F (21.1 °C), the pressure rating of Schedule 40 PVC is 330 psi (2275 kPa) whereas the pressure rating of Schedule 80 is 471 psi (3247 kPa). At a temperature of 110 °F (43.3 °C), the pressure ratings decrease by 50 percent to 165 and 235 psi (1138 and 1620 kPa), respectively. Further, at its maximum recommended service temperature of 140 °F (60 °C), the pressure ratings of the two types of PVC are 72 and 104 psi (496 and 717 kPa), respectively.

All PVC connections are glue welded, except for connections to the heat pump, pumps, coil, strainers, and rotameters, which require threaded fittings. These fittings are wrapped with teflon tape. Several ball-type shutoff valves are located throughout the PVC piping network to permit easier maintenance and service of major equipment. These valves are the Chemtrol Tru-Bloc™ TU Series type, constructed of PVC and equipped with integral unions on the inlet and exit ports to allow complete physical disconnection or removal of major equipment from the water loop, if necessary. A bronze Y-strainer with a manually removable and cleanable filter is installed immediately upstream of each pump to eliminate any large debris that may exist in the water system. Rotameters are Omega FL-550 in-line models and are installed downstream of each pump to monitor the volume flowrate of water.

3.2.5. Air Duct

Dimensionalized drawings of the new air duct system, are shown in Figures 3.14 and 3.15. The dimensions given on these sketches are strictly sheet metal to sheet metal. The new air duct is constructed of 20 gage, galvanized steel sheet metal with interior dimensions of 24" x 24". The duct is externally insulated with two (2) one inch layers of Armaflex black rubber insulation with a thermal conductivity of approximately 0.0208 Btu ft/hr ft² °F (0.0354 W/m K). The Armaflex insulation is only rated for service temperatures up to 200 °F (93.3 °C).

The new duct is connected air-tight to the existing wooden air duct constructed for the ASHRAE-sponsored research project. A neoprene sealed, sliding gate, shutoff damper is installed in each branch duct to isolate each air system.

Turning vanes are installed in the two (2) 90 degree duct elbows. In addition, there are three (3) 12" x 12" hinged and latchable, access doors provided in the side of the duct. These are located immediately upstream of the electric duct heater and upstream and downstream of the condenser test section, respectively, for service and maintenance. The air loop discharge is routed upward from the top of the control room approximately six feet and away from the supply fan air intake to avoid recirculation of the condenser exit air.

3.2.6. Temperature Measurement

In accordance with ANSI/ASHRAE Standard 41.1-1986, entitled "Standard Method for Temperature Measurement" [8], two independent and redundant methods are employed to measure the duct average air temperatures. Each of these methods is utilized at locations both upstream and downstream of the condenser coil test section.

The first way duct air temperatures are measured is by use of a bendable area averaging resistance thermometer. The actual thermometer is a Minco Products Model SS457PE averaging sensor with a copper encased, platinum wire sensing element and an element insertion length of 12 feet. The platinum sensing element has a calibrated resistance of 100 ohms at 32 °F (0 °C) and registers approximately 110 ohms at a temperature of 70 °F (21.1 °C). In order to reduce the potential for temperature measurement errors and fluctuations caused by self-heating of the RTD sensor, the excitation sent from the data logger is consistently less than the maximum of 5 milliamps as recommended by Minco. Since the air duct is only two feet square, the RTD sensing element is installed in a serpentine arrangement so that it uniformly crisscrosses the duct six times, per Figure 3.16. The primary aim of this sensor is to average out any temperature stratifications in the air duct upstream and downstream of the condenser. The temperature sensor is supported inside the duct by standard plumbing strap with longitudinal perforations. The finning effect and heat conduction as a result of the supporting structure is considered to be negligible.

The second way duct air temperatures are measured is by use of thermocouple grids. A thermocouple grid, equipped with nine (9) thermocouples arranged in a 3 x 3 matrix and each located at the center of an equal cross sectional area, is installed at locations approximately 24 inches upstream and downstream of the condenser core. The purpose of the thermocouple grid is to note the degree of temperature stratification by analyzing the nine individual readings on each side of the condenser. In addition, the grid serves as additional verification of the air temperature difference across the condenser coil. Figure 3.17 represents the layout of the individual thermocouples in the grid.

All thermocouple material is 30 gage, Type T, copper-constantan wire with teflon insulation, as supplied by Gordon Temperature Measurement of Illinois. For faster response time, 30 gage wire is employed rather than a larger diameter wire (e.g., 24 gage, 20 gage, etc.). Initially, mounting with 36 gage, Type T thermocouple wire was attempted; however, extreme difficulty in splicing and forming the measurement junctions led to selecting 30 gage material. Permanently bonded, measurement junctions have been formed on one end via an electric arc welder. Each thermocouple beaded junction faces into the incoming air stream and is positioned at least 1.5 inches (3.8 cm) away from any part of the grid support structure so as to minimize conduction of heat along the wires. The standard calibrated accuracy of the Type T thermocouples is ± 1.5 °F (0.83 °C) in the temperature range of -75 to 200 °F (-59 to 93.3 °C). Thermocouples operate on the principle that the electromotive force (emf) difference between the measuring junction and a reference point is proportional to the measured temperature. In this case, the data logger is used as a reference point. Calibration of the data logger ice point has been verified by monitoring the temperature of a liquid/crushed ice mixture in a Dewar flask.

Averaging the duct air temperatures as measured by either one of the two previously described methods is only correct provided that the air velocity distribution is uniform at the duct cross section containing the measurement. Although there are no velocity traverses currently made in the air duct, it is assumed that on the basis of accurate heat balances obtained, any errors introduced by nonuniformities in velocity are insignificant.

In general, temperature measurements in the airstream may also be subject to errors due to thermal radiation, particularly when the sensing elements are directly exposed or in direct line of sight to surfaces appreciably different in temperature from the airstream. Thermal radiation effects in the air duct, as a result of a "hot" condenser, were verified by comparing individual air temperature measurements of the RTD averaging sensor and of a single thermocouple for average condenser refrigerant temperatures between 120 °F (48.9 °C) and 170 °F (76.7 °C). The thermocouple was mounted to the RTD averaging probe surface so that the RTD behaved as a thermal shield, thus preventing the thermocouple measurement junction from "seeing" the radiant source, i.e., the condenser surface.

Comparison of individual measurements of each device for the stated range of condenser refrigerant temperatures failed to indicate any major influences of radiation. Specifically, the differences in air temperature noted were within the calibrated accuracy ranges of the sensors. Therefore, the necessity for shielding the air temperature sensors was ignored.

3.2.7. Pressure Measurement

On the air-side, the only pressure measurement recorded, other than those related to the air flow measurement chamber described in Section 3.4.2, is the differential pressure across the condenser coil. A Setra Systems Model 239 pressure transducer is employed for this measurement. The transducer has a differential pressure range of 0 to 2.5 inches w.g. (0 to 0.622 kPa) with an accuracy of 0.14 percent of the full scale or equivalently, about ± 0.35 inches w.g. (0.087 kPa). Its output is sent to the data acquisition system in the form of a 0 to 5 volts DC signal proportional to the measured differential pressure.

In order to sense the differential pressure properly, duct static pressure measurements are made in accordance with the suitable practices recommended in Section 7.2.1 of the ANSI/ASHRAE Standard 41.3-1989, entitled "Standard Method for Pressure Measurement" [7]. Burr-free, static pressure taps of 0.0625 inches diameter are located in the duct wall upstream and downstream of the condenser test section. Upstream and downstream, a total of six holes each are spaced uniformly around the three accessible sides of the duct in accordance with Figure 3.19. The taps are manifolded into a piezometer ring and routed to the high and low sides of the pressure transducer by using 1/4 inch (6.35 mm) nominal diameter plastic tubing.

Static tests with no air flow were performed in order to determine an appropriate voltage offset associated with the Setra differential pressure sensor. The results of one such test are shown in Figure 3.18. During testing, sensor measurements were recorded once per minute for over 840 minutes. As a result, an average voltage offset of +0.02345 volts DC or slightly less than 0.06 inches w.g. (0.015 kPa) was determined to be a reasonable value.

3.2.8. Humidity Measurement

Relative humidity of the air is measured upstream and downstream of the condenser coil test section by two (2) General Eastern RH-2 relative humidity transmitters. These instruments are duct-mounted and measure humidity by means of a resistance change through a bulk polymer film contained within a 9 inch element length. When humidity increases, ionization from the film molecular structure increases, decreasing the resistance. Conversely, when humidity decreases, ionization decreases, increasing the resistance. In contrast to capacitive and surface resistance type sensors, the bulk polymer sensor is resistant to surface and field effects, such as contamination, that may adversely affect accuracy. The sensor can measure relative humidity in the range from 20 to 95 percent with an accuracy of ± 2 percent. A 250-ohm precision resistor is wired in series with the humidity sensor current loop in order to furnish a 1 to 5 volts DC proportional output signal to the data logger. The transmitter is powered by a 24 volts DC power supply.

3.3. Condenser Test Section

The condenser test section is one of the most critical subsystems of the apparatus. This section consists of the mobile air conditioning or household refrigerator/freezer test condenser coil, and the air duct and refrigerant piping necessary to make it an integral part of the rest of the experimental apparatus. Figures 3.20 and 3.21 schematically indicate the refrigerant-side and air-side components that comprise the condenser test section.

Two of the major requirements of this section is that it be easily interchangeable and adequately sized so that several different condenser coil types and configurations may be accommodated. Easy interchangeability is accomplished on both the refrigerant-side and air-side. On the refrigerant-side, Swagelok QH Series, Quick-Connect valves with teflon seals are installed and axially aligned with the refrigerant piping. These valves or couplings are of two piece body construction and consistently maintain their 5000 psig (34,400 kPa) pressure rating over the temperature range of -10 to 250 °F (-23.3 to 121.1 °C) when coupled or uncoupled. When decoupling the quick-connects, it is first necessary to reduce the system pressure on the condenser-side to at least 50 psig (340 kPa). These valves were employed rather than flexible piping connections which are more

susceptible to leakage and not typically rated for as high of pressures. On the air-side, nuts and bolts connect the 30 to 36 inches long, transitioned sheet metal duct section to the air loop. In order to simplify the manual changeover of test sections, the condenser test sections are prefabricated with sheet metal and refrigerant piping so that they are ready for installation when needed.

The physical space allotted for the condenser test section permits accommodation of different condenser coil configurations. In particular, the apparatus is capable of running a typical household refrigerator condenser in any position. However, the test section is currently restricted to subjecting condensers to horizontal air flow.

Refrigerant pressure drop across the condenser test section is measured by comparing the individual measurements of the two absolute pressure transducers located at the condenser inlet and exit. These transducers are described in Section 3.1.7. Individual pressure transducers were employed rather than a single differential pressure transducer, since not a single vendor consulted could guarantee a differential pressure sensor's accuracy for the condition of gas on one side and liquid on the other side. Most differential pressure transducers are rated for gas-to-gas or liquid-to-liquid service only. Effects of separate phases on the transducer are apparently not well understood.

3.4. Instrumentation and Control

A description of the instrumentation employed to control the capacity of the refrigerant heater, electric duct heater and the heat pump is furnished in this Section. In addition, a description of the variable speed drive controllers for the supply fan and the water circulation pump is provided.

3.4.1. SCR Controllers

Silicon controlled rectifiers, or more commonly known as SCR's, are solid state, proportional power controllers employed to modulate the electric duct heater and refrigerant heater loads. The use of SCR controllers permits precise heating and temperature control to be achieved.

The electric duct heater utilizes two (2) Solitech, dual triac, three phase power controllers. These are the Model H3 Master and Model J3 Slave controllers, respectively, each rated for a load of 30 amps. The refrigerant heater utilizes two (2) Solitech, single triac, single phase power controllers for the evaporating and superheating sections, respectively. These are both the Model H Master controller, rated for a load of 28 amps.

In general, SCR controllers provide power control and extremely rapid cycling rates by employing triac semiconductor power switching. The logic and control circuits provide proportional control by establishing a basic time interval (in this case, 4 seconds) and energizing the heater for a percentage of each interval. The percentage of on-time varies in accordance with the demand sensed by a remote thermostat or manual dial potentiometer. At the beginning of each interval, the logic circuit analyzes the signal from a user-specified potentiometer or thermostat and determines the number of AC cycles or half-cycles required to supply the proper amount of heat. Using a zero voltage cross switching system to eliminate any radio frequency interference, the control circuit turns the triac(s) on for the appropriate number of cycles. At the end of each interval, the circuitry is retriggered to repeat this sequence. Slave units contain only the trigger and triac circuits. The control signal for the slave is derived from the master unit.

In the case of the Solitech power controllers, the basic time interval used is four seconds and each 60 Hertz subinterval is further subdivided into half cycles. Therefore, rather than only 240 cycles per interval (4 seconds multiplied by 60 cycles per second), the Solitech SCR's operate on 480 cycles per interval. This permits power input resolution as low as 1/480th of the total capacity.

Since there is a large amount of heat generation associated with SCR circuits, the triac(s) are mounted on a large finned heat sink to protect them against overheating. The triac(s) and heat sink are in close thermal contact; however, the heat sink is electrically isolated from the line voltage.

Figures 3.22 and 3.23 are block control diagrams of the refrigerant heater control system and the electric duct heater control system, respectively. The two SCR controllers associated with the refrigerant heater are mounted on the wall of the heater enclosure. These units are equipped with LED pilot lights that illuminate when the heater is on. The two SCR controllers associated with the electric duct heater are mounted on the inside of the terminal box. The heat sinks extend to the outside of the box. Although the SCR controllers are mounted within the duct heater terminal box and the heat sinks are oversized to protect them against overheating, the power controllers are actually designed for high ambient temperatures up to 186 °F (85.6 °C) at full load operation so that mounting them in terminal boxes is not normally a problem.

The refrigerant heater SCR controllers operate on 0 to 5 volts DC input signal from a remote control potentiometer circuit or alternately, the data acquisition system. The duct heater SCR controllers operate on a 0 to 5 volts DC signal received from the data acquisition system. An algorithm has been implemented to automatically control the electric duct heater based on the desired exit air temperature setpoint, and the actual exit air temperature. The SCR will energize the heater for the appropriate number of AC cycles in order to maintain the exit temperature.

3.4.2. Watt-Hour Transducers

As shown in Figure 3.22, on both the refrigerant heater evaporator and superheater circuits, individual watt-hour transducers are employed to permit an accurate measurement of the electric power consumed in the heaters and subsequently, the amount of heat input to the refrigerant. This measurement is important since it may serve as a additional check of the air-side and refrigerant-side heat balance computations. These watt-hour transducers are the Model H2BT, as manufactured by Ohio Semitronics.

Each watt-hour transducer provides a factory calibrated, 0 to 1 mA pulse output signal proportional to the electric power delivered to the load. The corresponding accuracy is ± 0.2 percent of the reading.

Since SCR controllers regulate the heat input by modulating the number of AC cycles, current transformers are wired downstream of each watt-hour transducer to provide the transient response necessary to accurately sense and compute the power consumption. The current transformers measure the amount of current delivered to the individual refrigerant heater sections. As previously stated, the watt-hour transducers output pulses which may be converted in to power based on the voltage output by the SCR controllers. By factory calibration, 100,000 pulses by the watt-hour transducer corresponds to 10 kw, or equivalently, 10 pulses is the same as 1 watt.

3.4.3. Variable Speed Drives

The flow rates through the supply air fan and one of the water circulation pumps are individually controlled by adjustable frequency, AC drives. The fan motor is 5 hp and is driven by an Emerson Electric, Prism Model 2950-8001 variable speed drive. The primary purpose of this drive is to allow variation of the air flow rate over a large range without the continual need for interchanging nozzles in the air flow measurement system. The pump motor is 1-1/2 hp and is driven by an Emerson Electric, Prism Model 2950-8000 variable speed drive. The main purpose of this drive is to allow variation of the chilled or hot water flow rate between the water storage tank and the coil, thereby the coil capacity and air temperature at the discharge of the coil.

Each variable speed drive is an adjustable frequency, adjustable voltage, AC motor controller specifically designed for centrifugal fans or pumps driven by three phase, 60 hertz squirrel cage induction motors. Its basic structure consists of a main control board containing the electronics for motor speed control with sequencing and fault protection logic. The controller main circuit boards are located in a forced air ventilated, louvered NEMA 1 housing, wall-mounted within about two feet of the controlled equipment and equipped with diagnostic lights to indicate controller operation. For each controller, a separate remote control operating station consisting of a start/stop switch and a manually adjustable, speedpot dial is wired and wall-mounted remotely from the main controller.

The fundamental principle of speed control for adjustable frequency drives is based on the following formula for a standard AC motor:

$$N_s = 120 \frac{F}{P}$$

where N_s is the synchronous motor speed in revolutions per minute (rpm), F is the frequency in cycles per second (cps), and P is the number of poles.

The number of poles of a particular motor is set in its design by the manufacturer. The adjustable frequency system controls the frequency applied to the motor. The motor speed is then proportional to this applied frequency. Control frequency is adjusted by means of a potentiometer mounted remotely from the control board logic. The frequency output of the controller is infinitely adjustable from the standard 60 Hertz to any lower value, thus making the speed range of the motor also infinitely adjustable.

3.4.4. Heat Pump Control

In cooling or in heating mode, the TETCO heat pump is wired to cycle on and off in order to maintain a constant tank water temperature. Since the range of water temperatures sustainable by the heat pump is large, two different water tank thermostatic controllers are necessary. Each of these 24 volt DC controllers consists of either a SPST or SPDT switching circuit contained in a NEMA 1 enclosure along with the manually adjustable setpoint and differential controls. A remote bulb, metal plated sensor is immersed in the storage tank to measure the water temperature.

The first temperature controller is a Johnson Controls/Penn Series A19 unit with a temperature setpoint range of -20 to 100 °F (28.9 to 37.8 °C) and a differential or deadband range of 3 to 12 °F (1.7 to 6.7 °C). This single-stage controller is intended for use when operating the heat pump in cooling mode. It is wired to cycle the heat pump off when the water tank temperature decreases below the temperature setpoint.

The second temperature controller is a two-stage, Johnson Controls/Penn Series A28 unit with an adjustable temperature setpoint of 60 to 140 °F (15.6 to 60 °C) and an adjustable differential of 3 to 12 °F (1.7 to 6.7 °C). This controller is furnished specifically for use when the heat pump is operated in heating mode. It is wired to cycle the heat pump off when the water tank temperature increases above the temperature setpoint.

3.4.5. Power Supplies

Instruments that require control voltage to operate are wired to one of two International Power, 24 volts DC regulated power supplies. One power supply is a Model IHA24-0.5 with a total amperage capacity of 0.5 amps or 500 milliamps. The other power supply is a Model IHC24-2.4 with a total amperage capacity of 2.4 amps or 2400 milliamps. Each unit is equipped with built-in open sense lead protection and regulated to 0.05% for any changes in load or 120 volts AC line voltage. Since the power supplies are open-frame construction, each unit is wall-mounted in its own dedicated sheet metal enclosure. These power supplies are also currently oversized to permit adding control items if necessary.

3.5. Data Acquisition System

The data acquisition system is comprised of the following components listed below:

1. A Campbell Scientific 21X Micrologger or data logger, whose responsibility it is to log and collect the measured data at user-specified time intervals and to send this data to the computer.
2. Three (3) Campbell Scientific, AM416 multiplexers, mounted remotely from the data logger, through which the data logger samples or scans a variety of sensor measurements.
3. A Campbell Scientific SC32A, optically isolated interface or communications bus through which data is transported from the data logger to the computer.
4. A MacPlus computer and hard disk which reads, displays and saves the measured data.

The interrelationship among the four components of the data acquisition system is shown in the block diagram, Figure 3.24.

A data logger program was developed to sample all the measurement channels in the three multiplexers and the data logger itself. The data logger has the capability to record and output data at a maximum rate of 80 items per second; however, due to the large number of measured variables during steady-state testing of the experimental facility (76 in all) and the associated programming space, the minimum sampling rate or execution interval that may be sustained is six seconds. The minimum output interval is 30 seconds.

A data acquisition program, written in TrueBasic™, has been formulated to allow two-way communication between the data logger and the MacPlus computer. In short, the data acquisition or communications program has two primary responsibilities. First, the program translates the data received from the data logger into information that may be monitored, saved and later analyzed on the MacPlus computer while the system is in operation. This information includes the sensor measurements of pressure, temperature, flow rates and heater energy consumption. Second, the communications software allows sending commands from the computer to the data logger. These commands include control input signals to the duct heater and refrigerant heater.

The communications software developed is interactive and during experimental data collection, permits displaying the measured information on the computer screen either in a raw format or on a schematic diagram, according to user preference. These format for these two screen display options are shown in Figures 3.25 and 3.26, respectively. In addition, the data acquisition program incorporates several audible alarms and warning messages to notify the user of potential, unsafe operating conditions with the apparatus. A full description of each these alarms as well as the recommended action associated with each alarm is contained in the "Startup, Operating, and Maintenance Instructions" in Appendix A, Section A.3.

The data acquisition program has been titled "DataMaster V1.1". A full copy of the subroutine based, computer source code is given in Appendix C. This source code was current as of July 1991 and is strictly based on testing with refrigerant CFC-12 by incorporating curve-fit polynomial functions for the saturated liquid transport properties. Some of the improvements to the source code, currently in development, include adding thermodynamic and transport property routines for refrigerant HFC-134a and organizing a data collection procedure for transient performance testing.

As identified previously, three multiplexers are employed in the data acquisition system. Each of these multiplexers may accommodate up to 32 different sensor measurements. Two of the multiplexers are remote-mounted at a wiring distance of approximately 12 feet from the data logger. One of these is dedicated to the 18 air duct thermocouple measurements while the other is reserved for the condenser surface thermocouple measurements. The final multiplexer, immediately adjacent to the data logger, accommodates the RTD sensors and pressure transducers. All other measurement devices are wired directly to the data logger.

The maximum control wiring distance associated with any of the apparatus measurement devices is approximately 18 to 20 feet. However, sensors specific to the condenser test section are at a wiring distance of less than 15 feet from the data acquisition system. In order to reduce the potential for electrical noise and other disturbances that may abruptly affect the data acquisition system, wiring with aluminum shielding is employed in nearly all instances, except for 24 volt DC wiring from the power supplies. Shielded wires have been properly grounded at the terminal point.

A data logger control algorithm utilizing "proportional plus integral" control has been written to permit automated control of the duct heater capacity and subsequently, the inlet air temperature to the condenser. This control scheme is based on the error between the measured exit air temperature from the duct heater (i.e., same as the condenser inlet air temperature) and the user-specified temperature setpoint. The data logger sends a 0 to 5 volts DC output signal to the electric duct heater that is proportional to the amount of energy required to heat the airstream to the desired temperature. Formerly, a "feed forward" control

system was incorporated to control the duct heater output. This algorithm scheme was based on four variables, namely, the nozzle inlet air temperature or duct heater inlet air temperature, the duct heater exit air temperature, the air mass flow rate, and the duct heater exit setpoint temperature. However, the "proportional plus integral" control was instituted in lieu of the "feed forward" control since more precise control of the condenser inlet air temperature was achieved.

Due to its recent success, the "proportional plus integral" or "PI" control scheme is currently being adapted to automate individual control of the refrigerant heater evaporating and superheating sections, respectively, for more precise control of the refrigerant inlet temperatures and pressures to the condenser. For the evaporating section, the control algorithm is based on the error between the heater-mounted thermocouple measurement at the end of the evaporating section and the user-specified setpoint temperature. For the superheating section, the algorithm is based on the error between the refrigerant temperature at the condenser inlet and the desired setpoint temperature input by the operator.

For a more complete description of the "proportional plus integral" control scheme, an ASHRAE reference [12] is recommended for further reading.

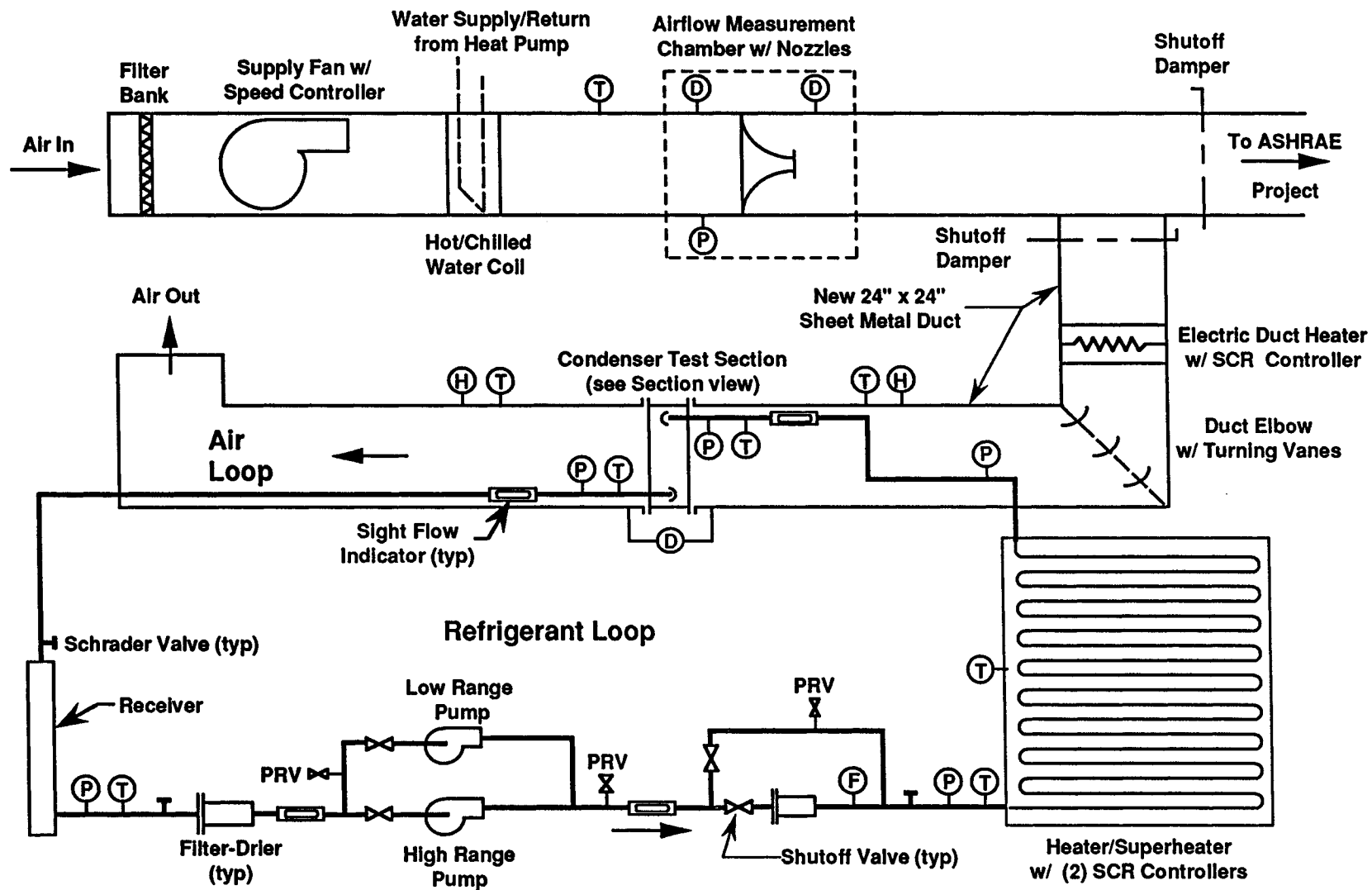


FIGURE 3.1
OVERALL SCHEMATIC OF EXPERIMENTAL APPARATUS

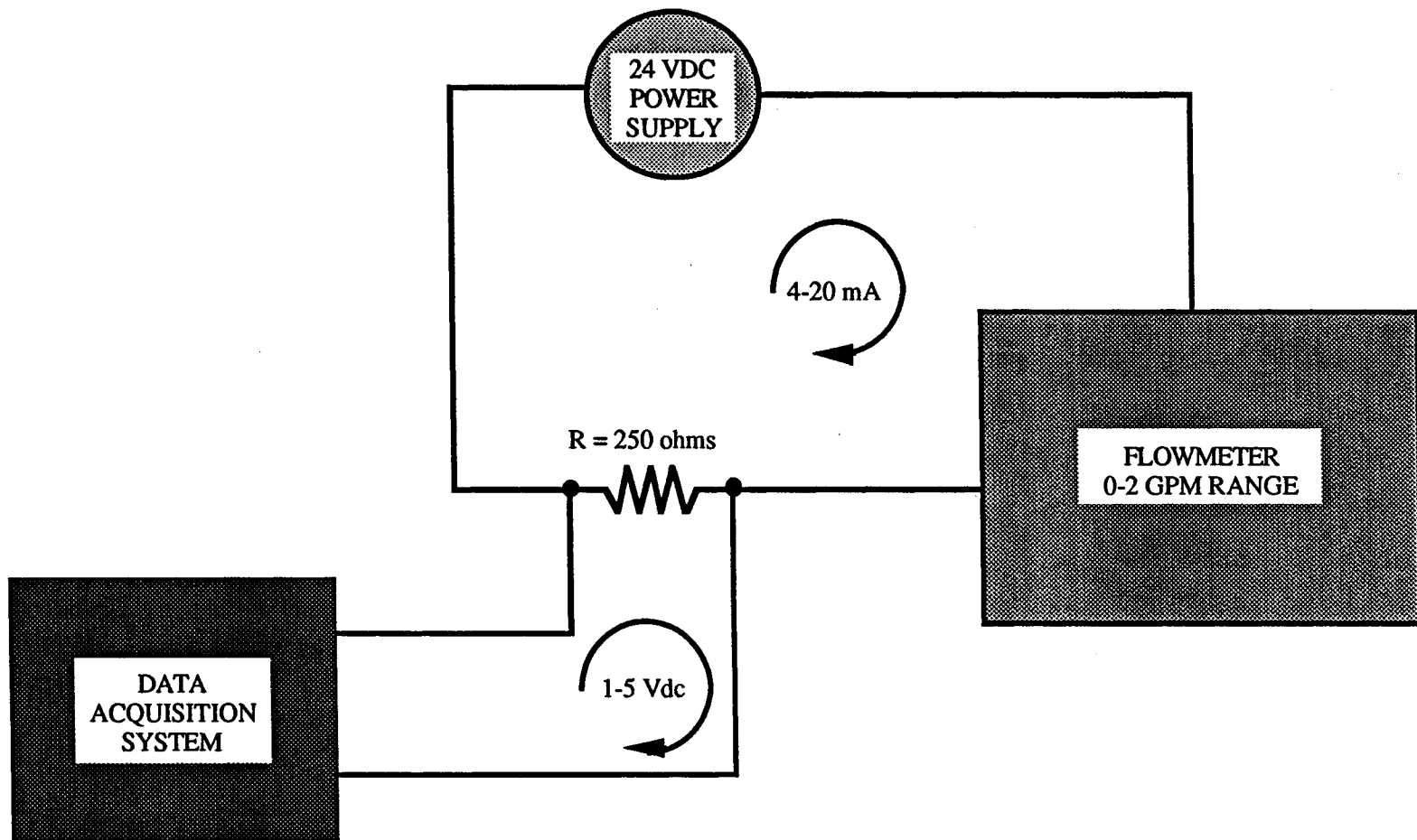


FIGURE 3.2
REFRIGERANT FLOWMETER WIRING DIAGRAM

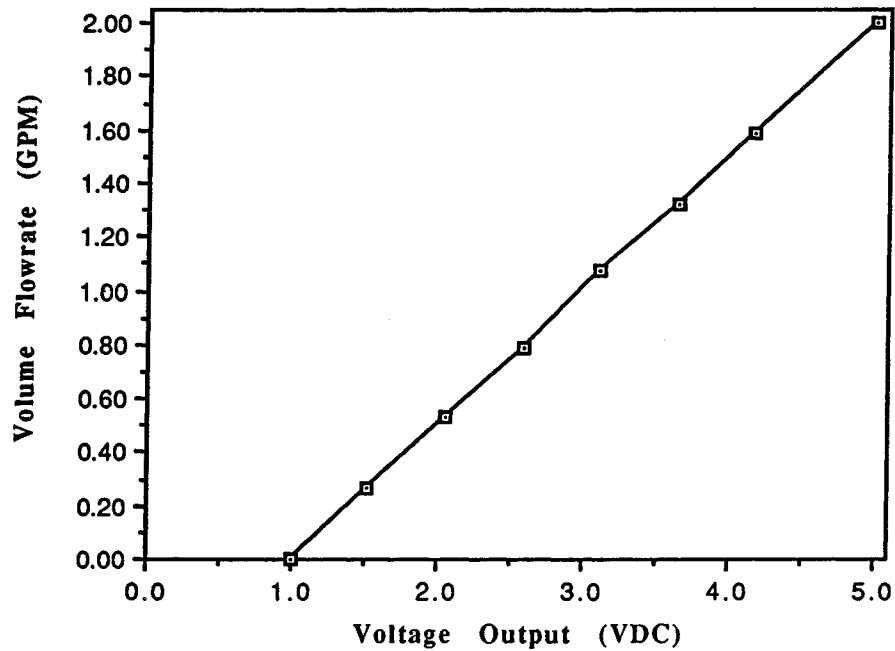


FIGURE 3.3
MAX MACHINERY FLOWMETER CALIBRATION
CONVERTED FROM 4-20 mA TO 1-5 vDC

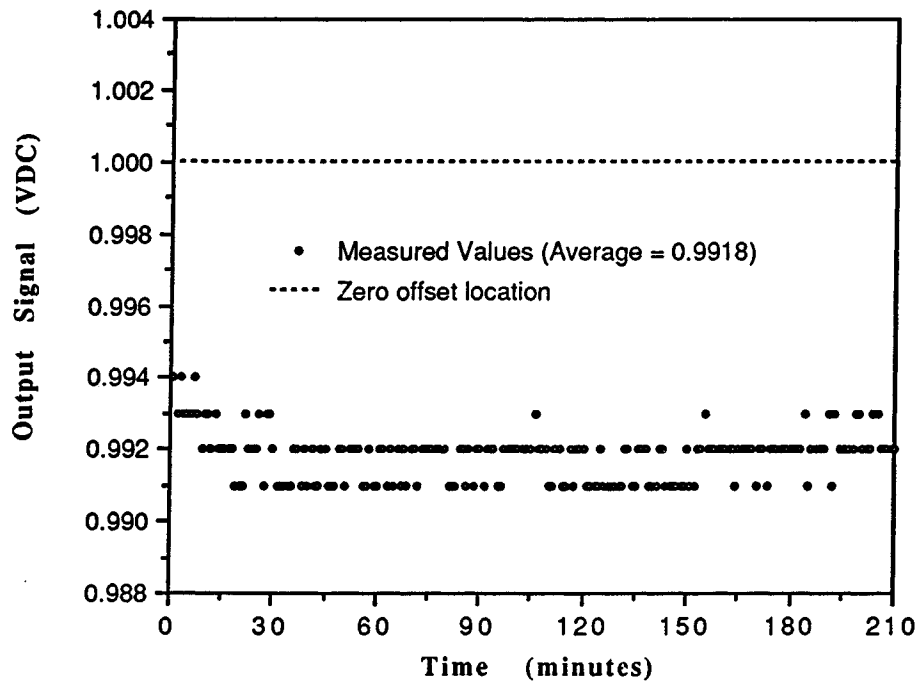


FIGURE 3.4
MAX MACHINERY FLOWMETER CALIBRATION
FIELD DETERMINATION OF VOLTAGE OFFSET

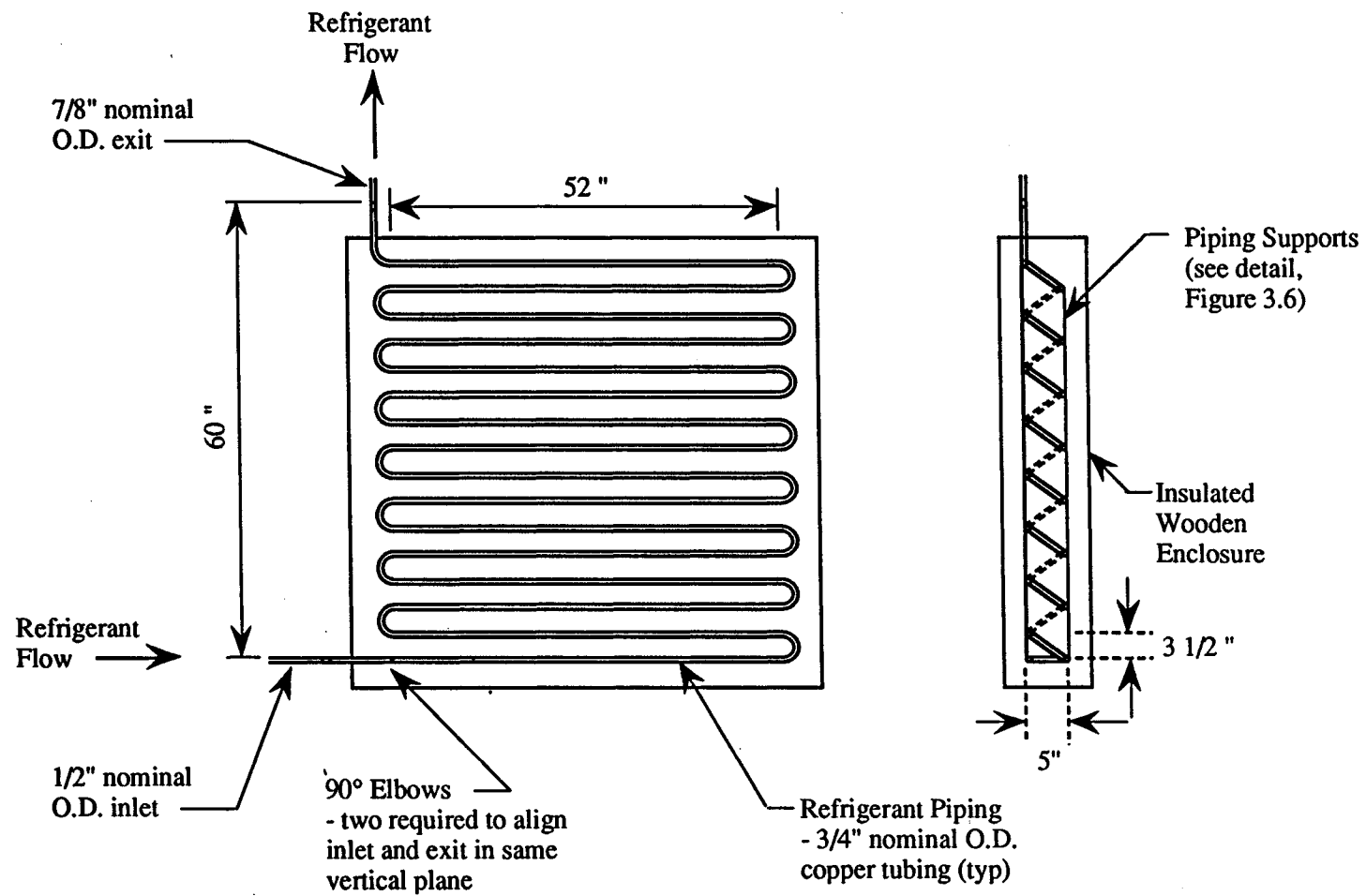


FIGURE 3.5
REFRIGERANT HEATER CONSTRUCTION

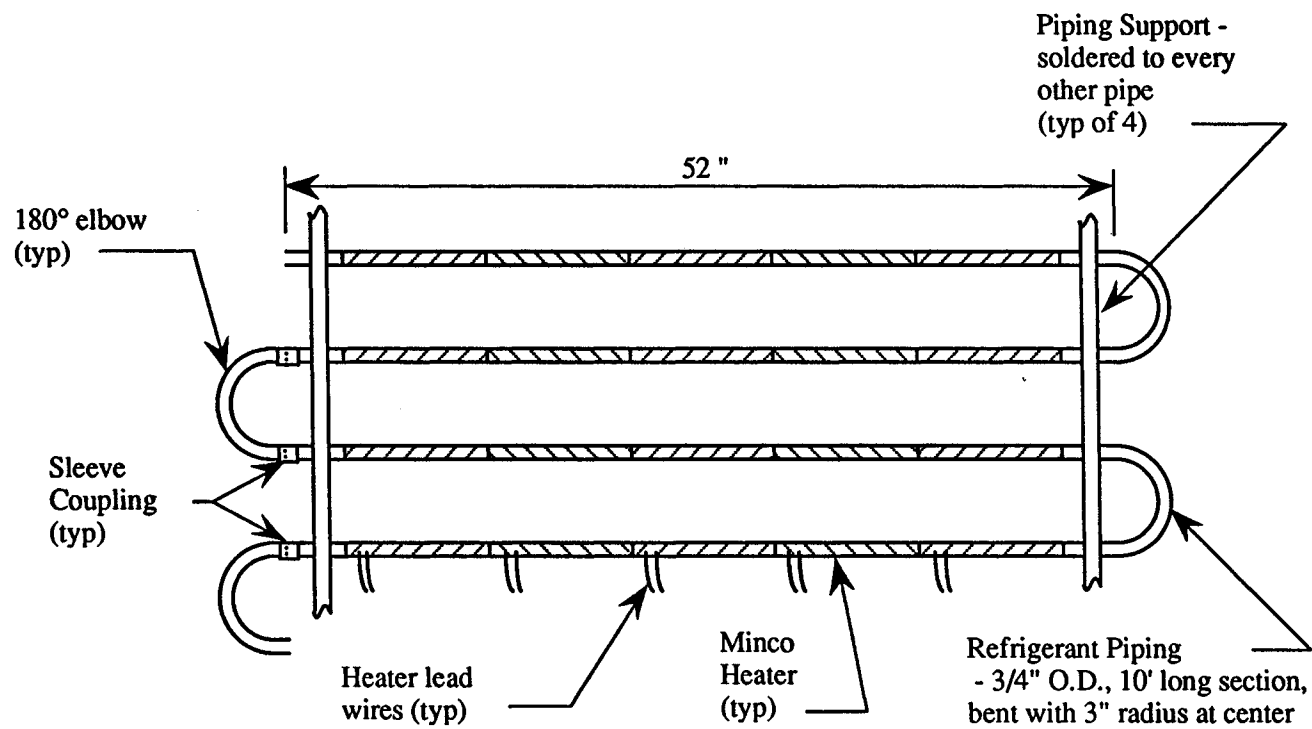


FIGURE 3.6
REFRIGERANT HEATER DETAIL

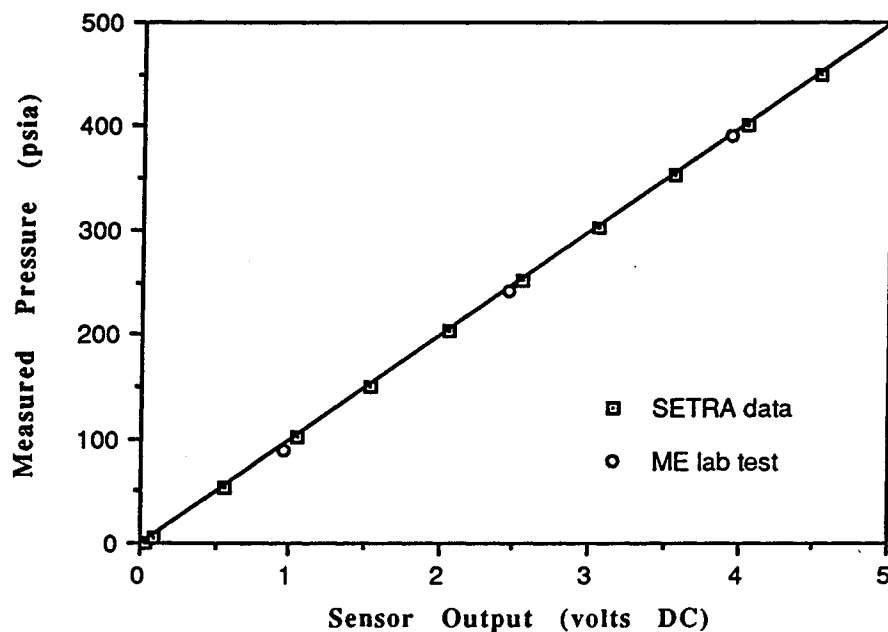


FIGURE 3.7
PRESSURE TRANSDUCER CALIBRATION DATA
SETRA MODEL 280E - SERIAL NO. 252389
CONDENSER EXIT PRESSURE

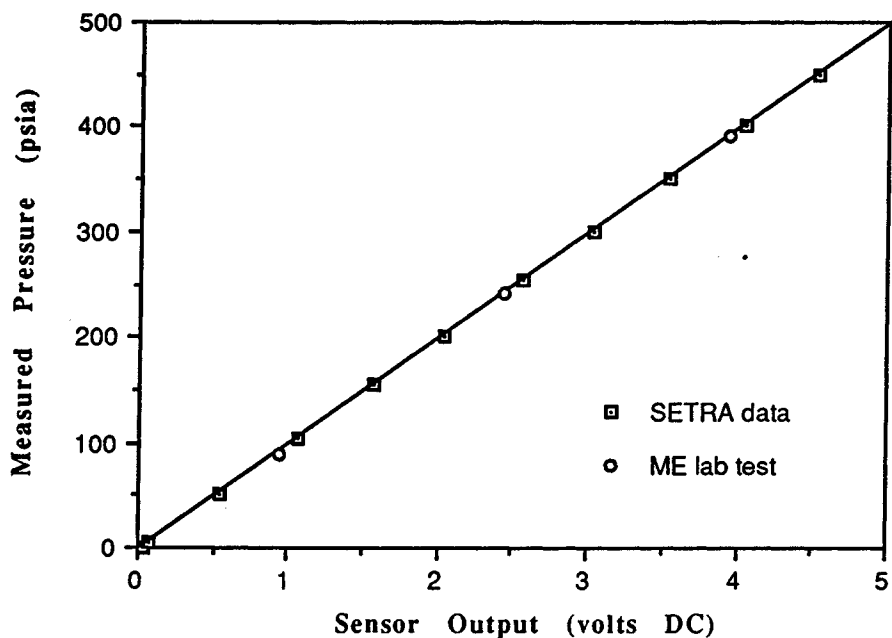


FIGURE 3.8
PRESSURE TRANSDUCER CALIBRATION DATA
SETRA MODEL 280E - SERIAL NO. 252390
HEATER INLET/PUMP EXIT PRESSURE

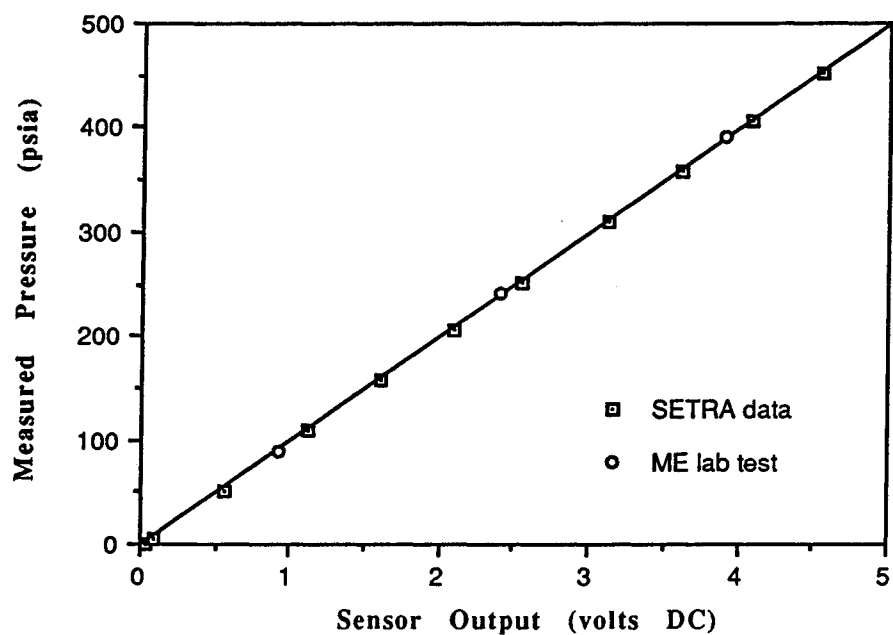


FIGURE 3.9
PRESSURE TRANSDUCER CALIBRATION DATA
SETRA MODEL 280E - SERIAL NO. 252391
CONDENSER INLET PRESSURE

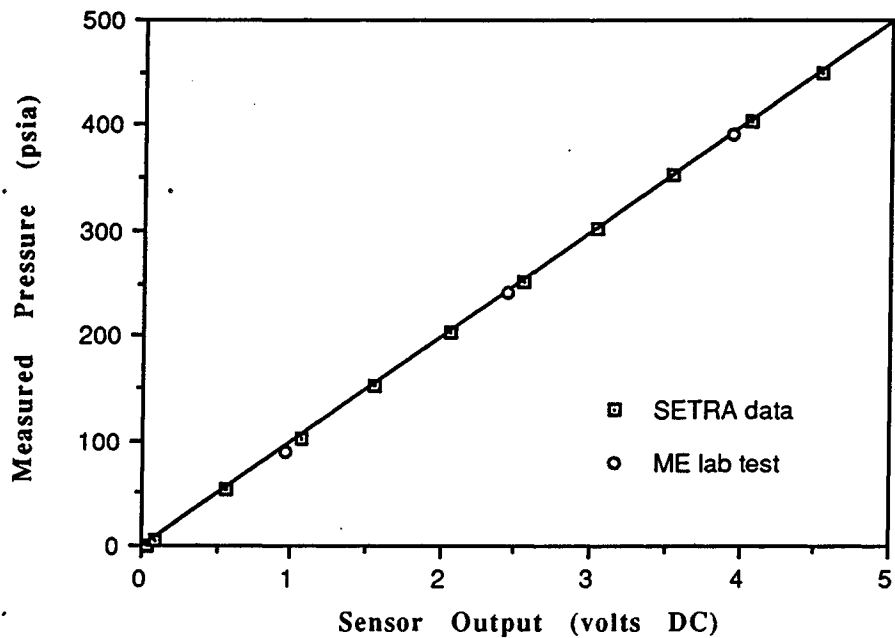


FIGURE 3.10
PRESSURE TRANSDUCER CALIBRATION DATA
SETRA MODEL 280E - SERIAL NO. 252392
RECEIVER EXIT/PUMP INLET PRESSURE

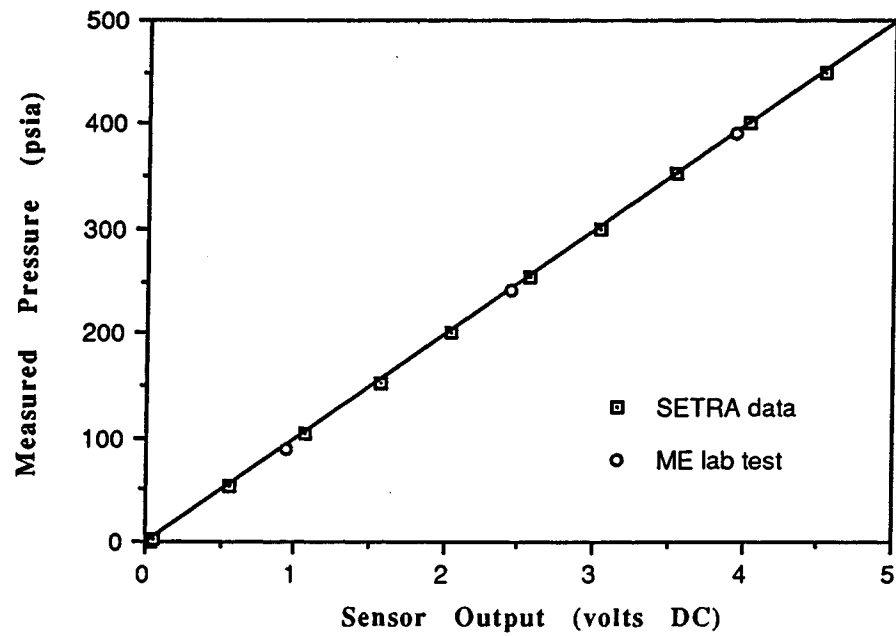


FIGURE 3.11
PRESSURE TRANSDUCER CALIBRATION DATA
SETRA MODEL 280E - SERIAL NO. 252393
HEATER EXIT PRESSURE

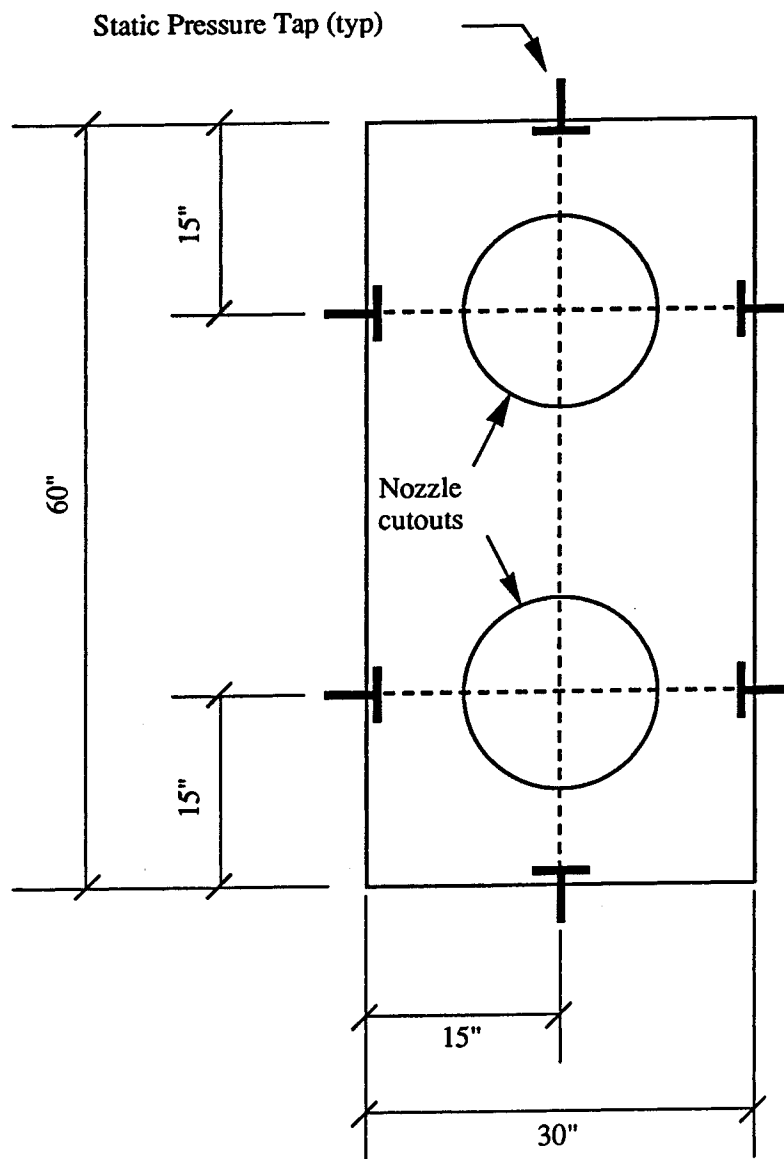


FIGURE 3.12
AIR FLOW MEASUREMENT CHAMBER
STATIC PRESSURE TAP LOCATIONS
UPSTREAM AND DOWNSTREAM OF NOZZLES

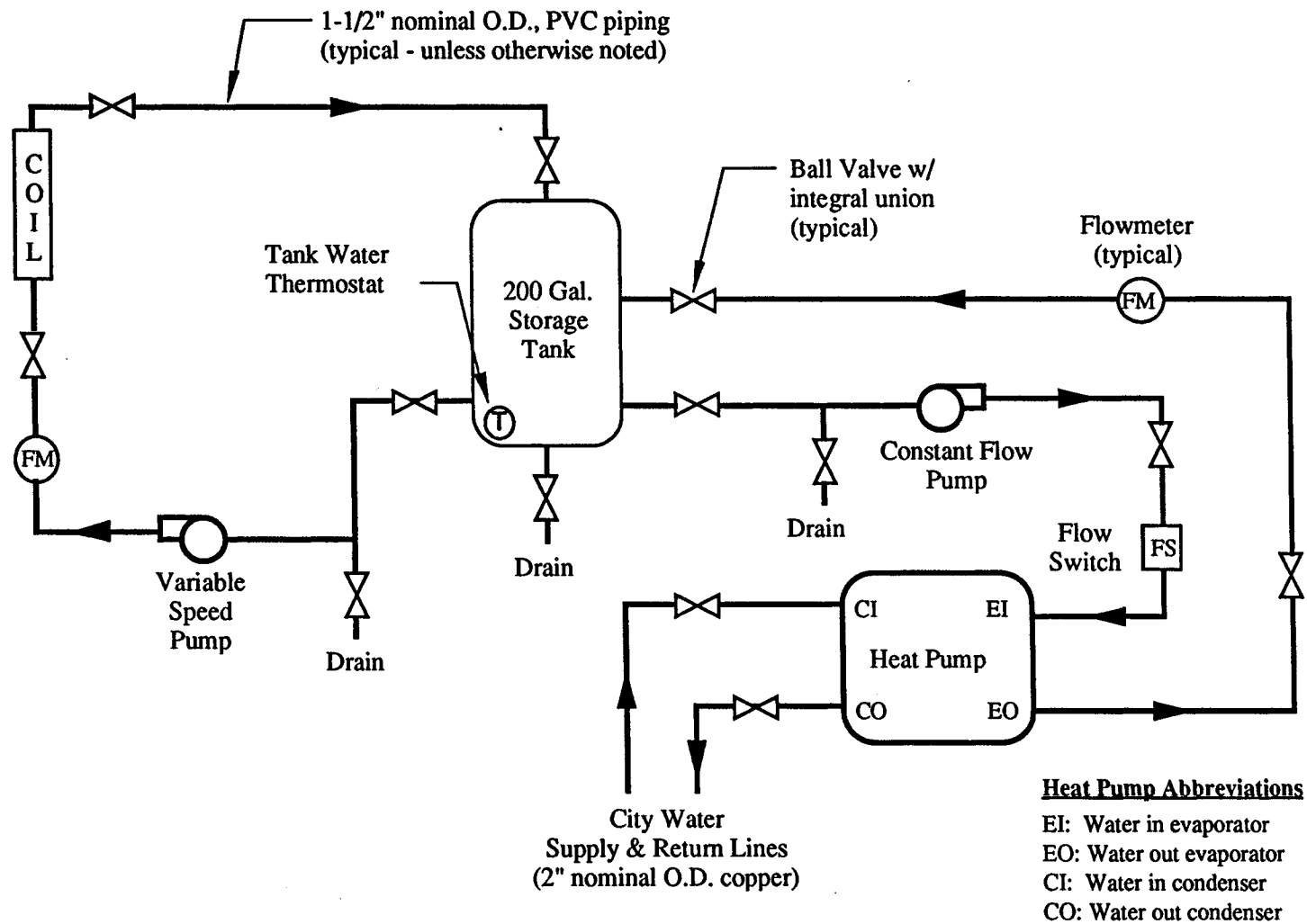


FIGURE 3.13
SCHEMATIC OF WATER LOOP PIPING

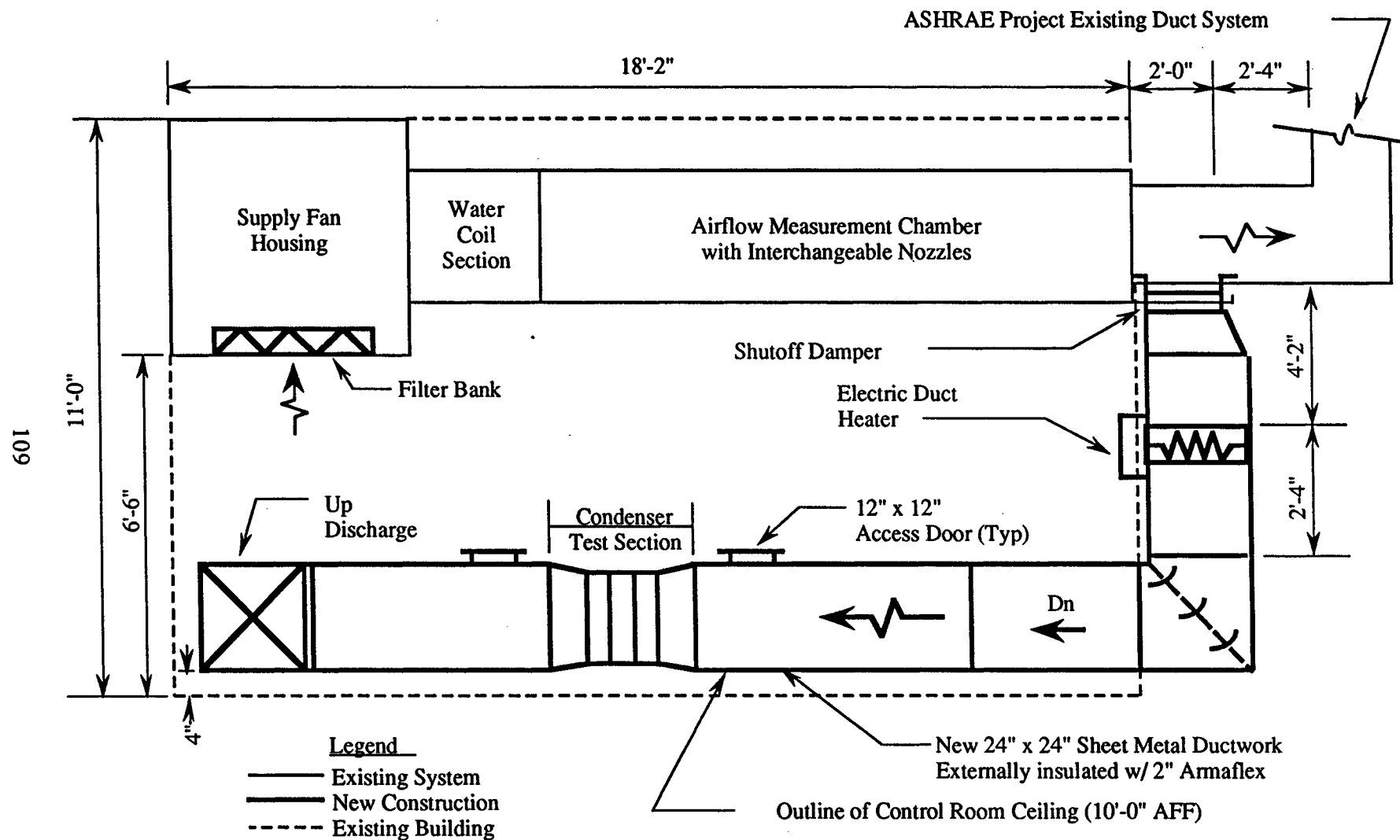


FIGURE 3.14
PLAN VIEW OF NEW AND EXISTING AIR LOOP ARRANGEMENT

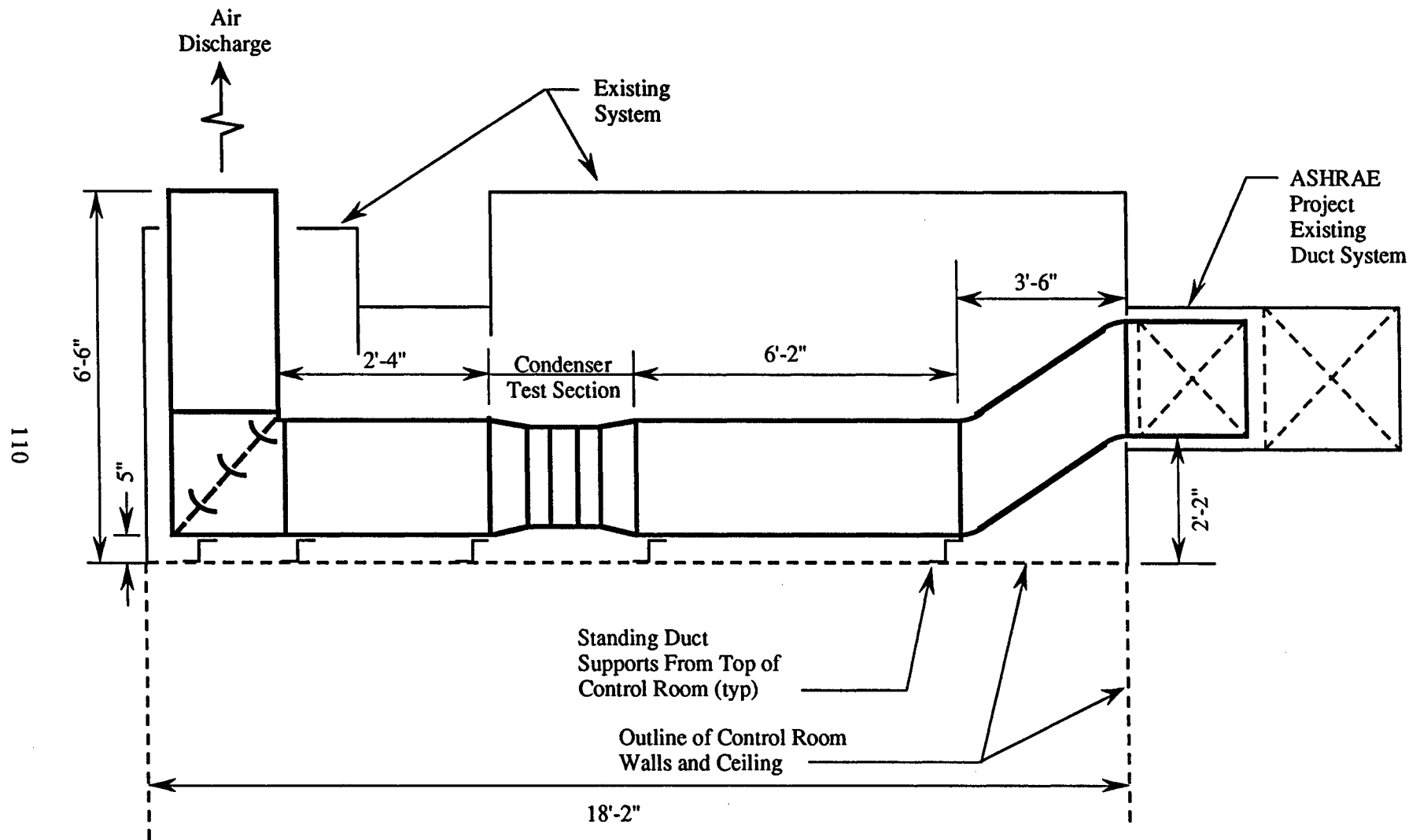


FIGURE 3.15
FRONT VIEW OF NEW AIR DUCT SYSTEM

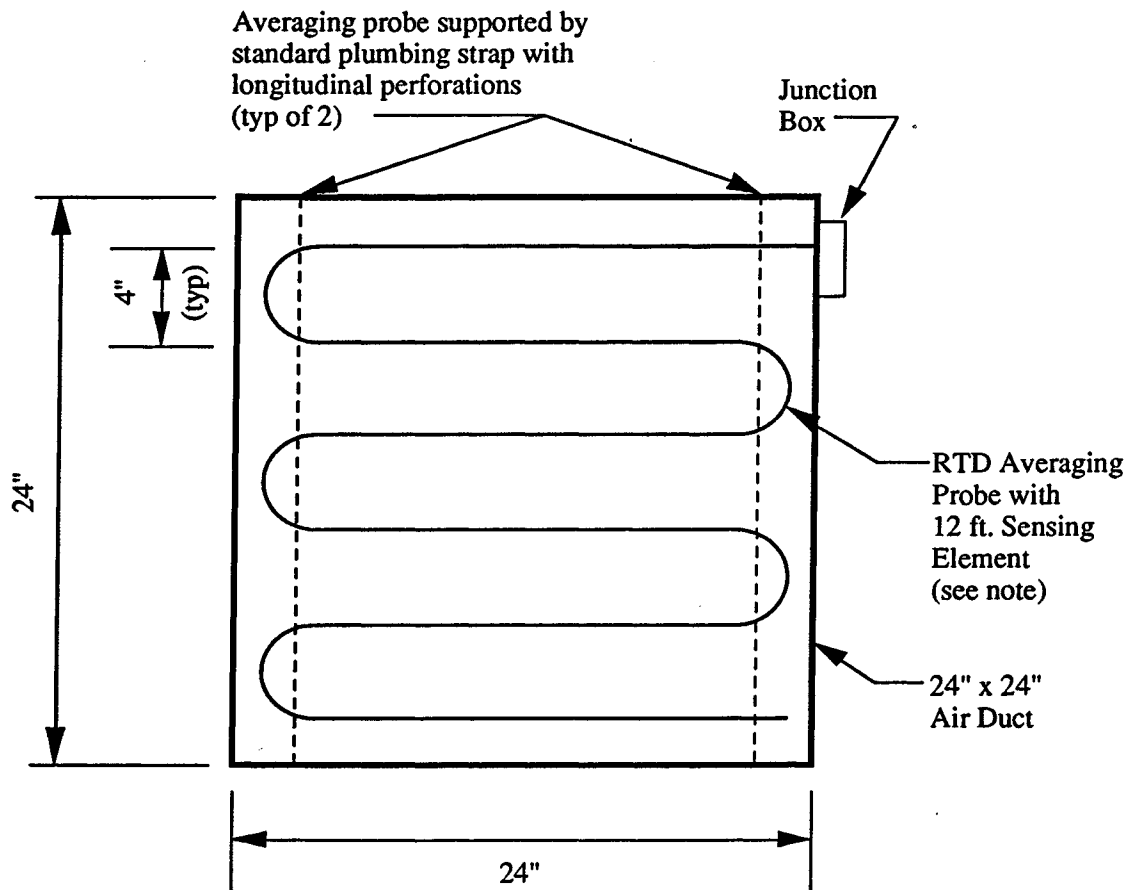
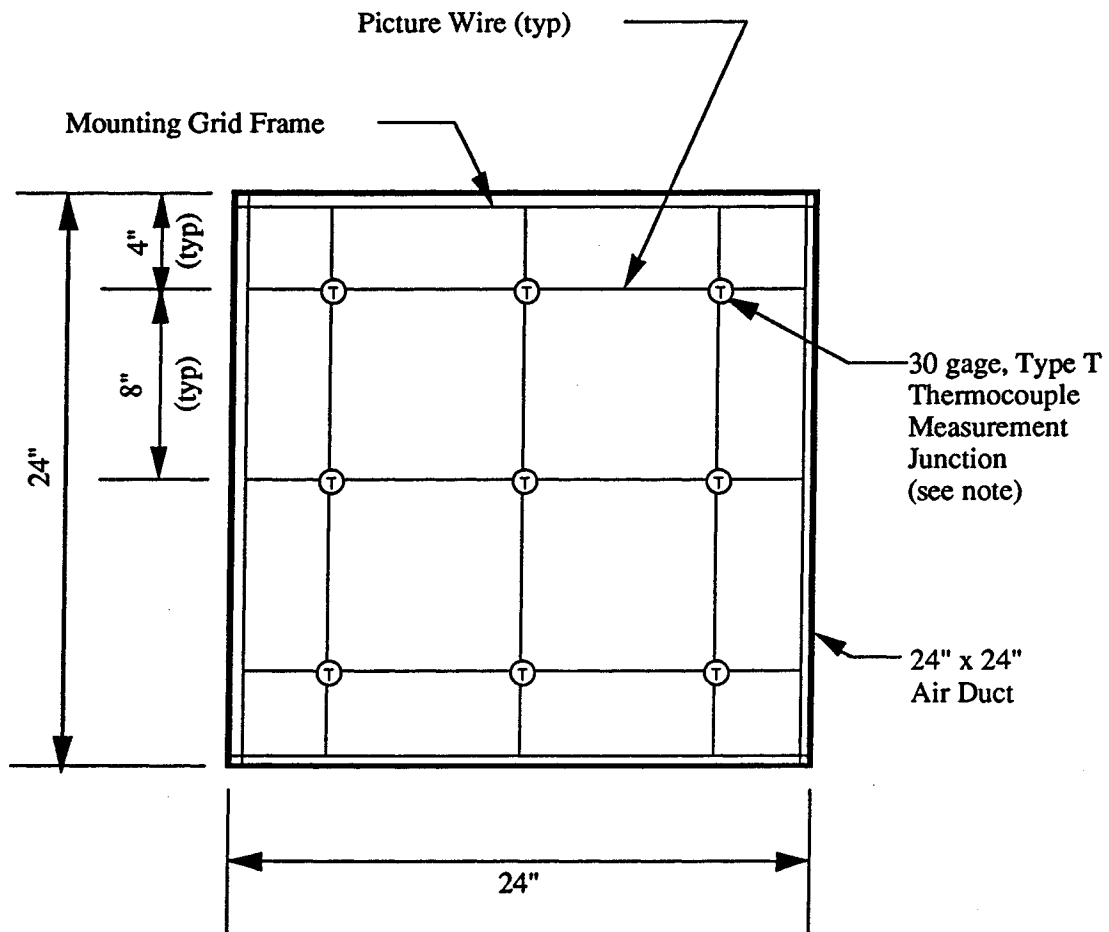


FIGURE 3.16
AIR DUCT TEMPERATURE MEASUREMENT
VIA RTD AVERAGING PROBE



NOTE:

Each thermocouple measurement junction (9 per grid) faces into the incoming air stream and is located at the center of a 8" x 8" cross sectional area.

**FIGURE 3.17
AIR DUCT TEMPERATURE MEASUREMENT
VIA THERMOCOUPLE GRID**

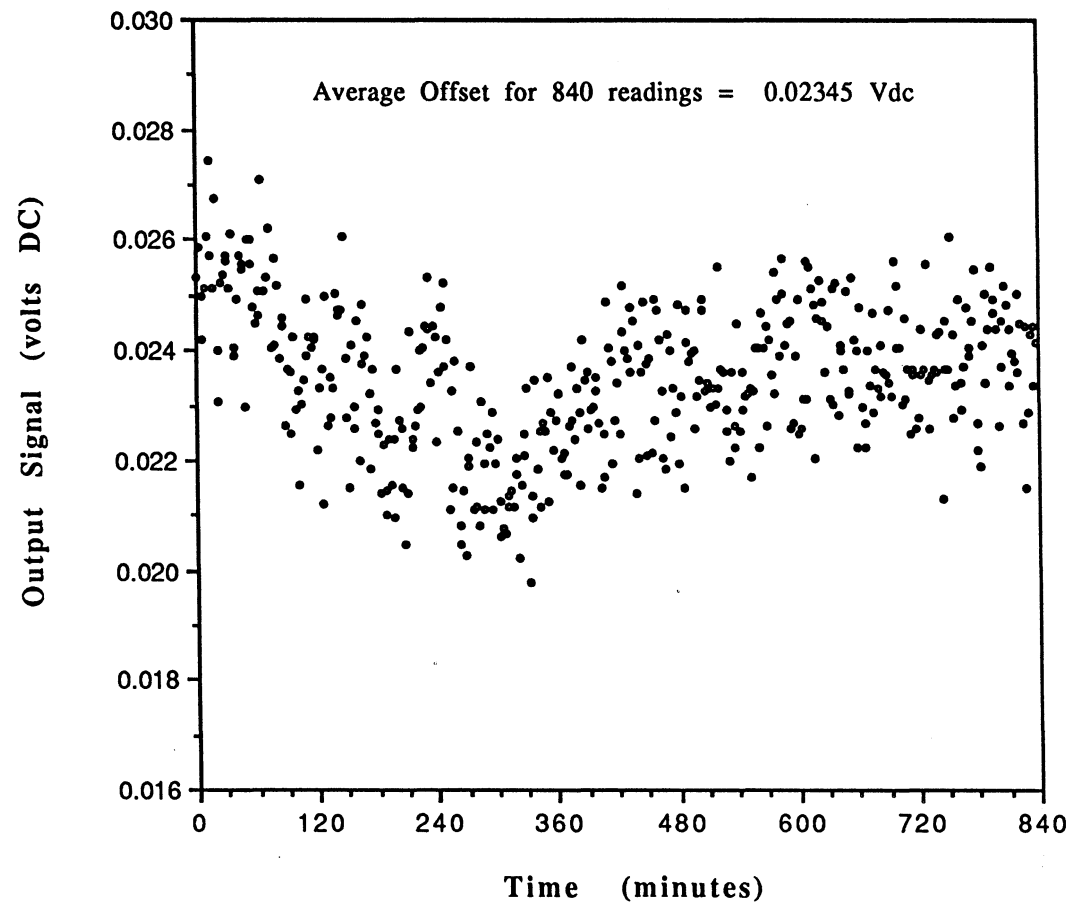


FIGURE 3.18
COIL AIR PRESSURE DROP SENSOR
FIELD DETERMINATION OF VOLTAGE OFFSET

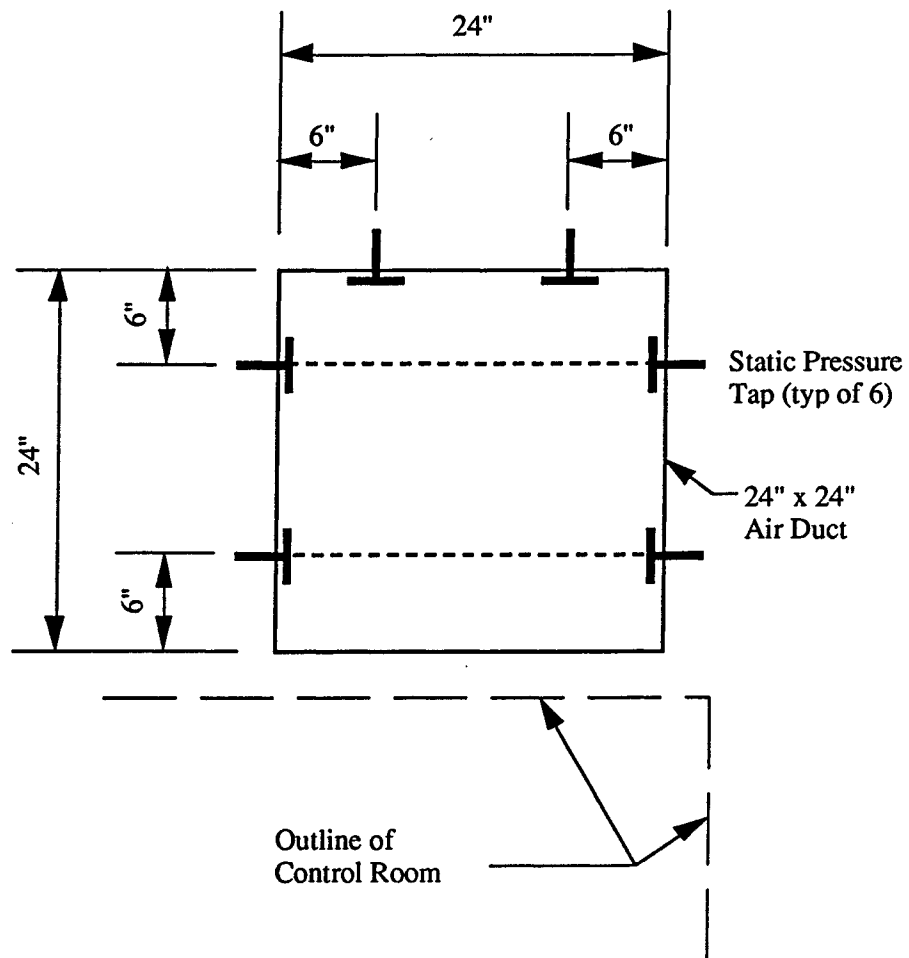


FIGURE 3.19
CONDENSER TEST SECTION
STATIC PRESSURE TAP LOCATIONS

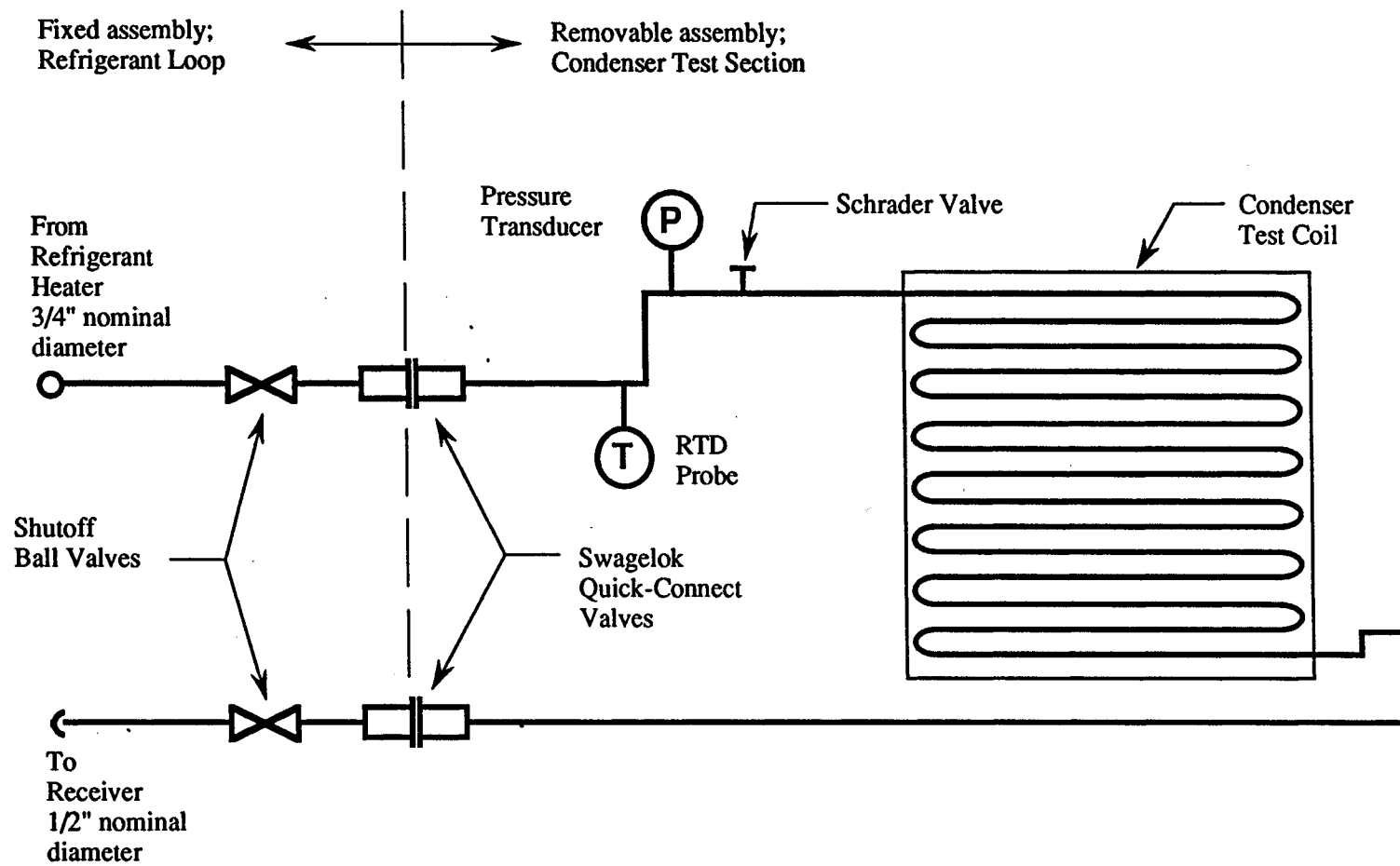


FIGURE 3.20
CONDENSER TEST SECTION
REFRIGERANT PIPING AND INSTRUMENTATION SCHEMATIC

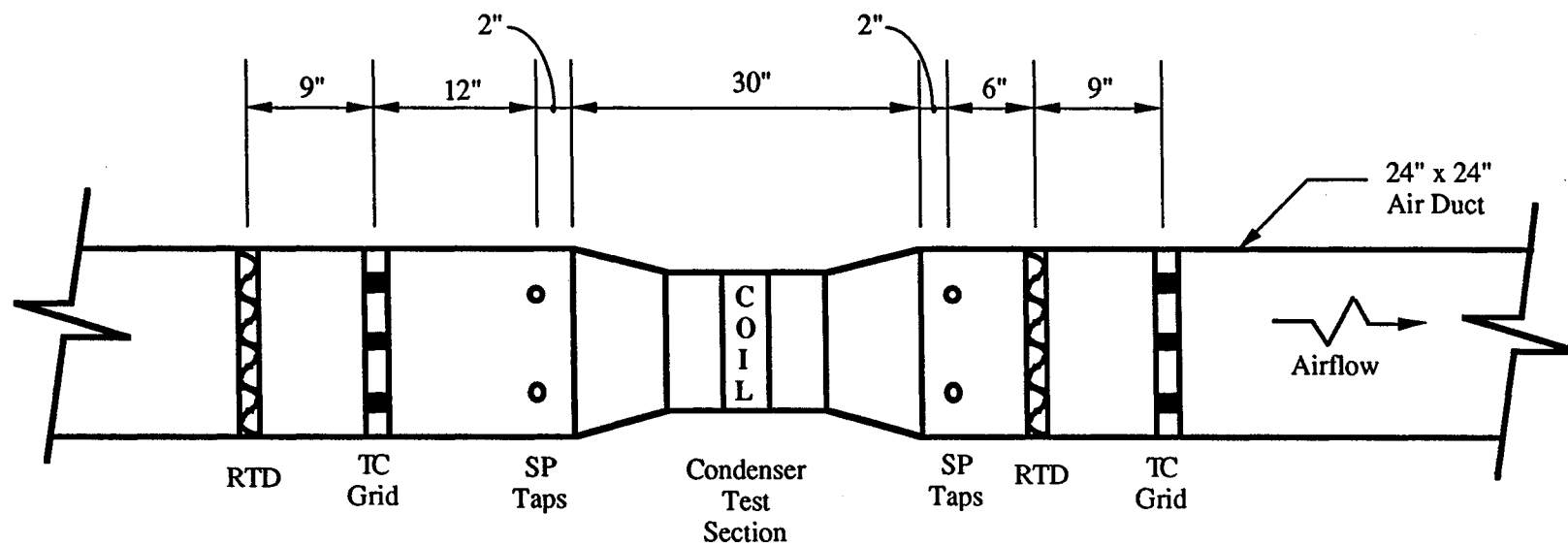


FIGURE 3.21
CONDENSER TEST SECTION
AIR DUCT SYSTEM AND INSTRUMENTATION

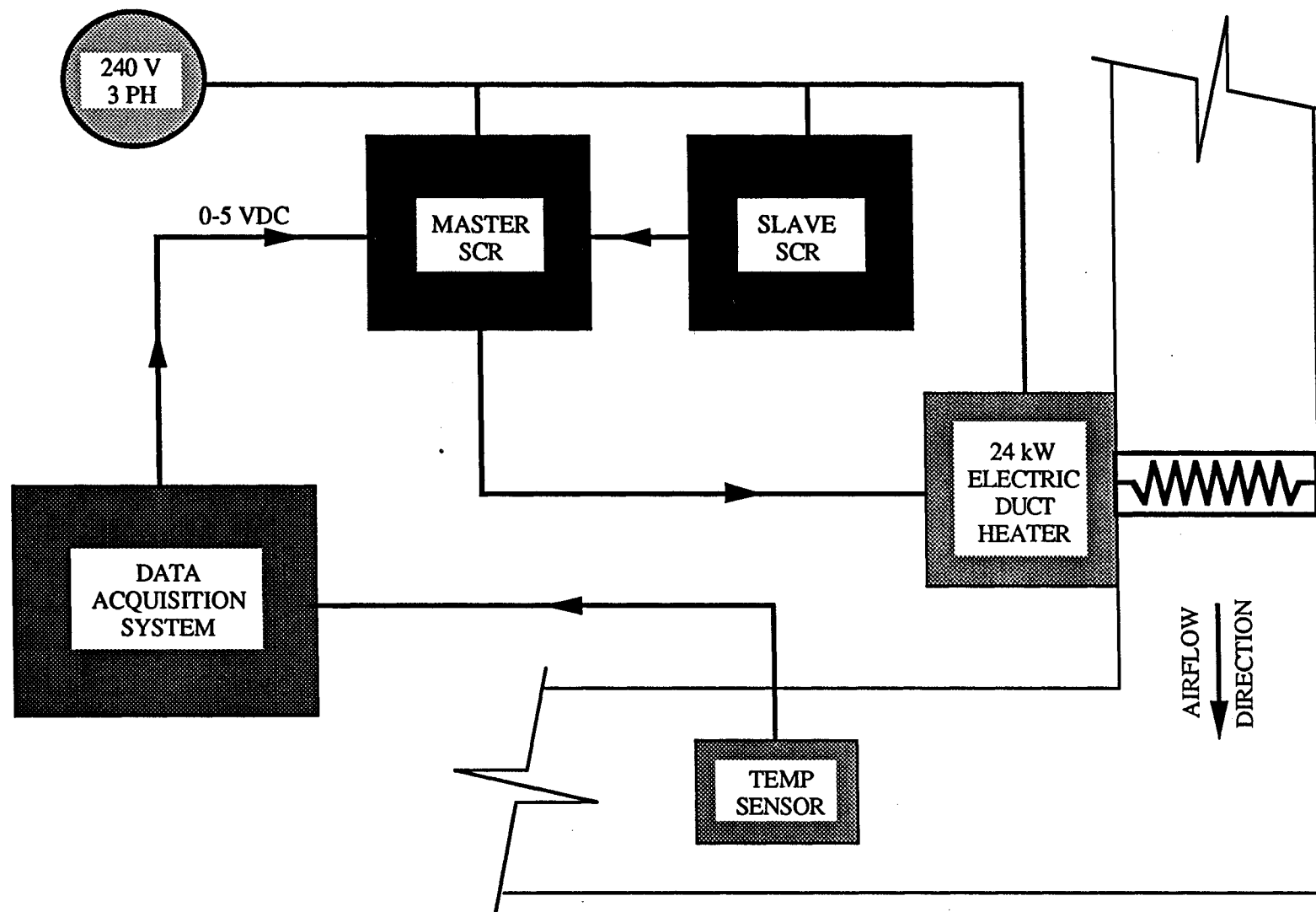


FIGURE 3.22
ELECTRIC DUCT HEAT CONTROL DIAGRAM

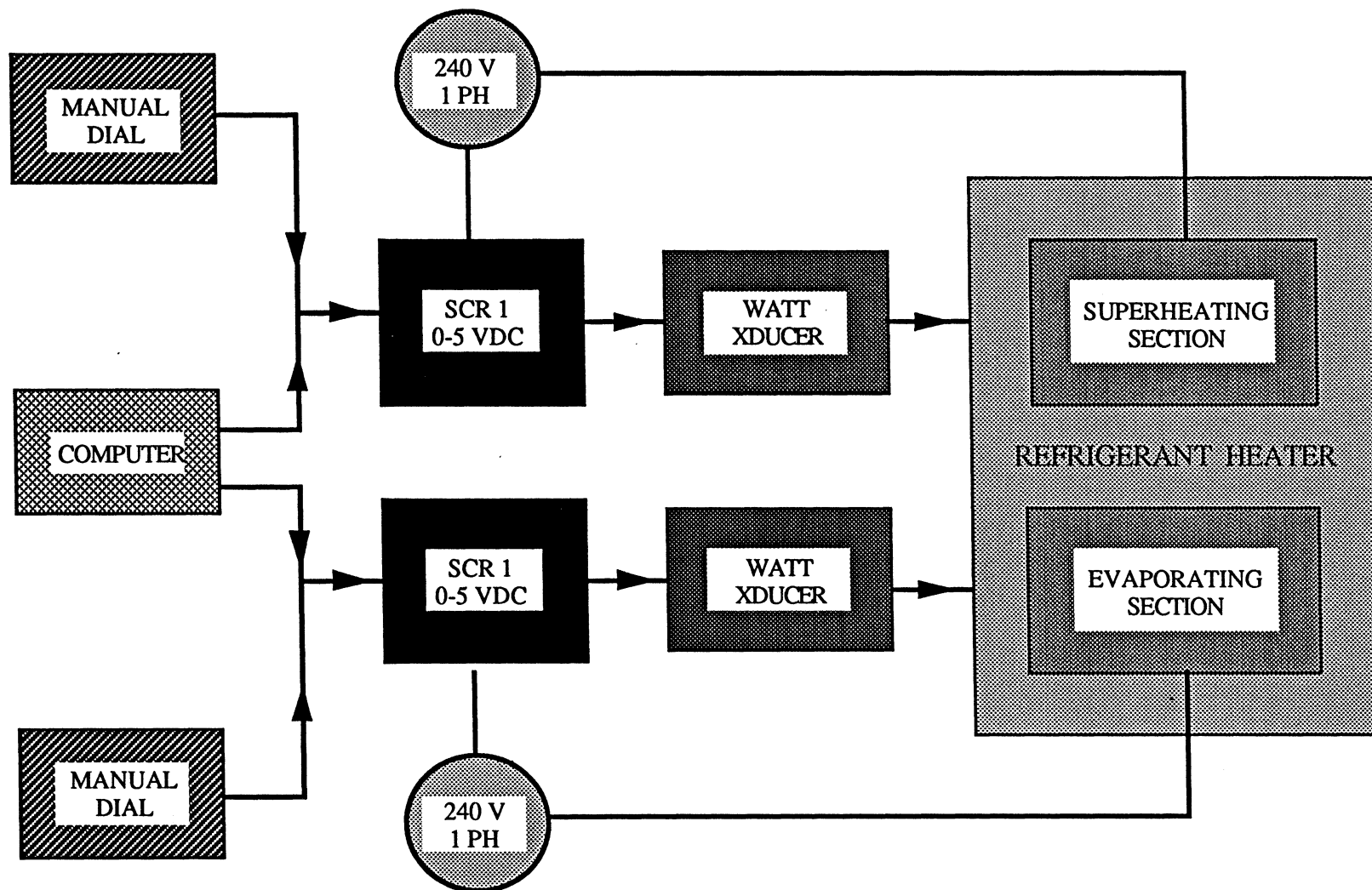


FIGURE 3.23
REFRIGERANT HEATER CONTROL DIAGRAM

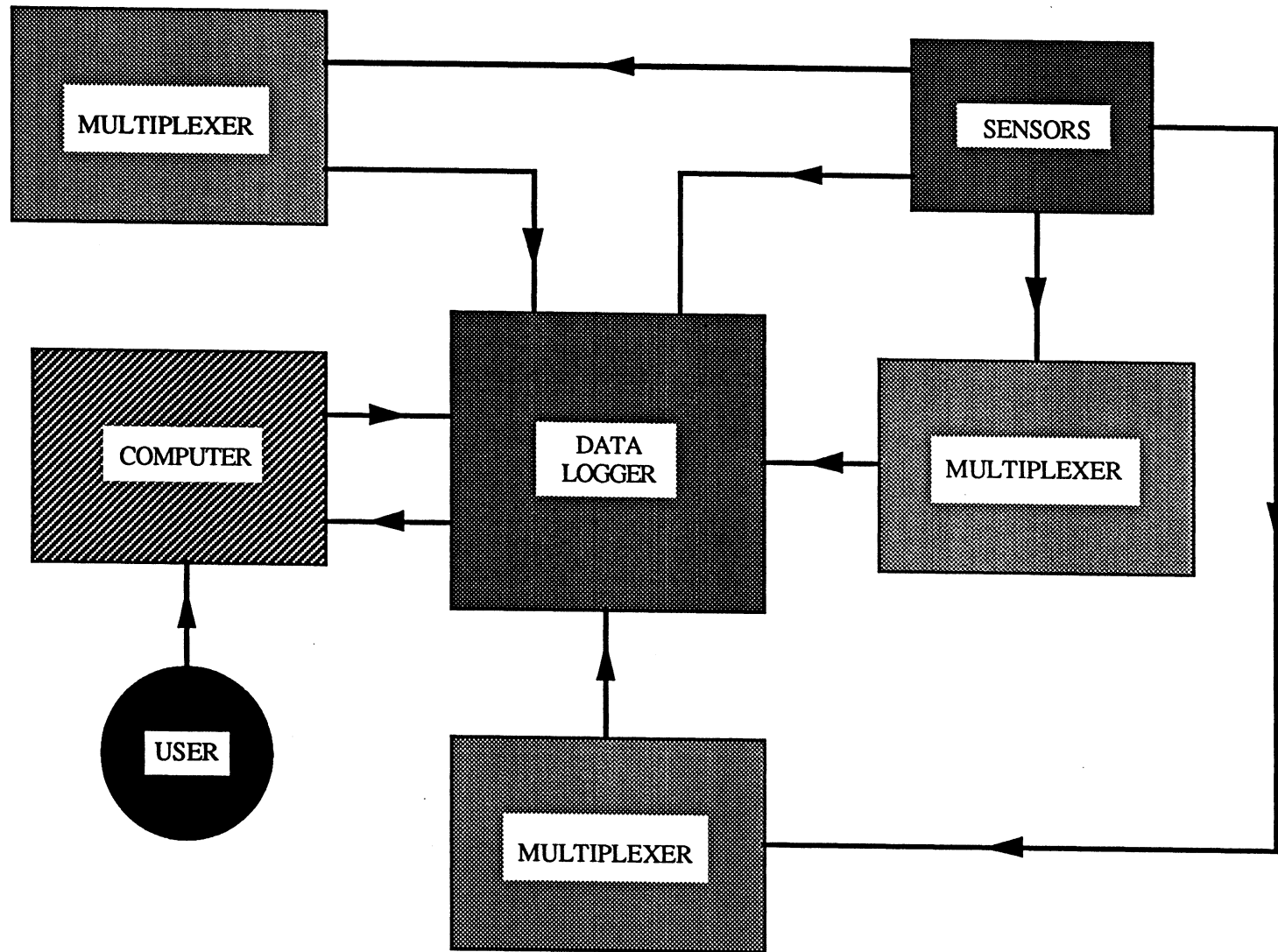


FIGURE 3.24
SCHEMATIC OF DATA ACQUISITION SYSTEM

STEADY STATE DATA COLLECTION MANAGER

REFRIGERANT TEMPERATURES		REFRIGERANT PHASE STATUS					
CONDENSER INLET:	87.900 °F	EVAPORATOR EXIT: 52.8° SUBCOOLED					
CONDENSER EXIT:	88.400 °F	CONDENSER INLET: 46.8° SUBCOOLED					
HEATER INLET:	83.900 °F	CONDENSER EXIT: 47.5° SUBCOOLED					
EVAPORATOR:	83.100 °F	CONDENSER TEMPERATURE CURVES					
CHILLER TANK:	77.100 °F	BAD: 7 MAX: 87.2 MIN: 86.2					
REFRIGERANT PRESSURES							
CONDENSER INLET:	207.410 PSIA						
CONDENSER EXIT:	210.420 PSIA						
HEATER INLET:	211.310 PSIA						
HEATER EXIT:	209.730 PSIA						
RECEIVER EXIT:	210.260 PSIA						
COIL P. DROP:	-3.010 PSI						
FLOWRATES							
REFRIG. VOLUME:	.000 GPM						
REFRIG. MASS:	.000 LB _m /HR						
AIR VOLUME:	50.000 CFM						
AIR MASS:	215.080 LB _m /HR						
REFRIGERANT INPUT WATTAGE		AIR DUCT GRID TEMPERATURES					
SUPERHEATER:	.000 WATTS	ENTERING			EXITING		
EVAPORATOR:	.000 WATTS	AVG: 85.8			AVG: 88.5		
AIR TEMPERATURES		86.1	86.2	86.2	88.8	88.7	88.7
CONDENSER INLET:	86.400 °F	85.7	85.7	85.7	88.5	88.5	88.5
CONDENSER EXIT:	88.500 °F	85.4	85.4	85.5	88.2	88.0	88.2
NOZZLE INLET:	85.200 °F						
AIR PRESSURES		OPTIONS MENU					
COIL P DROP:	-.007 IN HG	SAVE TOGGLE-----[S]					
NOZZLE P DROP:	.002 IN HG	DATALOGGER CFM SET---[D]					
NOZZLE STATIC:	.000 IN HG	TERMINATE TESTING----[T]					
		NEXT OPTIONS PAGE----[N]					
DATA LOGGER REF TEMP: 77.16 °F							
TIME: 2:22 PM POINTS SAVED: 0							
		WARNING STATUS					
		SAFELY OPERATING					



FIGURE 3.25
DATA ACQUISITION SYSTEM SCREEN DISPLAY OPTION
RAW DATA FORMAT

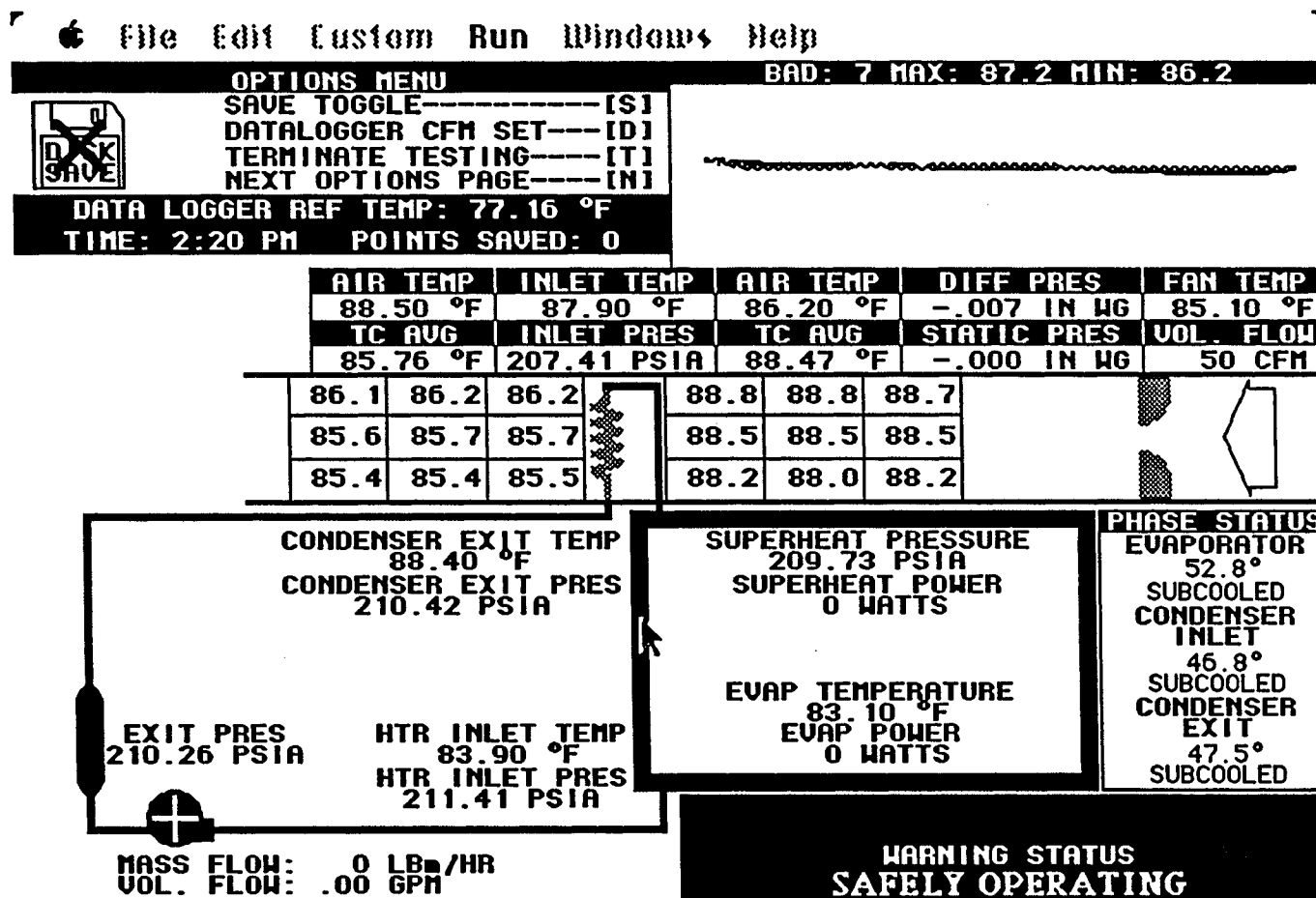


FIGURE 3.26
 DATA ACQUISITION SYSTEM SCREEN DISPLAY OPTION
 SCHEMATIC FORMAT

TABLE 3.1
FLOWMETER FACTORY CALIBRATION DATA
VOLUMETRIC FLOW RATE VERSUS CURRENT OUTPUT

Flow Rate (GPM)	Current Output (mA)	Percent Error (%)
2.00	20.00	0.0
1.59	16.70	+0.11
1.32	14.59	+0.15
1.07	12.47	+0.12
0.793	10.35	+0.09
0.528	8.23	0.0
0.264	6.11	0.0
0.0	4.00	0.0

TABLE 3.2
FLOWMETER FACTORY CALIBRATION DATA
VOLUMETRIC FLOW RATE VERSUS K-FACTOR

Flow Rate (cc/min)	K-Factor (pulses/cc)	Linearity Error (%)
10000	9.901015	-0.37
7000	9.929593	-0.08
3000	9.956405	+0.19
1000	9.974731	+0.37
300	9.964977	+0.27
100	9.960453	+0.23
30	9.943863	+0.06
10	9.849277	-0.89

NOTES FOR TABLE 3.1 AND TABLE 3.2

1. Average K-factor = 9.937873
2. Calibration data generated by Max Machinery, Inc. - 2/26/90.
2. Calibration based on kerosene as working fluid with absolute viscosity of 1.83 centipoise and specific gravity of 0.870 at a temperature of 21 C.
3. 1 cubic centimeter per minute (cc/min) = 2.64×10^{-4} gpm

TABLE 3.3
AIR FLOW CALIBRATION DATA FOR 1-INCH NOZZLE

FAN	***INCLINED MANOMETER RESULTS***			***PRESSURE TRANSDUCER RESULTS***			
SPEEDPOT	SP	DP	AIRFLOW	SP	DP	AIRFLOW	PERCENT
SETTING	(in. wg)	(in. wg)	(cfm)	(in. wg)	(in. wg)	(cfm)	DIFFERENCE
1.0	0.090	0.090	6.2	0.000	0.090	6.2	0.00%
1.5	0.220	0.220	9.8	0.141	0.216	9.7	1.03%
1.9	0.350	0.360	12.6	0.286	0.349	12.4	1.60%
2.0	0.380	0.380	13.0	0.311	0.369	12.8	1.55%
2.2	0.530	0.530	15.3	0.486	0.521	15.2	0.66%
2.5	0.685	0.690	17.5	0.661	0.684	17.5	0.00%
2.8	0.900	0.895	20.0	0.896	0.896	20.0	0.00%
3.0	0.820	0.810	19.0	0.816	0.813	19.1	0.52%
3.2	1.015	1.000	21.2	1.031	1.001	21.2	0.00%
3.5	1.110	1.120	22.4	1.156	1.123	22.4	0.00%
3.8	1.310	1.320	24.4	1.356	1.311	24.3	0.41%
4.0	1.530	1.550	26.4	1.606	1.537	26.3	0.38%
4.2	1.710	1.710	27.7	1.770	1.706	27.7	0.00%
4.5	1.930	1.940	29.6	2.006	1.906	29.3	1.02%

NOTES:

1. Air flow rate computations based on measured inlet air temperature of 72 F.
2. Nozzle calibration test performed 1/19/91.
3. Voltage offsets associated with Setra static and differential pressure transducers are -0.011 Vdc and +0.039 Vdc, respectively. These offsets verified by measurement before and after testing.

TABLE 3.4
AIR FLOW CALIBRATION DATA FOR 3-INCH NOZZLE

FAN	***INCLINED MANOMETER RESULTS****			***PRESSURE TRANSDUCER RESULTS***			
SPEEDPOT	SP	DP	AIRFLOW	SP	DP	AIRFLOW	PERCENT
SETTING	(in. wg)	(in. wg)	(cfm)	(in. wg)	(in. wg)	(cfm)	DIFFERENCE
1.0	0.105	0.100	60.3	0.006	0.097	59.4	1.50%
1.2	0.135	0.130	68.9	0.037	0.127	68.1	1.17%
1.5	0.195	0.190	83.5	0.110	0.186	82.7	0.96%
2.0	0.365	0.355	114.6	0.296	0.347	113.3	1.14%
2.2	0.470	0.465	131.3	0.423	0.459	130.4	0.69%
2.5	0.565	0.550	142.9	0.527	0.551	143.1	0.14%
3.0	0.795	0.780	170.4	0.792	0.787	171.2	0.47%
3.5	1.030	1.010	194.1	1.051	1.014	194.5	0.21%
3.8	0.950	0.935	186.7	0.968	0.935	186.7	0.00%
4.0	1.440	1.420	230.3	1.511	1.423	230.5	0.09%
4.5	1.790	1.760	256.4	1.856	1.756	256.0	0.16%
4.8	1.990	1.950	269.8	2.031	1.921	267.8	0.74%

NOTES:

1. Air flow rate computations based on measured inlet air temperature of 72 F.
2. Nozzle calibration test performed 1/19/91.
3. Voltage offsets associated with Setra static and differential pressure transducers are -0.011 Vdc and +0.039 Vdc, respectively. These offsets verified by measurement before and after testing.

TABLE 3.5
AIR FLOW CALIBRATION DATA FOR 4-INCH NOZZLE

FAN	***INCLINED MANOMETER RESULTS***			***PRESSURE TRANSDUCER RESULTS***			
SPEEDPOT	SP	DP	AIRFLOW	SP	DP	AIRFLOW	PERCENT
SETTING	(in. wg)	(in. wg)	(cfm)	(in. wg)	(in. wg)	(cfm)	DIFFERENCE
0.5	0.030	0.030	59.0	0.000	0.030	59.0	0.00%
1.0	0.090	0.085	100.1	0.004	0.081	97.7	2.43%
1.5	0.190	0.180	146.3	0.098	0.168	141.3	3.48%
1.8	0.280	0.270	179.6	0.206	0.258	175.6	2.25%
2.0	0.330	0.320	195.7	0.258	0.302	190.1	2.90%
2.2	0.440	0.425	225.8	0.399	0.419	224.2	0.71%
2.5	0.510	0.495	243.9	0.472	0.488	242.1	0.74%
2.8	0.600	0.580	264.1	0.574	0.571	262.1	0.76%
3.0	0.680	0.660	281.9	0.666	0.657	281.2	0.25%
3.2	0.750	0.725	295.5	0.751	0.721	294.7	0.27%
3.5	0.860	0.840	318.2	0.873	0.843	318.8	0.19%
3.6	0.970	0.940	336.7	1.002	0.937	336.1	0.18%
3.8	1.105	1.070	359.3	1.146	1.074	359.9	0.17%
4.0	1.285	1.250	388.4	1.333	1.244	387.5	0.23%
4.2	1.420	1.380	408.1	1.486	1.371	406.7	0.34%
4.3	1.510	1.460	419.8	1.578	1.448	418.0	0.43%
4.5	1.680	1.630	443.6	1.756	1.621	442.3	0.29%
4.8	1.775	1.720	455.6	1.866	1.706	453.7	0.42%
5.0	1.980	1.900	478.8	2.031	1.866	474.5	0.90%
5.2	2.050	2.000	491.3	2.106	1.931	482.7	1.77%

NOTES:

1. Air flow rate computations based on measured inlet air temperature of 80 F.
2. Nozzle calibration test performed 1/21/91.
3. Voltage offsets associated with Setra static and differential pressure transducers are -0.011 Vdc and +0.039 Vdc, respectively. These offsets verified by measurement before and after testing.

TABLE 3.6
AIR FLOW CALIBRATION DATA FOR 7-INCH NOZZLE

FAN	***INCLINED MANOMETER RESULTS***			***PRESSURE TRANSDUCER RESULTS***			
SPEEDPOT	SP	DP	AIRFLOW	SP	DP	AIRFLOW	PERCENT
SETTING	(in. wg)	(in. wg)	(cfm)	(in. wg)	(in. wg)	(cfm)	DIFFERENCE
1.0	0.040	0.030	181.2	0.001	0.025	165.2	9.24%
1.5	0.085	0.065	268.1	0.001	0.058	253.1	5.76%
2.0	0.160	0.130	380.5	0.076	0.117	360.8	5.31%
2.5	0.255	0.200	472.9	0.188	0.196	468.2	1.00%
2.8	0.320	0.260	539.8	0.260	0.249	528.2	2.17%
3.0	0.400	0.325	604.0	0.358	0.319	598.4	0.93%
3.5	0.530	0.440	703.4	0.508	0.430	695.4	1.14%
3.8	0.600	0.490	742.5	0.585	0.483	737.2	0.72%
4.0	0.710	0.575	804.7	0.718	0.591	815.8	1.37%
4.5	0.905	0.755	922.5	0.941	0.753	921.3	0.13%
4.8	1.025	0.860	984.8	1.084	0.857	983.0	0.18%
5.0	1.220	1.035	1080.5	1.311	1.041	1083.5	0.28%
5.5	1.265	1.080	1103.8	1.376	1.087	1107.2	0.31%
5.8	1.460	1.240	1182.7	1.571	1.241	1183.0	0.03%
6.0	1.740	1.480	1292.0	1.856	1.471	1287.8	0.33%
6.5	1.975	1.670	1372.1	2.086	1.641	1360.0	0.89%
6.8	2.040	1.770	1412.6	2.211	1.756	1406.7	0.42%
7.0	2.060	1.940	1478.9	2.381	1.896	1461.4	1.19%

NOTES:

1. Air flow rate computations based on measured inlet air temperature of 75 F.
2. Nozzle calibration test performed 1/21/91.
3. Voltage offsets associated with Setra static and differential pressure transducers are -0.011 Vdc and +0.039 Vdc, respectively. These offsets verified by measurement before and after testing.

TABLE 3.7
AIR FLOW CALIBRATION DATA FOR 10-INCH NOZZLE

FAN	***INCLINED MANOMETER RESULTS***			***PRESSURE TRANSDUCER RESULTS***			
SPEEDPOT	SP	DP	AIRFLOW	SP	DP	AIRFLOW	PERCENT
SETTING	(in. wg)	(in. wg)	(cfm)	(in. wg)	(in. wg)	(cfm)	DIFFERENCE
3.0	0.215	0.110	720.3	0.150	0.104	700.2	2.83%
3.5	0.315	0.160	870.0	0.278	0.159	867.3	0.31%
4.0	0.395	0.210	997.7	0.372	0.210	997.7	0.00%
4.5	0.525	0.310	1213.7	0.540	0.309	1211.7	0.16%
5.0	0.605	0.360	1308.4	0.640	0.359	1306.5	0.15%
5.5	0.690	0.435	1438.9	0.734	0.434	1437.1	0.13%
6.0	0.895	0.550	1618.6	0.950	0.555	1625.8	0.44%
6.5	1.120	0.670	1786.8	1.156	0.665	1780.0	0.38%
7.0	1.335	0.800	1952.6	1.366	0.794	1945.2	0.38%
7.5	1.510	0.935	2111.2	1.578	0.926	2100.7	0.50%
8.0	1.730	1.060	2247.6	1.811	1.053	2240.0	0.34%
8.5	1.950	1.200	2391.2	2.026	1.188	2379.0	0.51%

NOTES:

1. Air flow rate computations based on measured inlet air temperature of 80 F.
2. Nozzle calibration test performed 7/13/91.
3. Voltage offsets associated with Setra static and differential pressure transducers are -0.011 Vdc and +0.045 Vdc, respectively. These offsets verified by measurement before and after testing.

TABLE 3.8
AIR FLOW CALIBRATION DATA FOR 10-INCH AND 4-INCH NOZZLES

FAN	***INCLINED MANOMETER RESULTS****			***PRESSURE TRANSDUCER RESULTS***			
SPEEDPOT	SP	DP	AIRFLOW	SP	DP	AIRFLOW	PERCENT
SETTING	(in. wg)	(in. wg)	(cfm)	(in. wg)	(in. wg)	(cfm)	DIFFERENCE
3.0	0.190	0.085	1257.8	0.143	0.085	1257.9	0.01%
3.5	0.255	0.115	1465.5	0.222	0.114	1459.1	0.44%
4.0	0.355	0.165	1758.6	0.333	0.164	1753.3	0.30%
4.5	0.450	0.220	2033.3	0.432	0.216	2014.6	0.92%
5.0	0.560	0.280	2296.1	0.567	0.277	2283.6	0.55%
5.5	0.650	0.340	2532.0	0.666	0.334	2509.3	0.90%
6.0	0.810	0.430	2849.5	0.793	0.427	2839.5	0.35%
6.5	0.980	0.505	3089.2	1.040	0.498	3067.3	0.71%
7.0	1.135	0.600	3368.5	1.191	0.606	3385.2	0.49%
7.5	1.345	0.715	3678.3	1.398	0.709	3662.5	0.43%
8.0	1.450	0.855	4024.0	1.581	0.850	4011.2	0.32%
8.5	1.760	0.960	4263.3	1.866	0.958	4258.2	0.12%

NOTES:

1. Air flow rate computations based on measured inlet air temperature of 80 F.
2. Nozzle calibration test performed 7/13/91.
3. Voltage offsets associated with Setra static and differential pressure transducers are -0.011 Vdc and +0.045 Vdc, respectively. These offsets verified by measurement before and after testing.

TABLE 3.9
EXISTING WATER COIL GEOMETRICAL DATA

MANUFACTURER	Bohn Heat Transfer - Danville, Illinois
MODEL NUMBER	5WH 6A - 15T x 24
NOMINAL TUBE DIAMETER (in)	5/8
TUBE MATERIAL	Copper
NUMBER OF TUBES IN FACE	15
FINNED HEIGHT (in)	23.91
FINNED LENGTH (in)	24
FIN MATERIAL	Aluminum
COIL FACE AREA (sq ft)	3.98
NUMBER OF TUBE ROWS	6
FINS PER INCH	8
NUMBER OF CIRCUITS	1
NUMBER OF TUBE PASSES	12
HEADER CONNECTION SIZE (in)	1 - 5/8
CONNECTION LOCATIONS	Same end connections

* Refer to Table 3.10 and Bohn Technical Bulletin 3112 for coil performance data.

TABLE 3.10
EXISTING WATER COIL PERFORMANCE DATA (COOLING DUTY)

INPUT DATA:					
TITLE/ITEM	1	2	3	4	5
COIL GEOMETRY		(See Table 3.9)			
STANDARD AIR FLOW (scfm)	797	996	1195	1395	1594
ENTERING AIR DRY BULB (F)	70.0	70.0	70.0	70.0	70.0
ENTERING AIR WET BULB (F)	63.0	63.0	63.0	63.0	63.0
ENTERING WATER TEMP (F)	15.0	15.0	15.0	15.0	15.0
LEAVING WATER TEMP (F)	25.0	25.0	25.0	25.0	25.0
BAROMETRIC PRESSURE (in Hg)	29.921	29.921	29.921	29.921	29.921
OUTPUT DATA FOR 75/25 WATER/ETHYLENE GLYCOL SOLUTION					
TOTAL CAPACITY (BTUH)	69,900	84,900	98,800	112,200	124,700
SENSIBLE CAPACITY (BTUH)	39,600	47,800	55,400	62,500	69,100
LEAVING AIR DRY BULB (F)	24.6	26.1	27.6	29.0	30.4
LEAVING AIR WET BULB (F)	24.6	26.1	27.6	29.0	30.4
AIR PRESSURE DROP (in. wg)	0.17	0.24	0.31	0.39	0.47
FACE AIR VELOCITY (fpm)	200.0	250.0	299.9	350.1	400.1
WATER PRESSURE DROP (ft. H ₂ O)	5.1	7.2	9.4	11.7	14.0
WATER VELOCITY (fps)	2.32	2.81	3.26	3.69	4.09
WATER FLOW RATE (gpm)	14.8	18.1	21.0	23.9	26.6
OUTPUT DATA FOR 70/30 WATER/ETHYLENE GLYCOL SOLUTION					
TOTAL CAPACITY (BTUH)	69,000	83,800	97,900	111,000	123,300
SENSIBLE CAPACITY (BTUH)	39,100	47,200	54,800	61,700	68,200
LEAVING AIR DRY BULB (F)	25.2	26.7	28.1	29.5	30.9
LEAVING AIR WET BULB (F)	25.2	26.7	28.1	29.5	30.9
AIR PRESSURE DROP (in. wg)	0.17	0.24	0.31	0.39	0.47
FACE AIR VELOCITY (fpm)	200.0	250.0	299.9	350.1	400.1
WATER PRESSURE DROP (ft. H ₂ O)	5.3	7.5	9.8	12.1	14.5
WATER VELOCITY (fps)	2.33	2.82	3.28	3.71	4.11
WATER FLOW RATE (gpm)	14.9	18.1	21.2	24.0	26.7

* Results based exclusively on Bohn Heat Transfer, Water Coil Rating Report, V6.04. Data obtained 4/11/91.

CHAPTER 4

GRAPHICAL EVALUATION OF OUTSIDE HEAT TRANSFER COEFFICIENTS

In general, for a two fluid compact heat exchanger or shell and tube heat exchanger, the experimentally determined overall thermal resistance is related to the individual heat transfer resistances according to Equation (4.1).

$$\frac{1}{U_o A_o} = \frac{1}{U_i A_i} = \frac{1}{(\eta_s h A)_i} + R_{f,i} + R_w + R_{f,o} + \frac{1}{(\eta_s h A)_o} \quad (4.1)$$

Figure 4.1 depicts these individual resistances schematically in a thermal circuit as well as the two arbitrarily designated temperatures. The temperature gradient is the driving potential for the heat transfer rate. For a compact heat exchanger (e.g., an air-cooled condenser with plain finned tubes), the subscripts "i" and "o" in Equation (4.1) denote the tube inside and outside surfaces, respectively. Similarly, for a shell and tube heat exchanger (e.g., a water-cooled oil cooler with oil flowing inside the tubes and water on the shell-side), "i" and "o" represent the shell-side and tube-side, respectively, and are sometimes replaced by the subscripts "s" and "t". The three (3) subscripted R terms designate the fouling resistance on the inside or tube-side surface ($R_{f,i}$), the resistance of the tube wall (R_w), and the fouling resistance on the outside or shell-side surface ($R_{f,o}$), respectively. Finally, the two terms $(1/\eta_s h A)_i$ and $1/(\eta_s h A)_o$ correspond to the film resistance on each fluid side, including the overall surface efficiency parameter η_s to account for the presence of fins.

In the analysis of evaporating or condensing types of heat exchangers, the separation of individual heat transfer resistances from the overall resistance is extremely important in obtaining accurate heat transfer correlations that may be suitable for design purposes. Accurate design data for a heat transfer surface may be determined from Equation (4.1) if one of the thermal resistances is determined to be the "controlling resistance", i.e., its contribution to the total resistance is significantly higher or arbitrarily at least 80 percent higher than that of all other resistances combined. In addition, if all the thermal resistances are of the same order of magnitude and their values are known accurately, then the above equation may still be the optimum alternative in order to determine an

unknown resistance. However, in reality, most of the individual heat transfer resistances are either not known accurately or not known at all, so that Equation (4.1) may not be used directly to determine the unknown resistance or heat transfer coefficient.

The Wilson Plot technique is designed to be a useful means for determining the individual heat transfer coefficients in shell and tube and compact heat exchangers, in particular evaporators and condensers. In this Chapter, a general review of the classical Wilson Plot technique is presented as well as some of its limitations. In addition, various modifications to the original method are mentioned. Finally, the direct applicability of the techniques to the analysis of fin-and-tube heat exchangers, specifically air-cooled condensers, is conveyed.

4.1. Classical Wilson Plot

The original Wilson Plot technique was introduced by E.E. Wilson in 1915 [131], based on his experiments with a shell and tube condenser with unfinned circular tubes and with condensing steam on the shell-side and water on the tube-side. In general, Wilson neglected fouling resistances and related the total thermal resistance of the shell and tube heat exchanger to the individual resistances in the form written below:

$$\frac{1}{UA} = \frac{1}{h_t A_t} + \frac{1}{h_s A_s} + R_w \quad (4.1.1)$$

where the terms $(1/h_t A_t)$ and $(1/h_s A_s)$ denote the tube-side and shell-side resistances, respectively, and R_w represents the wall resistance.

The primary aims of Wilson's experiments were to determine the turbulent flow, single phase heat transfer coefficient for the water-side and the effects of water temperature and velocity on the overall heat transfer coefficient. From the results of his examination of the literature available at the time, Wilson inferred that the rate of heat transfer is most often described as some exponential function containing the circulating water velocity as a variable. Wilson modified this relationship by correcting for temperature-dependent fluid property variations and tube diameter and expressing the tube-side resistance in terms of a reduced

velocity variable, V_r . In terms of this reduced velocity parameter, the resistance balance on the heat exchanger was expressed as a function containing two unknowns, C_1 and C_2 , as indicated by Equation (4.1.2).

$$\frac{1}{U_o} = \frac{1}{C_1 V_r^{0.82}} + C_2 \quad (4.1.2)$$

where: C_1 = arbitrary constant

C_2 = sum of shell-side and wall resistances

$1/U_o$ = overall thermal resistance

In discussing Wilson's paper, Buckingham [131] showed that in effect Wilson's reduced velocity variable was the dimensionless parameter $\rho V D / \mu$. Buckingham called this quantity the "turbulence variable", known today as the Reynolds number.

Wilson proposed that by maintaining the shell-side resistance constant and varying the test fluid velocity during experiments, the above equation takes the straight line form expressed mathematically below and graphically in Figure 4.1.

$$y = mx + b \quad (4.1.3)$$

where: m = slope = C_1

x = independent variable = $1/V_r^{0.82}$

b = y-intercept = C_2

y = dependent variable = $1/U_o$

By subsequent extrapolation of the linear plot of $1/U_o$ versus $1/V_r^{0.82}$ to infinite fluid velocity, as shown in Figure 4.2, Wilson further interpreted that the tube-side resistance eventually becomes negligible, and the unknown shell-side resistance, including the wall resistance, may be easily determined by evaluating the y-intercept.

In Equation (4.1.2), Wilson utilized a trial and error procedure to determine the exponent of 0.82 on the velocity function such that his resulting plot provided the best linear fit to the given set of data points [64].

It should be noted that Wilson's shell-side resistance was based on condensing steam and that since the heat flux was not the same for all data points, then the shell-side resistance was not constant as presented in the original derivation.

4.2. Limitations of Classical Wilson Plot

Since first being introduced over 75 years ago, engineering researchers have continually employed the original Wilson Plot method or similar forms of the technique. Jameson [57] utilized the procedure to determine the effects of tube spacing on the overall heat transfer coefficient for air flowing across a staggered bank of helically finned tubes. Katz and Geist [58] implemented the technique to obtain condensing coefficients for a single row of finned tubes. Knudsen and Katz [65] incorporated the technique to develop tube-side and shell-side heat transfer correlations for concentric pipe heat exchangers. Williams and Katz [130] used the procedure to evaluate tube and shell side heat transfer coefficient correlations for shell and tube heat exchangers. Sheperd [102] employed the method in investigating the air-side performance of single-row, plain fin and tube coils. Rich [86] used the technique during his experiments to determine the effect of fin spacing on the overall heat transfer coefficients of multi-row, smooth plate and fin heat exchangers. Carnavos [26] employed the Wilson Plot techniques to develop correlating equations for the heat transfer coefficient and friction factor of internally finned tubes in turbulent flow. More recently, Holtzapfel, Allen and Lin [53] utilized the method in analyzing the heat transfer characteristics of spined pipe in crossflow. In addition, Fujii, et. al. [42] used the Wilson Plot procedure to evaluate the water-side resistance in extended surface compact heat exchangers with trapezoidal and perforated fins.

Despite all the documented usage of the procedure, there are several restrictions associated with the classical Wilson Plot method. These limitations are summarized in the eight paragraphs presented on the following pages. Items 1 thru 6 were basically outlined by Shah [98]. The remaining two items, 7 and 8, were not located in previously published literature, but are felt to be important.

1. The original technique is idealized by assuming the Reynold's number exponent in Equation (4.1.1) to be known as 0.82, prior to collecting and analyzing experimental data. For most heat exchanger surfaces, the exponent is unknown. For example, Shah [94] indicates that the value of the exponent may vary from 0.78 at $Pr=0.7$ to 0.90 at $Pr=100$ based on a fixed $Re=5 \times 10^4$ and utilizing the "best available" single-phase heat transfer coefficient correlations by Gnielinski [46] or Petukhov-Popov [82] for fully developed turbulent flow through a smooth circular, constant area tube. Therefore, in order to determine an accurate value of the Reynolds number exponent, a parameter estimation study may be required and the value of the exponent that provides the least error is the most appropriate one for the particular surface geometry under consideration.
2. The fluid flow rate on the side having the unknown heat transfer coefficient of interest and its log-mean average temperature must be maintained constant during experimental testing so that its thermal resistance remains constant. The suggested method for determination of the appropriate log-mean average temperature for a two-fluid compact heat exchanger is presented by Shah [97].
3. By considering only the circulating fluid velocity as the independent flow variable, fluid property variations are not taken into account on the fluid side having the unknown heat transfer coefficient. Figure 4.3 represents a modified Wilson Plot in which fluid property variations are included in the independent flow variable on the abscissa. The formulation of this variable is considered in more detail later in this Chapter.

4. All empirical data must be for a known range of Reynolds number and for one flow region only (e.g., fully developed turbulent flow) on the fluid side having the known heat transfer coefficient, or, alternatively, the Nusselt number correlation must be explicitly expressed by an equation with two unknowns. Figure 4.4 represents a typical Wilson Plot that results when the turbulent-to-laminar transition region is encountered during experimental testing [97]. Rather than a continuous linear relationship at low Reynolds numbers, a distinct change in the slope of the plot or a "knee" appears.
5. The presence of fins on either one or both fluid sides is not considered in the original analysis. The influence of fin efficiency on the individual thermal resistance(s) must be incorporated for extended surface heat exchangers.
6. Fouling on both sides of the heat exchanger surface must be kept constant or otherwise, nonfouling fluids must be used in order that the y-intercept is constant.
7. The original Wilson Plot approach was designed based on single-phase fluids on both sides of the heat exchanger. It would be convenient to extend the approach to include two-phase fluids since these exist in actual heat transfer equipment, such as in air-cooled condensers with refrigerant flow inside of the tubes.
8. Experimentally, due primarily to equipment limitations, it is impractical to attain the condition of infinite test fluid velocity such that the inverse of the velocity or Reynolds number terms go to zero on the Wilson Plot. Thus, in the idealized process, it is assumed that for a fixed Reynolds number or velocity term exponent, the plot may be linearly extrapolated outside the range of the experimentally known Reynolds number, in order to obtain an accurate value of the unknown heat transfer resistance.

Various modifications have been made to the original Wilson Plot technique in order to account for some of the aforementioned limitations. In particular, four research studies have proposed modifications, and these are reviewed in the next Section.

4.3. Modified Wilson Plot Techniques

As previously stated, four studies have documented so-called "modified" Wilson Plot methods, i.e., Young and Wall [135], Briggs and Young [24], Khartabil, Christensen and Richards [64], and Shah [98]. Each of these studies, summarized in this Section, have at least two characteristics in common. First, their derivations are based on the existence of single phase fluids on both sides of the heat exchanger. Second, their experimental analysis was performed with shell and tube heat exchangers. However, the individual Wilson Plot procedures discussed herein may also be extended to extended surface, compact heat exchangers as well as consideration of two phase fluids on one or both sides of the heat exchanger by application of the proper correlations.

In contrast to the generalized Equation (4.1.1) utilized by Wilson, the more specific expression for a shell and tube heat exchanger or a two fluid, compact heat exchanger that relates the experimentally determined overall thermal resistance to the individual heat transfer resistances is given by Equation (4.1) identified previously.

Since the original Wilson Plot procedure and the modified methods are based strictly on the use of single phase fluids on both sides of the heat exchanger, the single phase heat transfer coefficient(s) in Equation (4.1) may be replaced by one of the suitable Nusselt number expressions applicable to fully developed turbulent flow in a smooth circular tube. Some single phase, Nusselt number correlations are given in Section 2.2. Although any form may be implemented, one of the most common forms introduced is the following:

$$Nu = C_o Re^a Pr^b \left(\frac{\mu_w}{\mu_m} \right)^n \quad (4.3.1)$$

In Equation (4.3.1), the terms C_o , a , and b are geometrically dependent, empirical constants. In the majority of heat transfer literature, the Prandtl number exponent is taken as 0.4, although it is reasonable to include this exponent as a variable in a nonlinear regression analysis of experimental data.

The absolute viscosity ratio $(\mu_w/\mu_m)^n$ in Equation (4.3.1) takes into account the temperature dependent, fluid property effects for liquids and is often omitted from the equation since it is close to unity for most experimental data, particularly for liquids flowing within small diameter tubes. Similarly, for gases, the ratio should be replaced by the absolute temperature ratio function $(T_w/T_m)^n$. The value of exponent n may vary dependent on the flow situation and the fluid phase. For example, for laminar flow of liquids and gases, the values of n are -0.14 and 0, respectively. Other typical values may be found in Shah [97] or Kays and London [59]. The effect of variable properties of gases on the Nusselt number is also briefly discussed in Section 3.4.7. In the equations that follow, the property ratios are not included although it should be emphasized that the appropriate liquid or gas correction must be incorporated in actual practice, if warranted by the situation.

By definition of the dimensionless Nusselt number, the heat transfer coefficient may be determined from Equation (4.3.1) as follows:

$$Nu = hD_h/k = C_o Re^a Pr^{0.4} \quad (4.3.2)$$

$$h = C_o Re^a Pr^{0.4} (k/D_h) \quad (4.3.3)$$

where D_h is equal to the hydraulic diameter and k is the thermal conductivity of the fluid. The Reynolds number is based on the hydraulic diameter and the mean fluid velocity through the minimum free flow area. Use of the hydraulic diameter in lieu of the actual tube diameter allows convenient adaptation of the Nusselt expression to oval, rectangular and other non-circular tube geometries.

In their analysis of shell and tube heat exchangers, Young and Wall [135] proposed the use of the form of Equation (4.3.2) for both the shell-side and tube-side resistances. In addition, they assumed the Reynolds number exponents in each of these resistances to be known as 0.80.

With the foregoing substitutions and neglecting fouling resistances, the general overall thermal resistance equation becomes as shown below with two unknown constants, C_t and C_s .

$$\frac{1}{UA} - R_w = \frac{1}{C_t [Re^{0.8} Pr^{0.4} (k/D_h) A]_t} + \frac{1}{C_s [Re^{0.8} Pr^{0.4} (k/D_h) A]_s} \quad (4.3.4)$$

This equation possesses the form of a straight line with slope $1/C_t$ and with the y-intercept, which is the unknown shell-side resistance, equal to the entire second right-hand side term. Therefore, the two unknown constants may be determined directly from the graph of y versus x by linear extrapolation. The method of Young and Wall is obviously limited since it requires advance knowledge of the Reynolds number exponents for both the tube-side and shell-side resistances.

Briggs and Young [24] relaxed the requirement that the Reynolds number exponent for the shell-side resistance be known in advance. In this case, Equation (4.3.5) may be restated as follows:

$$\frac{1}{UA} - R_w = \frac{1}{C_t [Re^{0.8} Pr^{0.4} (k/D_h) A]_t} + \frac{1}{C_s [Re^d Pr^{0.4} (k/D_h) A]_s} \quad (4.3.5)$$

The unknowns to be determined in above equation are the constants C_t , C_s , and d . The wall resistance R_w is assumed to be known. Using their own experimental data, Briggs and Young formulated an iterative scheme consisting of two successive linear regressions in order to properly evaluate the three unknown parameters. However, the Briggs and Young method is also severely limited since it requires an "a priori" knowledge of the Reynolds number exponent on one side.

Khartabil, et. al. [64] introduced one of the most generalized, modified Wilson Plot methods that allows for the determination of all three resistances (shell-side, tube-side, and wall) by assuming appropriate laws for the tube-side and shell-side heat transfer coefficients. The form of their equation, containing five unknowns, is shown on the top of the next page, i.e., Equation (4.3.6).

$$\frac{1}{UA} - R_w = \frac{1}{C_t [Re^a Pr^{0.4} (k/D_h) A]_t} + \frac{1}{C_s [Re^d Pr^{0.4} (k/D_h) A]_s} \quad (4.3.6)$$

The five unknowns in the above expression are C_t , C_s , a , d , and R_w . Determining these five unknowns is complicated in that it requires a nonlinear regression analysis based on three sets of experimental data, and all test data must be in only one flow regime. In addition, the three sets of data must be collected such that a different resistance is dominant in each set. The advantage of this method is that it is applicable to experiments performed with augmented tube geometries (e.g., internally finned tubes) and other heat transfer surface geometries for which a heat transfer correlation for the wall resistance and for the inside heat transfer coefficient is unknown. Specifically, the analysis of Khartabil, et. al. [64] is based on a heat exchanger with internally fluted tubes.

Most recently, R.K. Shah [98] presents an overall review of the original and modified Wilson Plot techniques, and also introduces a modification to the Briggs and Young method that is recommended for determining the heat transfer correlation (or coefficient) on only one fluid side. His formulated expression is as follows:

$$\frac{1}{UA} - R_w = \frac{1}{C_s [Re^a Pr^{0.4} (k/D_h) A]_s} + C' \quad (4.3.7)$$

The above equation contains the three unknowns, C_s , C' and a , where C' may be defined as:

$$C' = \frac{1}{C_t [Re^a Pr^{0.4} (k/D_h) A]_t} = \frac{1}{(\eta_s h A)_t} \quad (4.3.8)$$

The unknown terms in Equation (4.3.7) may also be determined from a nonlinear regression analysis. The value of the Reynolds number exponent a is the one that best linearly fits the given experimental test data. In this case, the slope of the line is $1/C_s$ and the y-intercept is C' . The wall resistance is assumed to be known in the analysis. This method has the advantage that it does not require an "a priori" knowledge of the Reynolds number exponent. In this case, the thermal resistance on the fluid side having the unknown heat transfer coefficient is required to be kept constant. This is accomplished by maintaining the fluid flow rate and the bulk mean fluid temperature fixed for the complete range of

Reynolds number on the unknown side of interest. In addition, the fluid flow on the known side or tube-side must be for one flow regime.

4.4. Applicability of Methods to Finned Tube Surfaces

Although the foregoing derivations have been based primarily on shell and tube heat exchangers, the original and modified Wilson Plot methods are easily adaptable to the typical refrigerant-to-air, fin-and-tube heat exchanger surfaces used in mobile air conditioning and household refrigerator/freezer applications, e.g., an air-cooled, finned tube condenser with refrigerant flowing inside the tubes. In these cases, the air represents the shell-side fluid and the refrigerant represents the tube-side fluid. For single phase applications, it is reasonable to assume that the functional form of the convective heat transfer coefficient on the refrigerant-side is known by available correlations.

Since it is typically desirable to determine only the one unknown heat transfer coefficient (i.e., the air-side), the most applicable of the preceding techniques is that described by Shah [98]. Thus, experimental testing requires that the heat exchanger air-side conditions be maintained the same (i.e., the air flow rate, inlet air temperature and log-mean average air temperature) while the flow rate of single-phase refrigerant is systematically varied so that all test data is for one flow region only, e.g., fully developed turbulent flow.

Consideration of fouling resistance is important since it reduces heat exchanger performance by increasing the overall resistance to heat transfer. However, for most refrigerant-to-air, compact heat exchanger applications, the fouling resistances, $R_{f,i}$ and $R_{f,o}$, are negligibly small compared to the other heat transfer resistances. For example, according to the Standards of the Tubular Exchanger Manufacturer's Association [117] which contains information on fouling for various industrial fluids utilized in heat exchanger applications, the generally accepted fouling resistance associated with refrigerant liquids and oil-bearing refrigerant vapors are 0.001 and 0.002 hr ft² °F/Btu (0.0002 and 0.0004 m² °C/W), respectively. In addition, the fouling resistance given for industrial air is 0.002 hr ft² °F/Btu (0.0002 m² °C/W).

Although the basis for the TEMA tables is undocumented, their use is considered reliable due to their age and the fact that the values have not been challenged by other published sources [97]. In addition, if the values were overly conservative, then it is anticipated that complaints by designers would have forced revisions. Heat transfer textbooks often refer to the TEMA tables as a source for fouling coefficients.

Since the fouling resistances may be omitted, then Equation (4.1) reduces to the form of Equation (4.4.1) shown below.

$$\frac{1}{U_o A_o} = \frac{1}{U_i A_i} = \frac{1}{(\eta_s h A)_i} + R_w + \frac{1}{(\eta_s h A)_o} \quad (4.4.1)$$

Based on the experimental procedures briefly introduced, the last two terms on the right hand side, the wall resistance and the outside (or air-side) heat transfer resistance, essentially represent a constant. Subsequently, replacing these two terms by the constant C_2 yields the new functional relationship shown by Equation (4.4.2).

$$1/U_o A_o = (1/h A)_i + C_2 \quad (4.4.2)$$

where:

$$C_2 = R_w + \frac{1}{(\eta_s h A)_o} \quad (4.4.3)$$

It is also assumed in the above expression that no fins are employed on the inside of the tubes so that the tube-side surface efficiency is not important ($\eta_s = 1$).

Next, if the generalized expression for the inside heat transfer coefficient, Equation (4.3.2), is substituted into Equation (4.4.2), and assuming constant fluid properties then:

$$\frac{1}{U_o A_o} = \frac{1}{C_o \left[Re^a Pr^{0.4} (k/D_h) A_i \right]} + C_2 \quad (4.4.4)$$

Alternatively, the previous equation may be expressed as follows:

$$\frac{1}{U_o A_o} = \frac{1}{C_1} x + C_2 \quad (4.4.5)$$

Equations (4.4.4) and (4.4.5) represent the general form for the modified Wilson plot applicable to an air-cooled condenser with single phase refrigerant flowing inside of the tubes. This form is also that of a straight-line, as illustrated previously, but with slope $(1/C_1)$ and y-intercept C_2 . Remember that C_2 is also equivalent to the outside heat transfer resistance inclusive of the tube wall. Equations (4.4.4) and (4.4.5) are also different from the original Wilson plot linear relationship in that the independent variable plotted along the abscissa, x , is now composed of a combination of three independent variables:

$$x = \frac{1}{[Re^a Pr^{0.4} (k/D_h) A_i]} \quad (4.4.6)$$

The independent variables in Equation (4.4.6) are the Reynolds number (Re), Prandtl number (Pr), and thermal conductivity (k). If the temperature range over experimental testing is relatively small such that the effect of variable fluid properties is negligible, then essentially the only variable in the above equation is the Reynolds number.

The dependent variable, y , may be expressed in terms of either the outside or inside heat transfer surface area, shown respectively as:

$$y = \frac{1}{U_o A_o} = \frac{1}{U_i A_i} \quad (4.4.7)$$

Due to the complex surface geometries of finned tube heat exchangers, determination of the exact heat transfer surface area is sometimes a tedious and complicated task. The purpose of introducing the area term in the dependent variable is to make knowledge of the exact amount of heat transfer surface unimportant in this preliminary analysis.

4.5. Alternative Wilson Plot Methods with Two-Phase Fluids

The previous sections considered only the application where single phase fluids are used on both sides of the heat exchanger. In some cases, it may be desirable to generate Wilson Plot data based on two phase fluid flow on one side of the heat exchanger, e.g., an air-cooled condenser with refrigerant condensing inside of the tubes. However, documented adaptation of the Wilson Plot procedures to applications involving two phase flow on one fluid side are extremely limited compared to single phase applications. In the noted publication by Briggs and Young [24], modified Wilson Plot derivations were included for two different multi-phase flow conditions. One is based on refrigerant condensation on the outside of a single horizontal finned tube; whereas, the other is based on the same geometry, but refrigerant boiling on the inside of the tubes.

Although a number of two-phase flow correlations for the convective heat transfer coefficient are available for substitution into the generalized overall resistance balance expression, Equation (4.1), these correlations are considerably more complex than most single phase correlations. In addition, the presence of different flow regimes in multi-phase flow complicates attempts to accurately represent the two phase, heat transfer coefficient.

For the specific example of the air-cooled condenser with in-tube refrigerant condensation, Stoecker [110] points out that generally the heat transfer coefficient for air flowing over finned tube surfaces varies as the square root of velocity, that is:

$$h_o = C V^{0.5} \quad (4.5.1)$$

In this case, the abscissa of the Wilson Plot may be replaced by $1/V^{0.5}$ in order to achieve a straight line that may be extrapolated to the ordinate. In addition, experimental testing would have to be based on constant refrigerant-side conditions (i.e., flow rate and bulk mean temperature) and variable air velocities. However, there are inherent disadvantages to this method of analysis, primarily due to assuming the exponent.

Although $1/V^{0.5}$ may seem to be a reasonable approach or starting point, a better scheme to determine the outside or air-side heat transfer coefficient would be similar to the modified techniques outlined in the preceding section. For example, consider the general relationship by Shah, i.e., Equation (4.3.7). However, in this case, the test fluid that is varied during experimental testing is on the air-side, rather than the side having the known heat transfer coefficient, i.e., the refrigerant-side. Therefore, neglecting fouling resistances and internally finned tubes, and designating the fixed refrigerant-side and wall resistances as C_4 , the overall resistance balance may be expressed as:

$$\frac{1}{U_o A_o} = \frac{1}{(\eta_s h A)_o} + C_4 \quad (4.5.2)$$

where:

$$C_4 = R_w + \frac{1}{(hA)_i} \quad (4.5.3)$$

Substituting the form of Equation (4.3.2) in order to represent the outside heat transfer coefficient and assuming constant fluid properties (i.e., for a gas such as air, $T_w/T_m = 1$), then:

$$\frac{1}{U_o A_o} = \frac{1}{C_o [Re^a Pr^{0.4} (k/D_h) A_o]} + C_4 \quad (4.5.4)$$

The above equation represents another modified Wilson Plot with slope $1/C_o$ and y-intercept C_4 . However, this equation may not be accurately plotted without properly knowing the air-side Reynolds number exponent 'a'. The three unknown parameters in the above equation may also be solved by a nonlinear regression analysis. The Reynolds number exponent 'a' is the one that provides the best linear fit of the given set of experimental data.

After one has determined the unknown parameters in the Wilson Plot function, additional minor steps are required to evaluate the air-side thermal resistance. First, the y-intercept or refrigerant-side resistance must be determined directly from the nonlinear regression analysis. Second, in order to compute the air-side thermal resistance, the overall resistance (y) must be deducted by an amount equal to the constant C_4 . There is a different air thermal resistance corresponds to each air flow rate.

In terms of the physical properties of air, the previous equation may be expressed alternatively as follows:

$$\frac{1}{U_o A_o} = C_o \left[\left(\frac{\rho V D_h}{\mu} \right)^a \left(\frac{C_p \mu}{k} \right)^{0.4} \left(\frac{k}{D_h} \right) A_o \right]^{-1} + C_4 \quad (4.5.5)$$

In addition, if the temperature range during testing is small such that the fluid properties change insignificantly, then essentially the only variable in the above equation is the air velocity. By lumping the fluid property and remaining terms into a single constant C_3 , Equation (4.5.5) becomes:

$$\frac{1}{U_o A_o} = \frac{1}{C_3 V^a} + C_4 \quad (4.5.6)$$

Equation (4.5.6) identifies one of the general forms referenced by Stoecker [110], when the air velocity is the test flow variable. According to Stoecker, the value of 'a' is approximately 0.5. However, as stated in presenting the modified Wilson Plots of Section 4.3, assuming the Reynolds number exponent prior to examination of experimental data may lead to errors. A better approach is to perform a nonlinear regression analysis using experimentally determined UA values and Equation (4.5.5).

4.6. Other Analytical Air-Side Correlations

Although Stoecker notes that 0.5 is a reasonable value for the exponential velocity function, it is difficult to identify one particular numerical value for the exponent in Equation (4.5.5) or (4.5.6) that successfully validates heat transfer data. Based on the results of an additional literature search into this matter, the value of the exponent relating Nusselt number to Reynolds number or equivalently, relating the air-side convective heat transfer coefficient to the velocity may vary from approximately 0.40 to 0.75 depending on the flow situation (e.g., laminar or turbulent) and on the geometry (e.g., single cylinder, bank of bare tubes, or finned tubes).

According to the McAdams [68], for a single unfinned cylinder in crossflow with air and in the Reynolds number range of 1000 to 50,000, the air-side heat transfer coefficient may be directly expressed as:

$$h_a = 0.11 c_p \frac{G^{0.6}}{D_o^{0.4}} \quad (4.6.1)$$

Similarly, for any gas flowing normal to a single unfinned cylinder, the generalized Nusselt number expressions are given as follows.

For Reynolds numbers between 0.1 and 1000,

$$Nu = \frac{hD_o}{k} = 0.32 + 0.43 \left(\frac{GD_o}{\mu} \right)^{0.52} \quad (4.6.2)$$

and for Reynolds numbers between 1000 and 50,000,

$$Nu = \frac{hD_o}{k} = 0.24 \left(\frac{GD_o}{\mu} \right)^{0.6} \quad (4.6.3)$$

In Equations (4.6.2) and (4.6.3), G is the mass velocity of the fluid in units of mass per unit time and unit area, and D_o is the outside diameter of the cylinder.

For air flowing across staggered or in-line banks of unfinned circular tubes, Grimson [50] utilized the generalized relationship expressed by Equation (4.6.4) to correlate the experimental results of various investigators over the Reynolds number range of 2000 to 40,000.

$$Nu = C Re_d^m Pr^{1/3} \quad (4.6.4)$$

The values of C and m are constants, dependent on the geometry. Values of the Reynolds number exponent, m , range from roughly 0.554 to 0.752, where the Reynolds number is based on the mass air velocity through the minimum cross section. Values of C_1 range from approximately 0.070 to 0.590. Although Grimson originally derived the above correlating equation based on tube banks of 10 rows or more, he also introduced a row correction factor to allow evaluating the Nusselt

number in tube banks of 1 to 9 rows deep. Its value, also dependent on surface geometry and tube arrangement, ranges from 0.64 to 0.99.

Further, Zukauskas [136] presented additional information for air flow across bare tube bundles which is valid for wide ranges of Reynolds number and property variations. The correlating equation by Zukauskas takes the form:

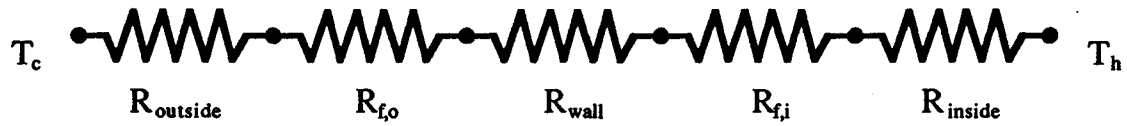
$$Nu = C Re_d^m Pr^{0.36} \left(\frac{Pr}{Pr_w} \right)^{1/4} \quad (4.6.5)$$

C and m are constants dependent on the tube arrangement, geometry and Reynolds number. The Prandtl number ratio accounts for variable fluid properties and is frequently omitted for gases. In the Reynolds number range of 10 to 2×10^5 , the value of the Reynolds number exponent, m, ranges from 0.40 to 0.63. The constant C ranges from 0.27 to 0.90 for the same range. In addition, since Zukauskas first derived the expression based on tube banks of 20 rows or more, he introduced a row correction factor for configurations with 1 to 19 rows that varies between 0.64 and 0.99.

Based on the correlations presented for various finned tube geometries in Section 2.4 and the relationship between Nusselt number and j-factor expressed by Equation (2.4.3), the values for the Reynolds number exponent range from approximately 0.595 to 0.681.

Finally, ARI Standard 410-87, entitled "Standards for Forced-Circulation Air-Cooling and Air-Heating Coils" [3], provides application rating data for three general coil types; namely, hot and cold water sensible heat coils, cooling and dehumidifying water coils, and volatile refrigerant cooling and dehumidifying coils. Based on laboratory test data for each coil, the air-side thermal resistance for a dry surface is linearly related to an exponential function of the standard free air velocity across the coil. In the three instances, the value of the velocity exponent ranges from approximately 0.52 to 0.56.

In summary, based on the previous documentation, it is extremely difficult to identify a single value for the velocity exponent that may be recommended for correlating the air-side convective heat transfer coefficient or Nusselt number. In the aforementioned examples, it has been demonstrated that the velocity or Reynolds number exponent may vary between 0.40 to 0.75, depending on the flow conditions and the surface geometry.



$$\begin{aligned}
 R_{\text{tot}} &= \text{Total thermal resistance to heat transfer} \\
 &= R_{\text{outside}} + R_{f,o} + R_{\text{wall}} + R_{f,i} + R_{\text{inside}}
 \end{aligned}$$

$$\begin{aligned}
 U_{\text{tot}} &= \text{Overall heat transfer coefficient} \\
 &= 1/R_{\text{tot}}
 \end{aligned}$$

where:

$$\begin{aligned}
 R_{\text{outside}} &= \text{outside thermal resistance} = \frac{1}{(\eta_s h A)_o} \\
 R_{f,o} &= \text{outside fouling resistance} \\
 R_{\text{wall}} &= \text{wall thermal resistance} \\
 R_{f,i} &= \text{inside fouling resistance} \\
 R_{\text{inside}} &= \text{inside thermal resistance} = \frac{1}{(\eta_s h A)_i}
 \end{aligned}$$

FIGURE 4.1
GENERAL RESISTANCE DIAGRAM
FOR A TWO FLUID HEAT EXCHANGER

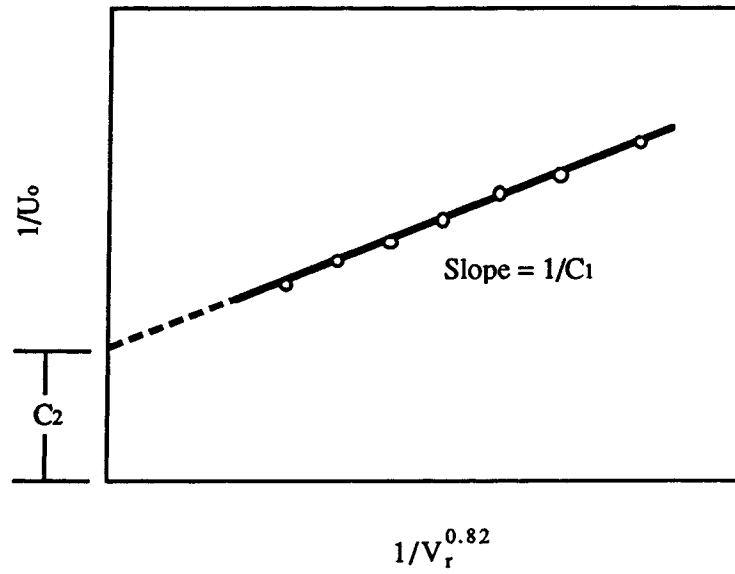


FIGURE 4.2
ORIGINAL WILSON PLOT TECHNIQUE

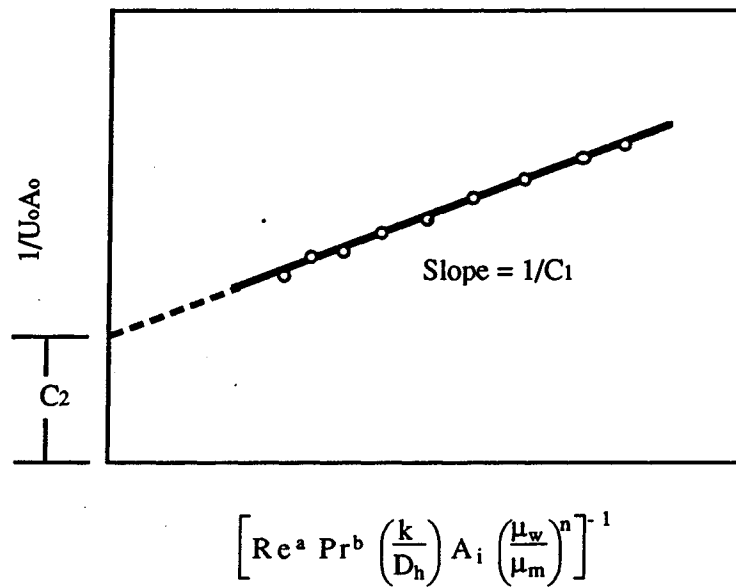


FIGURE 4.3
MODIFIED WILSON PLOT
TO ACCOUNT FOR VARIABLE PROPERTIES

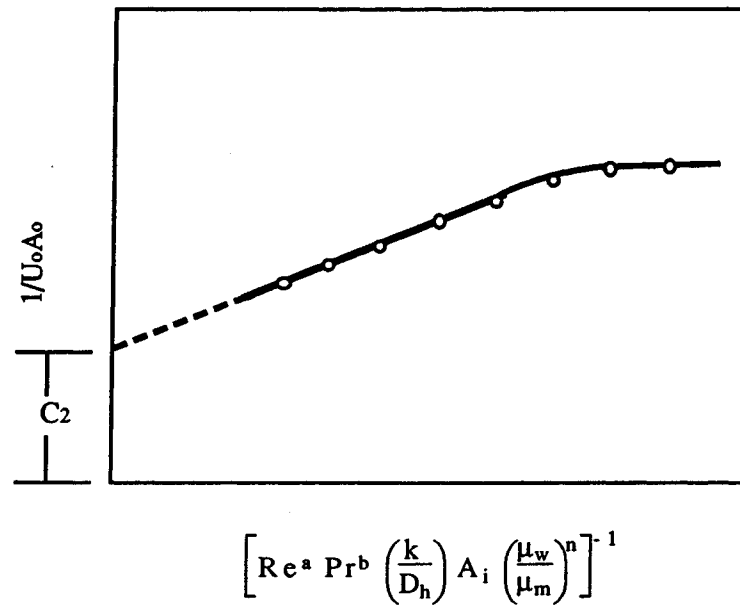
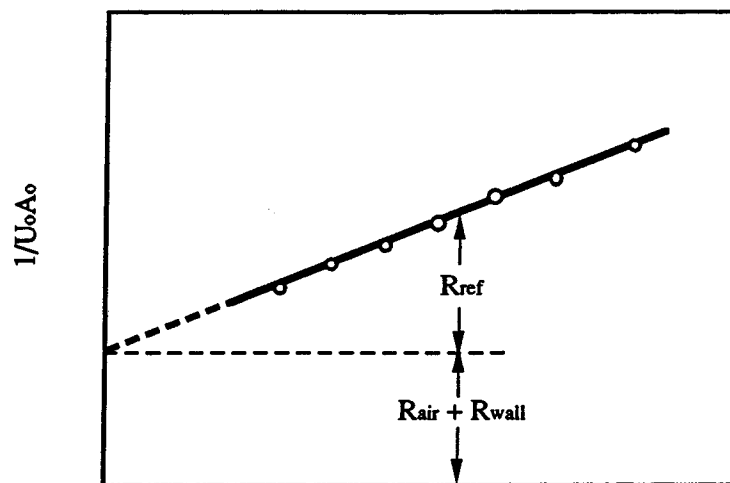
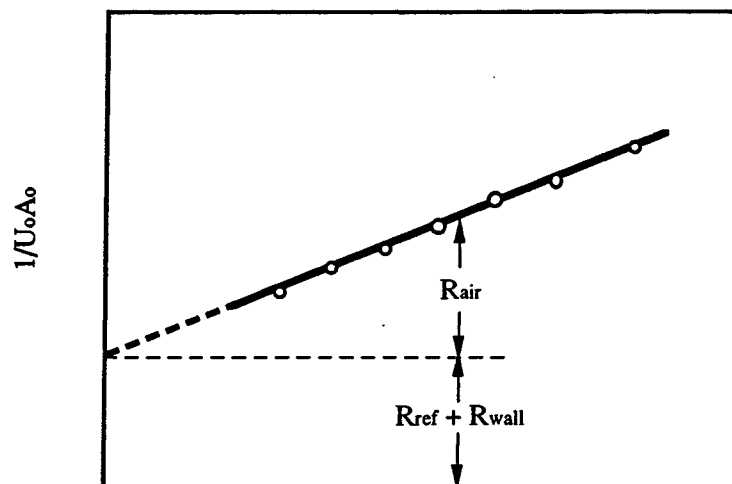


FIGURE 4.4
MODIFIED WILSON PLOT
INCLUDING TRANSITION AND LAMINAR FLOW REGIONS



$$\left[Re^a Pr^b \left(\frac{k}{D_h} \right) A_i \left(\frac{\mu_w}{\mu_m} \right)^n \right]^{-1}$$

FIGURE 4.5
MODIFIED WILSON PLOT TECHNIQUE
FOR AIR-COOLED CONDENSER
WITH SINGLE-PHASE REFRIGERANT FLOW



$$1/V_{air}^a$$

FIGURE 4.6
MODIFIED WILSON PLOT TECHNIQUE
FOR AIR-COOLED CONDENSER
WITH TWO-PHASE REFRIGERANT FLOW

CHAPTER 5

DATA REDUCTION TECHNIQUES

This chapter furnishes in detail how experimental data are reduced and analyzed in order to evaluate individual heat transfer coefficients and determine overall heat exchanger performance. Heat exchanger design and analysis methods are well documented in the literature; namely, by Shah [97,99], London [67], Kays & London [59] and generally, these methods fall into one of two different classifications. These are the "rating problem" and the "sizing problem". The data reduction methodology expounded herein essentially resembles the basic steps of the "rating problem", in which it is normally desired to evaluate both the heat transfer and pressure drop performance of an existing heat exchanger based on the exchanger construction type, flow arrangement, core dimensional data, surface geometry, and fluid flow conditions on both sides being known or specified.

Based on the experimental data collected for the given heat exchanger, the fundamental steps comprising the "rating problem" may be outlined as follows:

1. Compute condenser surface geometrical properties.
2. Determine bulk mean fluid temperatures and corresponding transport properties.
3. Calculate Reynolds numbers and surface basic characteristics.
4. Evaluate heat exchanger effectiveness and number of transfer units, NTU.
5. Compute overall UA values.
6. Determine individual heat transfer resistances and convective heat transfer coefficients.

The data reduction techniques employed may be considered in two parts. The first portion examines the reduction of raw test data into parameters important to the heat exchanger analysis. These parameters include condenser geometrical data (e.g., hydraulic diameter, heat transfer surface area, etc.) and also, air-side and refrigerant-side conditions at the inlet and exit of the condenser test section. The significant measured and computed variables at the condenser test section during

experimental testing are identified in Figure 5.1. The second part deals with how some of these parameters are further reduced in order to provide valuable information on heat exchanger surface basic data and overall performance, such as condenser heat rejection capacity, effectiveness and number of transfer units, overall heat transfer resistance, individual heat transfer resistances and coefficients, and j-factors.

5.1. Surface Geometrical Properties

Proper heat transfer and pressure drop analysis of any two-fluid, compact heat exchanger requires knowledge of the surface geometrical properties on each fluid side. Other than core dimensional data, flow and tube arrangements, and extended surface type, some of the more important geometrical properties include:

- minimum free flow area, $A_{\min}$
- primary and secondary surface areas, A_p and A_f
- total heat transfer surface area, A_o
- core frontal area, A_{fr}
- surface area density, based on overall volume, α
- hydraulic diameter, D_h
- flow lengths, L_1 , L_2 , and L_3
- ratio of minimum free flow area to frontal area, σ

The above properties may be determined directly from the heat exchanger manufacturer, although in some instances, the material may be deemed proprietary information and not disclosable. On the other hand, if field measurements of the exchanger face dimensions, flow depth, fin thickness and length, tube diameter, and other dimensional data are recorded, surface geometrical properties may be easily computed by available methods documented in this section. In the discussion that follows, procedures for determining surface geometrical properties are summarized by presenting specific relationships for the geometry of continuous plain fins on a circular tube array with either an in-line or staggered tube orientation [97].

Some common dimensions associated with staggered tube arrangements are defined in Figure 5.2. Additional dimensions, typical for both staggered and in-line tube arrangements are identified in Figure 5.3.

In general for any finned tube surface, the total heat transfer surface area consists of the primary and secondary surface areas. The primary surface area is equal to the sum of two parts: (1) the tube surface area less the area covered by the fins, and (2) the header plate surface area on each side:

$$A_p = \pi d_o L_2 (1 - \delta n_f) N_t + 2 \left(L_1 L_3 - \frac{\pi d_o^2}{4} N_t \right) \quad (5.1.1)$$

The secondary surface area, also called the extended or finned surface area, is equal to the total surface area of the fins, including the fin edge area:

$$A_f = 2 \left(L_1 L_3 - \frac{\pi d_o^2}{4} N_t + L_3 \delta \right) n_f L_2 \quad (5.1.2)$$

where $n_f L_2$ is equal to the total number of fins in the heat exchanger, N_f .

The minimum free flow area computation differs for in-line and staggered tube arrangements. For an in-line arrangement, the minimum free flow area is that area for a tube bank minus the area obstructed by the fins, expressed by Equation (5.1.3) below.

$$A_{\min} = \left[(S_t - d_o) L_2 - (S_t - d_o) \delta n_f L_2 \right] \frac{L_3}{S_t} \quad (5.1.3)$$

For a staggered arrangement, the $A_{\min}$ calculation is slightly more intensive, due to the presence of the parameter K_s , which accounts for the fact that the minimum free flow area may occur either through the front row (K_{s1}) or through the diagonals between tube rows (K_{s2})

$$A_{\min} = \left(\frac{L_3}{S_t} - 1 \right) K_{\min} L_2 - \left[(1 - \delta n_f) (S_t - d_o) \right] L_2 \quad (5.1.4)$$

In Equation (5.1.4), $K_{\min}$ is equal to the minimum of $2K_{s1}$ and $2K_{s2}$ defined below:

$$K_{s1} = \frac{1}{2} (1 - \delta n_f) (S_t - d_o) \quad (5.1.5)$$

$$K_{s2} = \left[\left(\frac{S_t}{2} \right)^2 + S_1^2 \right]^{1/2} - d_o - (S_t - d_o) \delta n_f \quad (5.1.6)$$

Subsequently, the hydraulic diameter, surface area density, and porosity on the fin-side may be calculated respectively as follow:

$$D = \frac{4 A_{\min} L_1}{A_o} = \frac{4\sigma}{\alpha} \quad (5.1.7)$$

$$\alpha = \frac{A_o}{L_1 L_2 L_3} \quad (5.1.8)$$

$$\sigma = \frac{A_{\min}}{A_{fr}} = \frac{A_{\min}}{L_2 L_3} \quad (5.1.9)$$

5.2. Air Flow Rate

It is necessary to know the air volumetric and mass flow rates for analysis purposes. As introduced in Section 3.2.2, an air flow measurement chamber consisting of interchangeable nozzles is provided in the air loop to permit accurate determination of fan air flow rate. The computations described herein are in accordance with ASHRAE Standard 41.2-1987, entitled "Standard Methods for Laboratory Air-Flow Measurement" [6] and ASHRAE Standard 51-1985, entitled "Laboratory Methods for Testing Fans for Rating" [5]. Tables 5.1 and 5.2 list the applicable formulas in English and SI units, respectively. The two relationships shown for the nozzle discharge coefficient, C_{noz} , are obtained directly from Bohanon [19].

During experimental testing, the following data is recorded at the air flow measurement chamber:

- Nozzle inlet air dry-bulb temperature, T_{db}
- Nozzle inlet static pressure, P_{s1}
- Nozzle differential pressure, ΔP
- Nozzle throat diameter(s), D_{noz}

In addition, the following four assumptions are incorporated in order to further define and simplify the air flow rate calculations:

- Nozzle inlet wet bulb temperature, T_{wb} , is the same as the measured dry bulb temperature, T_{db} ;
- Barometric pressure is equivalent to standard atmospheric pressure, i.e., $P_b = 29.921$ in. Hg (101.325 kPa);
- Since the air flow chamber cross-section is so large, the velocity pressure is essentially negligible so that the total pressure at the nozzle inlet, P_{tot} , is equal to the measured static pressure, P_{s1} .

Computer simulations of the tabulated air flow rate equations were performed to verify the first two assumptions listed; whereas, the third assumption has been previously verified in Section 3.2.2. Figures 5.2 and 5.3 illustrate the individual effects of barometric pressure and wet bulb temperature, respectively, on the mass flow rate of air over the dry bulb temperature range of 30 to 180 °F (-1.1 to 82.2 °C). Figure 5.4 displays the combined effects of variations in wet bulb temperature and barometric pressure on the air mass flow rate over the same temperature range. The range of air mass flow rate shown was selected arbitrarily and corresponds to a volumetric air flow rate of 1200 cfm (0.566 m³/sec) at a dry bulb temperature of 70 °F (21.1 °C) and a relative humidity of 50% (i.e., a humidity ratio of 0.0080).

Figure 5.2 displays that the air flow rate is directly proportional to the barometric pressure. Specifically, for a variation of 50 inches Hg in the barometric pressure and over the full temperature range, the air flow rate varies by slightly less than one percent. Figure 5.3 indicates that as the difference between the dry-bulb and

wet-bulb temperatures increases, the air flow rate increases. The effect is more pronounced at higher dry-bulb air temperatures. In particular, at a dry bulb temperature of 80 °F (26.7 °C), a wet bulb temperature of 60 °F (15.6 °C) increases the air flow rate by less than 0.75 percent. Likewise, at a dry-bulb temperature of 180 °F (82.2 °C), a wet bulb temperature of 160 °F (71.1 °C) increases the air flow by more than 1.5%.

Figure 5.4 illustrates the combined effects of the barometric pressure and wet-bulb temperature variations on the air mass flow rate at different dry-bulb temperatures. The plot shows that the assumed condition, namely the data set of "Tdb-Twb=0, Pbar=29.92", is bracketed by the conditions resulting in the lowest air flow rate and the highest air flow rate. At a dry-bulb air temperature of 80 °F (26.7 °C), the variation is approximately +1.8/-1.0 percent; whereas, at a dry-bulb air temperature of 180 °F (82.2 °C), the variation is +3.8/-1.0 percent.

Once the mass flow rate of air is known, the volume flow rate of air at any other location in the air loop may be easily determined by introducing the specific volume or density.

$$\dot{V} = \frac{\dot{m}}{\rho} = m v \quad (5.2.1)$$

where ρ and v are equal to the density and specific volume of air, respectively, evaluated at the temperature and humidity ratio associated with the desired location. For example, in order to determine the volumetric air flow rate across the condenser, the density or specific volume in Equation (5.2.1) must be evaluated at the measured condenser inlet temperature and the assumed humidity ratio.

5.3. Refrigerant Flow Rate

Volumetric flow rate of refrigerant is measured directly by the positive displacement flowmeter, described in Section 3.1.3. Equation (3.1.1) represents the volume flow rate of refrigerant in gpm as a function of the DC voltage output signal from the flowmeter transmitter.

Mass flow rate is directly related to the volume flow rate by the following expression:

$$\dot{m}_{ref} = (\rho \dot{V})_{ref} \quad (5.3.1)$$

where $\dot{V}$ is equal to the volume flow rate computed from Equation (3.1.1) and ρ is equal to the saturated liquid density of the refrigerant corresponding to the temperature measured at the inlet to the refrigerant heater and immediately downstream of the flowmeter. The curve-fit, fifth order polynomial shown by Equation (5.3.2) was formulated to determine the saturated liquid density, in lb_m/hr , of refrigerant CFC-12 as function of the measured temperature. The equation is based on 1989 ASHRAE Fundamentals data, and over the range of 20 to 200 °F (6.7 to 93.3 °C), the correlation is accurate within ± 0.01 percent.

$$\begin{aligned} \rho_{R12} = & 90.962 - (1.153e-1)T + (2.4393e-4)T^2 \\ & - (4.7292e-6)T^3 + (2.6436e-8)T^4 - (6.8294e-11)T^5 \end{aligned} \quad (5.3.2)$$

where in the above equation, the temperature T is in degrees Fahrenheit.

5.4. Condenser Temperature and Pressure Conditions

In addition to the flow rates of each fluid, the following air-side and refrigerant-side measurements of temperature and pressure, recorded at or near the condenser test section, are required for the heat exchanger analysis:

- Air inlet temperature measured by averaging RTD sensor, $T_{a,in,RTD}$
- Air inlet temperatures (9 total) measured by thermocouple grid, $T_{a,out,TC}$
- Air exit temperature measured by averaging RTD sensor, $T_{a,out,RTD}$
- Air exit temperatures (9 total) measured by thermocouple grid, $T_{a,out,TC}$
- Refrigerant inlet temperature, $T_{r,in}$, and pressure, $P_{r,in}$
- Refrigerant exit temperature, $T_{r,out}$, and pressure, $P_{r,out}$

5.5. Bulk Mean Temperature

In general, it is important to establish a representative flow length average or bulk mean fluid temperature on each side of a two-fluid heat exchanger so that fluid properties may be accurately evaluated. In this case, the air-side and refrigerant-side thermodynamic properties required are the density or specific volume, specific heat, viscosity, and thermal conductivity (see Section 5.6). Subsequently, these variables appear in further computations for the dimensionless Reynolds number, Prandtl number, Nusselt number, and Stanton number, so that proper evaluation is essential.

An arbitrary procedure for computing the bulk mean temperatures on the hot and cold fluid sides of a two-fluid compact heat exchanger was introduced by Shah [99] and is summarized in Table 5.3 for the two cases of $C_{\min}/C_{\max} < 0.5$ and $C_{\min}/C_{\max} > 0.5$. In this analysis, the refrigerant corresponds to the "hot fluid", whereas the air corresponds to the "cold fluid".

5.6. Fluid Properties

This section contains information on the thermodynamic and transport properties of refrigerants CFC-12 and HFC-134a and of air. Correlations valid for a wide range of temperatures are presented for some of the refrigerant properties. Air properties are presented graphically.

5.6.1. Refrigerant Properties

For single phase refrigerant R-12, fluid properties for saturated liquid and saturated vapor are conveniently tabulated in the latest edition of the ASHRAE Fundamentals Handbook [10]. In addition, Figures 5.5 thru 5.8 graphically portray some of the thermodynamic transport properties of saturated liquid refrigerant CFC-12. For comparative purposes, transport properties of refrigerant R-134a are also indicated on each graph. Refrigerant R-134a property data was obtained courtesy of General Electric.

Curve-fit polynomial functions were formulated for the different thermodynamic properties of saturated liquid refrigerant CFC-12 shown; namely, density, specific heat, thermal conductivity, and viscosity. Formulated and expressed in terms of

English units, these temperature-dependent relationships are presented below, along with the applicable temperature range for which each was developed and the associated accuracy.

For density, ρ , in units of lbm/ft³:

$$\begin{aligned}\rho = & 90.692 - 1.1536e-1T + 2.4393e-4T^2 \\ & - 4.7292e-6T^3 + 2.6436e-8T^4 - 6.8294e-11T^5\end{aligned}\quad (5.6.1)$$

The density equation was formulated based on a temperature range of 20 to 200 °F (-6.7 to 93.3 °C). It accurately represents the ASHRAE tabulated data to within ±0.01 percent.

For specific heat, c_p , in units of Btu/lb_m °F:

$$\begin{aligned}c_p = & 2.1702e-1 + 6.2472e-5T + 5.6563e-6T^2 \\ & - 9.1302e-8T^3 + 6.4795e-10T^4 - 1.2821e-12T^5\end{aligned}\quad (5.6.2)$$

This equation was derived for the temperature range of 0 to 180 °F (-17.8 to 82.2 °C), and correlates the ASHRAE Fundamentals data to within ±0.2 percent.

For thermal conductivity, k , in units of Btu/hr ft °F:

$$\begin{aligned}k = & 4.9076e-2 - 1.4916e-4T + 1.4145e-6T^2 \\ & - 2.0815e-8T^3 + 1.2403e-10T^4 - 2.5949e-13T^5\end{aligned}\quad (5.6.3)$$

This equation was derived for the temperature range of 0 to 230 °F (-17.8 to 110 °C). In the range of 0 to 200 °F (-17.8 to 93.3 °C), the equation correlates the ASHRAE data within ±1.0 percent; whereas, between 200 and 230 °F (93.3 and 110 °C), the equation agrees with the tabulated data to within 2.0 percent.

For absolute viscosity, μ , in units of lb_m/ft hr:

$$\begin{aligned}\mu = & 7.6714e-1 - 4.4136e-3T + 2.1114e-5T^2 \\ & - 7.8708e-8T^3 + 2.6750e-10T^4 - 7.9232e-13T^5\end{aligned}\quad (5.6.4)$$

The correlating equation for absolute viscosity was also derived for the temperature range of 0 to 230 °F (-17.8 to 110 °C). It agrees with the ASHRAE tabulated data to within ± 0.6 percent for temperatures up to 200 °F (93.3 °C) and to within ± 1.1 percent for temperatures between 200 and 230 °F (93.3 and 110 °C).

In Equations (5.6.1) thru (5.6.4), the temperature is expressed in degrees Fahrenheit. The foregoing equations may be easily converted to SI units by introducing the following four conversion factors below.

$$\begin{array}{llll} \rho: & 1 \text{ lbm/ft}^3 & = & 16.0 \text{ kg/m}^3 \\ c_p: & 1 \text{ Btu/lbm } ^\circ\text{F} & = & 4.19 \text{ kJ/kg K} \\ k: & 1 \text{ Btu/lbm ft } ^\circ\text{F} & = & 0.144 \text{ W/m K} \\ \mu: & 1 \text{ lbm/ft hr} & = & 0.413 \text{ m Pa sec} \end{array}$$

5.6.2. Air Properties

The air transport properties described in this Section were gathered from EES or Engineering Equation Solver, Version 2.55. EES is a computer equation based software useful for evaluating the thermodynamic and transport properties of several different fluids, including moist air. Air property data incorporated by EES is based on the tabulated and curve-fit information by Howell and Buckius [55] and Irvine and Liley [56].

Air transport properties are also typically influenced by the humidity ratio or relative humidity, in addition to the dry-bulb temperature. In order to properly establish the variation, for a temperature range of 30 to 200 °F (-1.1 to 93.3 °C), properties of air were evaluated and plotted at the humidity ratios of 0.0020, 0.0050, 0.0080, 0.0110, and 0.0140 lb_m moisture/lb_m dry air. These five humidity ratios correspond to 70 °F (21.1 °C) air at 10%, 30%, 50%, 70% and 90% relative humidity, respectively.

Figure 5.9 illustrates how the specific volume or inverse of density of air varies with dry-bulb temperature and humidity ratio.

Figure 5.10 indicates how the specific heat of air varies with dry-bulb temperature and humidity ratio. In the temperature range of 60 to 150 °F (15.6 to 65.6 °C), the specific heat of air is relatively constant at each value of the humidity ratio. These values range from 0.241 to 0.246 Btu/lb_m °F (1.010 to 1.031 kJ/kg K), a variation of more than two percent.

In comparison to the specific heat, the thermal conductivity and absolute viscosity of air are more influenced by dry bulb temperature than by humidity ratio. For the humidity ratio range of 0.0020 to 0.0140 lb_m moisture/lb_m dry air, the following tabulated values of thermal conductivity and viscosity are fairly representative.

Air Temperature (°F)	Thermal Conductivity (Btu/hr ft °F)	Absolute Viscosity (lb _m /ft hr)
30	0.014	0.041
50	0.014	0.043
100	0.015	0.046
150	0.017	0.049
200	0.018	0.052

Although the temperature behavior of these fluid properties is nonlinear in the range of 30 to 200 °F (1.1 to 93.3 °C), corresponding values of thermal conductivity and viscosity for individual temperatures in this range may be computed by interpolation with little error introduced.

5.7. Reynolds Number

The air-side Reynolds number is highly dependent on the finned tube surface geometry under consideration. It is calculated based on the hydraulic diameter, D_h , and the maximum air velocity through the flow passage, V_{max} (i.e., the mass velocity through the minimum free flow area, A_{min}). Specifically stated:

$$Re = \frac{\rho V_{max} D_h}{\mu} = \frac{GD_h}{\mu} \quad (5.7.1)$$

In Equation (5.7.1), G is equal to the mass velocity per unit free flow area, expressed as:

$$G = \dot{m}_a / A_{\min} \quad (5.7.2)$$

The refrigerant-side Reynolds number, represented by Equation (5.7.3), is based on the tube inside hydraulic diameter and the fluid velocity through the cross sectional flow area.

$$Re = \frac{\rho V D_h}{\mu} = \frac{(\dot{m}_r / A_c) D_h}{\mu} \quad (5.7.3)$$

In Equations (5.7.1) and (5.7.3), μ is equal to the absolute viscosity and must be evaluated at the appropriate bulk mean fluid temperature.

5.8. Hydraulic Diameter

Evaluation of the hydraulic diameter, D_h , on each fluid side is important since it directly influences the Reynolds number. Since finned or augmented surfaces are more rarely employed on the inside of tubes, determination of the appropriate hydraulic diameter for the Reynolds number on the refrigerant-side is typically not as complicated compared to that of the air-side. In general, the hydraulic diameter is defined as:

$$D_h = \frac{4 \text{ (Cross Sectional Flow Area)}}{\text{(Wetted Perimeter)}} \quad (5.8.1)$$

This definition of hydraulic diameter permits extension of the analysis to noncircular tube geometries. For circular tube geometries, the hydraulic diameter is simply equivalent to the inside diameter of the tube. For rectangular, triangular, oval and many other noncircular tube geometries, Shah [97] defines relationships for hydraulic diameter. For example, for a rectangular cross section of width '2a' and height '2b', the hydraulic diameter may be computed by

$$D_h = \frac{4ab}{a + b} \quad (5.8.2)$$

As another example, for an elliptical or oval cross section with major axis '2a' and minor axis '2b', the hydraulic diameter is defined as:

$$D_h = \frac{\pi b}{E(m)} \quad (5.8.3)$$

where:

$$m = 1 - \left(\frac{2b}{2a} \right)^2 \quad (5.8.4)$$

and $E(m)$ is the complete elliptic integral of the second kind.

5.9. Effectiveness and NTU

The ϵ -NTU method is employed in the both the single phase and two phase, steady-state, heat exchanger analysis in order to properly determine the overall conductance UA. It is important to recall several of the limitations associated with applying the ϵ -NTU method. In general, the mentioned limitations may also apply to any heat exchanger, although where noted, some are strictly related to compact heat exchanger surfaces.

1. The specific heat of each fluid must be constant throughout the heat exchanger so that the individual heat capacity rates, $(mc_p)_{\text{fluid}}$ are also constant.
2. Phase changes must be at or near a constant temperature. This corresponds to a phase-changing, single component fluid flowing at a constant pressure. In this case, the effective specific heat for the phase-changing fluid is infinity, i.e., $dT=0$ in $c_p=dh/dT$ and thus, $C_{\text{max}}= \text{infinity}$.

(or)

The temperature of the phase changing fluid must vary linearly with heat transfer during the condensation or boiling. In this case, the effective specific heat is constant and finite for the two phase fluid.

3. The overall heat transfer coefficient between the two fluids must be constant throughout the heat exchanger, including for the case of a phase-changing fluid.
4. For finned tube surfaces, the overall surface efficiency, η_o , must be uniform and constant.
5. The fluid flow rate on each side must be maintained fixed and distributed uniformly through the heat exchanger, particularly for compact surfaces. In addition, no temperature or velocity stratifications (particularly at the entrance of the exchanger), flow bypassing or flow leakage may exist in either stream.
6. The heat transfer surface area must be uniformly distributed on each fluid side.
7. Longitudinal heat conduction in the fluid and in the wall are negligible.

The first and second limitations are two of the most critical ones for any steady-state, heat exchanger analysis. These limitations essentially restrict the application of the ϵ -NTU method to single phase fluids on both sides or on only one side with a dominating thermal resistance. In the case of an air-cooled condenser with desuperheating, condensing and subcooling regions of refrigerant flow, the ϵ -NTU method may be applied by dividing the heat exchanger into segments so that the specific heats and heat capacities may be treated as constant. In addition, fluid properties for each of these segments must be evaluated at the appropriate bulk mean fluid temperature, described previously in Section 5.5.

5.9.1. Effectiveness Computation

Heat exchanger effectiveness is a dimensionless efficiency parameter defined as the ratio of the actual heat transfer rate, Q_{act} , in a given heat exchanger to the maximum possible heat transfer rate, Q_{max} . In similar terms:

$$\varepsilon = \frac{Q_{act}}{Q_{max}} \quad (5.9.1)$$

This maximum rate of heat transfer corresponds to that obtained by a counterflow heat exchanger with infinite surface area and operating with the same fluid flow rates and inlet conditions as the actual heat exchanger. Since the flow rate and inlet and exit temperatures of each fluid are known from the experimental data, Q_{act} and Q_{max} may be evaluated as described below.

The actual rate of heat transfer may be defined by an energy balance including either the refrigerant-side or air-side heat capacity rate, $(\dot{m}c_p)_{ref}$ and $(\dot{m}c_p)_{air}$, respectively, and the associated temperature difference.

$$Q_{act} = (\dot{m}c_p)_{ref} (T_{in} - T_{out})_{ref} \quad (5.9.2)$$

$$Q_{act} = (\dot{m}c_p)_{air} (T_{out} - T_{in})_{air} \quad (5.9.3)$$

The thermodynamically limited, maximum possible heat transfer rate would be achieved if the fluid with the smallest heat capacity rate, $(\dot{m}c_p)_{min}$, was flowing in a heat exchanger of infinite length such that it was cooled from the inlet temperature of the hot fluid (refrigerant) to the inlet temperature of the cold fluid (air). In mathematical form, this definition is expressed by Equation (5.9.4).

$$Q_{max} = (\dot{m}c_p)_{min} (T_{ref,in} - T_{air,in}) \quad (5.9.4)$$

Based on previous equations, the heat exchanger effectiveness may subsequently be re-expressed as the ratio of the actual temperature difference associated with the minimum heat capacity fluid to the maximum possible temperature change attainable. For example, if the refrigerant is the minimum heat capacity fluid, $C_{min} = (\dot{m}c_p)_{ref}$, then Equation (5.9.1) may be expressed by Equation (5.9.5), shown on the top of the next page.

$$\varepsilon = \frac{(T_{\text{ref,in}} - T_{\text{ref,out}})}{(T_{\text{ref,in}} - T_{\text{air,in}})} \quad (5.9.5)$$

Otherwise, if the air is the minimum heat capacity fluid, $C_{\text{min}} = (mc_p)_{\text{air}}$, then effectiveness may be computed from Equation (5.9.6).

$$\varepsilon = \frac{(T_{\text{air,out}} - T_{\text{air,in}})}{(T_{\text{ref,in}} - T_{\text{air,in}})} \quad (5.9.6)$$

5.9.2. NTU Calculation

The number of transfer units or NTU is a measure of the "heat transfer size" or "thermal size" of a heat exchanger. It is defined by Equation (5.9.7) as the ratio of the overall conductance to the minimum heat capacity.

$$\text{NTU} = \frac{UA}{C_{\text{min}}} \quad (5.9.7)$$

NTU is also directly related to the effectiveness. Several effectiveness-NTU relationships have been formulated and documented in the literature. Shah [97] presents a tabulated summary of available ε -NTU relations. Kays and London provide effectiveness-NTU data in graphical form, in addition to specific derivations. The appropriate relation is generally a function of the heat exchanger geometry, flow arrangement, and "mixing" criteria.

The individual finned tube surfaces in this full condenser research and analysis may typically be categorized as a single pass, crossflow heat exchanger with one fluid mixed and the other fluid unmixed. A "mixed" fluid may be characterized by a one-dimensional temperature change along the flow direction; while for an "unmixed" fluid, temperature gradients may exist both parallel and normal to the flow direction. In the refrigerant-to-air condenser, the refrigerant corresponds to the mixed fluid with temperature variations only along the flow direction and the air behaves as the unmixed fluid with temperature variations normal and longitudinal to the direction of refrigerant flow.

The following two functions relating NTU as a function of effectiveness are valid for the particular geometry, flow arrangement and mixing conditions presented.

For $C_{\min} = \text{unmixed fluid}$, $C_{\max} = \text{mixed fluid}$:

$$NTU = -\ln \left[\frac{1}{C_r} \ln (1 - C_r \epsilon) + 1 \right] \quad (5.9.8)$$

For $C_{\min} = \text{mixed fluid}$, $C_{\max} = \text{unmixed fluid}$:

$$NTU = -\frac{1}{C_r} \left[\ln [1 + C_r \ln (1 - \epsilon)] \right] \quad (5.9.9)$$

where in the above two equations, C_r is simply the ratio of fluid heat capacities:

$$C_r = \frac{C_{\min}}{C_{\max}} \quad (5.9.10)$$

For two-phase, condensing refrigerant flow within the air-cooled condenser tubes, the maximum heat capacity fluid (i.e., the refrigerant) possesses infinite specific heat. This is because $c_p = dh/dT$ and the refrigerant temperature is relatively constant in the condensation region. Therefore, since $C_{\max}$ is infinite, the heat capacity goes to zero. Further, the NTU for a condensing type heat exchanger with $C_r = 0$ is defined as:

$$NTU = -\ln (1 - \epsilon) \quad (5.9.11)$$

5.10. Overall Thermal Conductance, UA

Once the effectiveness and NTU have been calculated by the methods outlined in the previous section, the overall conductance may be calculated by rearranging Equation (5.9.7):

$$UA = (NTU)(C_{\min}) \quad (5.10.1)$$

In general, the overall thermal conductance is generally related to the individual heat transfer resistances by the following equation:

$$\frac{1}{U_o A_o} = \frac{1}{U_i A_i} = \frac{1}{(\eta_s h A)_o} + R_{f,o} + R_w + R_{f,i} + \frac{1}{(\eta_s h A)_i} \quad (5.10.2)$$

The subscripts "o" and "i" designate the air-side and refrigerant-side, respectively, and "w" denotes the tube wall. Since the fouling resistances associated with the air and refrigerant are insignificant compared to the total resistance [117], $R_{f,o}$ and $R_{f,i}$ may be neglected in the above relationship. In addition, since fins or a similar augmentation scheme is not employed on the tube inside surface, the surface efficiency is unity ($\eta_s = 1$) term may be dropped from the inside resistance term. Therefore, the expression for the inverse of the overall conductance, or equivalently, the overall thermal resistance may be simplified to Equation (5.10.3).

$$\frac{1}{U_o A_o} = \frac{1}{U_i A_i} = \frac{1}{(\eta_s h A)_o} + R_w + \frac{1}{(h A)_i} \quad (5.10.3)$$

5.11. Individual Heat Transfer Resistances

This section consists of individual descriptions of how to obtain the individual heat transfer resistances identified in Equation (5.10.3). These include the wall thermal resistance (R_w), the refrigerant-side thermal resistance (R_{ref}), and the air-side thermal resistance (R_{air}). Modified Wilson Plots generated to assist in evaluating the air-side and refrigerant-side thermal resistances are shown in Figures 5.13 and 5.14. The difference in these plots exists in the independent flow variable during testing and the refrigerant phase inside the condenser tubes.

5.11.1. Wall Resistance

The wall thermal resistance, R_w , may normally be omitted from the overall resistance balance equation since its value is negligibly small compared to the inside and outside convective heat transfer resistances. This is especially true for copper and aluminum tubes due to the associated high thermal conductivities and small wall thicknesses.

The wall resistance may be calculated by the following relation for a bank of smooth circular tubes.

$$R_w = \frac{\ln(D_o/D_i)}{2\pi k_w L N_t} \quad (5.11.1)$$

5.11.2. Refrigerant-Side Resistance

For experiments based on single-phase refrigerant flow inside of condenser tubes and a constant air flow rate across the condenser, the refrigerant-side resistance associated with each refrigerant mass flow rate or Reynolds number may be obtained from the Wilson Plot by deducting the corresponding overall thermal resistance ($1/UA$) by an amount equal to the air-side resistance, including the tube wall resistance. Figure 5.13 represents a typical modified Wilson Plot generated from data analysis based on single phase refrigerant flow. For a particular value of the independent variable, the refrigerant-side resistance may be defined by the following expression:

$$R_{ref} = (1/UA) - (R_{air} + R_{wall}) \quad (5.11.2)$$

where $(R_{air} + R_{wall})$ is equal to the y-intercept on the modified Wilson Plot shown in Figure 5.11.

For single-phase and two-phase refrigerant flow, an alternative means also exists for determining the inside convective heat transfer resistance. First, the refrigerant-side convective heat transfer coefficient (h) may be evaluated directly by using one of the appropriate single phase or two phase flow correlations documented in Chapter 2. Second, based on the total heat transfer area of the tube inside surface (A_i), the inside heat transfer resistance is equivalent to the following:

$$R_{ref} = \frac{1}{hA_i} \quad (5.11.3)$$

For experiments based on two phase refrigerant flow inside of the condenser tubes and with variations in the air flow rate across the condenser, the refrigerant-side resistance may be determined directly from the Wilson Plot by evaluating the y-intercept. Although the y-intercept in this instance includes the tube wall resistance also, the wall resistance may normally be neglected since it is an order of magnitude lower than the refrigerant-side resistance. Figure 5.14 is a typical modified Wilson Plot generated for data analysis based on two-phase refrigerant flow.

5.11.3. Air-Side Resistance

For single-phase refrigerant flow experiments with a constant air flow rate, the air-side resistance may be obtained directly from the Wilson Plot by evaluating the y-intercept and subsequently, neglecting the tube wall resistance, since it is an order of magnitude lower than the air-side resistance. Figure 5.13 and Equation (5.11.2) are general forms of the modified Wilson Plot for this case.

For two phase refrigerant flow experiments with systematic variations in the air flow rate, a different air-side resistance corresponds to each air flow rate. Thus, each value of the air-side resistance must be determined by deducting the corresponding overall thermal resistance ($1/UA$) by an amount equal to the y-intercept or refrigerant-side resistance. From Figure 5.12, it is clearly evident that for a given value of the independent flow variable, the air-side resistance may be expressed as follows:

$$R_{air} = (1/UA) - (R_{ref} + R_{wall}) \quad (5.11.4)$$

where $(R_{ref} + R_{wall})$ is equal to the y-intercept on the modified Wilson Plot shown in Figure 5.14.

5.12. Air-Side Heat Transfer Coefficient

The air-side convective heat transfer coefficient is related to the air-side thermal resistance by the expression:

$$R_{\text{air}} = \frac{1}{\eta_s h A} \quad (5.12.1)$$

where η_s is equal to the overall surface efficiency of the finned tube heat exchanger coil and A is the total air-side heat transfer surface area, including the primary (bare tube) and secondary (finned) surfaces. In this condenser analysis, the fin and surface efficiencies are analyzed on a one dimensional, steady-state basis. Consideration of efficiency in extended surface heat exchangers has been previously documented in Section 2.4.8.

Assuming that the air-side thermal resistance (R_{air}) is previously known by the appropriate Wilson Plot techniques described in Section 5.11, solving for the air-side convective heat transfer coefficient requires an iterative procedure since the overall surface efficiency is also an unknown parameter. The iterative procedure may be outlined in five steps as follows:

1. Guess a value for the air-side convective heat transfer coefficient, h_{air} .
2. Solve for the fin efficiency by using the appropriate relationship for the surface geometry being investigated.
3. Calculate the overall surface efficiency.
4. Compute a new value for the air-side convective heat transfer coefficient from Equation (5.12.1).
5. If the computed value in Step #4 is equal to the initial guess, then the resulting values obtained for the heat transfer coefficient and the efficiencies are correct. If the computed value in Step #4 does not match the initial guess, then guess a new value for h_{air} and repeat the procedure beginning from Step #2.

5.13. Colburn j-Factor

The Colburn modulus or j-Factor is defined as

$$j = St Pr^{2/3} \quad (5.13.1)$$

or, replacing the Stanton Number by its definition

$$St = \frac{Nu}{Re Pr} \quad (5.13.2)$$

it follows that

$$j = \frac{Nu}{Re Pr^{1/3}} \quad (5.13.3)$$

The three dimensionless parameters appearing in Equation (5.13.3) and the Stanton number relationship are individually defined as

$$Nu = \text{Nusselt Number} = \frac{h_o D_h}{k} \quad (5.13.4)$$

$$Re = \text{Reynolds Number} = \frac{GD_h}{\mu} \quad (5.13.5)$$

$$Pr = \text{Prandtl Number} = \frac{c_p \mu}{k} \quad (5.13.6)$$

$$St = \text{Stanton Number} = \frac{h}{G c_p} \quad (5.13.7)$$

It is clearly evident that for relatively constant fluid properties, the j-factor is simply dependent on the air-side convective heat transfer coefficient and the mass velocity through the minimum free flow area. In the previous relationships, the air transport properties k , c_p and μ are evaluated at the air-side bulk mean air temperature.

5.14. Heat Transfer and Energy Balances

Independent of whether single-phase or two-phase refrigerant is flowing inside the condenser tubes, the heat gained by the air may be determined by the following relationship.

$$Q = (\dot{m}c_p)_{\text{air}} (T_{\text{air,out}} - T_{\text{air,in}}) \quad (5.14.1)$$

Alternatively, for single-phase refrigerant flow, the heat loss by the refrigerant may also be determined by the expression:

$$Q = (\dot{m}c_p)_{\text{ref}} (T_{\text{ref, in}} - T_{\text{ref, out}}) \quad (5.14.2)$$

Or, equivalently, in terms of the inlet and exit enthalpies, the heat loss may also be related as

$$Q = \dot{m}_{\text{ref}} (h_{f, \text{in}} - h_{f, \text{out}}) \quad (5.14.3)$$

where h_f is equivalent to the enthalpy of saturated liquid refrigerant evaluated at the condenser inlet and exit temperatures, respectively. Tabulated values of enthalpy are available in the latest edition of the ASHRAE Fundamentals Handbook.

For two-phase refrigerant flow with regions of desuperheating, condensing and subcooling, the heat transfer from the refrigerant to the air may be calculated on a "total" basis or a "region" basis, accordingly.

For the total heat transfer rate, Q_{tot} :

$$Q_{\text{tot}} = \dot{m} (h_{\text{in}} - h_{\text{out}}) \quad (5.14.4)$$

where it is important to consider the temperature and pressure conditions at the inlet and exit in order to evaluate the refrigerant enthalpies.

$$\begin{aligned} h_{\text{in}} &= h_{\text{in}}(P_{\text{in}}, T_{\text{in}}) \\ h_{\text{out}} &= h_{\text{out}}(P_{\text{out}}, T_{\text{out}}) \end{aligned}$$

Similarly, the individual heat transfer rates in the refrigerant desuperheating, condensing and subcooling regions may be identified and computed separately.

$$Q_{\text{desuperheat}} = (\dot{m}c_p)_{\text{ref}}(T_{\text{in}} - T_{\text{sat}})_{\text{ref}} = \dot{m}_{\text{ref}} (h_{\text{in}} - h_g)_{\text{ref}} \quad (5.14.5)$$

$$Q_{\text{subcooled}} = (\dot{m}c_p)_{\text{ref}}(T_{\text{sat}} - T_{\text{out}})_{\text{ref}} = \dot{m}_{\text{ref}} (h_f - h_{\text{out}})_{\text{ref}} \quad (5.14.6)$$

$$Q_{\text{cond}} = \dot{m}_{\text{ref}} (h_g - h_f)_{\text{ref}} = \dot{m} h_{fg} \quad (5.14.7)$$

where h_f and h_g are evaluated at the condensation temperature corresponding to the saturation pressure.

Ideally, an energy balance on the two fluid streams requires that the heat loss from the refrigerant be equal to the heat gain by the air, i.e.

$$Q_{\text{air}} = Q_{\text{ref}} \quad (5.14.8)$$

Unfortunately, due to nonideal circumstances, such as conduction losses through the equipment, the heat transfer is not always exactly equivalent. For these instances, the percent difference may be determined by the following relationship:

$$\text{Percent Difference} = \frac{Q_{\text{air}} - Q_{\text{ref}}}{\frac{Q_{\text{air}} + Q_{\text{ref}}}{2}} \times 100 \quad (5.14.9)$$

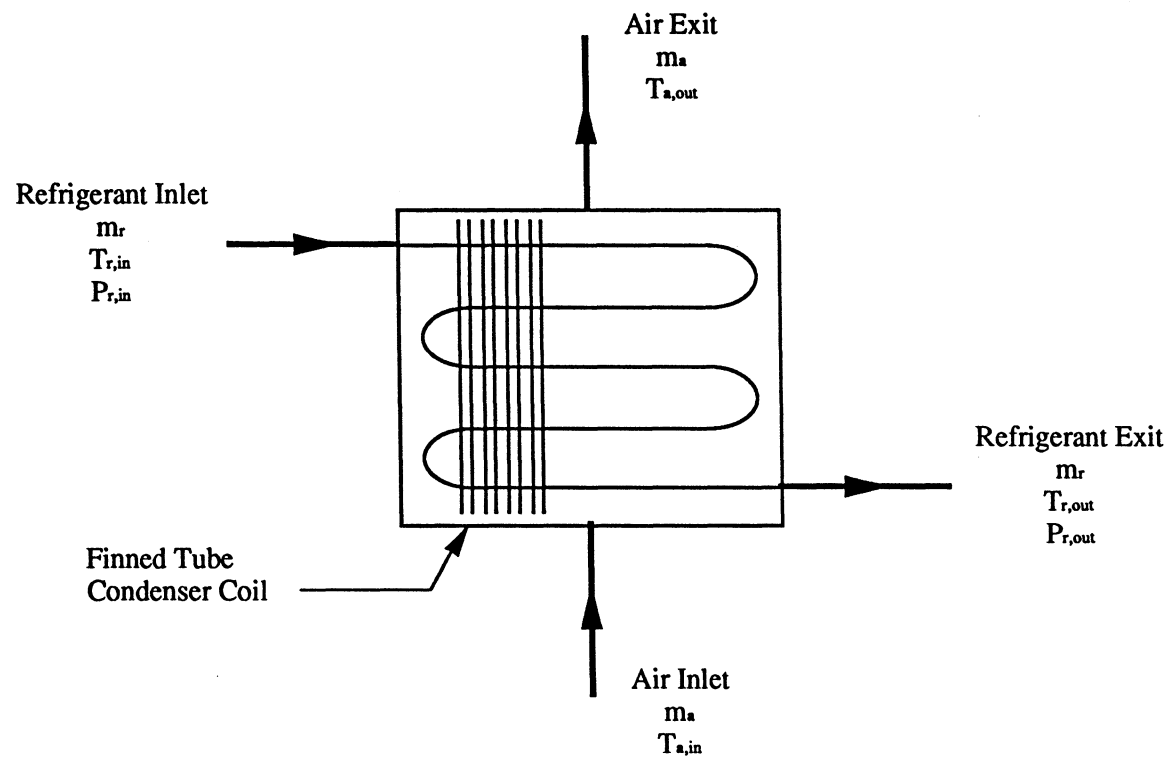
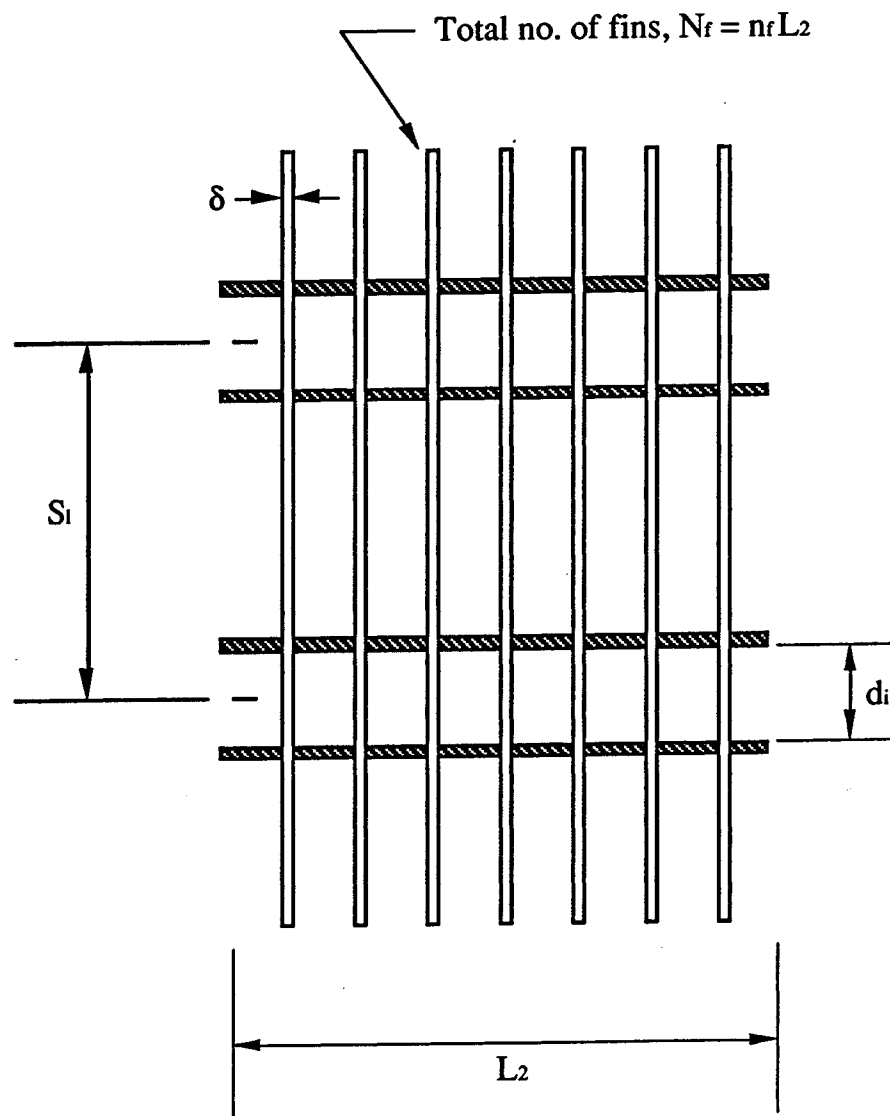


FIGURE 5.1
TYPICAL TEST CONDENSER
WITH SIGNFICANT MEASURED AND CALCULATED TEST PARAMETERS





SECTION A-A

FIGURE 5.3
SECTION VIEW OF FINNED TUBE GEOMETRY

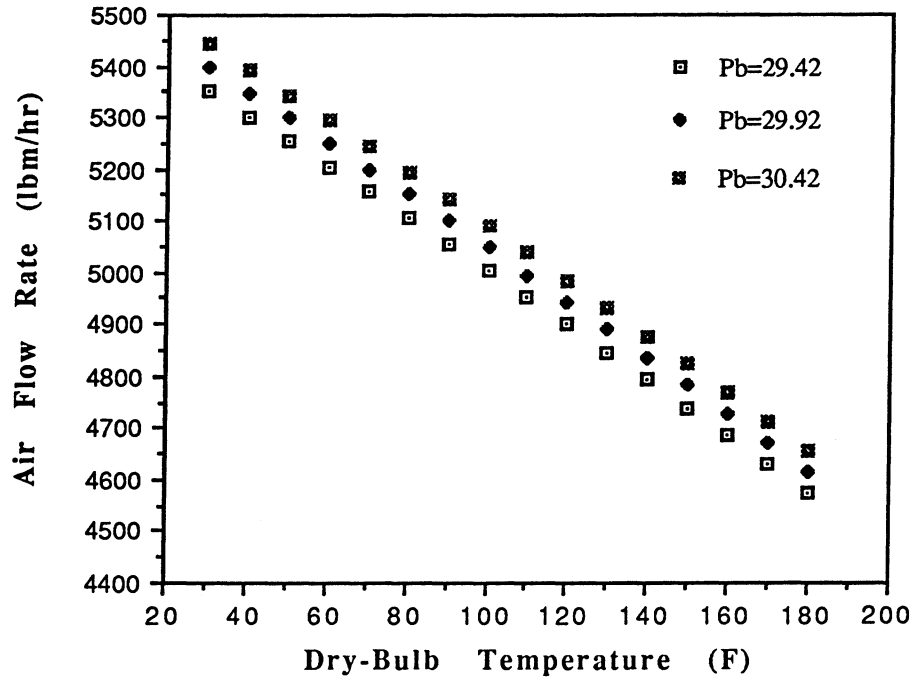


FIGURE 5.4
AIR FLOW RATE VARIATION WITH BAROMETRIC PRESSURE

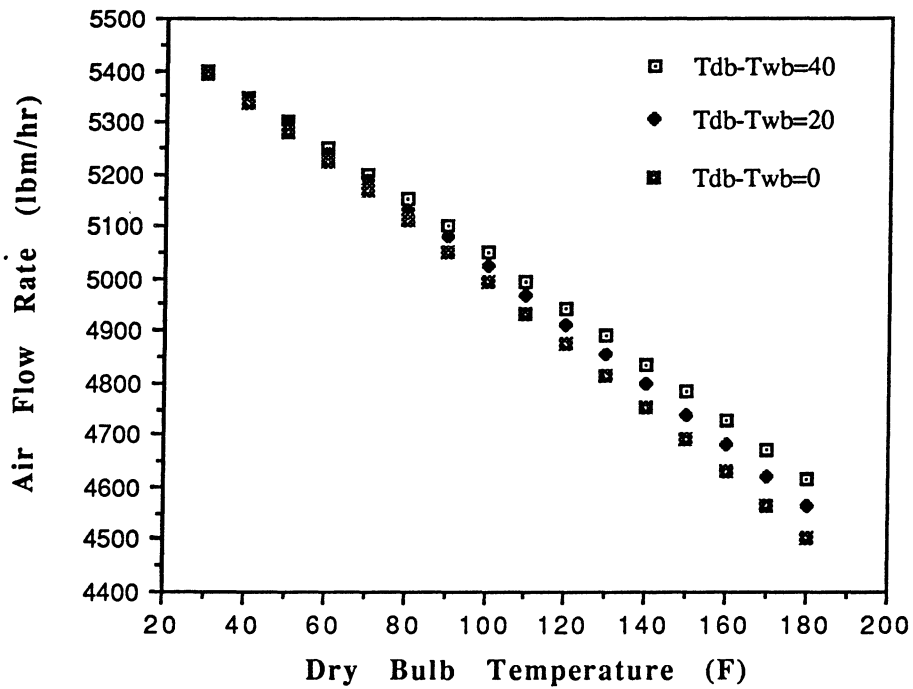


FIGURE 5.5
AIR FLOW RATE VARIATION WITH WET BULB TEMPERATURE

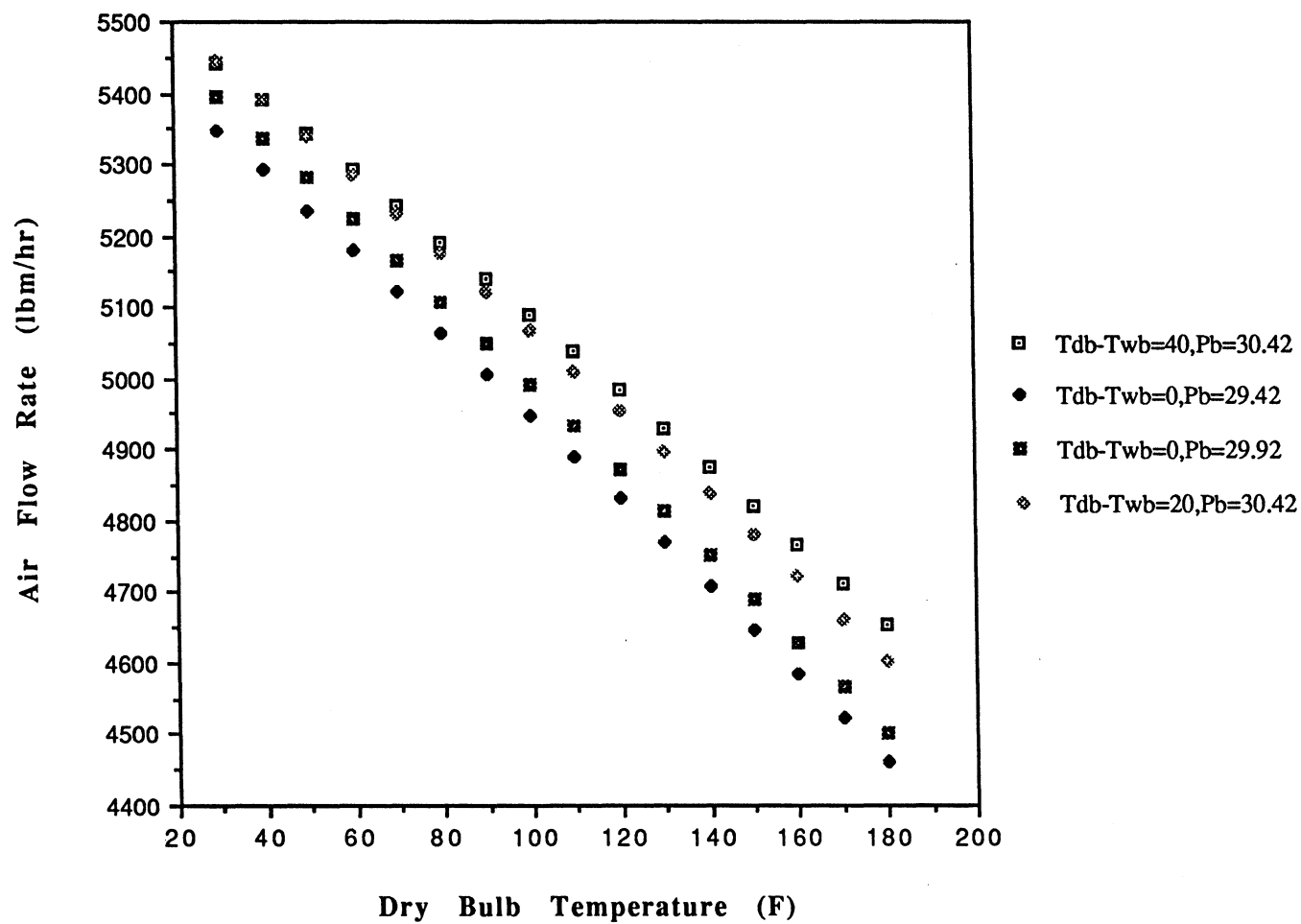


FIGURE 5.6
AIR FLOW RATE VARIATION
WITH BAROMETRIC PRESSURE AND WET BULB TEMPERATURE

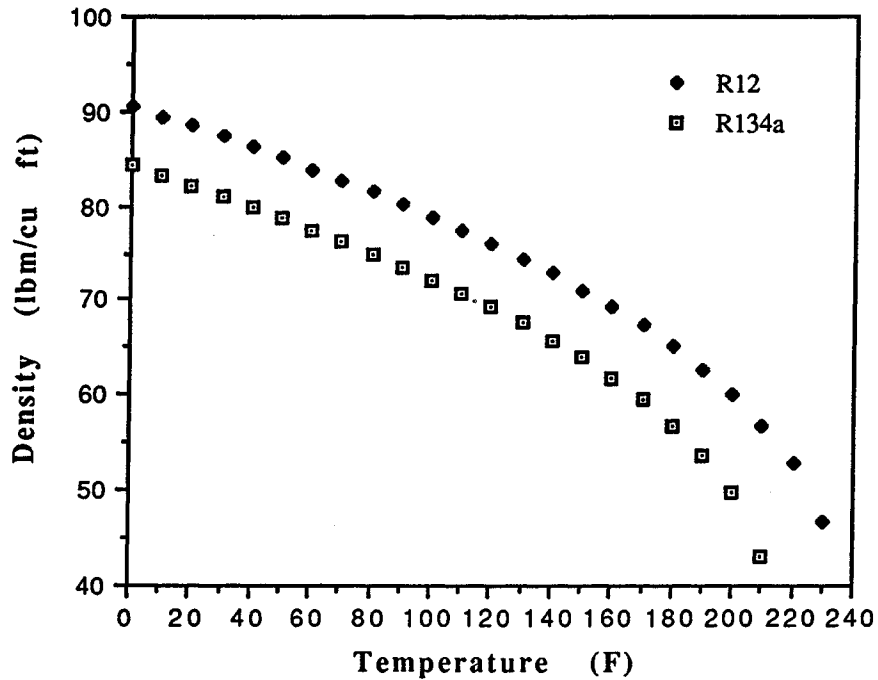


FIGURE 5.7
SATURATED LIQUID DENSITY
OF R12 & R134A VS TEMPERATURE

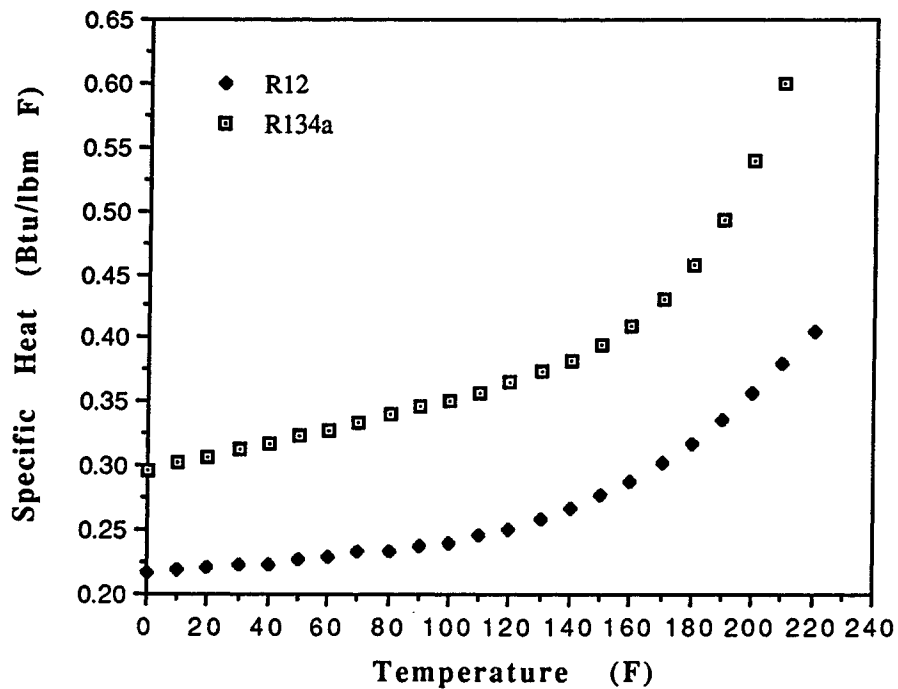


FIGURE 5.8
SATURATED LIQUID SPECIFIC HEAT
OF R12 & R134A VS TEMPERATURE

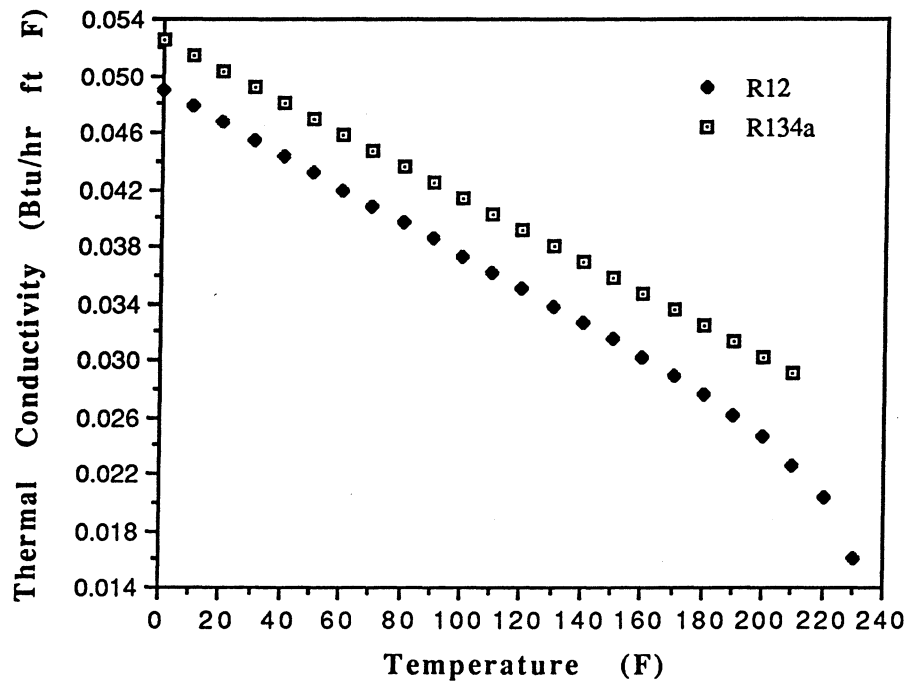


FIGURE 5.9
SATURATED LIQUID THERMAL CONDUCTIVITY
OF R12 & R134A VS TEMPERATURE

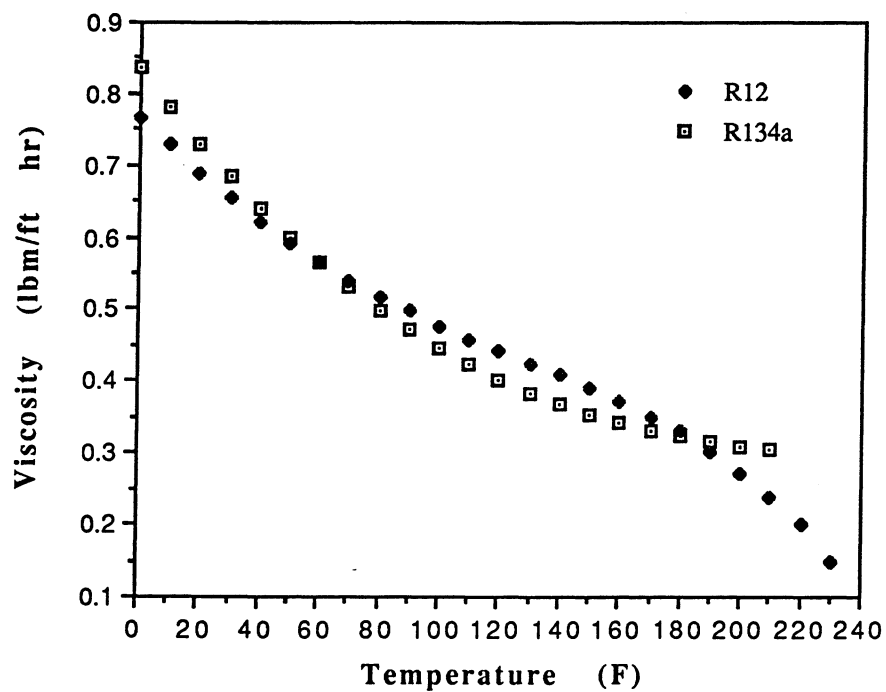


FIGURE 5.10
SATURATED LIQUID VISCOSITY
OF R12 & R134A VS TEMPERATURE

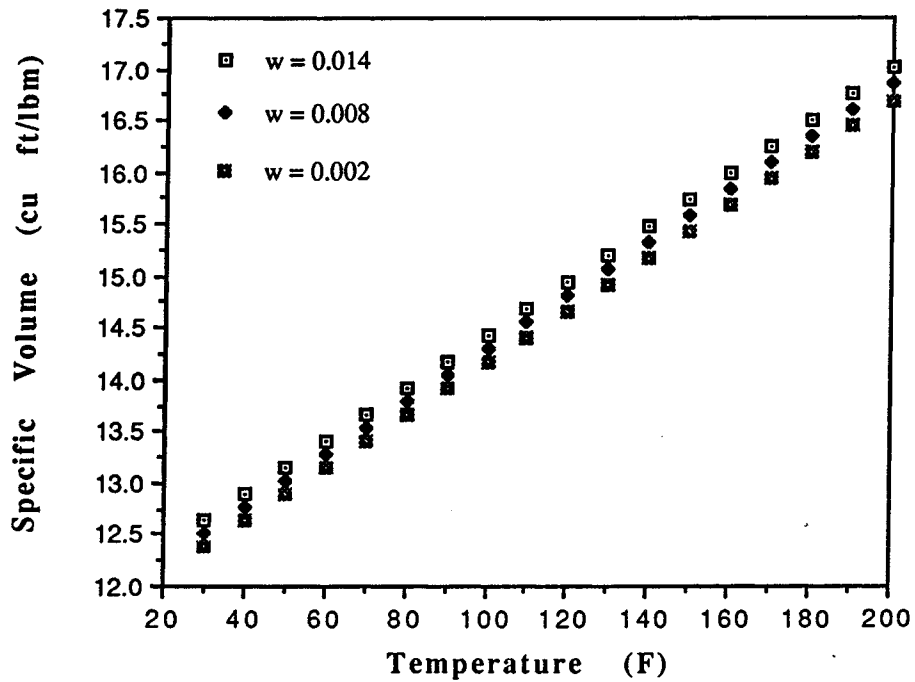


FIGURE 5.11
SPECIFIC VOLUME VS TEMPERATURE FOR AIR

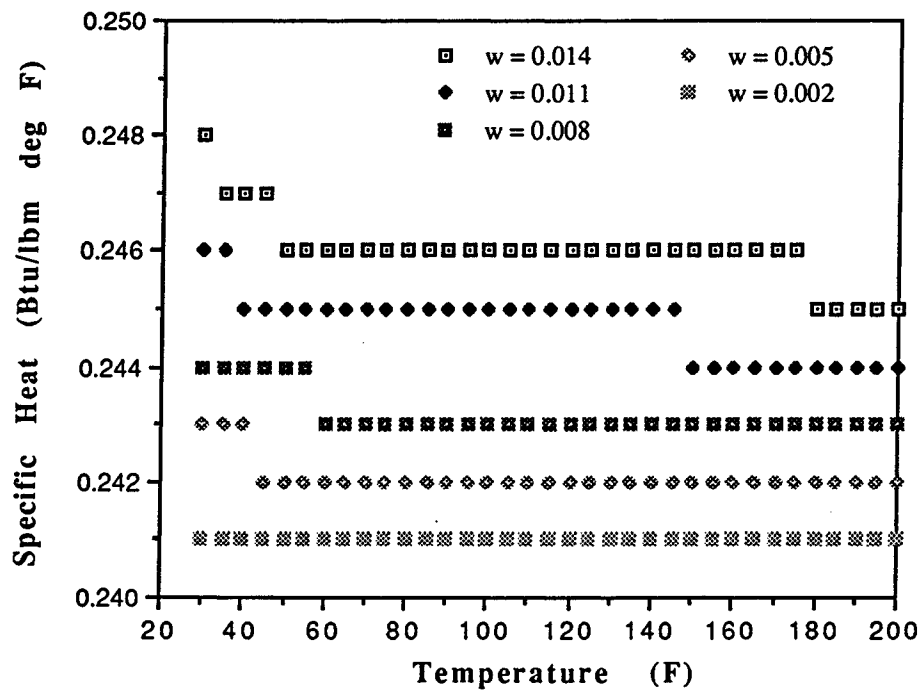


FIGURE 5.12
SPECIFIC HEAT VS TEMPERATURE FOR AIR

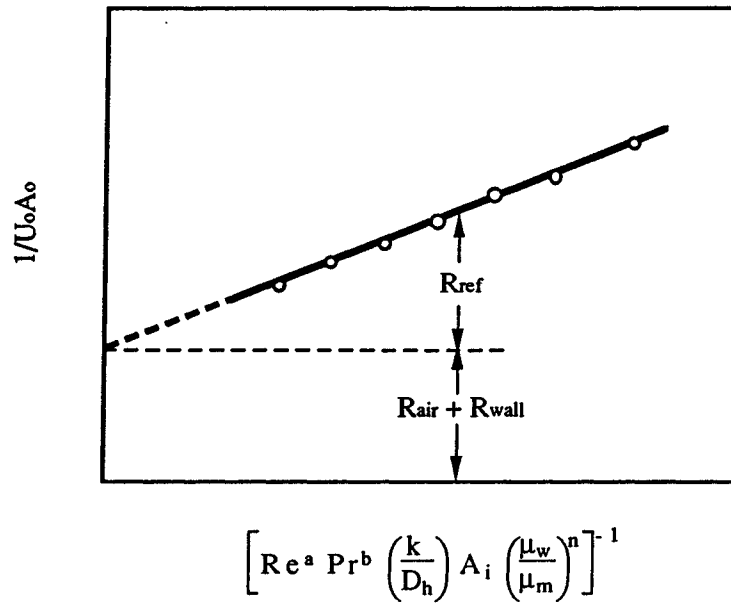


FIGURE 5.13
MODIFIED WILSON PLOT TECHNIQUE
FOR AIR-COOLED CONDENSER
WITH SINGLE-PHASE REFRIGERANT FLOW

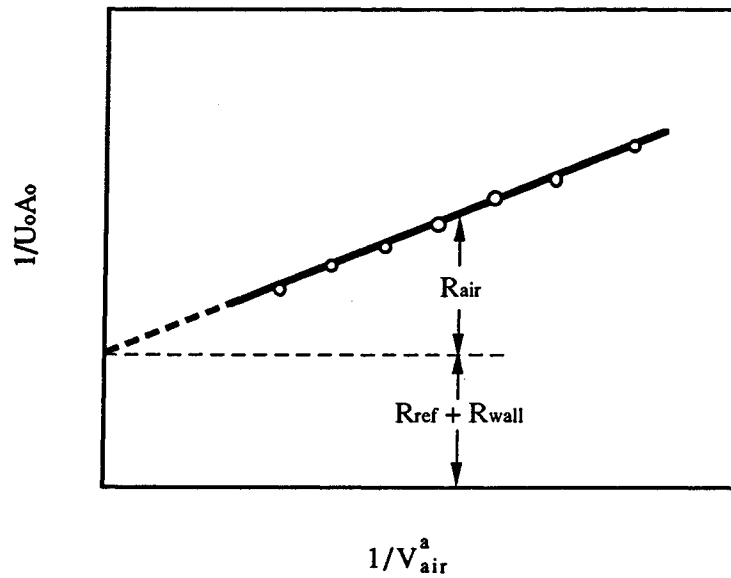


FIGURE 5.14
MODIFIED WILSON PLOT TECHNIQUE
FOR AIR-COOLED CONDENSER
WITH TWO-PHASE REFRIGERANT FLOW

TABLE 5.1
AIR FLOW RATE COMPUTATION - ENGLISH UNIT SYSTEM

Parameter	Formula	Units
Sat'd Vapor Pressure of Air at Fan Inlet	$p_e = 0.000296 T_{wb}^2 - 0.0159 T_{wb} + 0.41$	inches Hg
Partial Vapor Pressure of Air at Fan Inlet	$p_p = p_e - p_b \left(\frac{T_{db} - T_{wb}}{2700} \right)$	inches Hg
Density of Air at Fan Inlet	$\rho_n = \frac{70.73001 (p_b - 0.378 p_p)}{R(T_{db} + 459.7)}$	$\frac{lb_m}{ft^3}$
Density of Air at Nozzle Inlet	$\rho_{in} = \rho_{fan} \left(\frac{P_{in} + 13.63 P_b}{13.63 P_b} \right)$	$\frac{lb_m}{ft^3}$
Alpha Ratio	$\alpha = \frac{P_{s2} + 13.63 P_b}{P_{s1} + 13.63 P_b}$	Dimensionless
Expansion Factor	$Y = 1 - 0.548 (1 - \alpha)$	Dimensionless
Reynolds Number	$Re = 1,363,000 D_{noz} \sqrt{\Delta P \rho_{in}}$	Dimensionless
Nozzle Discharge Coefficient	For $L/D = 0.5$ $C_{noz} = 0.9986 + \frac{7.006}{\sqrt{Re}} + \frac{134.6}{Re}$ For $L/D = 0.6$ $C_{noz} = 0.9986 + \frac{6.688}{\sqrt{Re}} + \frac{131.5}{Re}$	Dimensionless
Volume Flow Rate of Air	$Q = 1096 Y (\Sigma C A_{noz}) \sqrt{\Delta P / \rho_{in}}$	$\frac{ft^3}{min}$
Mass Flow Rate of Air	$m = 60 \rho_o Q$	$\frac{lb_m}{hr}$

TABLE 5.2
AIR FLOW RATE COMPUTATION - SI UNIT SYSTEM

Parameter	Formula	Units
Sat'd Vapor Pressure of Air at Fan Inlet	$p_e = 0.00325 T_{wb}^2 - 0.0186 T_{wb} + 0.692$	kPa
Partial Vapor Pressure of Air at Fan Inlet	$p_p = p_e - p_b \left(\frac{T_{db} - T_{wb}}{1500} \right)$	kPa
Density of Air at Fan Inlet	$\rho_n = \frac{1000 (p_b - 0.378 p_p)}{R(T_{db} + 273.2)}$	$\frac{kg}{m^3}$
Density of Air at Nozzle Inlet	$\rho_{in} = \rho_{fan} \left(\frac{P_{in} + 1000 P_b}{1000 P_b} \right)$	$\frac{kg}{m^3}$
Alpha Ratio	$\alpha = \frac{P_{s2} + 1000 P_b}{P_{s1} + 1000 P_b}$	Dimensionless
Expansion Factor	$Y = 1 - 0.548 (1 - \alpha)$	Dimensionless
Reynolds Number	$Re = 70,900 D_{noz} \sqrt{\Delta P \rho_{in}}$	Dimensionless
Nozzle Discharge Coefficient	For L/D = 0.5 $C_{noz} = 0.9986 + \frac{7.006}{\sqrt{Re}} + \frac{134.6}{Re}$ For L/D = 0.6 $C_{noz} = 0.9986 + \frac{6.688}{\sqrt{Re}} + \frac{131.5}{Re}$	Dimensionless
Volume Flow Rate of Air	$Q = Y (\Sigma C A_{noz}) \sqrt{2 \Delta P / \rho_{in}}$	$\frac{m^3}{sec}$
Mass Flow Rate of Air	$m = \rho_o Q$	$\frac{kg}{hr}$

TABLE 5.3
BULK MEAN FLUID TEMPERATURE DETERMINATION

$C_{\max} = \text{hot fluid, } C_{\min} = \text{cold fluid}$	$C_{\max} = \text{cold fluid, } C_{\min} = \text{hot fluid}$
$\frac{C_{\min}}{C_{\max}} < 0.5$	
$T_{h,m} = \frac{1}{2} (T_{h,i} + T_{h,o})$ $T_{c,m} = T_{h,m} - \Delta T_{lm}$ $\Delta T_{lm} = \frac{(T_{h,m} - T_{c,i}) - (T_{h,m} - T_{c,o})}{\ln[(T_{h,m} - T_{c,i})/(T_{h,m} - T_{c,o})]}$	$T_{c,m} = \frac{1}{2} (T_{c,i} + T_{c,o})$ $T_{c,m} = T_{h,m} + \Delta T_{lm}$ $\Delta T_{lm} = \frac{(T_{h,i} - T_{c,m}) - (T_{h,o} - T_{c,m})}{\ln[(T_{h,i} - T_{c,m})/(T_{h,o} - T_{c,m})]}$
$\frac{C_{\min}}{C_{\max}} \geq 0.5$	
$T_{h,m} = \frac{1}{2} (T_{h,i} + T_{h,o})$	$T_{c,m} = \frac{1}{2} (T_{c,i} + T_{c,o})$

* For an air-cooled condenser, the refrigerant is the "hot" fluid and the air is the "cold" fluid.

CHAPTER 6

EXPERIMENTAL TESTING

Baseline experimental data with oil-free, refrigerant CFC-12 has recently been collected and analyzed for one condenser test section as part of ACRC Project SP03. Several different steady-state experiments were performed at a wide range of air-side and refrigerant-side conditions. Testing was based on either single phase or two phase refrigerant flowing inside of the condenser tubes. The main purpose of these experiments was to establish the individual heat transfer resistances for the finned tube condenser, in particular the air-side resistance, in addition to the dimensionless j-factors. In addition, this baseline testing assists in identifying the accuracy, repeatability, reproducibility, and confidence level associated with the experimental data collected.

This chapter summarizes the condenser experimental test envelope and analysis techniques employed for establishing the individual resistances to heat transfer. In addition, in the last section of this chapter, a discussion of the experimental test envelope developed for steady-state, condenser performance testing at typical mobile air conditioning and household refrigerator/freezer systems is documented.

6.1. Test Condenser

The heat exchanger used for baseline testing is a finned-tube automotive condenser as manufactured by Harrison Radiator of Lockport, New York. Harrison Radiator is a division of General Motors Company. In general, the automotive condenser is all-aluminum construction with two rows of wavy finned, 3/8-inch (9.53 mm) nominal diameter tubes arranged in a staggered configuration. Other than the tube inside diameter, given as 0.331 inches (8.41 mm) based on personal communications with one of the Company's product engineers [34], no dimensional data or surface basic data was available from GM/Harrison Radiator since it was deemed proprietary information.

Field measurements were recorded to determine the condenser geometrical properties. These results are shown in Table 6.1. All of the dimensions shown including the fin spacing, fin thickness, transverse and longitudinal tube pitches and tube outside diameter are based on the average of ten (10) individual measurements taken at different locations on each side of the condenser with a micrometer.

According to Czajkowski [34], although the tube wall thickness for the 3/8 inch (9.53 mm) nominal diameter tube condenser is approximately 0.028 inches (0.71 mm), the suggested value for the tube inside diameter, 0.331 inches (8.41 mm), is based on adjustments incorporated due to the mechanical expansion processes employed when attaching the fins. Similarly, for the GM/Harrison Radiator, 1/4 inch (6.35 mm) nominal diameter tube condenser, the nominal tube wall thickness is 0.0165 inches (0.42 mm) and the recommended value for the tube inside diameter is 0.230 inches (5.84 mm).

From its inlet port to the exit port, the GM/Harrison Radiator automotive condenser coil consists of 15 tube passes in each tube row and a total of 30 return bends. In order to best approximate the temperature of the refrigerant flowing within the condenser, thermocouples are mounted on each of the tube return bends, in addition to the inlet and outlet connections, for a total of 32 condenser thermocouple measurements. These thermocouples are 30 gage, Type T, copper-constantan with teflon insulation and with beaded measurement junctions formed by an electric arc welder. Thermocouple wire, purchased from Gordon Temperature Measurement of Richmond, Illinois, and is identical to that wire utilized for the duct air temperature measurements, described in Section 3.2.6.

Individual thermocouple beads are attached to the condenser aluminum tube surface via thermally conductive epoxy. Thermocouple leads are routed immediately adjacent and in contact with the condenser aluminum tube surface for approximately 1.5 inches (3.8 cm) so that a portion of the wire in the vicinity of the measurement junction is at the same surface temperature. Further, a 1/4-inch (6.35 mm) thick piece of Armaflex insulation is also placed over the thermocouple measurement junction and lead wires.

Epoxy was selected as the most appropriate fastening means after continual attempts with soldering to the condenser aluminum surface failed. The epoxy utilized is Omegabond™ 101, as produced by Omega Engineering of Connecticut. It is specifically designed for attaching beaded wire thermocouples to metal surfaces. The two-part adhesive possesses a thermal conductivity of 7.2 Btu in./hr ft² °F (1.04 W/m K) and a maximum continuous temperature rating of 275 °F (136 °C). The Omega epoxy cures in four hours at room temperature. Other epoxies and greases that have higher thermal conductivities were evaluated; however, most of these are not a permanent type of fastening means or they require as much as 4 to 6 times the curing time as the Omegabond™ 101.

The primary purpose of the thermocouple measurements is to detect approximate temperature changes of the refrigerant flowing through the condenser. In particular, for two phase testing, the thermocouple measurements provide a means for determining the length of the coil in the superheating, condensing and subcooling regions. However, since the distance between individual return bends is approximately 24 inches, there is some error associated with exactly where a change in the refrigerant flow region occurs. Specifically, for the GM/Harrison Radiator condenser with thermocouples located at 16 locations per tube row, changes in flow regions for regions may only be estimated within 1/16 or 6.25 percent by analyzing the experimental results.

The thermocouples are mounted on the tube bends rather than the finned tube surface for a couple of reasons. First, due to the close condenser fin spacing, it is extremely more difficult to secure and insulate the thermocouples on the tube surface between the return bends (i.e., between adjacent fins) rather than on the bends themselves. Second, since their actual mounting location is outside of the airstream, the thermocouples will not interfere with the condenser heat transfer performance.

6.2. Single Phase Testing

A total of seven (7) different sets of experimental data, including not less than thirteen (13) data points per set, was gathered during testing of the full condenser apparatus with single phase refrigerant flowing within the condenser tubes and air in crossflow over the extended surface. As prescribed by Shah [98] in his presentation of the modified Briggs and Young procedure, experimental testing required maintaining the condenser air-side conditions constant (i.e., the air flow rate and the bulk mean air temperature) while varying the refrigerant-side flow rate from a minimum to a maximum for a fixed amount of heat input. Subsequently, the inlet and exit conditions of the refrigerant also varied as its mass flow rate was altered. It should be noted that a constant heat input corresponds to a fixed condenser heat rejection capacity during steady state operation.

In order to establish a full range of test conditions, single phase experiments were performed at three different air flow rates ranging from approximately 600 to 1500 cfm (0.283 to 0.708 m³/sec) and two different condenser inlet air temperatures of 80 and 100 °F (26.7 and 37.8 °C). The range of volumetric air flow rates corresponds to an air-side Reynolds number range of roughly 168 to 392. In addition, some test data was duplicated in order to establish repeatability in the results. Table 6.2 contains a summary of the different test conditions employed.

For the test conditions identified in Table 6.2, the corresponding refrigerant-side Reynolds numbers range from approximately 1.18×10^4 to 5.74×10^4 and Prandtl numbers range from about 3.10 to 3.30. In addition, refrigerant temperatures at the condenser inlet vary from 112 to 192 °F (44.4 to 88.9 °C).

Based on experimental data collected, overall condenser UA values were calculated by the appropriate effectiveness-NTU relations presented in Section 5.9 and individual heat transfer resistances were determined in accordance with the modified Wilson Plot procedures introduced in Chapter 4 and Section 5.11. The primary purpose of the experimental procedure was to eliminate the refrigerant-side resistance so that the resulting modified Wilson Plots provided the air-side thermal resistance directly (i.e., the y-intercept). Since the refrigerant flow varied for each test point, an inside or refrigerant-side resistance exists for each data point. These resistances may be easily determined by deducting the

appropriate total UA value by an amount equal to the air-side thermal resistance, as documented in Section 5.11.

The main advantage associated with single phase testing and analysis is that the majority of published literature to-date concerning the utilization of the original and modified Wilson Plot techniques is based on testing with single phase fluids on both sides of a heat exchanger. Therefore, the experimental methodologies and data reduction techniques associated with single-phase testing and the incorporation of Wilson Plots are rather well established. Another advantage of single-phase testing is that the idealizations associated with the effectiveness-NTU computations are more directly applicable.

There are two disadvantages with respect to single phase testing. First, compared to two phase refrigerant flow, testing range limitations are often a result of high system operating pressures. Second, since the experimental procedures require variation in the refrigerant-side conditions and fixed air-side conditions and a single value of the air-side heat transfer coefficient corresponds to each individual air flow rate, more time is necessary to collect and analyze single phase data for the purposes of obtaining a single air-side resistance value.

One additional item shown in Table 6.2 is the condenser heat rejection rate for each set of single phase refrigerant flow tests. This heat rejection rate shown is equal to the heat gained by the air stream. The heat transfer from the refrigerant was consistently less than the air-side heat transfer rate during all testing and its magnitude may be determined by deducting the air-side heat transfer rate by an amount equivalent to the average percent error. The rates of heat transfer tabulated range from 6700 to 12,400 Btu/hr (0.196 to 0.363 kW) and the average percent error varies from 1.8 to 13.8 percent. In general, the average percent error in the fluid heat transfer rate increases as the condenser air flow rate decreases and in addition, as the condenser heat rejection capacity increases. This is most likely due to the larger degree of temperature stratification that exists at these lower air flow rates and corresponding condenser air face velocities.

6.3. Two Phase Testing

The main function of testing with two phase refrigerant flow on the inside of the condenser tubes was to validate the single phase results. However, in contrast to single phase experiments, procedures involved maintaining the refrigerant-side conditions the same (i.e., the refrigerant flow rate and the bulk mean refrigerant temperature) while varying the air flow rate over a wide range in order to eliminate the air-side resistance. In this case, the resulting Wilson Plots provide the inside thermal resistance directly (i.e., the y-intercept). Further, in order to determine the air-side resistance corresponding to each air flow rate, the overall resistance must be deducted by an amount equal to the inside annular flow resistance.

A summary of the conditions employed during experiments with two phase refrigerant flow through the condenser is shown in Table 6.3. A total of four different tests were conducted at bulk mean air temperatures ranging from roughly 82 to 110 °F (27.8 to 43.3 °C). Condenser air flow rates varied from 695 to 1355 cfm (0.328 to 0.640 m³/sec). This range of air flow rates corresponds to an air-side Reynolds number range of approximately 255 to 505.

Some of the advantages of single phase testing have previously been documented. The primary advantage of two phase testing is that since the air flow rates are varied during experimental testing, a single resulting Wilson Plot provides more than one value of the air-side thermal resistance. Essentially, for the same time period, single phase experimental testing may lead to only one value of the air-side thermal resistance; whereas, two phase testing may result in at least three values.

On the other hand, due to its complexity - namely, the existence of three different flow regions of superheating, condensing and subcooling that each possess a different UA value and that must each be accounted for only a limited amount of information is published which considers two phase flow on one side of the heat exchanger and applicable Wilson Plot techniques. Ideally, the UA computation must be exercised for each flow region in order to obtain a representative value.

In this research analysis, the UA value computation was idealized by assuming that the heat transfer resistance or thermal conductance for the two phase refrigerant flow region (i.e., the condensing region) is a constant and uniformly distributed throughout the entire condenser, including the single phase desuperheating and subcooling regions. During experimental testing, precautions were exercised in order to minimize the amounts (i.e., the tube lengths) of desuperheating and subcooling in the condenser and to make the foregoing assumption more applicable.

The range of condenser heat rejection rates during experimental testing with two phase refrigerant flow is approximately 23,000 to 56,000 Btu/hr (0.674 to 1.64 kW) and is shown in Table 6.3. The values indicated are for the full condenser, not just the two phase flow region. Similar to the single phase testing, the heat transfer rate shown is applicable to the air-side heat transfer only. In order to evaluate the refrigerant-side heat transfer, the range of heat transfer indicated for each data set must be deducted by an amount equivalent to the average percent error, also given in Table 6.3. In comparison with the heat transfer rates encountered with single phase refrigerant flow, the rates of heat transfer for two phase refrigerant flow experiments are 2 to 5 times larger. As a result and since the air flow rates are also consistently higher in most instances, the average percent error associated with the energy balances is much less.

6.4. Condenser Steady-State Test Envelope

This section examines the experimental test envelope established for current and future steady-state performance testing of individual automotive and household refrigerator/freezer system condenser coils utilizing refrigerants CFC-12 and HFC-134a. First, however, a brief discussion of the typical refrigerant-side and air-side operating conditions in mobile air conditioning and household refrigerator/freezer systems is documented.

6.4.1. Conditions in Mobile Air Conditioning Condensers

The refrigeration capacity of an automotive air conditioning system must be adequately sized in order to provide large amounts of cooling and reduce the vehicle interior temperature to comfortable levels in a short period of time. Generally, the most critical operating condition occurs at engine idle under high load conditions. These conditions are commonly referred to as the "pull-down" period. The air conditioning system must also consistently maintain the selected temperature and comfort level at reasonable humidity during all operating conditions and in all environments.

Since space, weight and power consumption are considered premiums in mobile air conditioning systems and generally directly related to cost, the condenser package and performance design must be established carefully by evaluating all the known and predicted factors. For example, high head pressures as a result of an undersized condenser will increase internal pressure drop, reduce compressor capacity and increase power consumption. Conversely, an oversized condenser may produce inadequate cooling by producing condensing temperatures significantly less than the engine compartment temperature which in turn may result in boiling of the refrigerant in the liquid lines passing through the engine compartment prior to entering the evaporator.

Based on a survey of the literature available [11,12] and responses from a questionnaire sent to several of the automotive condenser coil manufacturers; some typical information regarding condenser operating and geometrical characteristics was gathered. Specifically, it was discovered that automotive condenser heat rejection capacities may range anywhere from 6000 to 54000 Btu/hr (1.8 to 15.8 kW) although most systems are commonly designed for a capacity between 20,000 to 36000 Btu/hr (5.2 to 10.5 kW). For the refrigerant-side, flow rates often range from approximately 50 to 900 lb_m/hr (0.0063 to 0.1134 kg/sec). In addition, condenser inlet and exit conditions may contain anywhere from 0 to 100 °F (-17.8 to 55.6 °C) degrees of superheat and from 5 to 30 °F (2.8 to 16.7 °C) degrees of subcooling, respectively, based on condensing pressures of 80 to 400 psig (552 to 2758 kPa). Refrigerant condensing temperatures are generally 10 to 30 °F (5.6 to 16.7 °C) higher than the entering ambient air temperature. Air-side design conditions typically involve ambient air temperatures from roughly 30 to 110 °F (-1.1 to 43.3 °C) and fan air flow rates of 300 to 3000 cfm (0.142 to 1.416

m³/sec). Fans are normally located downstream of the condenser in order to "draw-thru" cooling air without the addition of motor heat. Table 6.4 contains a summary of the information solicited from three automotive condenser coil manufacturers concerning their own internal testing standards and condenser geometries. These manufacturers include Ford Motor Company, Acustar-Dayton, which produces condensers for Chrysler automobiles, and Harrison Radiator, a division of General Motors Company. Unfortunately, slightly more than 50 percent of the information requested was considered non-proprietary. Any "N.A." shown in Table 6.4 refers to an item that was "not answered" or considered proprietary information by the particular condenser manufacturer.

6.4.2. Conditions in Household Refrigerator/Freezer Condensers

In general, refrigeration systems for household refrigerator/freezer systems must be designed to maintain desired food and ice preservation temperatures of 32 to 39 °F (0 to 4 °C) and 0 to 8 °F (-18 to 8 °C), respectively. As a result of rising energy costs and government regulatory actions, overall appliance efficiency has become one of the most critical parameters affecting system design. Control of noise and vibration are also important, especially with the need for refrigerating compressors and forced-air cooling fans having higher efficiency and capacity. In particular, the ability to provide sufficient heat dissipation at peak-load conditions, adequate storage volume that prevents high discharge pressures during pulldown, good refrigerant drainage to minimize off-cycle losses and the time required for equalization of system pressures, and an easily cleanable external surface to retard dust and lint accumulation, are four (4) of the most important design factors for forced-draft and natural draft condensers.

According to the latest edition of the ASHRAE Equipment Volume [11], and telephone conversations with engineers Mr. Bob Cushman of Amana Refrigeration and Mr. Joe Ziegler of General Electric, information was collected on the actual performance and geometry of household refrigerator/freezer condensers. Condenser heat rejection capacities average approximately 800 to 1600 Btu/hr (0.23 to 0.47 kW) with most household sizes rarely exceeding 2000 Btu/hr (0.59 kW). In addition, refrigerant mass flow rates typically range from as low as 8 lb_m/hr to as high as 20 lb_m/hr (0.001 to 0.0025 kg/sec); whereas, air flow rates range from roughly 80 to 250 cfm (0.0378 to 0.118 m³/sec). Standard rating

conditions of household refrigerator/freezers are expressed at a condenser saturation temperature of 130 °F (54.4 °C) and an ambient air temperature of 70 °F (21.1 °C).

ANSI/AHAM Standard HRF-1-1986 [2] describes in detail experimental procedures for determining the performance of household refrigerator/freezer systems, including the testing at ambient air conditions of 55 °F, 70 °F, 90 °F, and 110 °F (12.8 °C, 21.1 °C, 32.2 °C, and 43.3 °C, respectively).

6.4.3. Condenser Test Plan

As a result of the aforementioned data from literature searches and personal communications with automotive and household refrigerator/freezer condenser manufacturers, the experimental test envelopes were carefully formulated. Table 6.5 outlines the ranges of individual test parameters for steady state performance, condenser baseline testing with CFC-12. Table 6.6 provides the ranges of each test parameter for more in-depth condenser testing with the alternative refrigerant HFC-134a.

In setting up the condenser steady state test plan, the following geometric and fluid flow parameters were selected as the independent variables:

1. Application
(e.g., automotive or household refrigerator)
2. Condenser Geometry
(e.g., circular or flat tubes, plain or wavy fins, etc.)
3. Refrigerant Type
(e.g., CFC-12 or HFC-134a)
4. Air Mass Flow Rate
5. Refrigerant Mass Flow Rate
6. Inlet Air Temperature
7. Inlet Degrees of Refrigerant Superheat

Essentially, for a given condenser and refrigerant type, items 4 thru 7 are the independent parameters varied during experimental testing. Originally, the refrigerant condensing temperature was chosen as the independent flow variable in lieu of the inlet degrees of refrigerant superheat at the condenser inlet. However, it is much easier to control the inlet degrees of superheat by adjusting the refrigerant heater, since the refrigerant condensing temperature is actually influenced by the flow rates and inlet conditions associated with each fluid.

Although some of the numerical values for the parameters shown in Tables 6.4 and 6.5 are outside the ranges stated in Sections 6.4.2 and 6.4.3, the main reason for their inclusion is based on the requests from several participating members of the ACRC Industrial Advisory Board.

TABLE 6.1
TEST CONDENSER GEOMETRY DATA

Core Dimensions, W x H x D	23.125 inch x 16.75 inch x 1.625 inch	587 mm x 425 mm x 41 mm
Nominal Tube Diameter, D	3/8 inch	9.53 mm
Outside Tube Diameter, d_o	0.400 inch	10.2 mm
Inside Tube Diameter, d_i	0.331 inch	8.41 mm
No. of Tube Rows, N_r	2	2
No. of Tube Passes, N_p	15	15
Tube Length per Row, L	35 ft	10.7 m
Fin Density	18 fins/inch	709 fins/mm
Fin Thickness, δ	0.005 inch	0.127 mm
Tranverse Tube Pitch, S_t	0.96 inch	24.4 mm
Longitudinal Tube Pitch, S_l	0.60 inch	15.2 mm
Frontal Area, A_{fr}	2.690 ft ²	0.2499 m ²
Minimum Free Flow Area, A_o	1.428 ft ²	0.1327 m ²
Hydraulic Diameter, D_h	0.0055 ft	0.00168 m
Total Heat Transfer Area, A_{tot}	140.8 ft ²	13.08 m ²
Finned Surface Area, A_f	134.6 ft ²	12.50 m ²
Free Flow Area/Frontal Area, σ	0.53	0.53
Heat Transfer Area/Tot. Volume, α	386 ft ² /ft ³	1265 m ² /m ³
Fin Area/Total Area	0.96	0.96

TABLE 6.2
CLASSIFICATION OF EXPERIMENTAL DATA (SINGLE PHASE)

Case No.	No. of Data Points	Condenser Test Conditions						
		Air Flow (cfm)	Air Velocity (ft/min)	Inlet Air Temp (°F)	Bulk Mean Air Temp (°F)	Heat Exchange (Btu/hr)	Heat Xfer Average Pct. Error	R12 Flow Range (lb _m /hr)
A-1	14	1506	560	79.7	83.4	12,000	1.8%	510-1100
B-1	16	1066	396	80.0	85.2	12,400	5.8%	450-1040
C-1	21	1050	390	79.9	85.0	11,800	5.6%	390-1070
D-1	13	1025	381	79.7	84.1	10,000	4.2%	450-1060
E-1	50	1050	390	79.9	84.8	11,500	5.3%	390-1070
F-1	15	1073	399	110.1	114.6	10,900	12.1%	495-1030
G-1	20	680	253	80.7	85.4	7900	13.8%	415-1100
H-1	13	570	212	79.1	84.1	6700	10.7%	350-1060

TABLE 6.3
CLASSIFICATION OF EXPERIMENTAL DATA (TWO PHASE)

Case No.	No. of Data Points	Condenser Test Conditions						
		Air Flow Range (cfm)	Air Velocity (ft/min)	Inlet Air Temp (°F)	Bulk Mean Air Temp (°F)	Heat Exchange (Btu/hr)	Heat Xfer Average Pct. Error	R12 Flow Rate (lb _m /hr)
A-2	9	790-1270	294-472	89.6	110.3	31,100-47,100	1.4%	952
B-2	8	755-1140	281-425	55.4	82.8	52,000-55,400	0.9%	920
C-2	5	695-1170	258-435	73.3	86.0	23,100-24,400	2.1%	410
D-2	8	770-1355	286-504	60.8	82.5	48,100-50,000	1.2%	945

TABLE 6.4
RESULTS OF PERSONAL COMMUNICATIONS WITH AUTOMOTIVE CONDENSER MANUFACTURERS

CONDENSER MANUFACTURER			
Company Name	Ford Motor Company	Acustar-Dayton	Harrison Radiator,
Location	Detroit, Michigan	Dayton, Ohio	Division of GM
Contacts	J. Grimes/B. Diedrich	K. Fouts/P. Bathla	R. Flower/R. Shah
Phone	(313) 323-6905	(513) 224-2411	(716) 439-3023
AIR-SIDE CHARACTERISTICS			
Ambient Air Temperatures	110 °F	30-110 °F	80-120 °F
Air Flow Rates	800-2000 cfm	300-3000 cfm	1800-13,800 lb _m /hr
Maximum Face Velocity	1000 fpm	N.A.	N.A.
Maximum Pressure Drop	Proprietary	2 inches w.g.	2 inches w.g.
Fan Data	Draw-thru	Draw-thru.	Draw-thru
REFRIGERANT-SIDE CHARACTERISTICS			
Inlet Conditions	180-200 °F	80-400 psig & 0-100 °F superheat	100-180 °F
Exit Conditions	N.A.	80-400 psig & 5-30 °F superheat	N.A.
Refrigerant Flow Rates	200-550 lb _m /hr	50-500 lb _m /hr	180-900 lb _m /hr
Maximum Pressure Drop	Proprietary	N.A.	As minimum as possible
Refrigerants Tested	R-12	R-12 and Alternatives	R-12 and Alternatives
CONDENSER COIL			
Heat Rejection Capacity	10,000-35,000 Btu/hr	N.A.	6000-54,000 Btu/hr
Design Pressure (see below)	Per SAE J639	Per SAE J639	Per SAE J639
Geometry Type	Tube & Fin	Tube & Fin	Tube & Fin
Core Dimensions or Size	N.A.	36"(W) x 30"(H) x 3"(D)	N.A.
Test Mounting Arrangement	In-vehicle	N.A.	N.A.
Refrigerant Connections	Package dependent	N.A.	Package dependent
Condenser Surface Temps	No	N.A.	Mount TC's on tube bends

* At least two times the system pressure relief valve setting per SAE Recommended Practice J639.

TABLE 6.5
PROPOSED STEADY STATE TEST PLAN
REFRIGERANT CFC-12 BASELINE DATA

Coil Application	Mobile Air Conditioning			Household Refrigerators/Freezers	
No. of Coils Tested	1			1	
Finned Tube Geometry	Circular Tubes			Circular Tubes	
Inlet Air Temperature (°F)	80	110	150	70	90
Refrigerant Inlet Degrees of Superheat (°F)	10	100		10	100
Air Flow Rate (cfm)	400	1000	1600	80	150
Refrigerant Mass Flow Rate (lb _m /hr)	300	600		8	18

TABLE 6.6
PROPOSED STEADY STATE TEST PLAN
REFRIGERANT HFC-134a DATA

Coil Application	Mobile Air Conditioning			Household Refrigerators/Freezers		
No. of Coils Tested	2			1		
Finned Tube Geometry	Circular Tubes		Flat Tubes	Circular Tubes		
Inlet Air Temperature (°F)	80	110	150	70	90	
Refrigerant Inlet Degrees of Superheat (°F)	10		100	10	100	
Air Flow Rate (cfm)	400	1000	1600	80	150	
Refrigerant Mass Flow Rate (lb _m /hr)	200	400	600	8	13	18

CHAPTER 7

EXPERIMENTAL RESULTS AND DISCUSSION

This chapter presents and discusses the air-side and refrigerant-side heat transfer data obtained as a result of the baseline experimental testing described in Tables 6.2 and 6.3. Unless otherwise noted, experimental results, tabulated values, and plotted data in this Chapter are presented in English units only.

7.1. Single Phase Results

In order to obtain the condenser air-side thermal resistance, experimental data based on single phase refrigerant CFC-12 flowing inside of the condenser tubes was first correlated by the form of the following modified Wilson Plot expression:

$$\frac{1}{U_o A_o} = \frac{1}{[C_1 \text{ Re}^a \text{ Pr}^{0.4} (k/D_h) A_i]} + C_2 \quad (7.1.1)$$

where the individual terms may be categorized as:

$$[\text{Re}^a \text{ Pr}^b (k/D_h) A_i]^{-1} = \text{independent variable}$$

$$1/U_o A_o = \text{dependent variable}$$

$$C_1, C_2, a = \text{unknown or arbitrary parameters}$$

As introduced in Chapter 4, the two unknown parameters, C_1 and C_2 , represent $1/(\text{slope})$ and the y -intercept, respectively, on a linear plot of the dependent variable versus the independent variable, i.e., a modified Wilson Plot. In order to determine these two parameters, as well as the unknown Reynolds number exponent, a nonlinear regression analysis was performed on each of the eight (8) sets of single phase refrigerant flow, experimental data categorized in Table 6.2. Further, utilizing fixed Reynolds number exponents of 0.7, 0.8 and 0.9, the parameter estimation process was repeated to evaluate the two remaining unknown terms, C_1 and C_2 in Equation (7.1.1). The results from these analyses are also summarized in the next Sections.

Essentially, the nonlinear regression procedure determines the most optimum values of the unknown parameters that best fit the given set of experimental data. A computer program, written in TrueBasic™ was developed to automate the nonlinear regression process. A summary of the nonlinear least squares (NLLS) procedure, including a computer source code listing, is contained in Appendix B.

In the following sections, results of the parameter estimation process for different conditions are given in tabulated form. Two of the columns in each of these tables are labeled "ΣR" and "Error Range", respectively. The sum of the residuals squared column, labeled "ΣR", is indicative of the accuracy of the resulting linear fit through the given set of data and is a suitable tool for comparing data different experimental data sets. In this analysis, it is defined as:

$$\Sigma R = [(1/UA)_{\text{exp}} - (1/UA)_{\text{NLLS}}]^2 \times 10^{10} \quad (7.1.2)$$

The "Error Range" column is indicative of the maximum positive and negative percent error between the (1/UA) values obtained experimentally and the (1/UA) values obtained from the nonlinear regression analysis.

7.1.1. NLLS Results Based on Unknown Reynolds No., Exponent

The results of the parameter estimation process based on an unknown Reynolds number exponent are shown in Table 7.1 for each set of experimental data. The table indicates a variation in the Reynolds number exponent 'a' from 0.55 to 1.007 and a variation in the air-side thermal resistance (C₂) from 3.9261E-04 to 1.5283E-03 hr ft² °F/Btu (6.9122E-05 to 2.6907E-04 m² K/W) for seven of the experimental data sets. Case "G-1" failed to converge to a solution. This is primarily due to the large amount of scatter or error associated with this experimental data compared to the other data sets. The inconsistencies in the results is demonstrative of the sensitivity associated with the experimental data and the nonlinear regression routine.

Experimental data for cases "B-1" thru "F-1" was obtained at approximately the same condenser volumetric air flow rate of 1050 cfm ($0.497 \text{ m}^3/\text{sec}$) with a variation of +2.2/-2.4 percent. This air flow rate corresponds to an air-side Reynolds number of 270 and a condenser face air velocity of 390 ft/min (1.981 m/sec). The average air-side thermal resistance for these five experiments conducted at approximately the same air flow rate is $7.0429\text{E-}04 \text{ hr ft}^2 \text{ }^\circ\text{F/Btu}$ ($1.2400\text{E-}04 \text{ m}^2 \text{ K/W}$). The variation from this mean value is +19.4/-44.3 percent. Assuming that Case "C-1" may be neglected because it has a much lower Reynolds number exponent and air-side thermal resistance compared to the other four data sets, the average heat transfer resistance increases to $7.8221\text{E-}04 \text{ hr ft}^2 \text{ }^\circ\text{F/Btu}$ ($1.3771\text{E-}04 \text{ m}^2 \text{ K/W}$) and the variation from this mean value is reduced to ± 7.5 percent.

Case "F-1" is applicable for a condenser inlet air temperature of $110 \text{ }^\circ\text{F}$ ($43.3 \text{ }^\circ\text{C}$) compared to the other four similar cases which are applicable for a condenser inlet air temperature of $80 \text{ }^\circ\text{F}$ ($26.7 \text{ }^\circ\text{C}$). Comparing the results for the two inlet air temperatures appears to indicate that air-side thermal resistance is not heavily influenced by temperature in the range of 80 to $110 \text{ }^\circ\text{F}$ (26.7 to $43.3 \text{ }^\circ\text{C}$).

Experimental data for case "H-1" was gathered at a condenser air flow rate of 570 cfm ($0.269 \text{ m}^3/\text{sec}$), which corresponds to an air-side Reynolds number of 210 and a condenser face air velocity of 212 ft/min (1.076 m/sec). As anticipated, comparison of the air-side thermal resistance for this lower air flow rate with those at the higher air flow rates of 1050 cfm ($0.496 \text{ m}^3/\text{sec}$) indicates that the resistance to heat transfer increases as the air flow rate is decreased.

Figures 7.1 through 7.6 graphically illustrate the correlation of experimental data sets by Equation (7.1.1) utilizing two different Reynolds number exponents; namely, an exponent of 0.8 and the optimum value of the exponent obtained from the nonlinear regression analysis and shown in Table 7.1. Results for an independent variable exponent of 0.8 is also shown since this condition corresponds to the original Wilson Plot procedure.

7.1.2. NLLS Results Based on Two (2) Unknown Parameters

Tables 7.2, 7.3, and 7.4 provide the results of the parameter estimation study for fixed Reynolds number exponents of 0.7, 0.8, and 0.9, respectively. The two unknown parameters in these tables are C_1 and C_2 . Figures 7.8 thru 7.15 graphically portray the linear plots of the dependent variable versus the independent variable for each experimental data set and for each Reynolds number exponent. By maintaining the Reynolds number exponent at a fixed value, another constraint is added to the parameter estimation process, but it is much easier to detect trends in the results as discussed herein.

The nonlinear regression results indicate that the Wilson Plot y-intercept or air-side thermal resistance is directly related to the Reynolds number exponent, increasing by 13.1 to 23.8 percent for a Reynolds number exponent increase from 0.7 to 0.8 and increasing by 9.2 to 15.3 percent for a Reynolds number exponent increase from 0.8 to 0.9. The results also show that as the Reynolds number exponent is increased, the slope of the linear plot increases.

Examination of the sum of the residuals for each set of experimental data indicates that the linear fit associated with two unknown parameters is not quite as accurate as when three unknown parameters were estimated. However, for each experimental data set, the accuracy of the linear fit improves as the Reynolds number exponent approaches the optimum value identified previously in Section 7.1.1 and Table 7.1. By further comparison of the error ranges, the accuracy of the linear fits associated with Reynolds number exponents of 0.8 and 0.9 tend to agree best with the fits of Table 7.1.

As previously stated in Section 7.1, experimental data for cases "B-1" thru "F-1" was collected at approximately the same air flow rate. Evaluation of the air-side thermal resistance for these five cases at a Reynolds number exponent of 0.7 yields an average value of $6.0293\text{E-}04 \text{ hr ft}^2 \text{ }^\circ\text{F/Btu}$ ($1.0615\text{E-}04 \text{ m}^2 \text{ K/W}$) with a variation from this mean value of +11.7/-7.7 percent. For a Reynolds number exponent of 0.8, the average air-side resistance is $7.0837\text{E-}04 \text{ hr ft}^2 \text{ }^\circ\text{F/Btu}$ ($1.2471\text{E-}04 \text{ m}^2 \text{ K/W}$) with a variation from this mean value of +7.6/-6.1 percent. For a Reynolds number exponent of 0.9, the average air-side resistance is $7.9297\text{E-}04 \text{ hr ft}^2 \text{ }^\circ\text{F/Btu}$ ($1.3961\text{E-}04 \text{ m}^2 \text{ K/W}$) with a variation from this mean value of +5.0/-5.2 percent.

In Section 7.1.1, it was verified that the air-side thermal resistance increases as the air flow rate decreases by comparing values of the heat transfer resistance at 570 cfm (0.269 m³/sec) and 1050 cfm (0.496 m³/sec). In Tables 7.2, 7.3, and 7.4, it is clearly evident that this trend is also true by comparison of the air-side thermal resistance values obtained at volumetric air flow rates of 570, 680, 1050 and 1506 cfm (0.269, 0.321, 0.496 and 0.711 m³/sec, respectively).

7.1.3. NLLS Results Based on Unknown Prandtl Number Exponent

Although the Prandtl number exponent is fixed in Equation (7.1.1) as 0.4, its value may also be determined for the given sets of experimental data by the parameter estimation process. In this instance, an alternative form of the correlating equation may be written as follows:

$$\frac{1}{U_o A_o} = \frac{1}{[C_1 Re^a Pr^b (k/D_h) A_i]} + C_2 \quad (7.1.3)$$

According to Equation (7.1.3), four unknown parameters (C_1 , C_2 , a , b) exist when the Prandtl number exponent is included as an unknown. As a result of applying the nonlinear least squares procedure to the eight sets of experimental data, in only three instances did the optimization process converge to a solution. This is primarily because the problem is too unconstrained and sensitive to initial guesses to permit easy access to a solution. In addition, the error associated with the experimental data, combined with the sensitivity of the NLLS routine, is a contributing factor that promotes the divergence in five of the eight data sets.

In order to clearly identify the influence of the Prandtl number exponent, two different fixed values, 0.333 and 0.4, were incorporated into Equation (7.1.1) and the nonlinear regression analysis was repeated for each experimental data set to determine optimum values of the three remaining parameters (C_1 , C_2 , a) for each experimental data set. These particular values for the Prandtl number exponent were selected in preference to others due to their consistent appearance throughout the literature for many of the empirical single phase heat transfer coefficient correlations for forced convection condensation inside tubes. As a result of the analysis, the sum of the residuals for the Prandtl number exponent of 0.4 was consistently less than that for an exponent of 0.333. Therefore, a fixed

Prandtl number exponent of 0.4 in lieu of 0.333 was incorporated in Equation (7.1.1) due to the evidence of a better fit with the larger exponent.

7.1.4. Comparison of Modified and Original Wilson Plots

The original Wilson Plot technique utilizes the form of the correlating equation given by Equation (7.1.4) below:

$$\frac{1}{U_o A_o} = \frac{1}{C_1 V^a} + C_2 \quad (7.1.4)$$

In comparison with the modified Wilson Plot function given by Equation (7.1.1), while the dependent variable ($1/U_o A_o$), the slope ($1/C_1$) and the y-intercept (C_2) are the same in both Equations (7.1.1) and (7.1.4), the independent variable in Equation (7.1.4) is $1/V^a$, where V is the circulating fluid or refrigerant in-tube velocity. Essentially, this equation form neglects the effects of temperature dependent fluid properties and surface geometry by including only the dimensional refrigerant flow velocity as the independent variable rather than the dimensionless Reynolds number as in Equation (7.1.1).

The units of velocity utilized in the single phase refrigerant flow analysis are feet per hour. Since the Wilson Plot y-intercept or air-side heat transfer resistance is independent of the units of velocity, any units may be implemented in Equation (7.1.4), e.g., feet per minute, feet per second, etc. However, the magnitude of the slope ($1/C_1$) is dependent on the units specified for the velocity.

A separate nonlinear regression analysis was carried out on each of the eight sets of experimental data to determine the unknown parameters in Equation (7.1.4), specifically, C_1 , C_2 , and a . These results are presented in Table 7.5. In addition, another nonlinear regression analysis was performed to determine the constants C_1 and C_2 based on fixed velocity function exponents of 0.7, 0.8 and 0.9. These results are shown in Tables 7.6, 7.7, and 7.8, respectively. Since these results are independent of the Prandtl number, the Prandtl number exponent 'b' column in Tables 7.5 thru 7.8 is omitted.

A comparison of the air-side resistance values obtained by the original and modified Wilson Plot techniques, Equations (7.1.1) and (7.1.4), respectively, is presented in Table 7.9. The modified Wilson Plot technique yields values for the air-side thermal resistance that are consistently higher than those values related to the original Wilson Plot procedure. The percent error between the two values obtained for each data set is minimized for an exponent of 0.9, ranging from approximately 1.0 to 3.3 percent for the eight experimental data cases. However, the accuracy progressively worsens as the exponent is decreased. For an exponent of 0.8, the range of percent error is 1.9 to 5.9 percent; and for an exponent of 0.7, the range of percent error is 3.4 to 11.0 percent.

7.1.5. Refrigerant-Side Convective Heat Transfer Coefficients

During experimental testing with single phase fluids on both sides of the condenser, it has been emphasized that the air-side resistance was maintained fixed, while the refrigerant-side resistance varied with the mass flow rate and the bulk mean temperature of the refrigerant. From the individual Wilson Plots, refrigerant-side convective heat transfer coefficients were obtained for several of the experimental data sets. Tables 7.10 thru 7.21 show these results. In addition, these tables consist of a comparison between the experimental heat transfer coefficients and the convective heat transfer coefficients obtained by utilizing the single phase flow correlations of Dittus-Boelter, Petukhov-Popov, and Gnielinski. Results are tabulated for Reynolds number exponents of 0.8 and 0.9.

The results shown in Tables 7.10 thru 7.21 indicate that the higher Reynolds number exponent leads to a larger value of the experimental heat transfer coefficient on the refrigerant-side. This is anticipated since it has been previously verified that the higher Reynolds number exponent increases the y-intercept on the Wilson Plot and hence, the air-side thermal resistance. An increase in the air-side resistance lowers the refrigerant-side resistance obtained experimentally, thereby increasing the heat transfer coefficient.

For a Reynolds number exponent of 0.9, the experimental results for cases "B-1", "C-1", "D-1", "F-1" and "H-1" are correlated most accurately by the Gnielinski and Petukhov-Popov correlations in comparison with the Dittus-Boelter correlation. Overall, for a Reynolds number exponent of 0.9, the Petukhov-Popov and Gnielinski expressions correlate roughly 90 percent of the experimental data within an error of 9.5 and 10.7 percent, respectively.

For a Reynolds number exponent of 0.8, the classical Dittus-Boelter expression typically correlates the single phase data more accurately than both the Petukhov-Popov and Gnielinski correlations. In particular, this fact is most detectable for cases "A-1", "B-1", "D-1" and "H-1". Overall, the Dittus-Boelter correlation agrees with 62 percent of the data in these four cases within an error of 10 percent; whereas, the Gnielinski and Petukhov-Popov correlations only agree with 24 percent of the data within 10 percent.

For all of the tabulated results shown in Tables 7.10 thru 7.21, the Dittus-Boelter equation consistently predicts values of the refrigerant-side convective heat transfer coefficient that are 15 to 20 percent lower than those obtained from the Gnielinski and Petukhov-Popov correlations.

7.1.6. Colburn Modulus/j Factors

The individual values for the Colburn modulus or j-factor that resulted from the experiments with single phase refrigerant flow are shown in Figure 7.16 for Reynolds number exponents of 0.7, 0.8, and 0.9, respectively. The plotted j-factors are based on the air-side thermal resistances determined from the modified Wilson Plot techniques (i.e., Equation 7.1.2 and Tables 7.2 thru 7.4).

The results indicate that the j factor increases as the Reynolds number exponent decreases. This is intuitive since the j factor is directly proportional to the air-side convective heat transfer coefficient. As the Reynolds number exponent increases, it has been proven earlier that the air-side resistance or the inverse of the heat transfer coefficient increases. Figure 7.16 also generally shows that the j-factor is inversely proportional to the air-side Reynolds number. Similarly, as the mass velocity of the air through the minimum free flow area of the finned

tube heat exchanger decreases, the j-factor or air-side convective heat transfer coefficient increases.

In Figure 7.17, the single phase results for the three Reynolds number exponents are plotted along with the best available correlation found in Kays and London [59]. The Kays and London data is from Figure 10-91 for a plate-fin heat exchanger with two rows of 3/8 inch (9.53 mm) nominal diameter tubes in a staggered configuration. Since the plot only encompasses an air-side Reynolds number range from 400 to 10,000, the Kays and London data has been linearly extrapolated to cover the lower range of Reynolds number related to the experiments described in this document. In addition, the Kays and London information was selected as a basis for comparison, since the particular geometry most closely resembles the automotive test condenser coil in this analysis.

7.2. Two Phase Results

Since good success was demonstrated by implementing the original Wilson Plot correlating equation (see Section 7.1.4), the same form was applied for the experimental data based on two phase refrigerant flow inside of the condenser tubes. The form of the correlating expression is given by Equation (7.2.1) below.

$$\frac{1}{U_o A_o} = \frac{1}{C_1 V^a} + C_2 \quad (7.2.1)$$

In this instance, the velocity in Equation (7.2.1) is the circulating air velocity, since it is the flow rate variable during experimental testing. The constant C_2 or Wilson Plot y-intercept is equal to the refrigerant-side resistance, since it is maintained fixed during the experiments by holding the refrigerant mass flow rate and the bulk mean temperature constant.

The nonlinear least squares (NLLS) routine was modified in order to correlate each of the four sets of two phase experimental data by the form of Equation (7.2.1). The results of the parameter estimation procedure for each experimental two phase data set are described in Sections 7.2.1 and 7.2.2.

7.2.1. NLLS Results Based on Unknown Velocity Exponent

The results of the parameter estimation study to determine the optimum values of C_1 , C_2 , and a in Equation (7.2.1) are shown in Table 7.22 for each two phase experimental data set.

7.2.2. NLLS Results Based on Two (2) Unknown Parameters

The parameter estimation study was repeated by assuming a fixed velocity exponent in Equation (7.1.4) and calculating the optimum values of C_1 and C_2 . Individual evaluations using velocity exponents of 0.4, 0.5, 0.55, 0.6, and 0.7 were performed. The five velocity exponents incorporated fall within the range of velocity exponents identified in Section 4.5 for relating the air-side heat transfer resistance to the Reynolds number or velocity.

For an exponent of 0.4, the resulting refrigerant-side resistances from Equation (7.2.1) are either negative or negligibly small in each case. Results for exponents of 0.4, 0.5, 0.55, 0.6 and 0.7 are furnished in Tables 7.23, 7.24, 7.25, 7.26 and 7.27, respectively. Figures 7.18 thru 7.21 illustrate the linearized Wilson Plots for velocity exponents of 0.5, 0.6, and 0.7, respectively.

The four sets of two phase results consistently indicate that air-side thermal resistance is insignificantly affected by the value of the velocity exponent in the range of air velocities considered. However, as the exponent is increased from 0.5 to 0.7, the refrigerant-side resistance or the y-intercept on the Wilson Plot also increases. In addition, as the air velocity increases, the air-side thermal resistance increases, which implies that the air-side convective heat transfer coefficient also decreases.

7.2.3. Condensing Fraction Influence

The data analysis based on two phase refrigerant flow assumes that the overall thermal resistance obtained for the condensing or two phase flow region is also representative of the overall $1/UA$ for the entire heat exchanger, i.e., including the single phase refrigerant flow desuperheating and subcooling regions. This assumption is incorporated in order to simplify the effectiveness-NTU

computations. The condenser length in the two phase region may be predicted by investigating the condenser thermocouple measurements on each return bend.

For the four sets of experimental data collected for two phase refrigerant flow, the estimated condensing fraction varied from approximately 68.8 to 93.8 percent of the total condenser tube length. These values were utilized in generating the air-side thermal resistance values and j-factors described in this Chapter. Figures 7.22 thru 7.25 each present a comparison of the resulting Wilson Plots for over-estimating or under-estimating the condensing fraction by approximately ± 6.25 percent. The plots reveal that the refrigerant-side heat transfer resistance or the y-intercept increases when the estimated condensing fraction is lowered. However, the impact of the different condensing fractions on the results is roughly within ± 8 percent.

7.2.4. Colburn Modulus/j Factors

Since the air flow rate was varied during experiments with two phase refrigerant flow inside of the condenser tubes, an air-side resistance was determined for each air flow rate associated with each experimental data set.

7.3. Comparison/Correlation of Single Phase and Two Phase Results

Figures 7.26 thru 7.32 show different plots of the j factor versus the air-side Reynolds number for the experimental data based on single phase and two phase refrigerant flow. The main difference between these plots is in the Reynolds number and velocity function exponents employed for the single phase and two phase data analysis, respectively. The three graphs that most accurately represent the data are Figures 7.26 thru 7.28. Several j-factor versus Reynolds number plots, as well as the associated correlating equation, are further described in the following paragraphs. The Reynolds number in each plot and correlation is based on the hydraulic diameter.

In Figure 7.26, the single phase data shown is based on a Reynolds number exponent of 0.9; whereas, the two phase flow data presented is based on a velocity exponent of 0.5. The following expression, derived by a nonlinear regression procedure, correlates all the data of Figure 7.26 within an accuracy of +10.6/-9.8 percent:

$$j = 0.23265 \text{ Re}_D^{-0.5237} \quad (7.3.1)$$

In addition, Equation (7.9) correlates all of the data with an average percent error of 5.7 percent and agrees with ninety percent of the experimental data within an accuracy of ± 9.0 percent.

In Figure 7.27, the single phase data displayed is based on a Reynolds number exponent of 0.8; whereas, the two phase flow data shown is based on an exponent of 0.55. The following function correlates all the data of Figure 7.27 within an accuracy of +11.8/-10.7 percent:

$$j = 0.20369 \text{ Re}_D^{-0.4829} \quad (7.3.2)$$

In comparison with Equation (7.3.1), the above expression correlates all of the data with an average percent error of 6.4 percent and agrees with ninety percent of the experimental data within ± 9.7 percent.

In Figure 7.28, the single phase data shown is based on a Reynolds number exponent of 0.8; whereas, the two phase flow data presented is based on an exponent of 0.6. The following equation correlates all the data of Figure 7.28 within +20.8/-12.7 percent:

$$j = 0.164265 \text{ Re}_D^{-0.4332} \quad (7.3.3)$$

In comparison with the previous two correlating equations, Equation (7.3.3) correlates all of the experimental data with an average percent error of 6.0 percent and agrees with ninety percent of the data within ± 10.0 percent.

In Figure 7.29, the single phase data depicted is based on a Reynolds number exponent of 0.9; whereas, the two phase data shown is based on an exponent of 0.55. The following correlation agrees with the experimental data of Figure 7.29 within an error of +19.2/-12.6 percent and an average percent error of 6.3 percent.

$$j = 0.190497 R_{\phi}^{-0.4756} \quad (7.3.4)$$

Ninety percent of the data presented in Figure 7.29 is accurately correlated to within ± 10.7 percent by Equation (7.3.4).

In Figure 7.30, the single phase data shown is based on a Reynolds number exponent of 0.8; whereas, the two phase data presented is based on an exponent of 0.5. The following correlation agrees with the experimental data of Figure 7.30 within an error of +13.0/-13.7 percent and an average percent error of 7.3 percent.

$$j = 0.25076 R_{\phi}^{-0.5321} \quad (7.3.5)$$

Ninety percent of the data displayed in Figure 7.30 is accurately correlated to within ± 12.3 percent.

The remaining plots, Figures 7.31 and 7.32, represent the j-factor versus Reynolds number experimental results for additional combinations of the two independent variable exponents. By inspection and in comparison with Figures 7.26 thru 7.30, it is clearly evident that it is more difficult to correlate the data in these remaining plots into a single, more accurate expression.

Figures 7.33 thru 7.37 display the single phase and two phase experimental data along with the generated correlations, i.e., Equations (7.3.1) thru (7.3.5), respectively, based on the different combinations of the Reynolds number and velocity function exponents.

7.4. Comparison of Experimental Data with Available Correlations

Figure 7.38 portrays a graphical comparison of the most accurate correlation of the experimental data, Equation (7.3.1), with the empirical data of Kays & London, Figure 10-91 [59], and the wavy fin correlation of Webb [129], i.e., Equation (2.4.23) applicable for Graetz numbers larger than 25. Although the Kays and London information is applicable for a geometrically similar extended surface with straight fins, it was selected as a basis of comparison in this analysis, since it most closely matches the geometry of the GM/Harrison Radiator automotive test condenser coil.

According to Figure 7.38, the Kays and London data is consistently higher than the experimental correlation by a factor of 1.45 to 1.60, depending on the air-side Reynolds number. As the Reynolds number increases, the difference between the experimental data and the Kays and London data also increases. The experimental correlation agrees more accurately with the Webb wavy fin correlation. Over the range of air-side Reynolds numbers shown, the two correlations differ by an error of +12.3/-15.0 percent while ninety percent of the data agrees within an accuracy of ± 7.9 percent. The average error between the two correlations is approximately ± 4.9 percent.

It should also be noted that Webb's wavy fin correlation is valid for Graetz numbers between 5 and 180 and for geometrical parameters, identified in Section 2.4.2. The Graetz number for the single phase and two phase experimental testing, described in this document, is approximately 50 and the geometrical parameters fall within the constraints of the well-verified correlation.

7.5. Effect of Variations in h on the Heat Exchange

In order to further determine whether the empirically-derived j -factor correlation, Equation (7.3.1), may be suitable for condenser design purposes, it is important to determine the influence of variations in the heat transfer coefficient on the condenser heat rejection capacity. Subsequently, although an oversized condenser may provide the equivalent heat rejection capacity required for a specific application, the same final result may be approached by attempting to improve the convective heat transfer resistance.

In his discussion of the full condenser computer simulation results, Ragazzi [85] studies the effects of varying the air-side heat transfer coefficient and the effects of varying the refrigerant-side heat transfer coefficient on the condenser total heat rejection capacity.

According to the plotted information by Ragazzi, the results may be summarized in the table below.

<u>% Change in h_{air} or h_{ref}</u>	<u>% Change in Q_{air}</u>	<u>% Change in Q_{ref}</u>
-50	-27.4	-15.1
-30	-13.1	-6.1
-10	-3.0	-1.2
0 (Base Case)	0	0
+10	1.9	1.1
+30	4.7	2.8
+50	6.4	4.1
+100	8.9	6.2

In the above table, h_{air} and h_{ref} are equal to the air-side and refrigerant-side convective heat transfer resistances, respectively, and Q_{air} and Q_{ref} are equal to the air-side and refrigerant-side heat exchange rates, respectively. In addition, the base case represents average operating conditions (i.e., an air inlet temperature of 110 °F (43.3 °C), an air flow rate of 1000 cfm (0.472 m³/sec), refrigerant superheat of 10 °F (5.6 °C), and a refrigerant mass flow rate of 400 lb_m/hr (0.0504 kg/sec).

According to its definition, the Colburn modulus is directly related to the air-side heat transfer coefficient. Therefore, the previous tabulated results may also be interpreted in terms of the j-factor. The published "percent change" information indicates that doubling the air-side convective heat transfer coefficient from the base case results in approximately a 9 percent increase in the total heat rejection capacity of the condenser; whereas, doubling the refrigerant-side convective heat transfer coefficient from the base case results in approximately a 6 percent increase in the total capacity. Thus, a large amount of error in the heat transfer coefficient does not significantly affect the capacity prediction if the actual value for the heat transfer coefficient is larger than the base case. Further, the data

indicates that a ± 10 percent error in the experimental j-factor correlation, i.e., Equation (7.3.1), results in a +1.9/-3.0 percent error in the air-side heat transfer and a +1.1/-1.2 percent error in the refrigerant-side heat transfer.

The behavior of the plot of percent change in the heat rejection capacity versus the change in the heat transfer coefficient is analogous to the shape of the effectiveness-NTU curves typically found in Kays and London [59]. Recall that for a condensing type of heat exchanger, effectiveness is related to NTU by the following expression:

$$\varepsilon = 1 - e^{-NTU} \quad (7.5.1)$$

In addition, effectiveness is a function of the heat transfer rate and the number of transfer units (NTU) is a function of the heat transfer coefficient. Thus, as the effectiveness of a heat exchanger increases, the effect of changes in the heat transfer coefficient on the total capacity diminishes.

TABLE 7.1
PARAMETER ESTIMATION RESULTS BASED ON THREE (3) UNKNOWNNS
PRANDTL NO. EXPONENT = 0.4

Case	C ₁	C ₂	a	b	ΣR	% Error
A	2.7486E-03	7.5185E-04	1.007	0.4	5.044	+1.1/-0.64
B	8.4152E-03	7.5232E-04	0.9007	0.4	1.458	+0.39/-0.42
C	2.2815E-01	3.9261E-04	0.5468	0.4	11.760	+0.71/-0.92
D	3.1750E-03	8.4134E-04	0.9990	0.4	24.847	+2.5/-1.4
E	6.4538E-03	8.1160E-04	0.9308	0.4	251.05	+2.4/-2.8
F	4.0674E-02	7.2358E-04	0.7531	0.4	2.261	+0.62/-0.41
G	N/A	N/A	N/A	0.4	N/A	N/A
H	3.8669E-03	1.5283E-03	0.9887	0.4	48.795	+2.6/-2.1

TABLE 7.2
PARAMETER ESTIMATION RESULTS BASED ON TWO (2) UNKNOWNNS
REYNOLDS NO. EXPONENT = 0.7/PRANDTL NO. EXPONENT = 0.4

Case	C ₁	C ₂	a	b	ΣR	% Error
A	5.0256E-02	4.7235E-04	0.7	0.4	7.808	+1.3/-0.59
B	5.4844E-02	5.5632E-04	0.7	0.4	3.029	+0.35/-0.64
C	5.7596E-02	6.2732E-04	0.7	0.4	13.915	+0.64/-1.0
D	5.2377E-02	5.6608E-04	0.7	0.4	28.508	+2.1/-1.7
E	5.5453E-02	5.9127E-04	0.7	0.4	271.3	+2.3/-3.0
F	6.6829E-02	6.7366E-04	0.7	0.4	2.308	+0.66/-0.39
G	6.7473E-02	7.0614E-04	0.7	0.4	1868.2	+11.4/-11.0
H	5.3997E-02	7.5333E-04	0.7	0.4	61.950	+2.2/-1.8

TABLE 7.3
PARAMETER ESTIMATION RESULTS BASED ON TWO (2) UNKNOWNNS
REYNOLDS NO. EXPONENT = 0.8/PRANDTL NO. EXPONENT = 0.4

Case	C ₁	C ₂	a	b	ΣR	% Error
A	1.9739E-02	5.8481E-04	0.8	0.4	6.335	+1.2/-0.53
B	2.1686E-02	6.6519E-04	0.8	0.4	1.862	+0.37/-0.53
C	2.2955E-02	7.3644E-04	0.8	0.4	17.827	+0.89/-1.1
D	2.0755E-02	6.7692E-04	0.8	0.4	26.523	+2.3/-1.6
E	2.2006E-02	7.0087E-04	0.8	0.4	257.3	+2.3/-2.9
F	2.6162E-02	7.6243E-04	0.8	0.4	2.298	+0.58/-0.43
G	2.7001E-02	7.9899E-04	0.8	0.4	1844.2	+11.1/-11.2
H	2.1907E-02	8.7993E-04	0.8	0.4	54.488	+2.0/-1.8

TABLE 7.4
PARAMETER ESTIMATION RESULTS BASED ON TWO (2) UNKNOWNNS
REYNOLDS NO. EXPONENT = 0.9/PRANDTL NO. EXPONENT = 0.4

Case	C ₁	C ₂	a	b	ΣR	% Error
A	7.6555E-03	6.7432E-04	0.9	0.4	5.396	+1.1/-0.55
B	8.4684E-03	7.5181E-04	0.9	0.4	1.458	+0.39/-0.42
C	9.0370E-03	8.2326E-04	0.9	0.4	23.844	+1.2/-1.3
D	8.1211E-03	7.6900E-04	0.9	0.4	25.273	+2.4/-1.5
E	8.6269E-03	7.8819E-04	0.9	0.4	251.4	+2.3/-2.8
F	1.0106E-02	8.3258E-04	0.9	0.4	2.633	+0.59/-0.48
G	1.0664E-02	8.7225E-04	0.9	0.4	1820.5	+11.2/-9.9
H	8.7755E-03	9.8006E-04	0.9	0.4	50.063	+2.4/-1.4

TABLE 7.5
PARAMETER ESTIMATION RESULTS BASED ON THREE (3) UNKNOWNNS
(1/VELOCITY) = INDEPENDENT VARIABLE

Case	C ₁	C ₂	a	b	ΣR	% Error
A	0.11681	7.6375E-04	1.044	-	5.034	+1.1/-0.63
B	0.28320	7.6797E-04	0.9461	-	1.438	+0.39/-0.40
C	3.0141	4.8675E-04	0.6427	-	12.542	+0.75/-0.94
D	0.11586	8.6603E-04	1.055	-	25.391	+2.6/-1.4
E	1.0706	6.3952E-04	0.7765	-	213.82	+2.1/-2.7
F	1.2567	7.2883E-04	0.7531	-	2.252	+0.62/-0.41
G	N/A	N/A	N/A	-	N/A	N/A
H	0.17487	1.0637E-03	1.017	-	48.619	+2.6/-2.1

TABLE 7.6
PARAMETER ESTIMATION RESULTS BASED ON TWO (2) UNKNOWNNS
1/(VELOCITY) = INDEPENDENT VARIABLE/EXPONENT = 0.7

Case	C ₁	C ₂	a	b	ΣR	% Error
A	1.7370	4.2061E-04	0.7	-	9.918	+1.4/-0.78
B	1.8824	5.0521E-04	0.7	-	4.710	+0.38/-0.70
C	1.9762	5.7689E-04	0.7	-	12.982	+0.70/-1.0
D	1.8213	5.1714E-04	0.7	-	32.774	+2.1/-1.8
E	1.9054	5.3904E-04	0.7	-	215.47	+2.1/-2.9
F	2.2859	6.5066E-04	0.7	-	2.376	+0.68/-0.37
G	2.3922	6.7833E-04	0.7	-	1892.0	+11.5/-10.3
H	1.9176	7.1465E-04	0.7	-	68.744	+2.3/-2.0

TABLE 7.7
PARAMETER ESTIMATION RESULTS BASED ON TWO (2) UNKNOWN
1/(VELOCITY) = INDEPENDENT VARIABLE/EXPONENT = 0.8

Case	C ₁	C ₂	a	b	ΣR	% Error
A	0.80666	5.5056E-04	0.8	-	7.494	+1.3/-0.57
B	0.88120	6.3150E-04	0.8	-	2.590	+0.36/-0.58
C	0.93243	7.0323E-04	0.8	-	15.857	+0.74/-1.1
D	0.85238	6.4692E-04	0.8	-	29.205	+2.2/-1.7
E	0.89516	6.6652E-04	0.8	-	213.98	+2.1/-2.7
F	1.0595	7.4813E-04	0.8	-	2.298	+0.60/-0.42
G	1.1258	7.8070E-04	0.8	-	1862.4	+11.3/-10.2
H	0.91548	8.5430E-04	0.8	-	58.008	+2.2/-1.8

TABLE 7.8
PARAMETER ESTIMATION RESULTS BASED ON TWO (2) UNKNOWN
1/(VELOCITY) = INDEPENDENT VARIABLE/EXPONENT = 0.9

Case	C ₁	C ₂	a	b	ΣR	% Error
A	0.36897	6.5171E-04	0.9	-	5.893	+1.2/-0.49
B	0.40632	7.2970E-04	0.9	-	1.552	+0.38/-0.46
C	0.43346	8.0152E-04	0.9	-	21.399	+1.1/-1.2
D	0.39294	7.4777E-04	0.9	-	26.801	+2.4/-1.6
E	0.41431	7.6569E-04	0.9	-	218.15	+2.2/-2.5
F	0.48369	8.2399E-04	0.9	-	2.556	+0.57/-0.47
G	0.52201	8.6026E-04	0.9	-	1834.2	+11.1/-10.0
H	0.42993	9.6312E-04	0.9	-	51.338	+2.4/-1.9

TABLE 7.9
COMPARISON OF AIR-SIDE THERMAL RESISTANCES
OBTAINED FROM MODIFIED AND ORIGINAL WILSON PLOT TECHNIQUES

Case	Exponent = 0.7			Exponent = 0.8			Exponent = 0.9		
A-1	4.7235	4.2061	11.0%	5.8481	5.5056	5.9%	6.7432	6.5171	3.3%
B-1	5.5632	5.0521	9.2%	6.6519	6.3150	5.1%	7.5181	7.2970	2.9%
C-1	6.2732	5.7689	8.0%	7.3644	7.0323	4.5%	8.2326	8.0152	2.6%
D-1	5.6608	5.1714	8.6%	6.7692	6.4692	4.4%	7.6900	7.4777	2.8%
E-1	5.9127	5.3904	8.8%	7.0087	6.6652	4.9%	7.8819	7.6569	2.8%
F-1	6.7366	6.5066	3.4%	7.6243	7.4813	1.9%	8.3258	8.2399	1.0%
G-1	7.0614	6.7833	3.9%	7.9899	7.8070	2.3%	8.7225	8.6026	1.4%
H-1	7.5333	7.1465	5.1%	8.7993	8.5430	2.9%	9.8006	9.6312	1.7%

$R_{air} = (\text{Tabulated Value}) \times 10^{-4}$ in units of [hr ft² F/Btu]. Divide R_{air} by 5.68 to obtain units of [m² K/W].

TABLE 7.10
COMPARISON OF REFRIGERANT-SIDE HEAT TRANSFER COEFFICIENTS
CASE A-1 DATA SET - REYNOLDS NUMBER EXPONENT OF 0.8

Experiment	Dittus-Boelter	Error	Petukhov	Error	Gnielinski	Error
137.70	143.08	3.8%	172.48	22.4%	173.83	23.2%
154.33	160.40	3.9%	193.61	22.6%	195.68	23.6%
171.69	179.68	4.5%	217.21	23.4%	220.09	24.7%
171.87	179.10	4.1%	216.54	23.0%	219.40	24.3%
185.82	191.64	3.1%	231.97	22.1%	235.35	23.5%
194.36	199.91	2.8%	242.12	21.9%	245.83	23.4%
197.86	204.94	3.5%	248.38	22.6%	252.30	24.2%
208.10	211.84	1.8%	256.92	21.0%	261.13	22.6%
214.05	223.87	4.5%	271.81	23.8%	276.51	25.5%
220.64	230.18	4.2%	279.72	23.6%	284.68	25.3%
231.80	243.48	4.9%	296.24	24.4%	301.75	26.2%
245.99	257.72	4.7%	314.10	24.3%	320.20	26.2%
250.47	262.75	4.8%	320.39	24.5%	326.70	26.4%

TABLE 7.11
COMPARISON OF REFRIGERANT-SIDE HEAT TRANSFER COEFFICIENTS
CASE A-1 DATA SET - REYNOLDS NUMBER EXPONENT OF 0.9

Experiment	Dittus-Boelter	Error	Petukhov	Error	Gnielinski	Error
148.83	143.08	3.9%	172.48	14.7%	173.83	15.5%
168.45	160.40	4.9%	193.61	13.9%	195.68	15.0%
189.34	179.10	5.6%	217.21	13.7%	219.40	14.7%
189.56	179.68	5.4%	216.54	13.3%	220.09	14.9%
206.67	191.64	7.5%	231.97	11.5%	235.35	13.0%
217.29	199.91	8.3%	242.12	10.8%	245.83	12.3%
221.68	204.94	7.8%	248.38	11.4%	252.30	12.9%
234.61	211.84	10.2%	256.92	9.1%	261.13	10.7%
242.19	223.87	7.9%	271.81	11.5%	276.51	13.2%
250.66	230.18	8.5%	279.72	11.0%	284.68	12.7%
265.18	243.48	8.5%	296.24	11.1%	301.75	12.9%
283.90	257.52	9.7%	314.10	10.1%	320.20	12.0%
289.89	262.75	9.8%	320.39	10.0%	326.70	11.9%

TABLE 7.12
COMPARISON OF REFRIGERANT-SIDE HEAT TRANSFER COEFFICIENTS
CASE B-1 DATA SET - REYNOLDS NUMBER EXPONENT OF 0.8

Experiment	Dittus-Boelter	Error	Petukhov	Error	Gnielinski	Error
137.63	130.29	5.5%	157.07	13.2%	157.89	13.7%
152.39	142.95	6.4%	172.46	12.4%	173.82	13.1%
162.77	153.36	6.0%	185.14	12.9%	186.93	13.8%
171.76	162.36	5.6%	196.17	13.3%	198.34	14.4%
181.40	171.87	5.4%	207.82	13.6%	210.39	14.8%
195.14	183.89	5.9%	222.63	13.2%	225.70	14.5%
206.64	195.09	5.8%	236.44	13.5%	239.98	14.9%
211.13	199.50	5.7%	241.90	13.6%	245.62	15.1%
220.45	207.93	5.8%	252.33	13.5%	256.40	15.1%
226.82	214.20	5.7%	260.10	13.7%	264.42	15.3%
233.57	220.25	5.9%	267.60	13.6%	272.18	15.3%
242.50	227.78	6.3%	276.97	13.3%	281.86	15.0%
249.10	235.29	5.7%	286.33	13.9%	291.53	15.7%
252.53	241.60	4.4%	294.16	15.2%	299.61	17.1%
258.88	246.14	5.0%	299.88	14.7%	305.53	16.5%
265.55	252.31	5.1%	307.59	14.7%	313.50	16.6%

TABLE 7.13
COMPARISON OF REFRIGERANT-SIDE HEAT TRANSFER COEFFICIENTS
CASE B-1 DATA SET - REYNOLDS NUMBER EXPONENT OF 0.9

Experiment	Dittus-Boelter	Error	Petukhov	Error	Gnielinski	Error
148.36	130.29	13.0%	157.07	5.7%	157.89	6.2%
165.65	142.95	14.7%	172.46	4.0%	173.82	4.8%
177.99	153.36	14.9%	185.14	3.9%	186.93	4.9%
188.80	162.36	15.1%	196.17	3.8%	198.34	4.9%
200.51	171.87	15.4%	207.82	3.6%	210.39	4.8%
217.43	183.89	16.7%	222.63	2.4%	225.70	3.7%
231.80	195.09	17.2%	236.44	2.0%	239.98	3.5%
237.48	199.50	17.4%	241.90	1.8%	245.62	3.4%
249.33	207.93	18.1%	252.33	1.2%	256.40	2.8%
257.51	214.20	18.4%	260.10	1.0%	264.42	2.6%
266.24	220.25	18.9%	267.60	0.5%	272.18	2.2%
277.91	227.78	19.8%	276.97	0.3%	281.86	1.4%
286.61	235.29	19.7%	286.33	0.1%	291.53	1.7%
291.17	241.60	18.6%	294.16	1.0%	299.61	2.9%
299.63	246.14	19.6%	299.88	0.1%	305.53	1.9%
308.61	252.31	20.1%	307.59	0.3%	313.50	1.6%

TABLE 7.14
COMPARISON OF REFRIGERANT-SIDE HEAT TRANSFER COEFFICIENTS
CASE C-1 DATA SET - REYNOLDS NUMBER EXPONENT OF 0.8

Experiment	Dittus-Boelter	Error	Petukhov	Error	Gnielinski	Error
131.41	115.20	13.1%	138.55	5.3%	138.70	5.4%
147.25	129.77	12.6%	156.24	5.9%	157.03	6.4%
160.12	145.54	9.5%	175.47	9.1%	176.93	10.0%
164.76	149.87	9.5%	180.77	9.3%	182.41	10.2%
181.05	163.06	10.5%	196.90	8.4%	199.09	9.5%
193.82	173.83	10.9%	210.08	8.1%	212.72	9.3%
205.93	183.04	11.8%	221.38	7.2%	224.39	8.6%
209.86	186.30	11.9%	225.56	7.2%	228.73	8.6%
211.74	189.38	11.1%	229.21	7.9%	232.49	9.3%
220.82	198.08	10.9%	240.10	8.4%	243.76	9.9%
232.33	205.94	12.0%	249.68	7.2%	253.65	8.8%
232.99	209.09	10.8%	253.69	8.5%	257.80	10.1%
238.73	213.79	11.0%	259.40	8.3%	263.69	9.9%
243.31	219.70	10.2%	266.76	9.2%	271.30	10.9%
247.69	220.81	11.5%	268.21	8.0%	272.80	9.6%
257.36	226.70	12.7%	275.55	6.8%	280.38	8.6%
261.03	232.45	11.6%	282.61	7.9%	287.68	9.7%
266.95	237.52	11.7%	288.97	7.9%	294.25	9.7%
282.02	249.11	12.4%	303.46	7.3%	309.21	9.2%
288.94	256.70	11.8%	312.99	8.0%	319.07	9.9%

TABLE 7.15
COMPARISON OF REFRIGERANT-SIDE HEAT TRANSFER COEFFICIENTS
CASE C-1 DATA SET - REYNOLDS NUMBER EXPONENT OF 0.9

Experiment	Dittus-Boelter	Error	Petukhov	Error	Gnielinski	Error
141.18	115.20	20.3%	138.55	1.9%	138.70	1.8%
159.63	129.77	20.6%	156.24	2.1%	157.03	1.6%
174.87	145.54	18.3%	175.47	0.3%	176.93	1.2%
180.42	149.87	18.5%	180.77	0.2%	182.41	1.1%
200.13	163.06	20.4%	196.90	1.6%	199.09	0.5%
215.85	173.83	21.6%	210.08	2.7%	212.72	1.5%
230.97	183.04	23.2%	221.38	4.2%	224.39	2.9%
235.93	186.30	23.5%	225.56	4.5%	228.73	3.1%
238.32	189.38	22.9%	229.21	3.9%	232.49	2.5%
249.88	198.08	23.1%	240.10	4.0%	243.76	2.5%
264.73	205.94	25.0%	249.68	5.9%	253.65	4.3%
265.58	209.09	23.8%	253.69	4.6%	257.80	3.0%
273.06	213.79	24.3%	259.40	5.1%	263.69	3.5%
279.07	219.70	23.8%	266.76	4.5%	271.30	2.8%
284.85	220.81	25.3%	268.21	6.0%	272.80	4.3%
297.71	226.70	27.1%	275.55	7.7%	280.38	6.0%
302.63	232.45	26.2%	282.61	6.8%	287.68	5.1%
310.62	237.52	26.7%	288.97	7.2%	294.25	5.4%
331.21	249.11	28.3%	303.46	8.7%	309.21	6.9%
340.79	256.70	28.1%	312.99	8.5%	319.07	6.6%

TABLE 7.16
COMPARISON OF REFRIGERANT-SIDE HEAT TRANSFER COEFFICIENTS
CASE D-1 DATA SET - REYNOLDS NUMBER EXPONENT OF 0.8

Experiment	Dittus-Boelter	Error	Petukhov	Error	Gnielinski	Error
147.22	145.86	0.9%	175.41	17.5%	176.82	18.3%
156.74	130.07	18.6%	156.18	0.4%	156.92	0.1%
166.56	166.41	0.1%	200.55	18.5%	202.83	19.6%
186.96	184.85	1.1%	223.19	17.7%	226.24	19.0%
195.38	153.44	24.0%	184.66	5.6%	186.40	4.7%
202.08	193.41	4.4%	233.67	14.5%	237.06	15.9%
210.60	208.22	1.1%	252.07	17.9%	256.09	19.5%
224.05	220.30	1.7%	266.96	17.5%	271.46	19.1%
234.91	234.70	0.1%	284.96	19.3%	290.07	21.0%
243.23	248.85	2.3%	302.61	21.8%	308.30	23.6%
243.23	244.66	0.6%	297.37	20.0%	302.89	21.8%
253.71	255.10	0.5%	310.43	20.1%	316.38	22.0%
262.60	248.22	5.6%	301.72	13.9%	307.37	15.7%

TABLE 7.17
COMPARISON OF REFRIGERANT-SIDE HEAT TRANSFER COEFFICIENTS
CASE D-1 DATA SET - REYNOLDS NUMBER EXPONENT OF 0.9

Experiment	Dittus-Boelter	Error	Petukhov	Error	Gnielinski	Error
160.06	145.86	9.3%	175.41	9.2%	176.82	10.0%
171.37	130.07	27.4%	156.18	9.3%	156.92	8.8%
183.17	166.41	9.6%	200.55	9.1%	202.83	10.2%
208.15	184.85	11.9%	223.19	7.0%	226.24	8.3%
218.64	153.44	35.0%	184.66	16.9%	186.40	15.9%
227.08	193.41	16.0%	233.67	2.9%	237.06	4.3%
237.89	208.22	13.3%	252.07	5.8%	256.09	7.4%
255.20	220.30	14.7%	266.96	4.5%	271.46	6.2%
269.37	234.70	13.8%	284.96	5.6%	290.07	7.4%
280.37	248.85	11.9%	302.61	7.6%	308.30	9.5%
280.37	244.66	13.6%	297.37	5.9%	302.89	7.7%
294.39	255.10	14.3%	310.43	5.3%	316.38	7.2%
306.43	248.22	21.0%	301.72	1.5%	307.37	0.3%

TABLE 7.18
COMPARISON OF REFRIGERANT-SIDE HEAT TRANSFER COEFFICIENTS
CASE F-1 DATA SET - REYNOLDS NUMBER EXPONENT OF 0.8

Experiment	Dittus-Boelter	Error	Petukhov	Error	Gnielinski	Error
198.28	141.99	33.1%	173.22	13.5%	157.89	22.7%
236.38	166.66	34.6%	203.85	14.8%	173.82	30.5%
250.38	176.98	34.4%	216.78	14.4%	186.93	29.0%
259.45	182.33	34.9%	223.39	14.9%	198.34	26.7%
278.28	191.36	37.0%	234.75	17.0%	210.39	27.8%
281.13	196.67	35.4%	241.36	15.2%	225.70	21.9%
298.97	205.37	37.1%	252.33	16.9%	239.98	21.9%
305.05	212.51	35.8%	261.26	15.5%	245.62	21.6%
318.00	219.34	36.7%	269.87	16.4%	256.40	21.4%
333.44	226.85	38.0%	279.31	17.7%	264.42	23.1%
340.32	233.53	37.2%	287.76	16.7%	272.18	22.2%
346.76	236.48	37.8%	291.47	17.3%	281.86	20.6%
355.74	241.68	38.2%	298.09	17.6%	291.53	19.8%
373.47	248.35	40.2%	306.50	19.7%	299.61	21.9%
376.03	251.93	39.5%	311.02	18.9%	305.53	20.7%

TABLE 7.19
COMPARISON OF REFRIGERANT-SIDE HEAT TRANSFER COEFFICIENTS
CASE F-1 DATA SET - REYNOLDS NUMBER EXPONENT OF 0.9

Experiment	Dittus-Boelter	Error	Petukhov	Error	Gnielinski	Error
182.85	141.99	25.2%	173.22	5.4%	157.89	14.7%
214.78	166.66	25.2%	203.85	5.2%	173.82	21.1%
226.27	176.98	24.4%	216.78	4.3%	186.93	19.0%
233.65	182.33	24.7%	223.39	4.5%	198.34	16.3%
248.81	191.36	26.1%	234.75	5.8%	210.39	16.7%
251.09	196.67	24.3%	241.36	4.0%	225.70	10.7%
265.22	205.37	25.4%	252.33	5.0%	239.98	10.0%
270.00	212.51	23.8%	261.26	3.3%	245.62	9.5%
280.09	219.34	24.3%	269.87	3.7%	256.40	8.8%
292.00	226.85	25.1%	279.31	4.4%	264.42	9.9%
297.27	233.53	24.0%	287.76	3.3%	272.18	8.8%
302.17	236.48	24.4%	291.47	3.6%	281.86	7.0%
308.97	241.68	24.4%	298.09	3.6%	291.53	5.8%
322.25	248.35	25.9%	306.50	5.0%	299.61	7.3%
324.15	251.93	25.1%	311.02	4.1%	305.53	5.9%

TABLE 7.20
COMPARISON OF REFRIGERANT-SIDE HEAT TRANSFER COEFFICIENTS
CASE H-1 DATA SET - REYNOLDS NUMBER EXPONENT OF 0.8

Experiment	Dittus-Boelter	Error	Petukhov	Error	Gnielinski	Error
111.54	105.08	6.0%	125.04	11.4%	124.54	11.0%
142.60	133.82	6.4%	159.89	11.4%	160.65	11.9%
162.57	153.49	5.7%	183.94	12.3%	185.55	13.2%
178.21	162.56	9.2%	195.06	9.0%	197.06	10.0%
188.18	174.88	7.3%	210.20	11.1%	212.72	12.2%
190.13	183.53	3.5%	220.88	15.0%	223.76	16.3%
204.01	185.23	9.6%	222.97	8.9%	225.93	10.2%
204.77	190.75	7.1%	229.77	11.5%	232.96	12.9%
213.53	201.32	5.9%	242.88	12.9%	246.52	14.3%
231.19	220.00	5.0%	266.05	14.0%	270.47	15.7%
243.49	227.76	6.7%	275.74	12.4%	280.49	14.1%
255.96	237.28	7.6%	287.61	11.6%	292.75	13.4%
263.32	254.37	3.5%	308.97	16.0%	314.83	17.8%

TABLE 7.21
COMPARISON OF REFRIGERANT-SIDE HEAT TRANSFER COEFFICIENTS
CASE H-1 DATA SET - REYNOLDS NUMBER EXPONENT OF 0.9

Experiment	Dittus-Boelter	Error	Petukhov	Error	Gnielinski	Error
119.64	105.08	13.0%	125.04	4.4%	124.54	4.0%
156.12	133.82	15.4%	159.89	2.4%	160.65	2.9%
180.38	153.49	16.1%	183.94	2.0%	185.55	2.8%
199.84	162.56	20.6%	195.06	2.4%	197.06	1.4%
212.46	174.88	19.4%	210.20	1.1%	212.72	0.1%
214.95	183.53	15.8%	220.88	2.7%	223.76	4.0%
232.87	185.23	22.8%	222.97	4.3%	225.93	3.0%
233.86	190.75	20.3%	229.77	1.8%	232.96	0.4%
245.34	201.32	19.7%	242.88	1.0%	246.52	0.5%
268.96	220.00	20.0%	266.05	1.1%	270.47	0.6%
285.94	227.76	22.7%	275.74	3.6%	280.49	1.9%
303.08	237.28	24.4%	287.61	5.2%	292.75	3.5%
313.45	254.37	20.8%	308.97	1.4%	314.83	0.4%

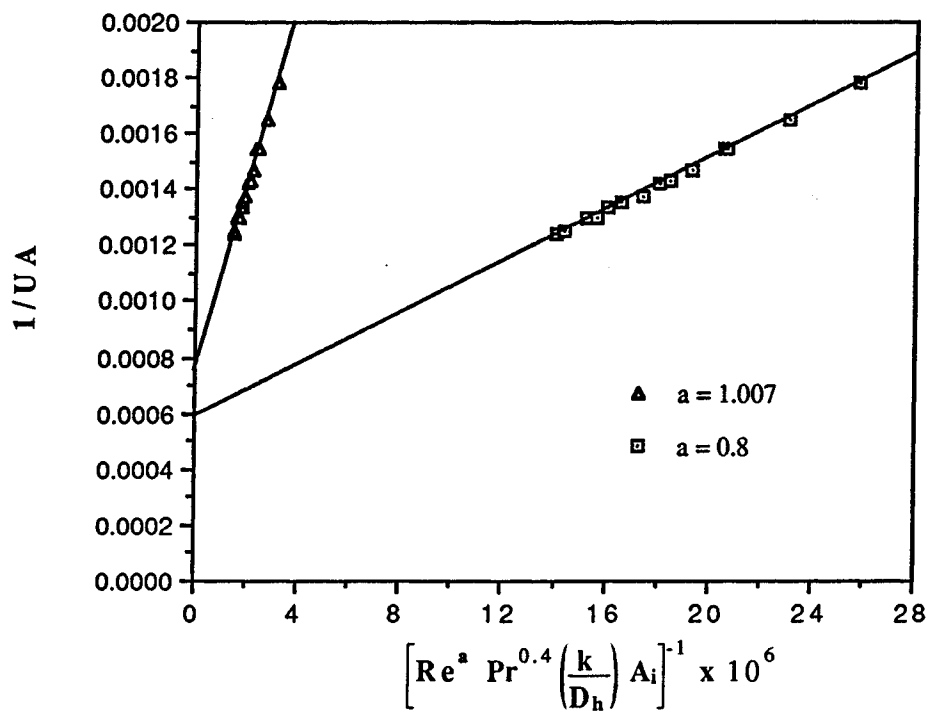


FIGURE 7.1
SINGLE-PHASE WILSON PLOT COMPARISON (CASE A-1 DATA SET)

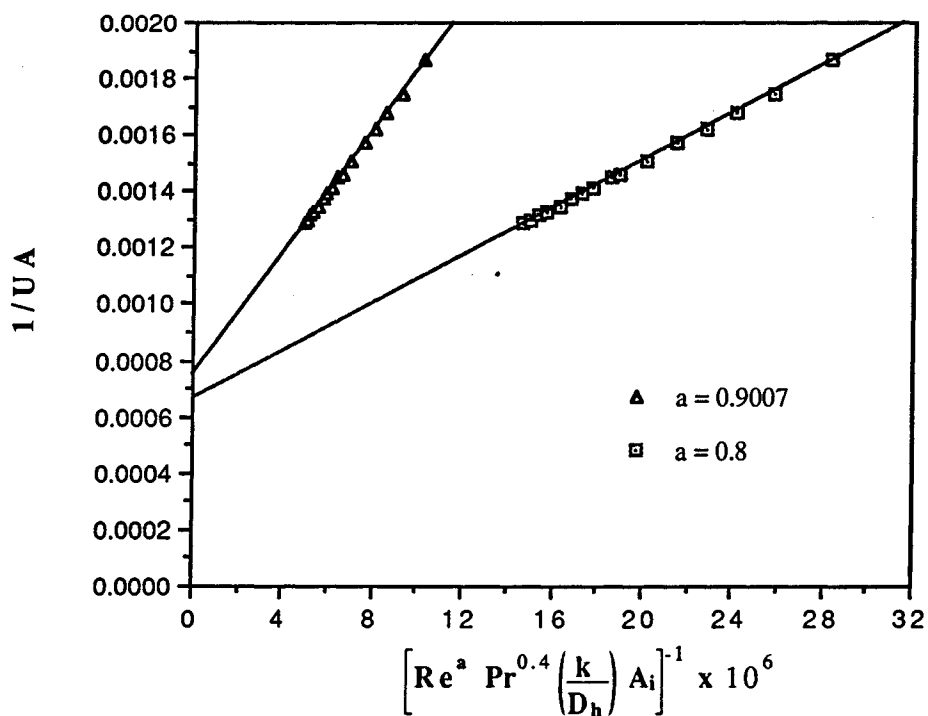


FIGURE 7.2
SINGLE-PHASE WILSON PLOT COMPARISON (CASE B-1 DATA SET)

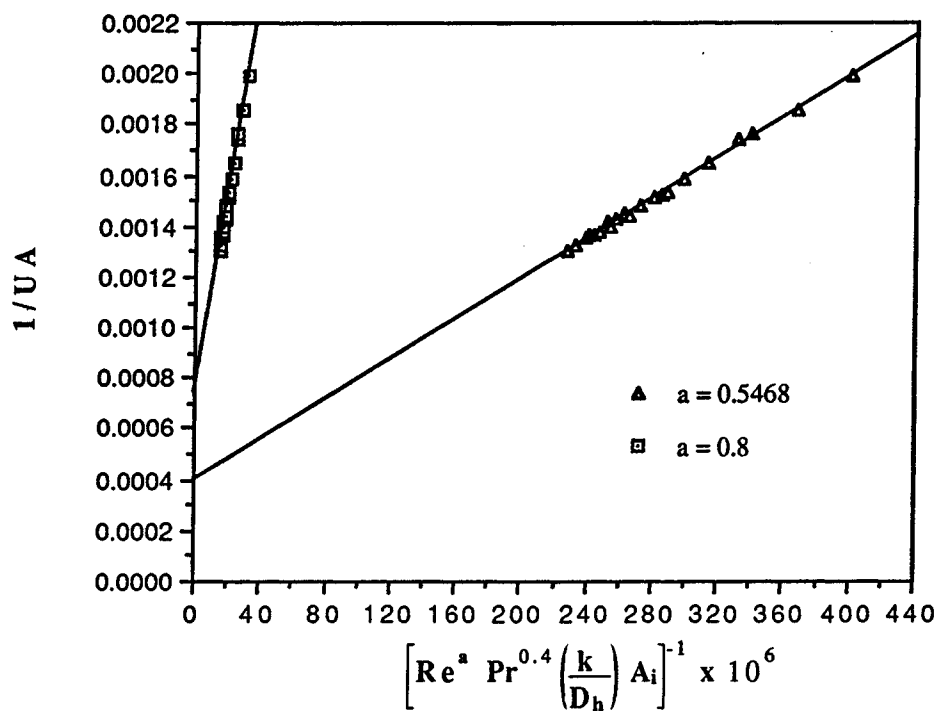


FIGURE 7.3
SINGLE-PHASE WILSON PLOT COMPARISON (CASE C-1 DATA SET)

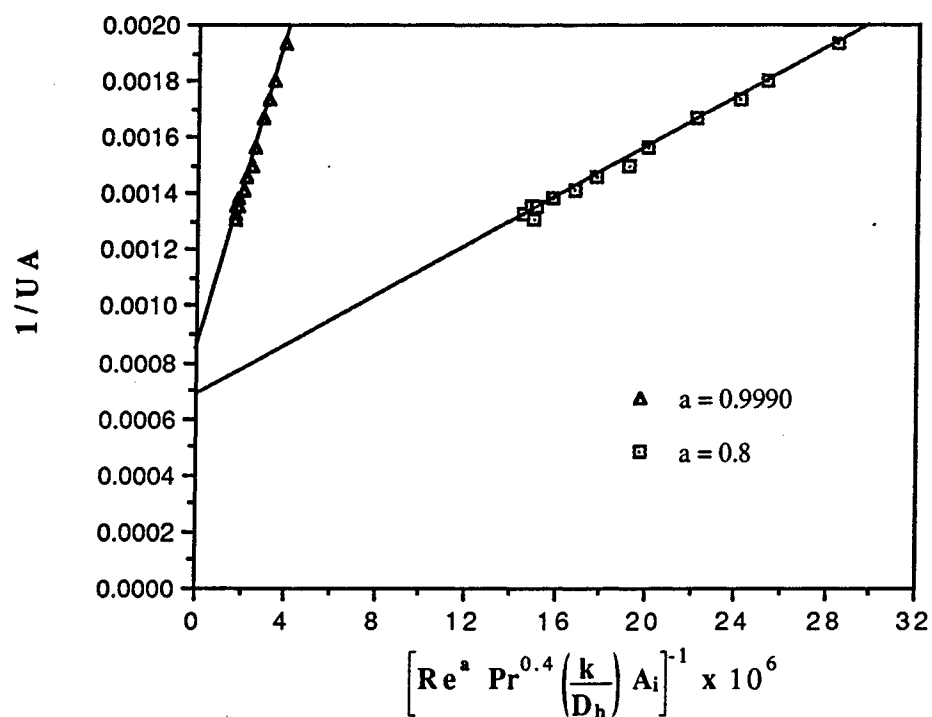


FIGURE 7.4
SINGLE-PHASE WILSON PLOT COMPARISON (CASE D-1 DATA SET)

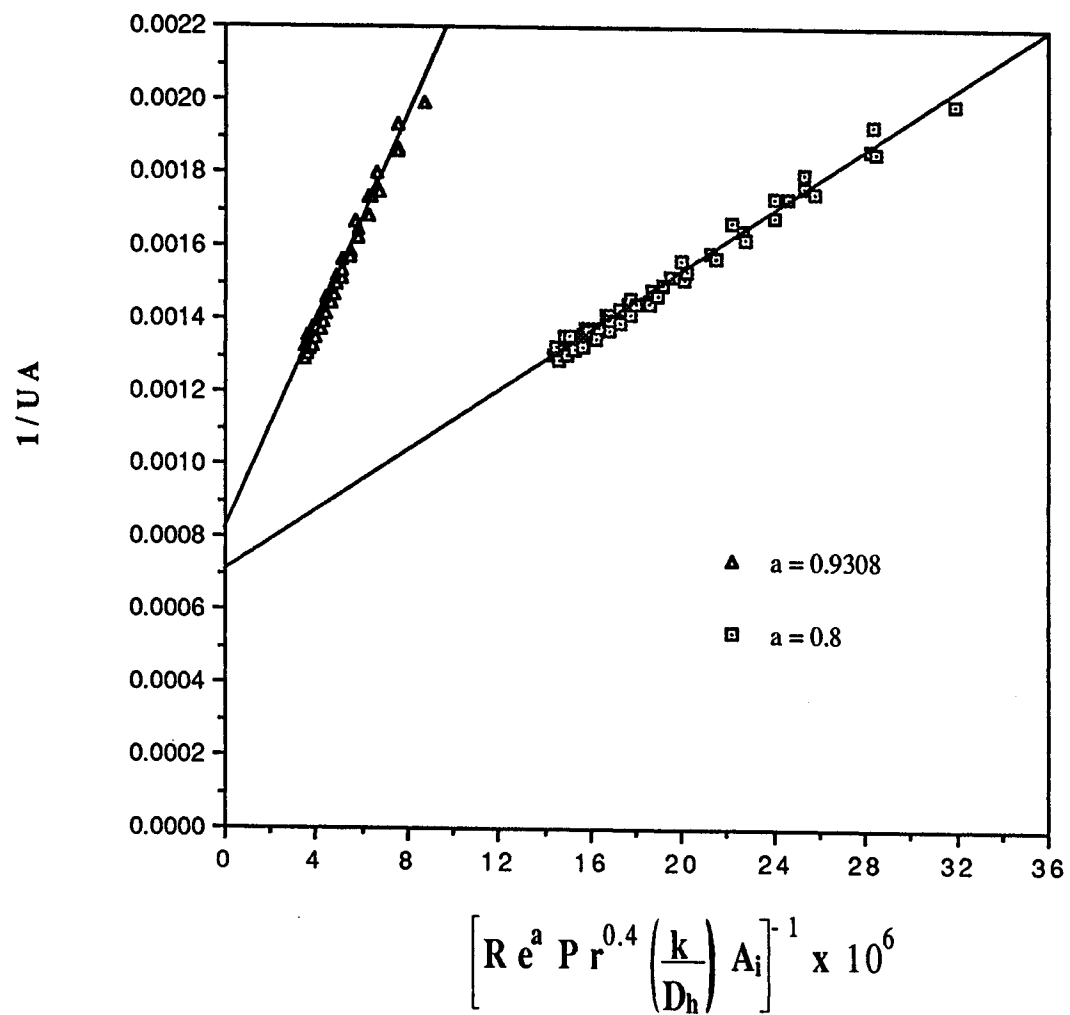


FIGURE 7.5
SINGLE-PHASE WILSON PLOT COMPARISON (CASE E-1 DATA SET)

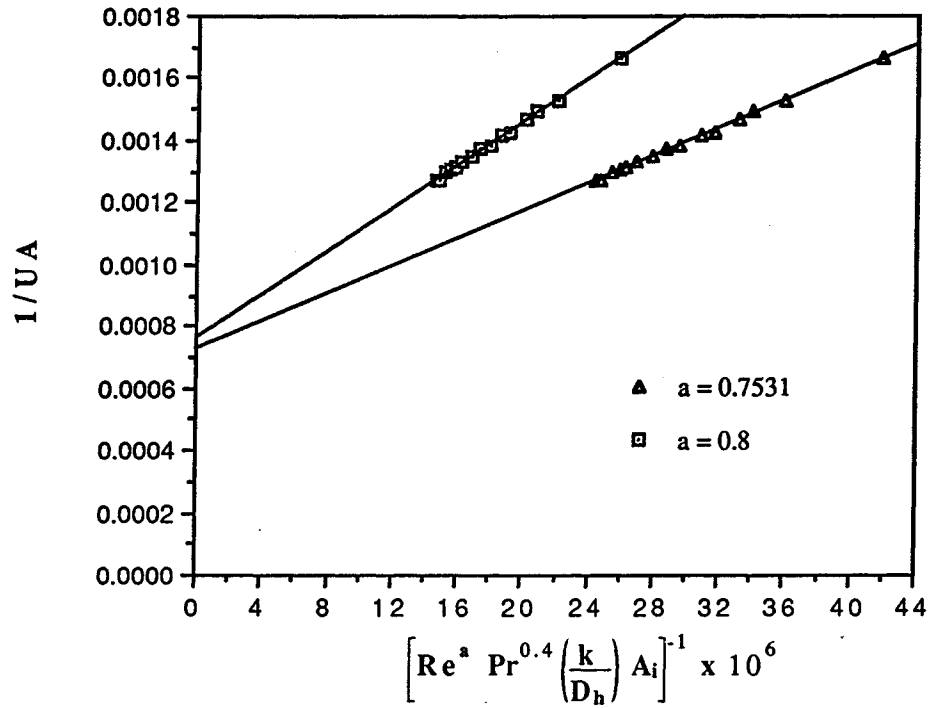


FIGURE 7.6
SINGLE-PHASE WILSON PLOT COMPARISON (CASE F-1 DATA SET)

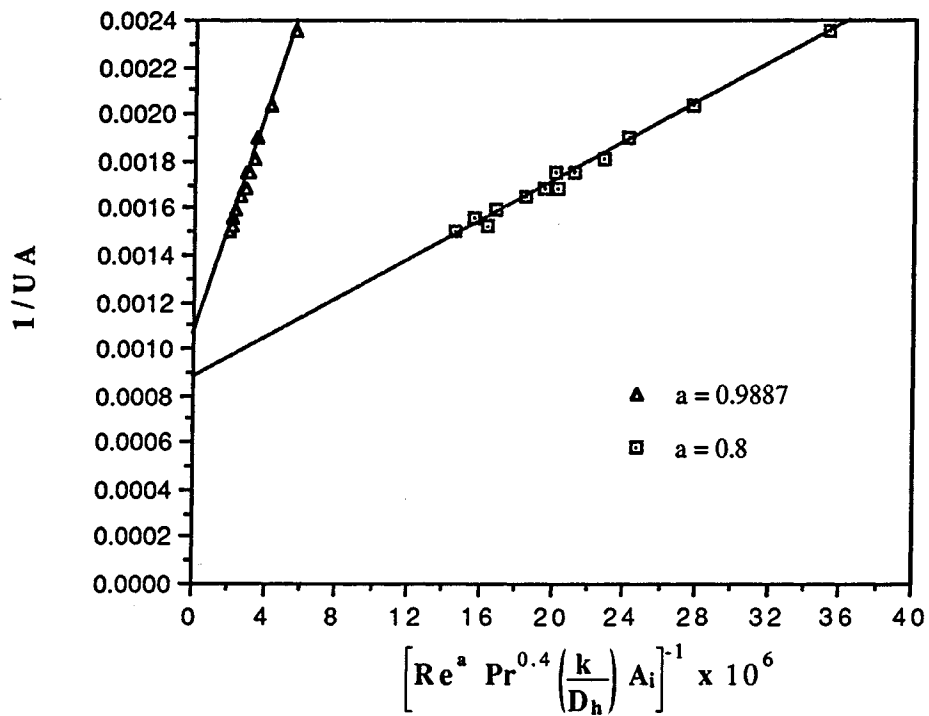


FIGURE 7.7
SINGLE-PHASE WILSON PLOT COMPARISON (CASE H-1 DATA SET)

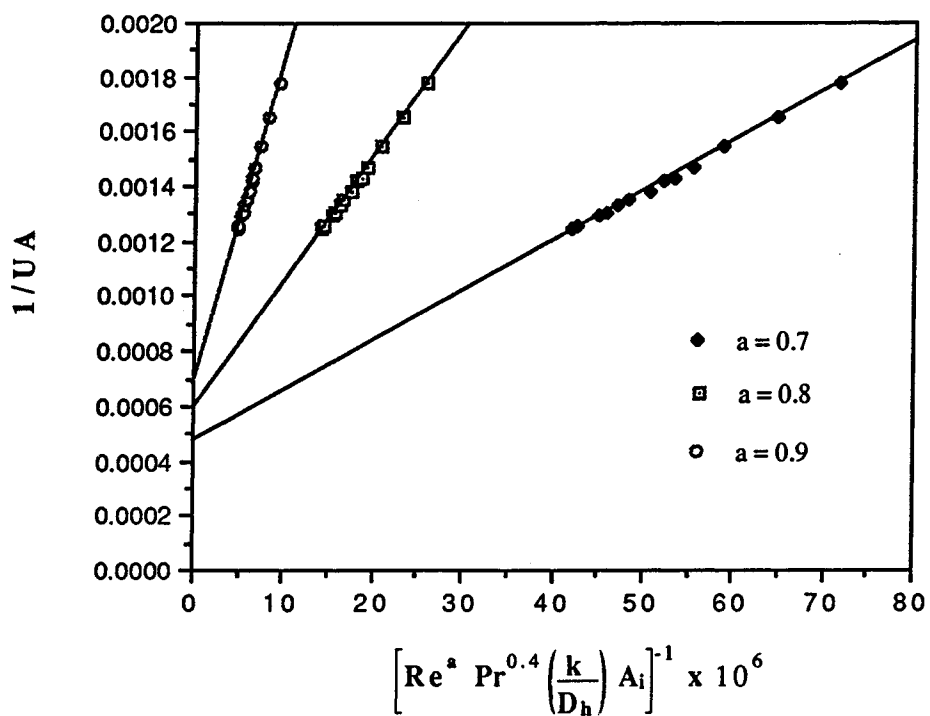


FIGURE 7.8
WILSON PLOT EXPONENT SENSITIVITY (CASE A-1 DATA SET)

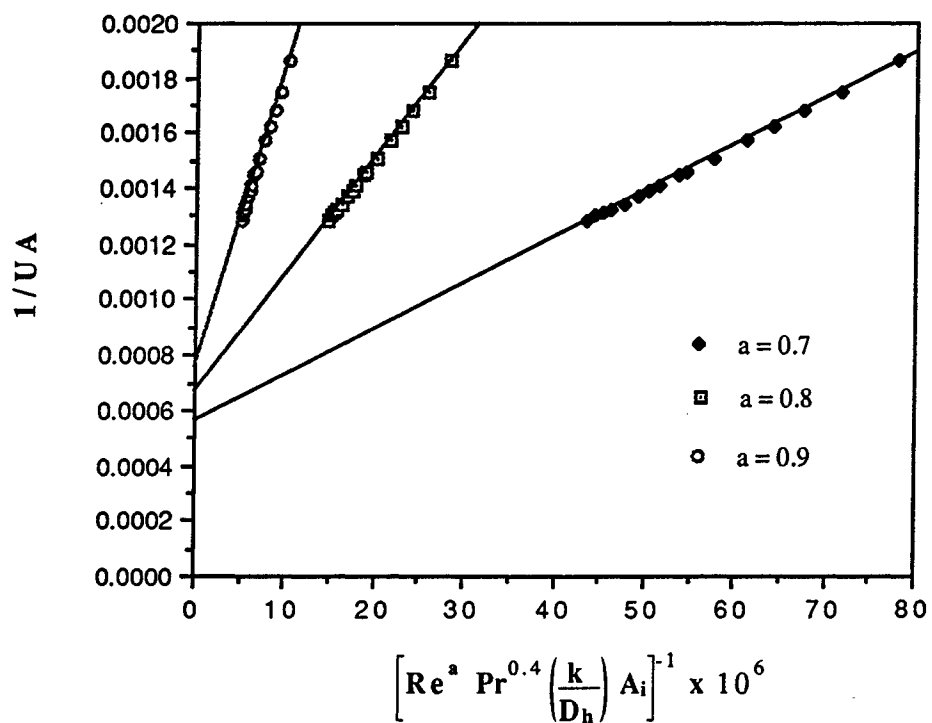


FIGURE 7.9
WILSON PLOT EXPONENT SENSITIVITY (CASE B-1 DATA SET)

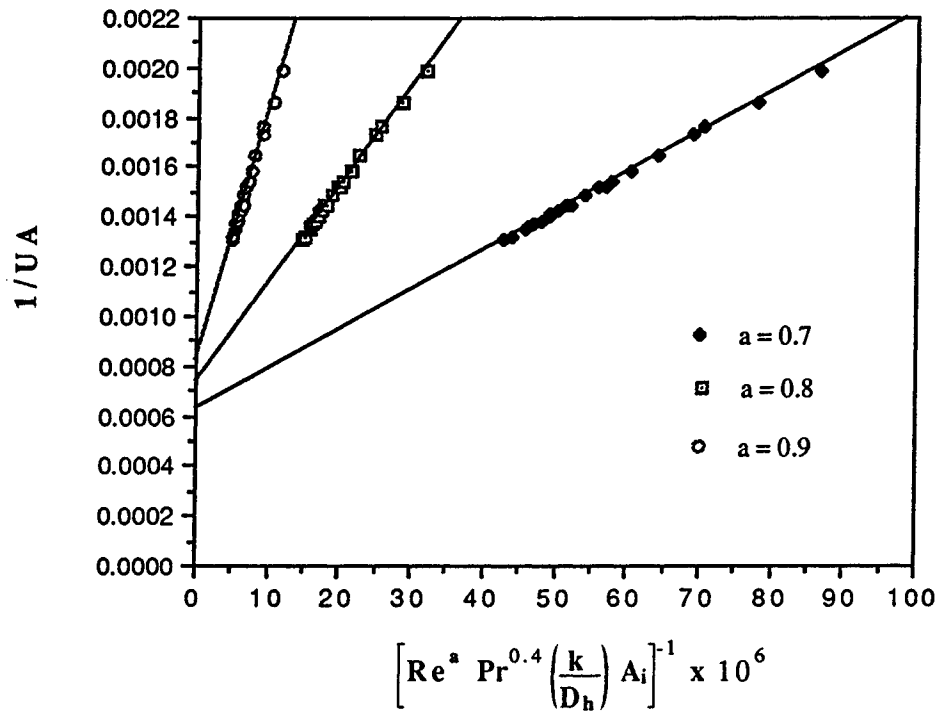


FIGURE 7.10
WILSON PLOT EXPONENT SENSITIVITY (CASE C-1 DATA SET)

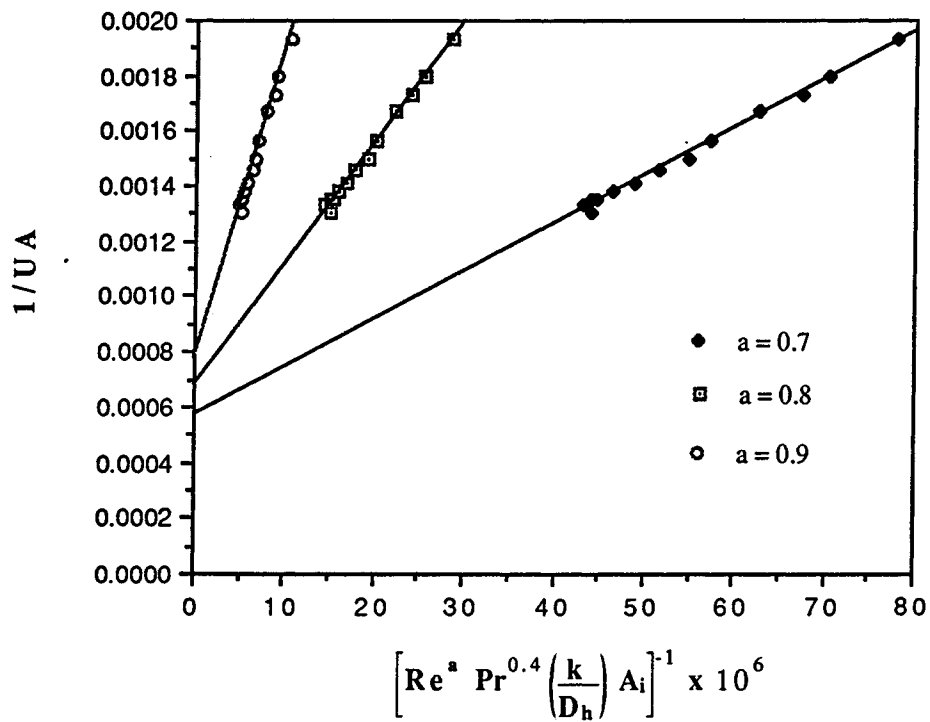


FIGURE 7.11
WILSON PLOT EXPONENT SENSITIVITY (CASE D-1 DATA SET)

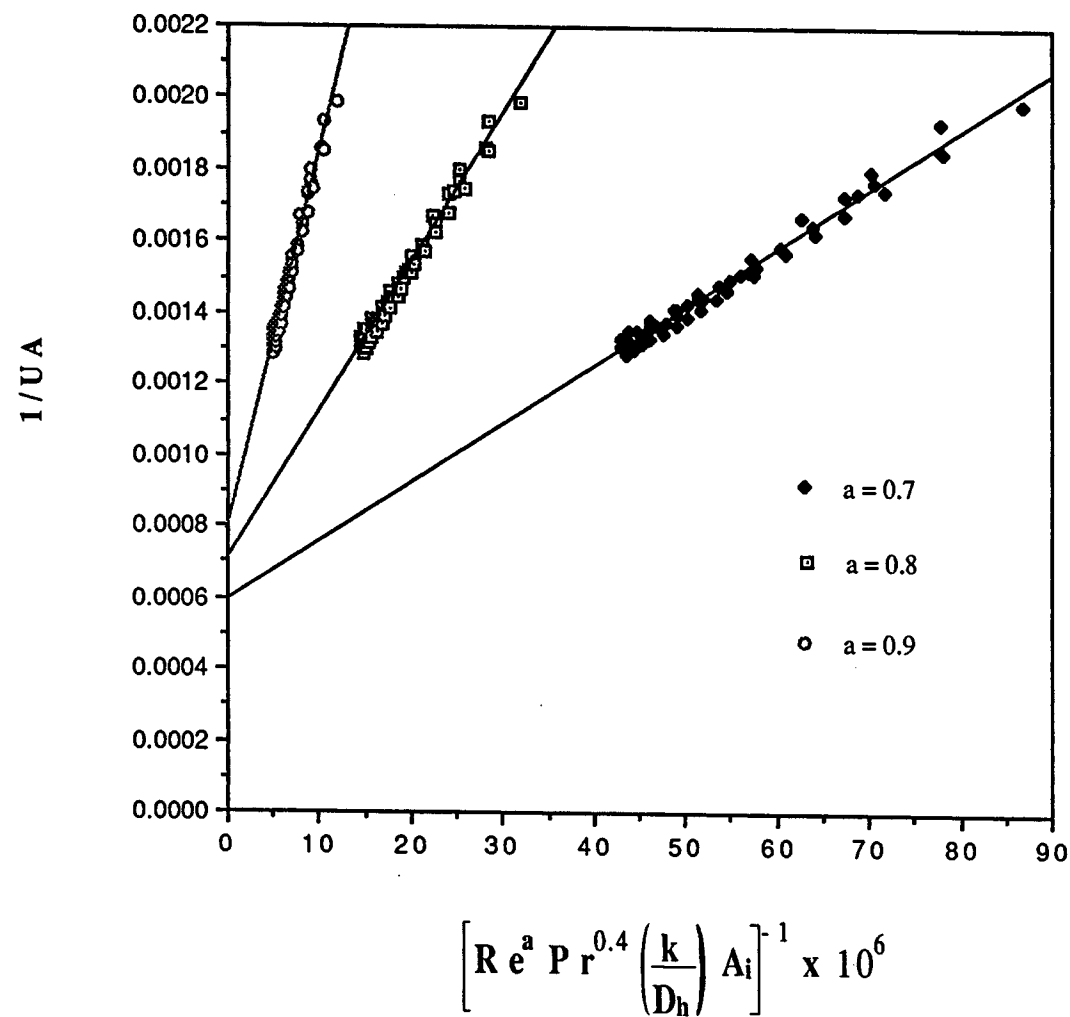


FIGURE 7.12
WILSON PLOT EXPONENT SENSITIVITY (CASE E-1 DATA SET)

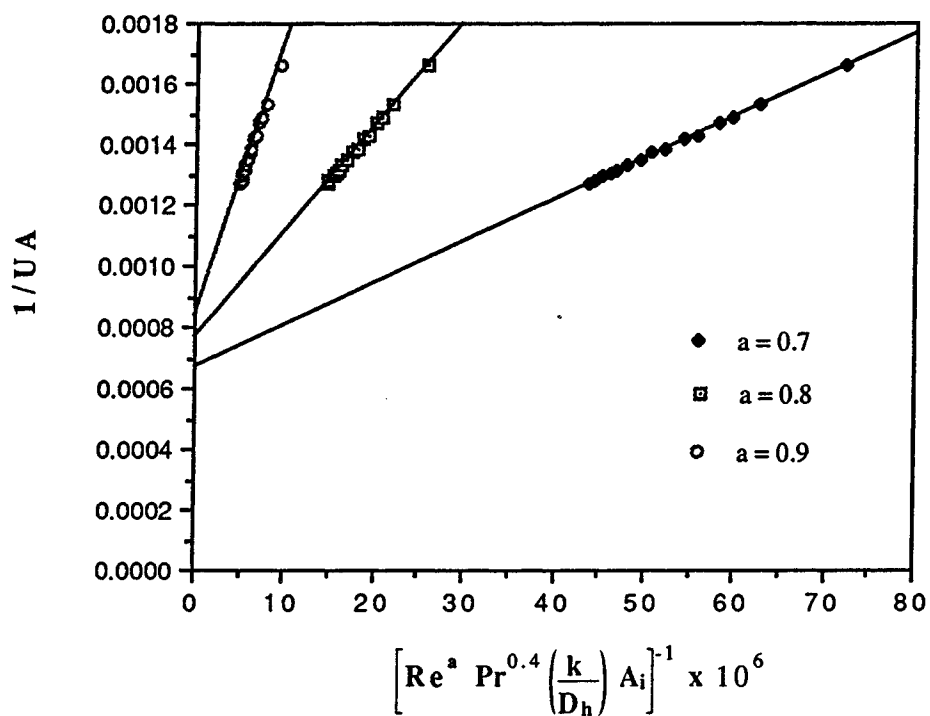


FIGURE 7.13
WILSON PLOT EXPONENT SENSITIVITY (CASE F-1 DATA SET)

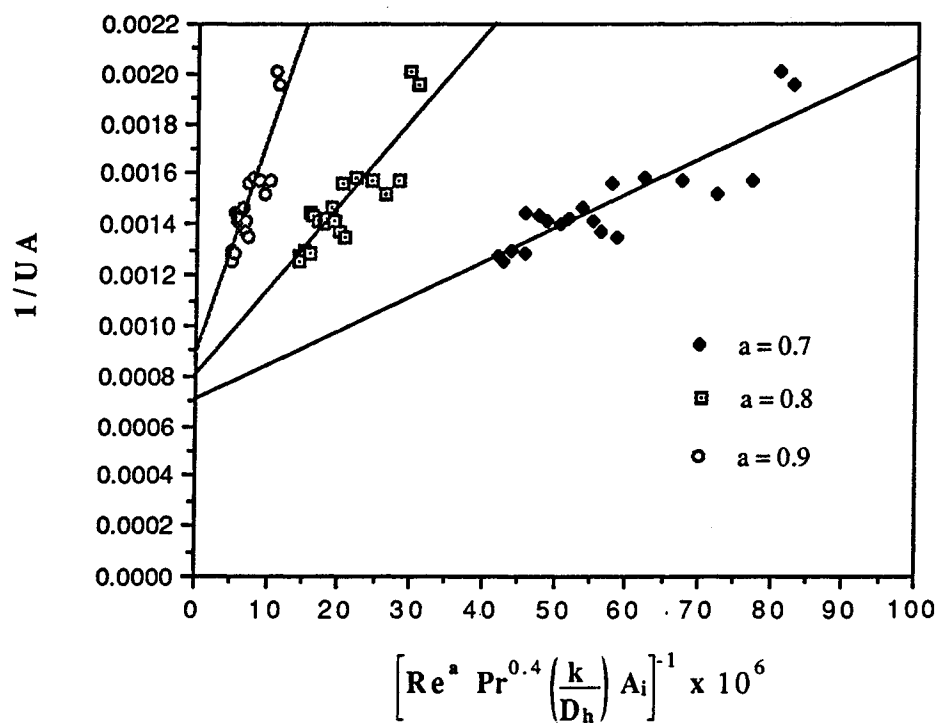


FIGURE 7.14
WILSON PLOT EXPONENT SENSITIVITY (CASE G-1 DATA SET)

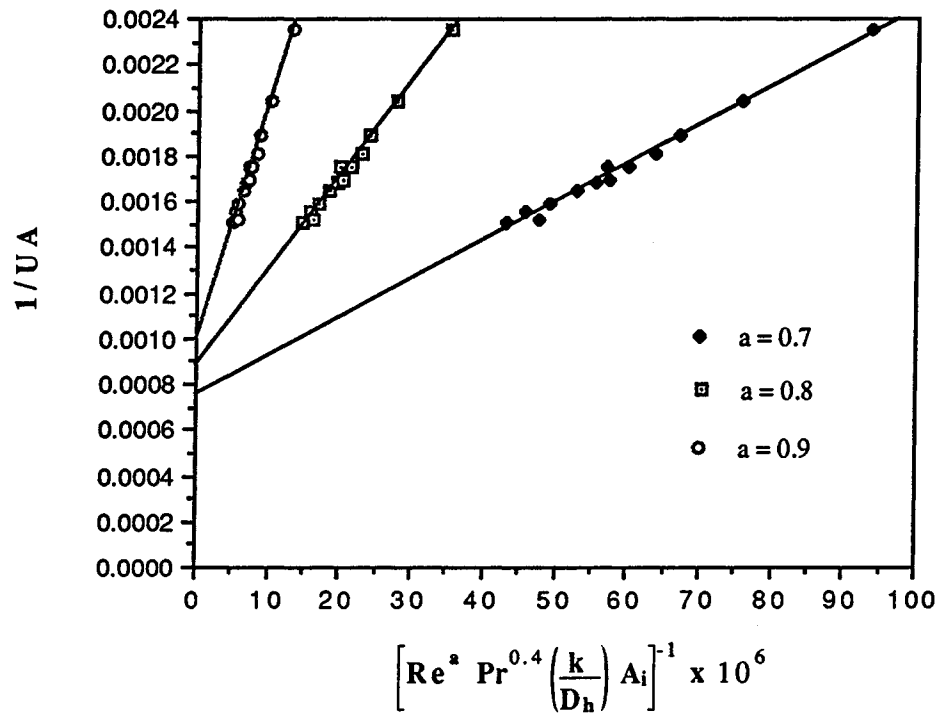


FIGURE 7.15
WILSON PLOT EXPONENT SENSITIVITY (CASE H-1 DATA SET)

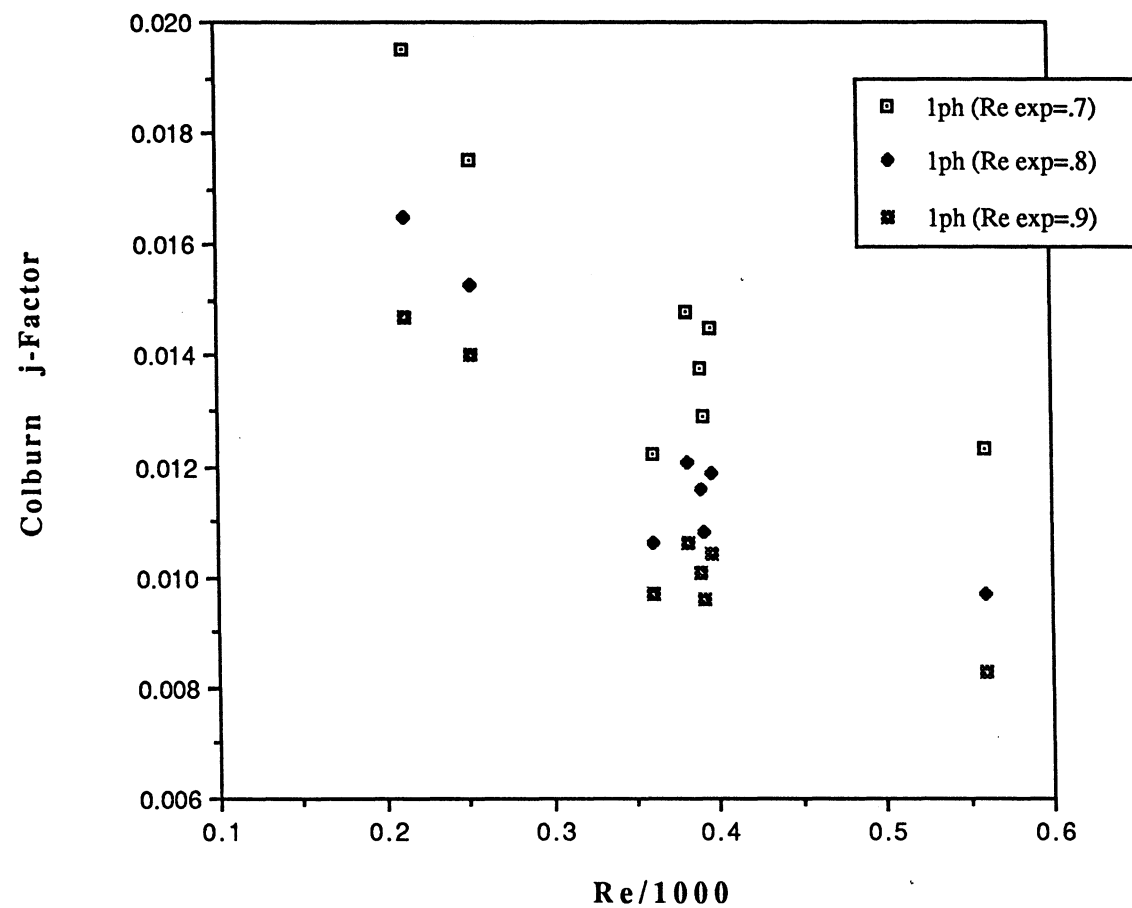


FIGURE 7.16
j-FACTOR VS REYNOLDS NUMBER
SINGLE PHASE DATA - REYNOLDS NO. EXPONENT SENSITIVITY

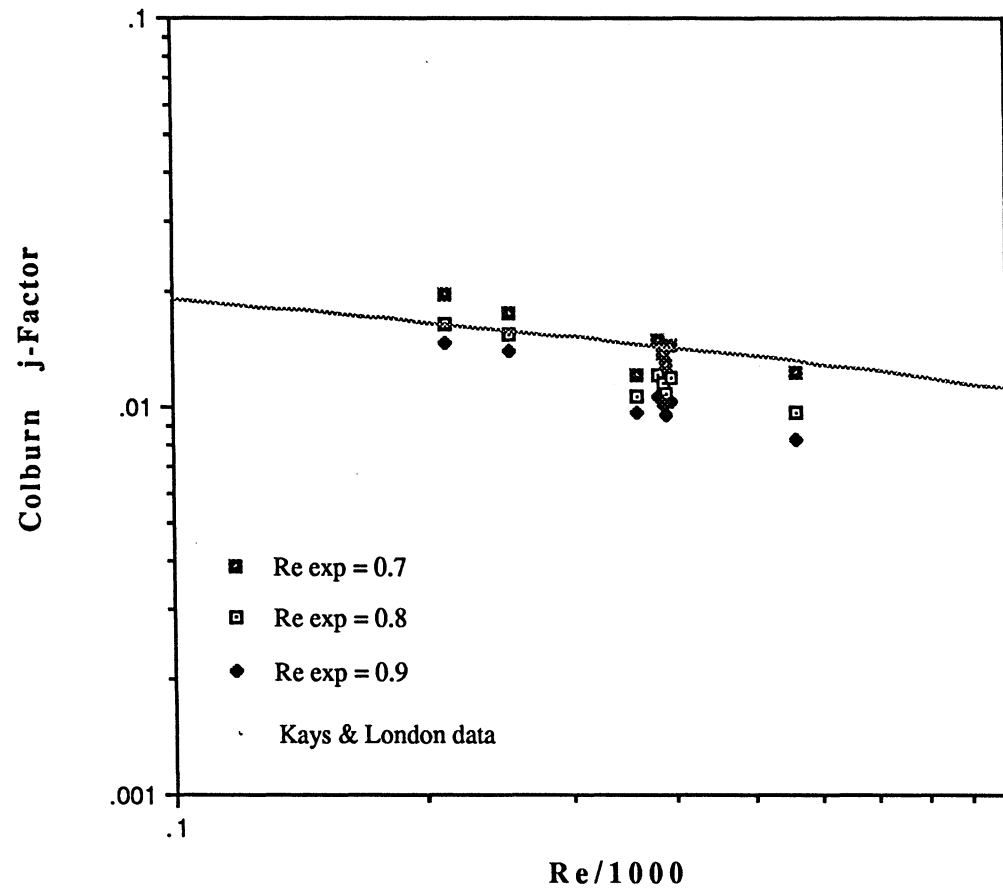


FIGURE 7.17
 j-FACTOR VS REYNOLDS NUMBER
 SINGLE PHASE DATA COMPARISON WITH KAYS & LONDON DATA

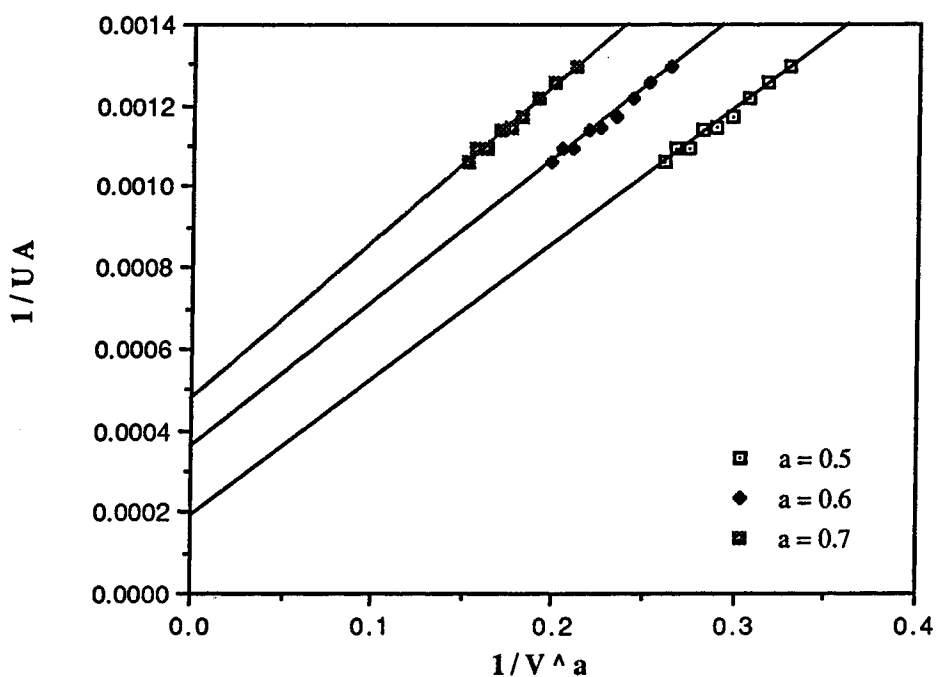


FIGURE 7.18
WILSON PLOT - TWO PHASE DATA (CASE A-2)

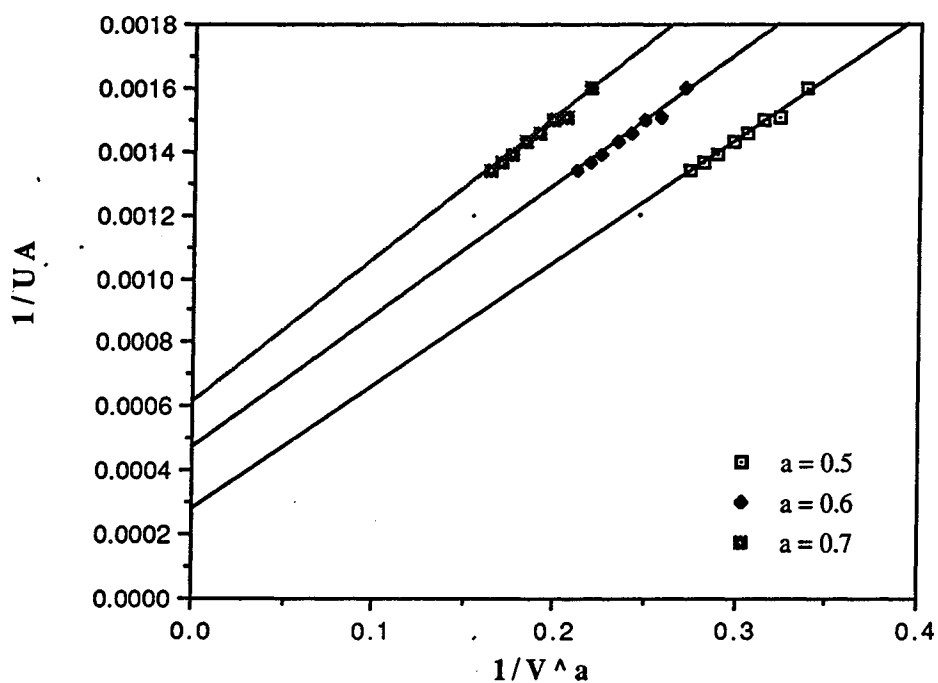


FIGURE 7.19
WILSON PLOT - TWO PHASE DATA (CASE B-2)

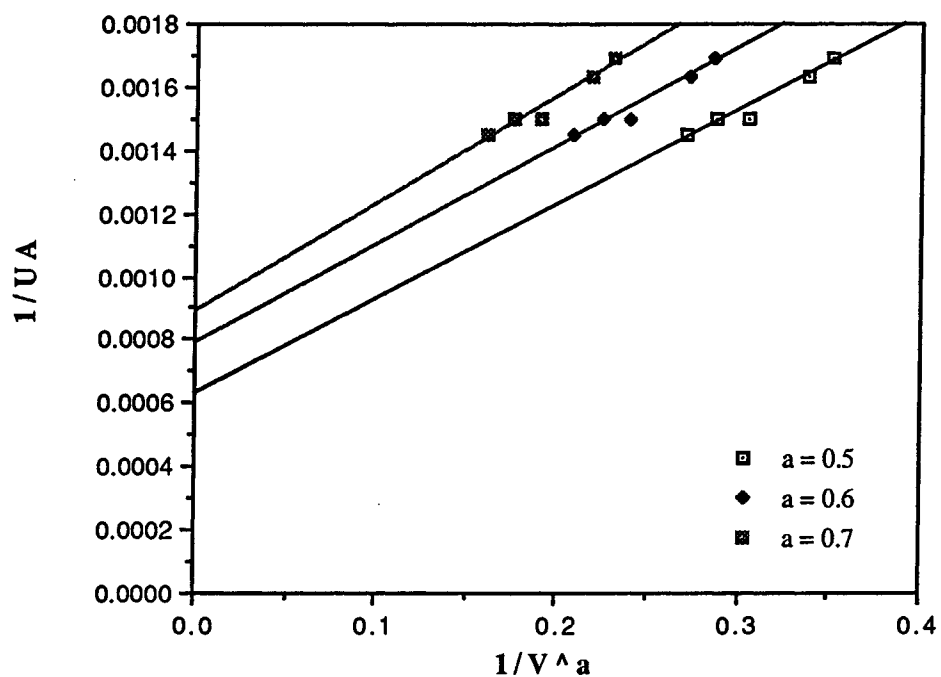


FIGURE 7.20
WILSON PLOT - TWO PHASE DATA (CASE C-2)

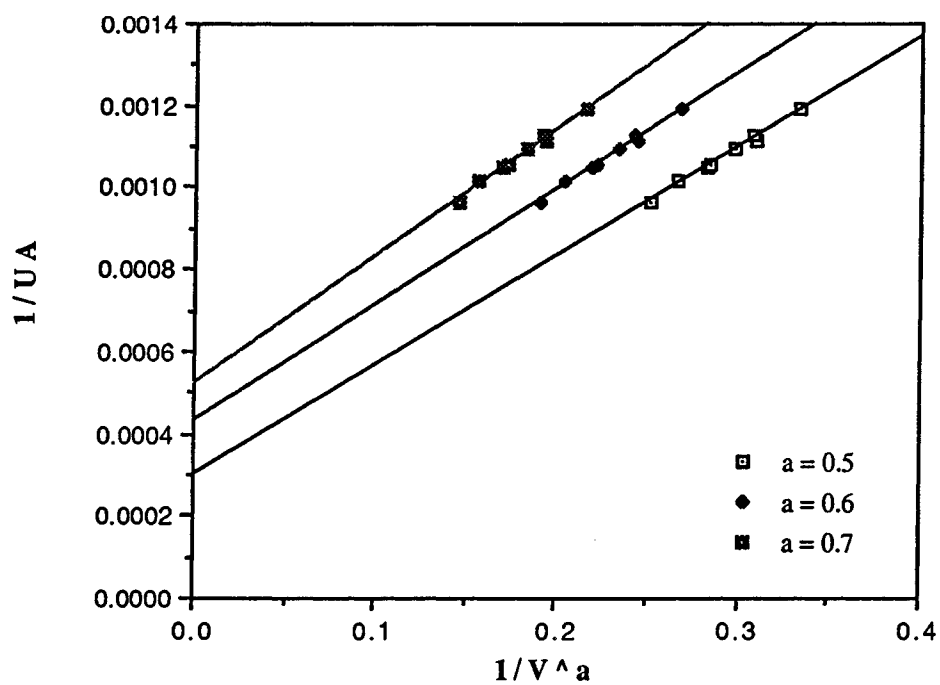


FIGURE 7.21
WILSON PLOT - TWO PHASE DATA (CASE D-2)

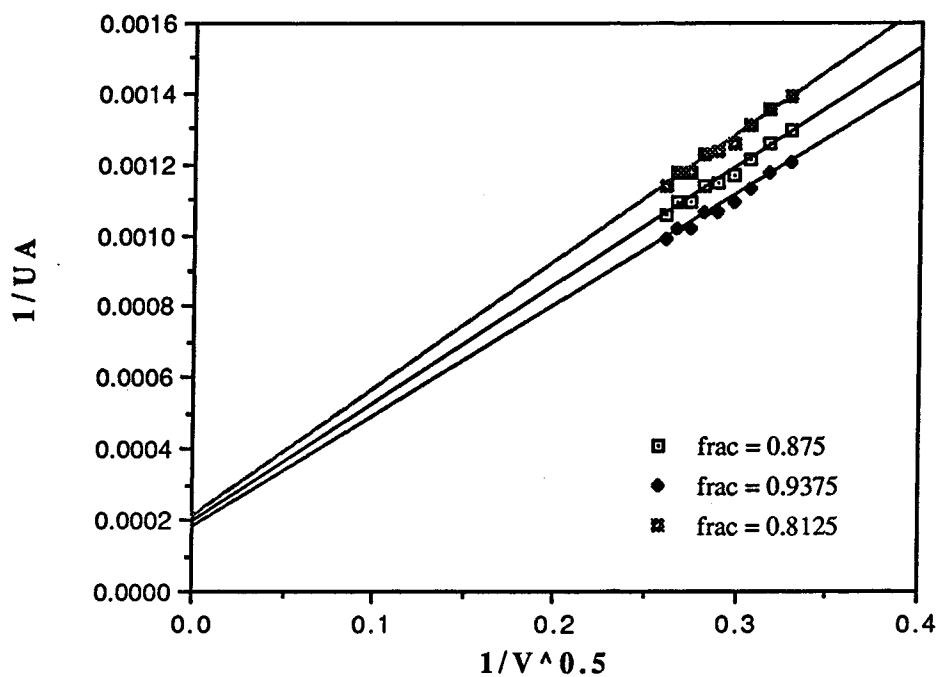


FIGURE 7.22
CONDENSING FRACTION INFLUENCE (CASE A-2 DATA)

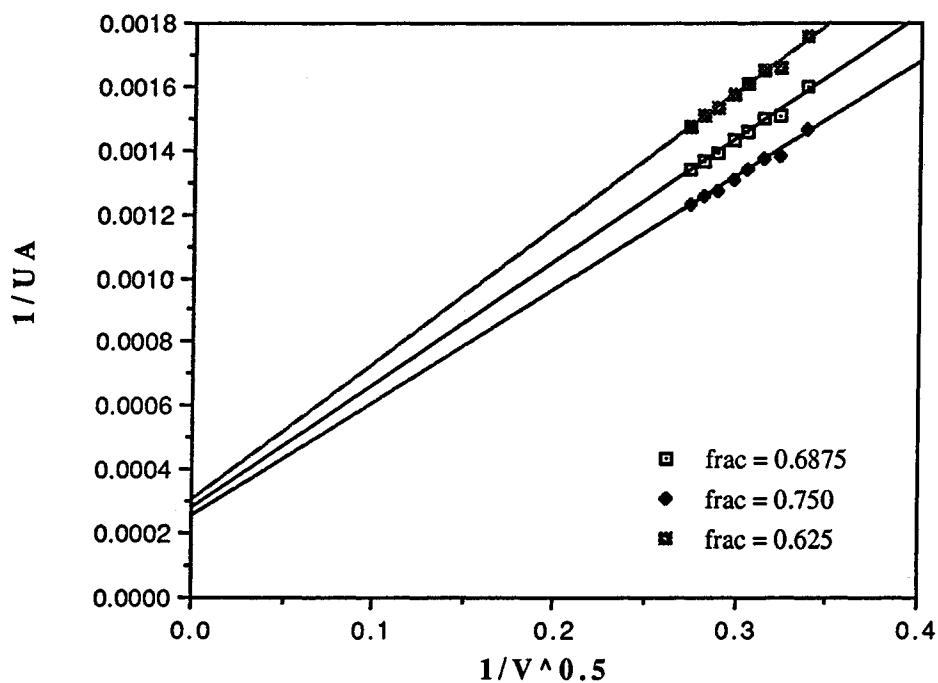


FIGURE 7.23
CONDENSING FRACTION INFLUENCE (CASE B-2 DATA)

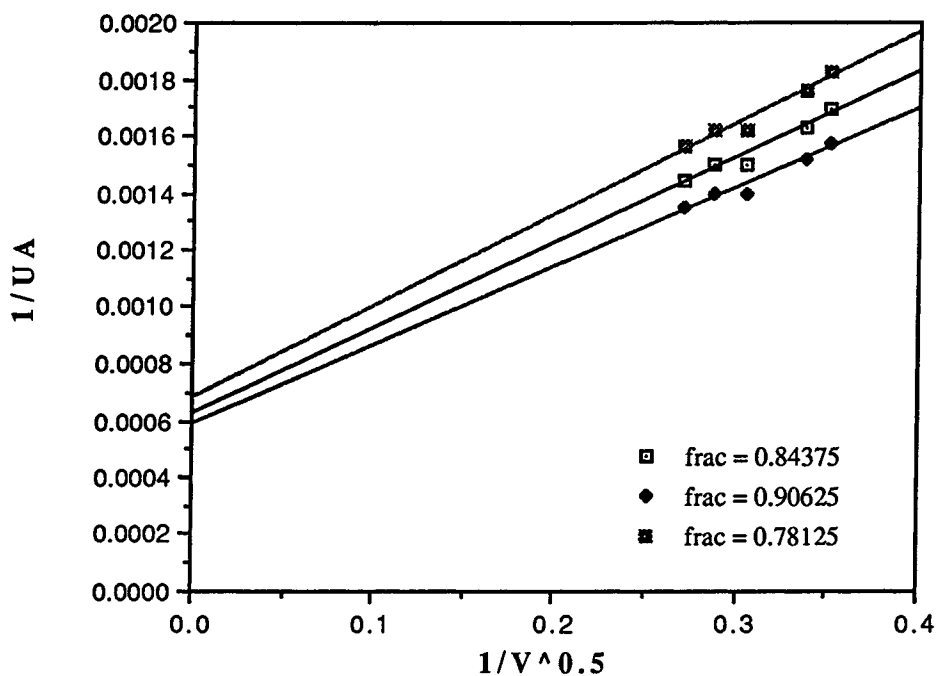


FIGURE 7.24
CONDENSING FRACTION INFLUENCE (CASE C-2 DATA)

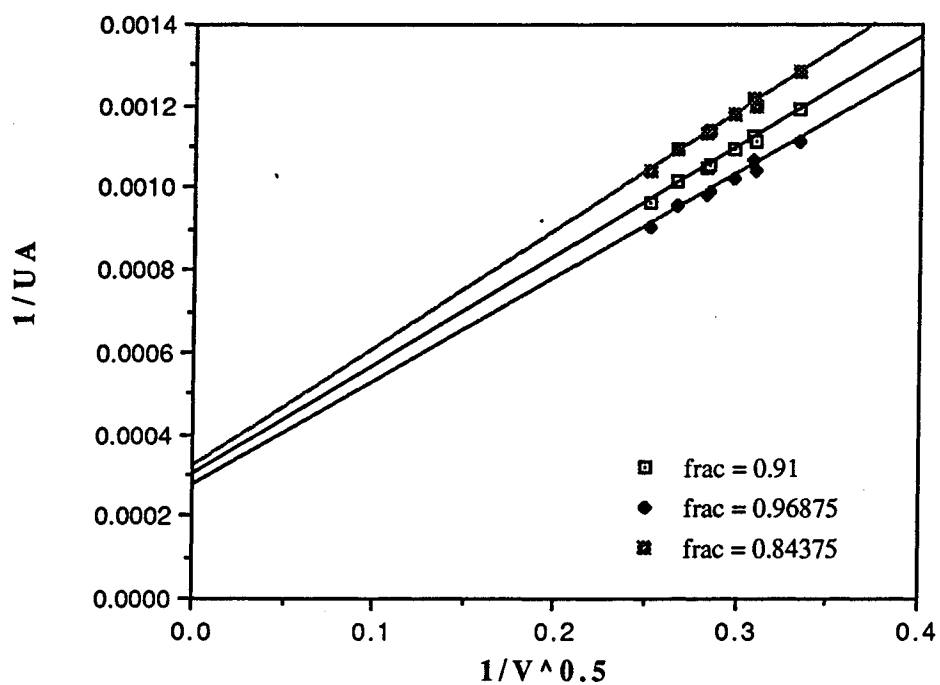


FIGURE 7.25
CONDENSING FRACTION INFLUENCE (CASE D-2 DATA)

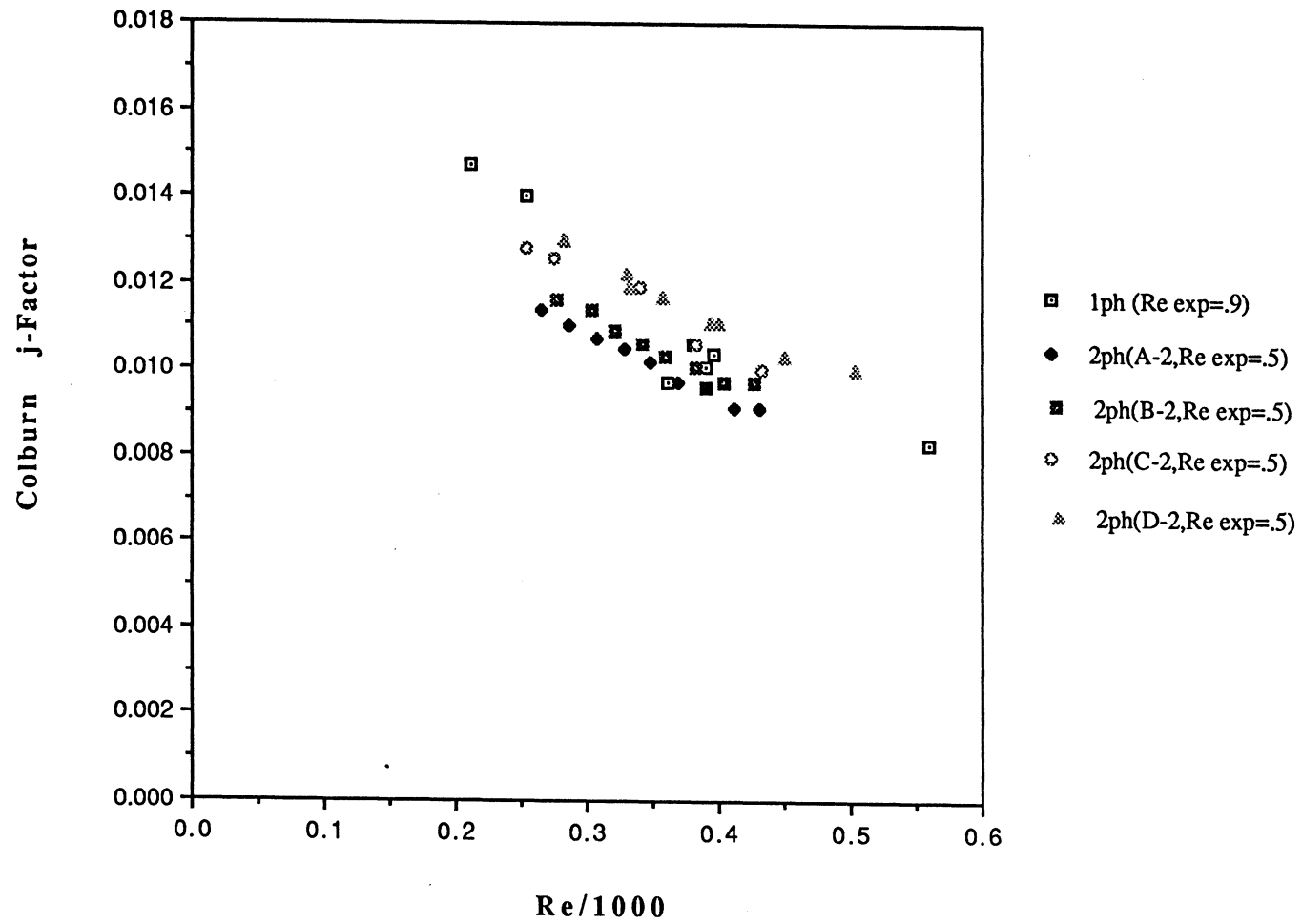


FIGURE 7.26
CORRELATION OF j , Re DATA

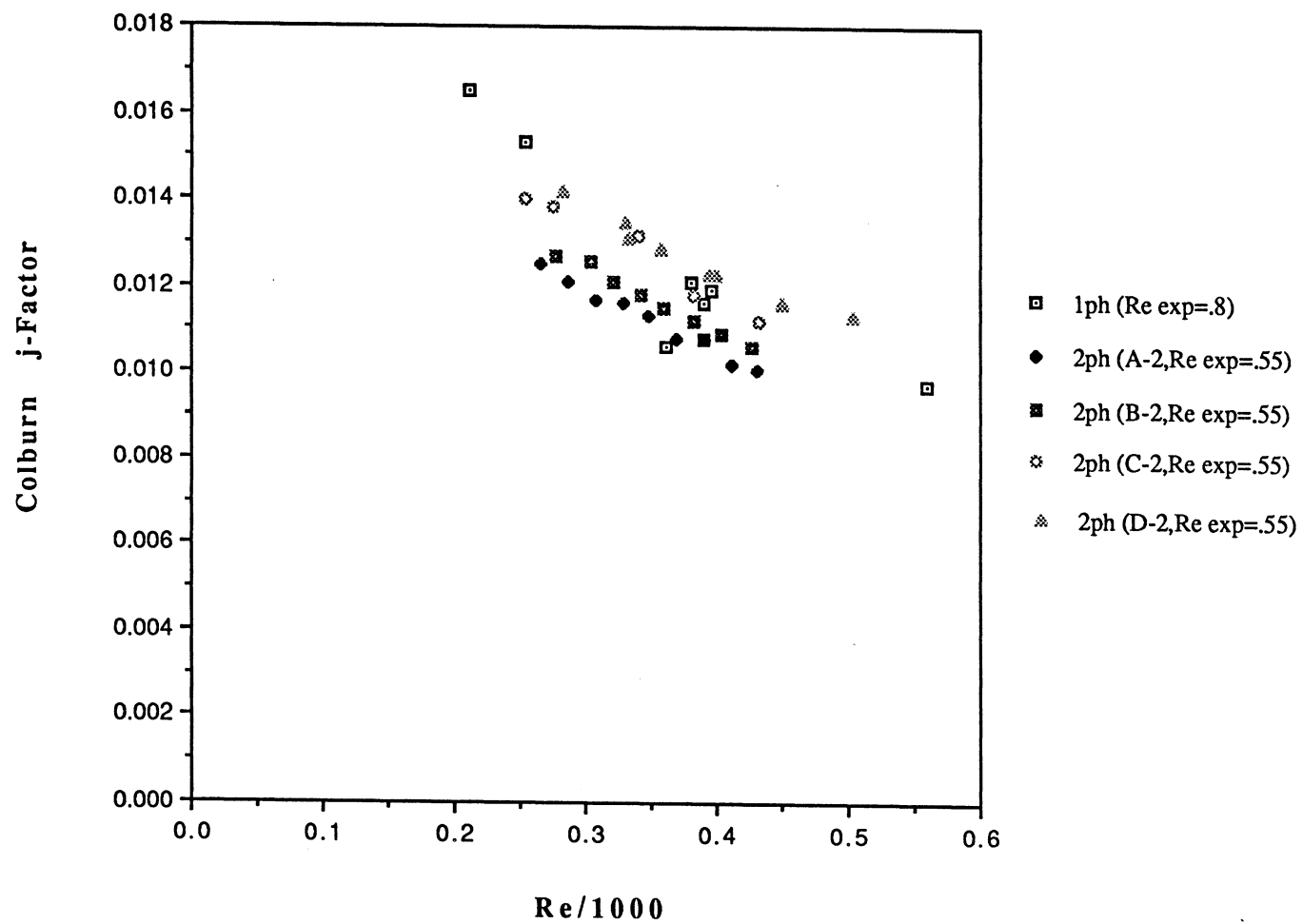


FIGURE 7.27
CORRELATION OF j , Re DATA

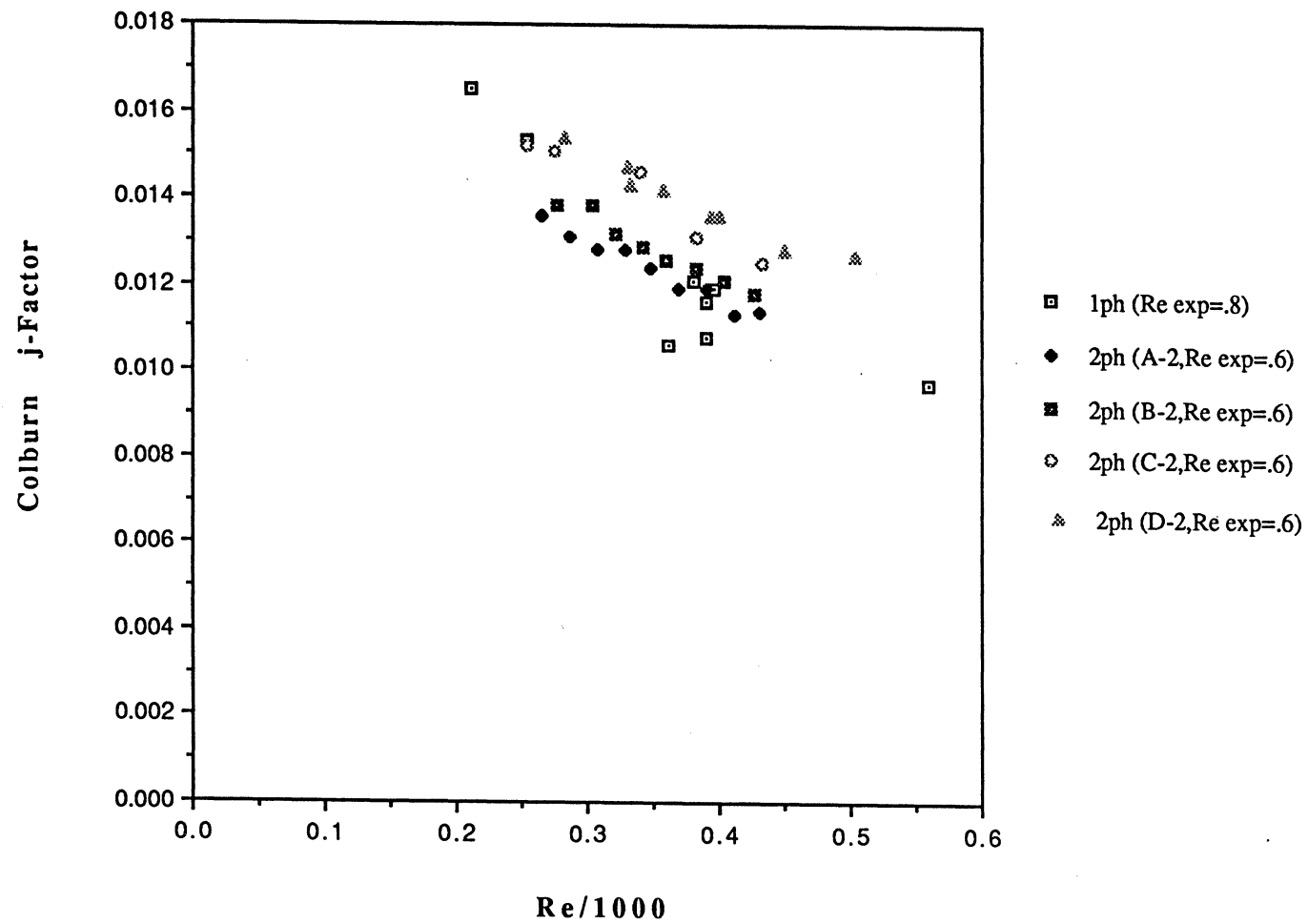


FIGURE 7.28
CORRELATION OF j , Re DATA

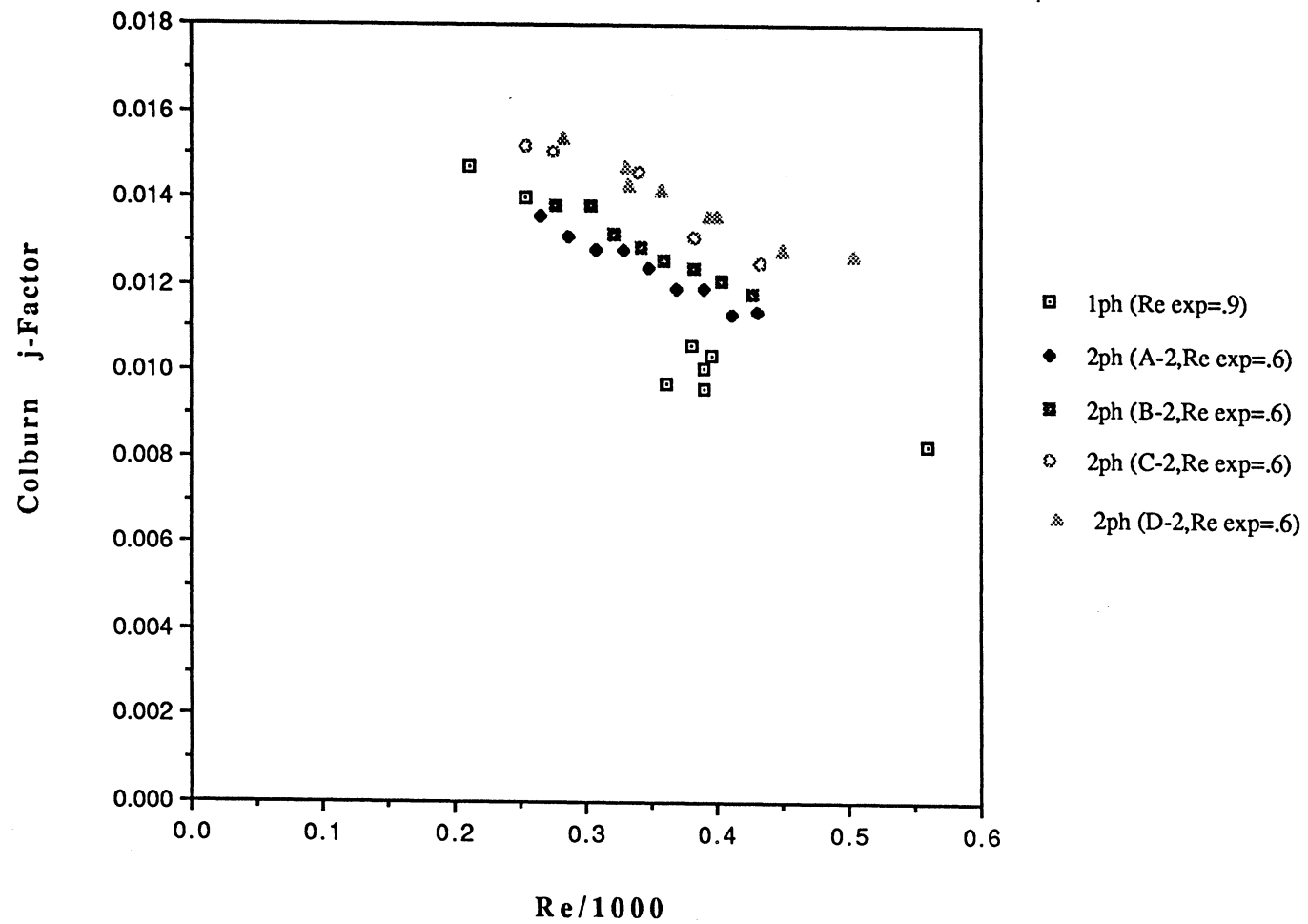


FIGURE 7.29
CORRELATION OF j , Re DATA

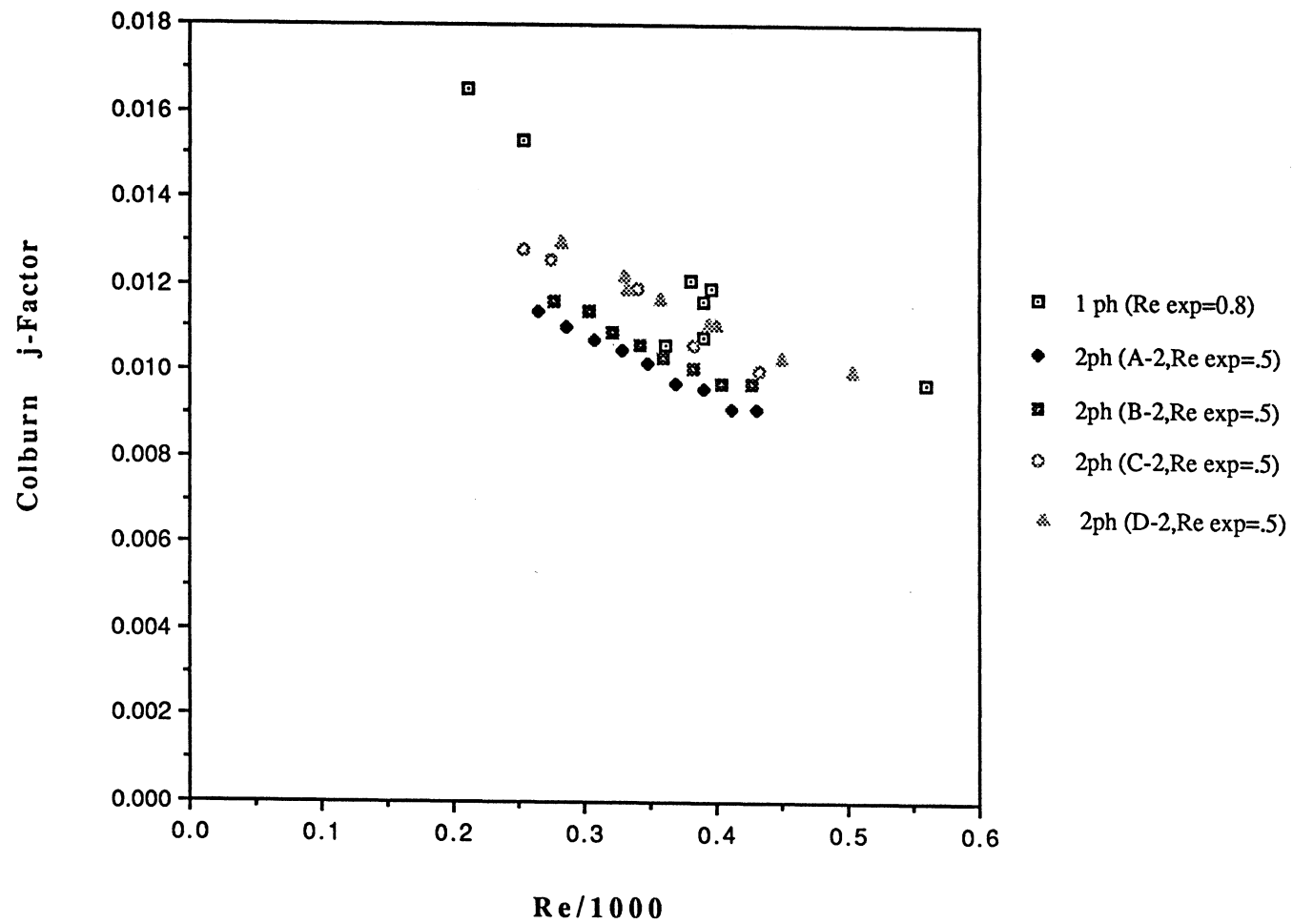


FIGURE 7.30
CORRELATION OF j , Re DATA

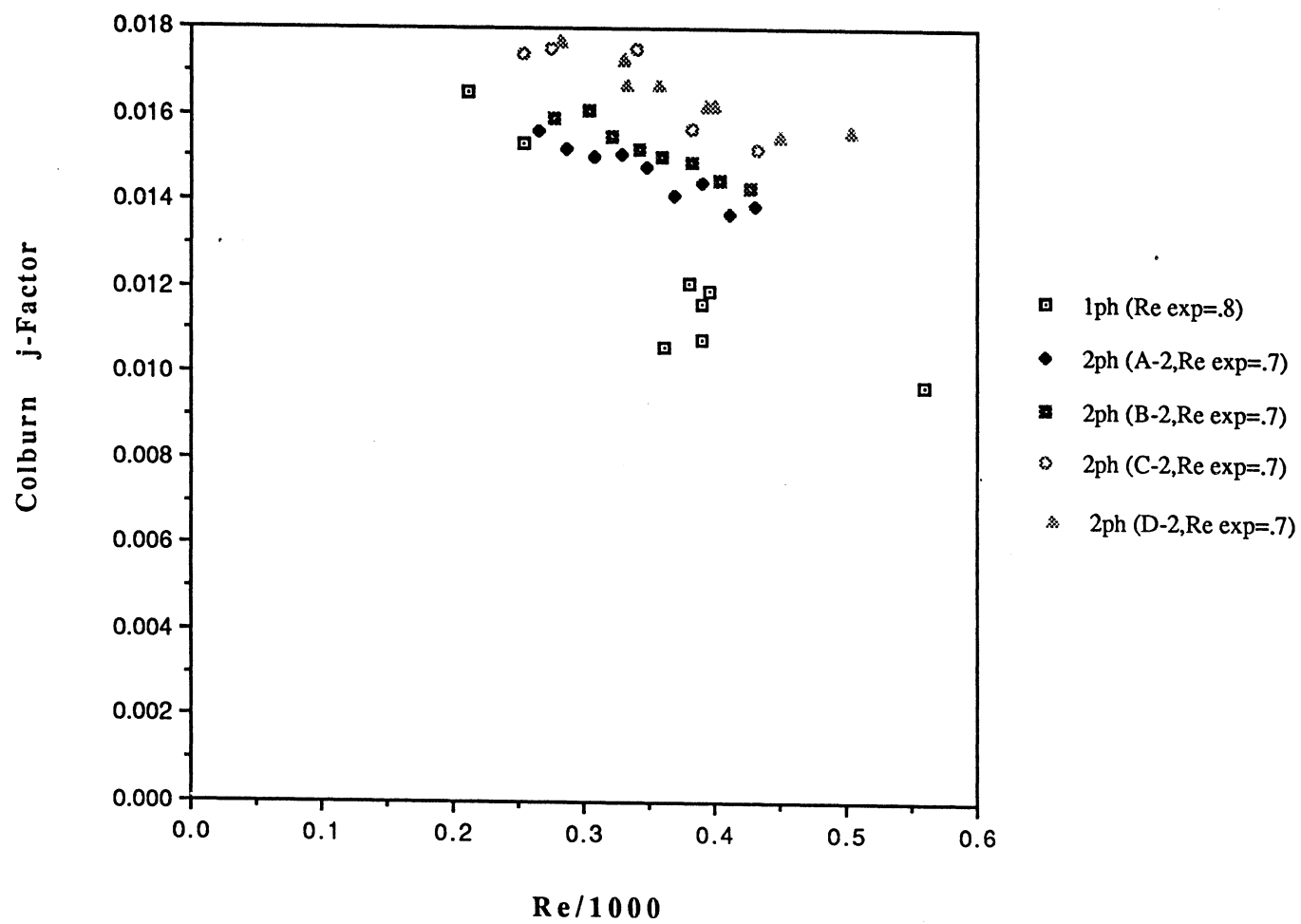


FIGURE 7.31
CORRELATION OF j , Re DATA

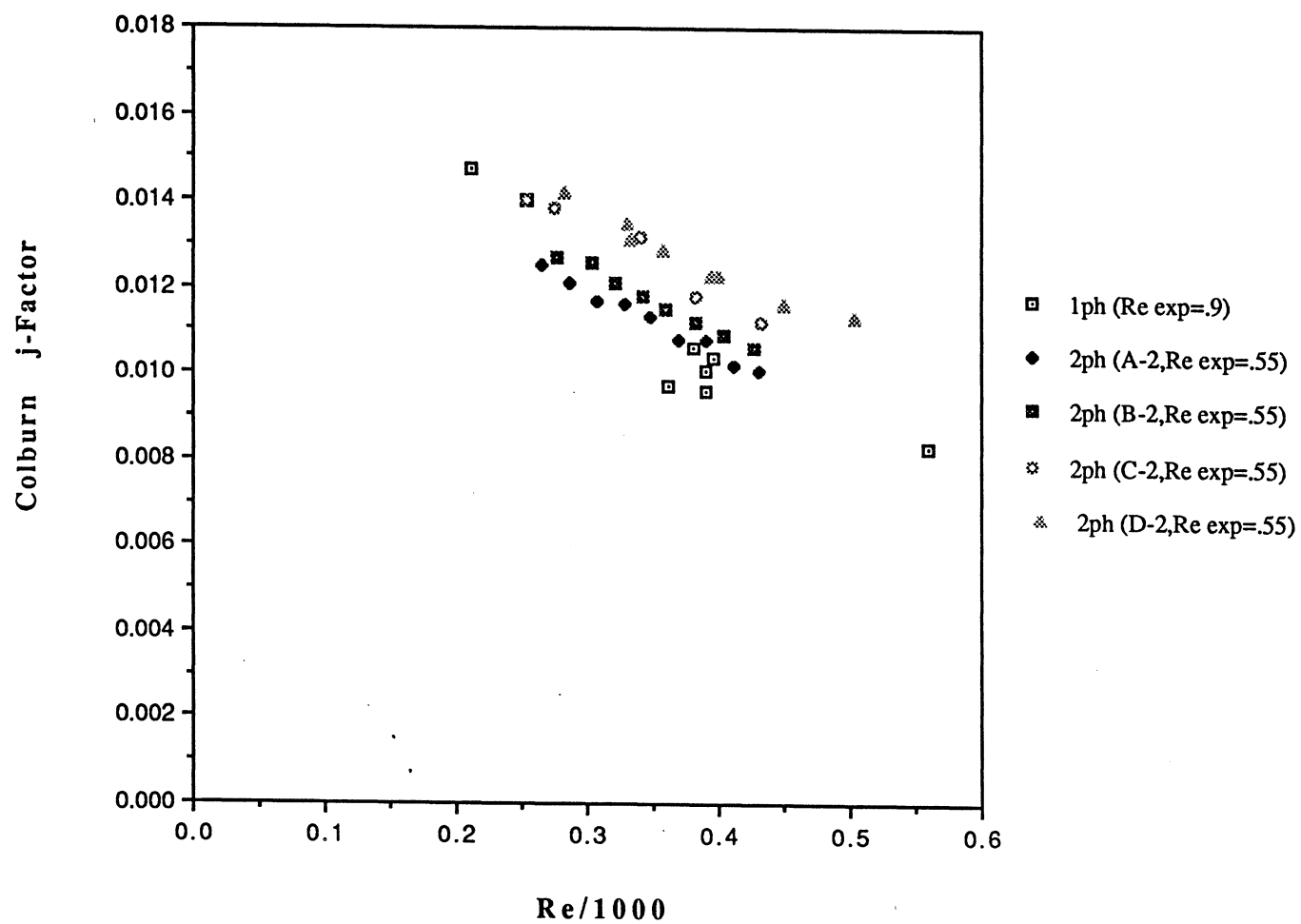


FIGURE 7.32
CORRELATION OF j , Re DATA

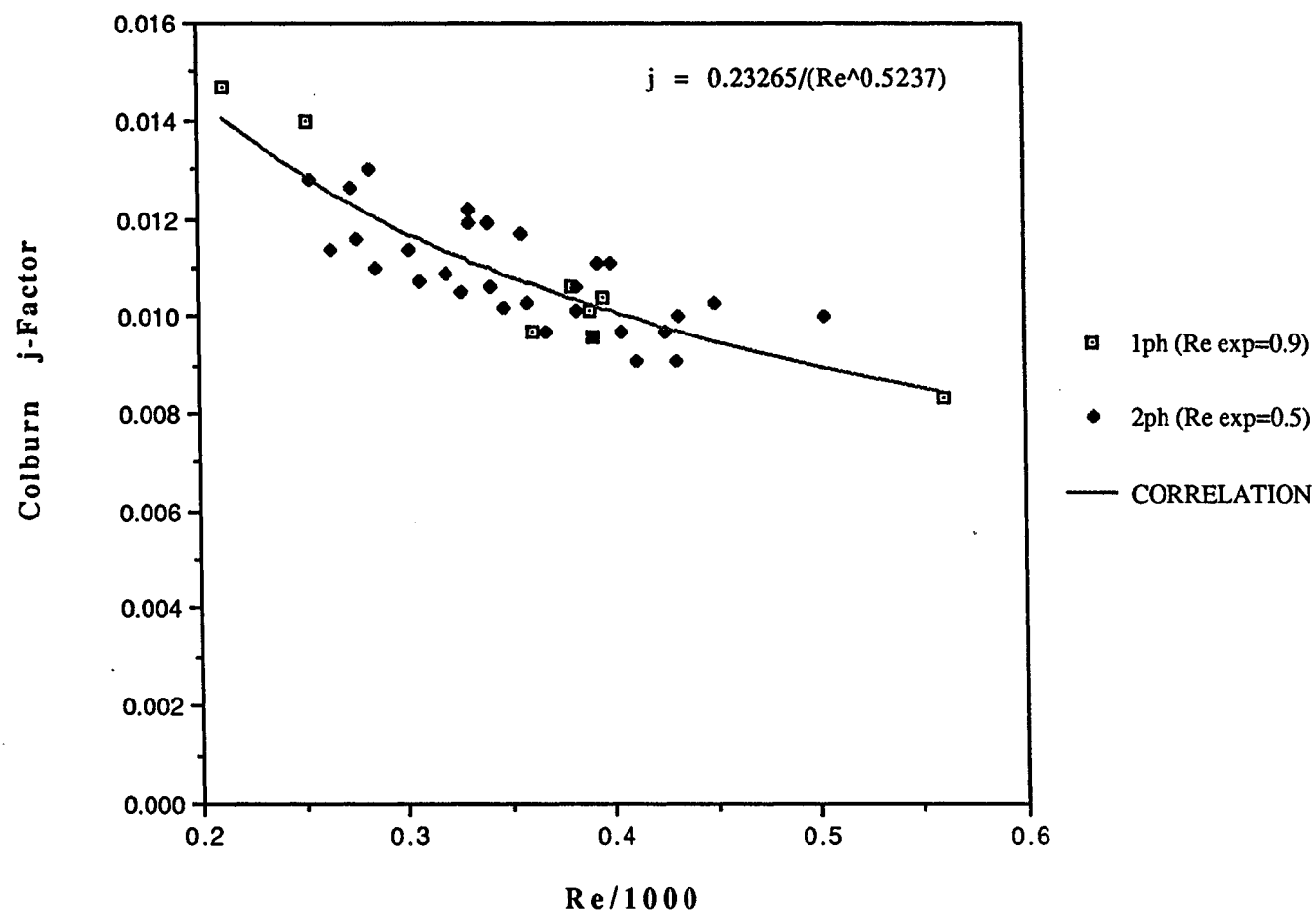


FIGURE 7.33
CORRELATION OF j , Re DATA
FOR REYNOLDS NO. EXPONENTS OF 0.9 & 0.5

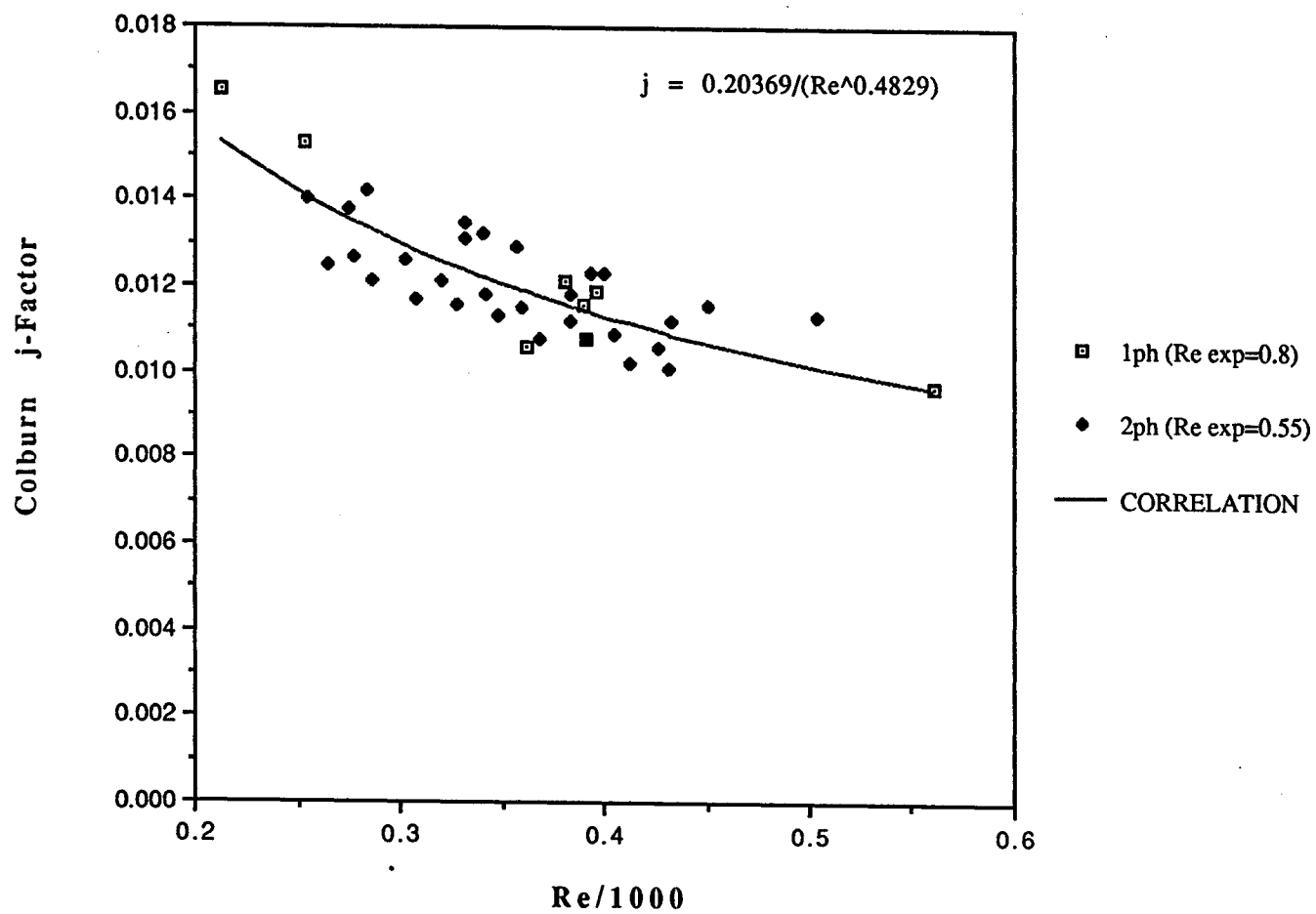


FIGURE 7.34
CORRELATION OF j , Re DATA
FOR REYNOLDS NO. EXPONENTS OF 0.8 & 0.55

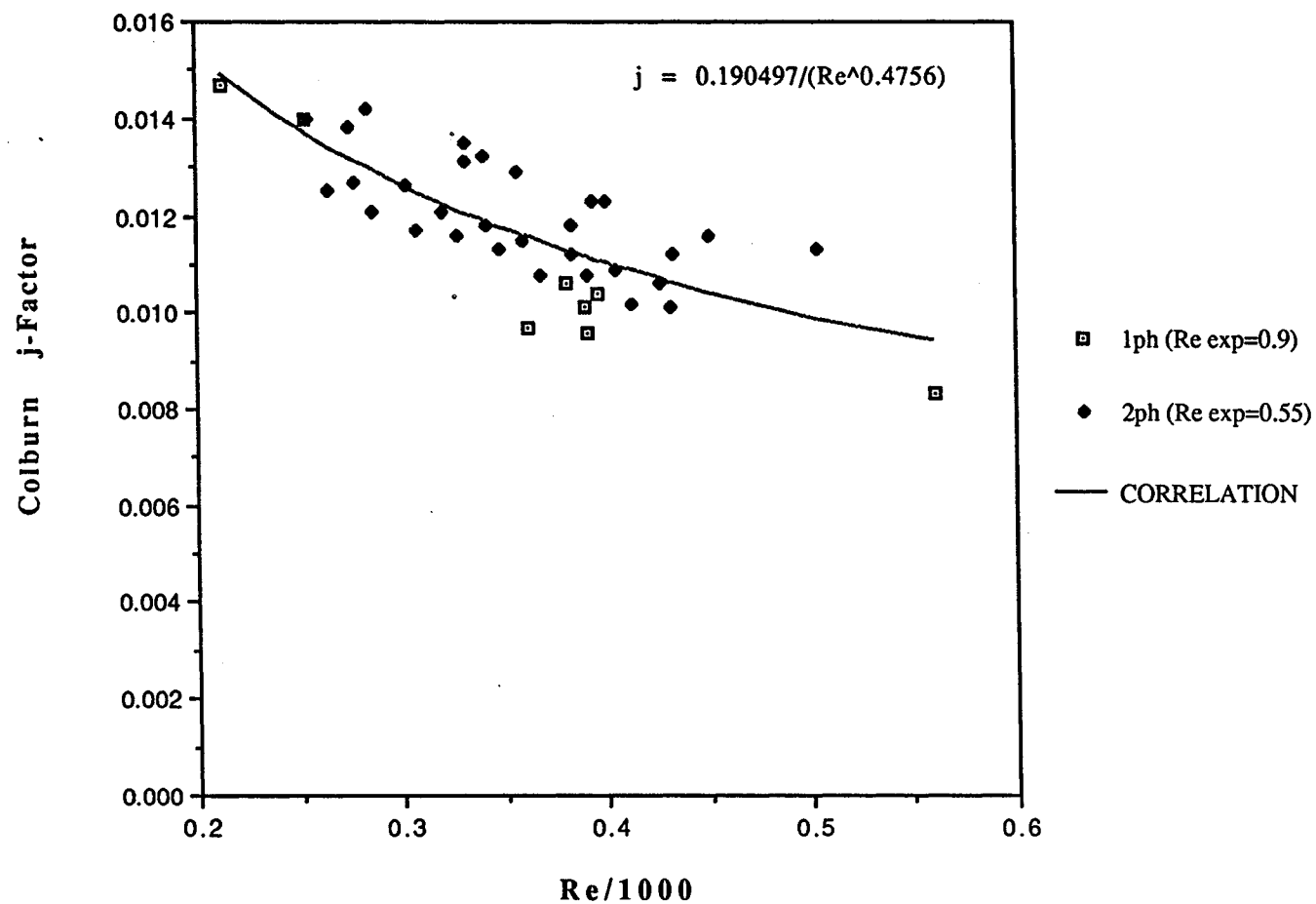


FIGURE 7.35
CORRELATION OF j , Re DATA
FOR REYNOLDS NO. EXPONENTS OF 0.9 & 0.55

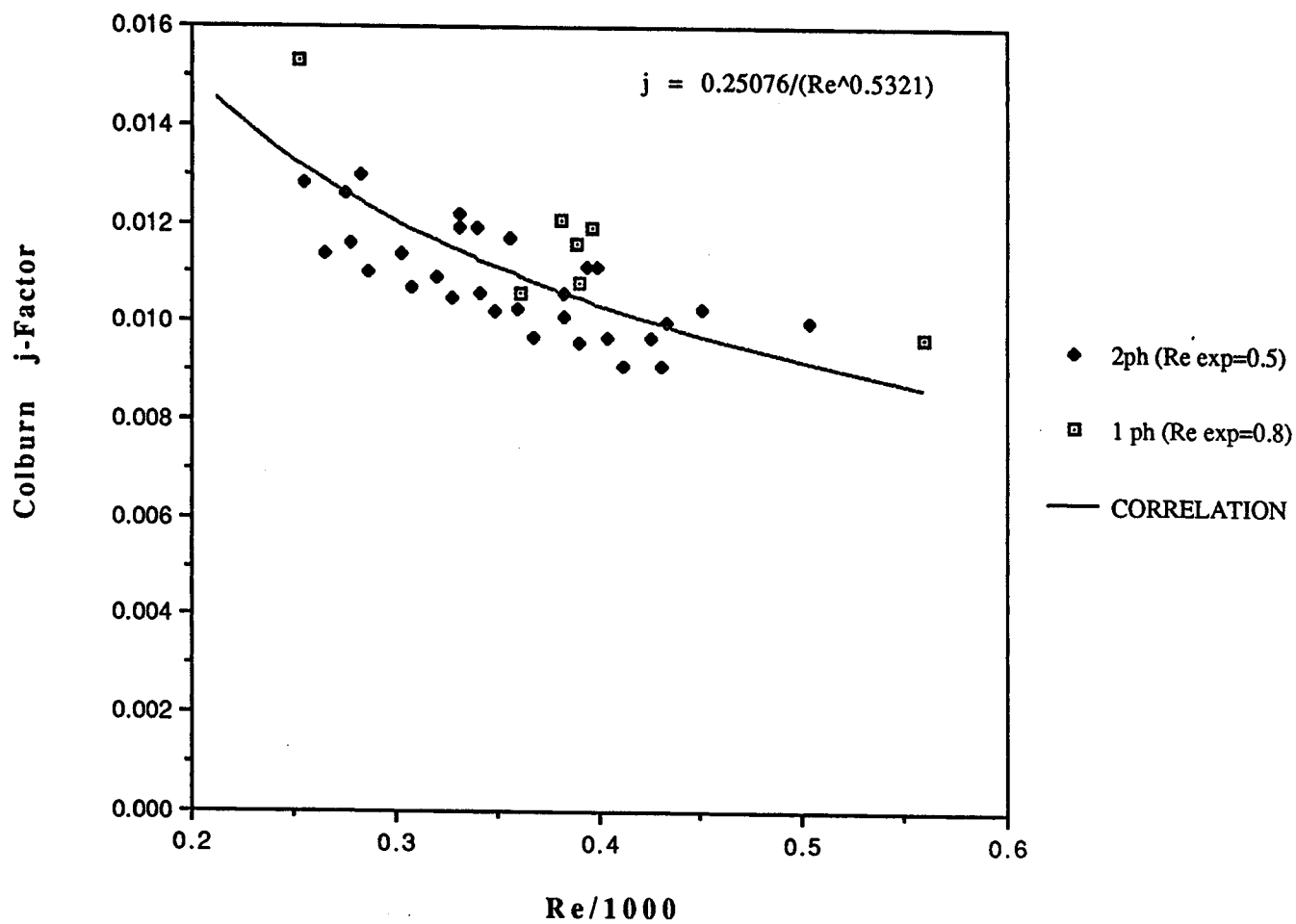


FIGURE 7.36
CORRELATION OF j , Re DATA
FOR REYNOLDS NO. EXPONENTS OF 0.8 & 0.5

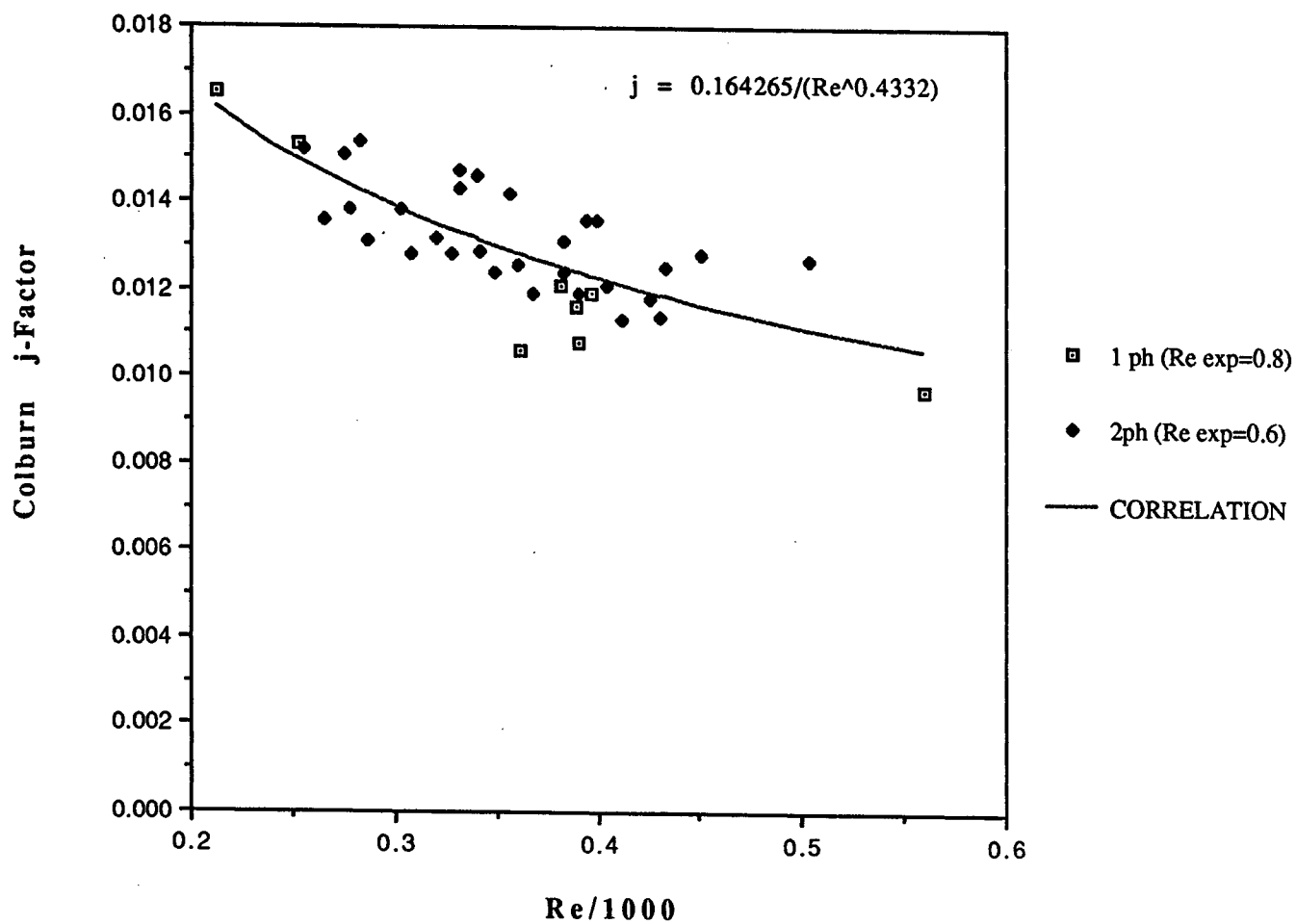


FIGURE 7.37
CORRELATION OF j , Re DATA
FOR REYNOLDS NO. EXPONENTS OF 0.8 & 0.6

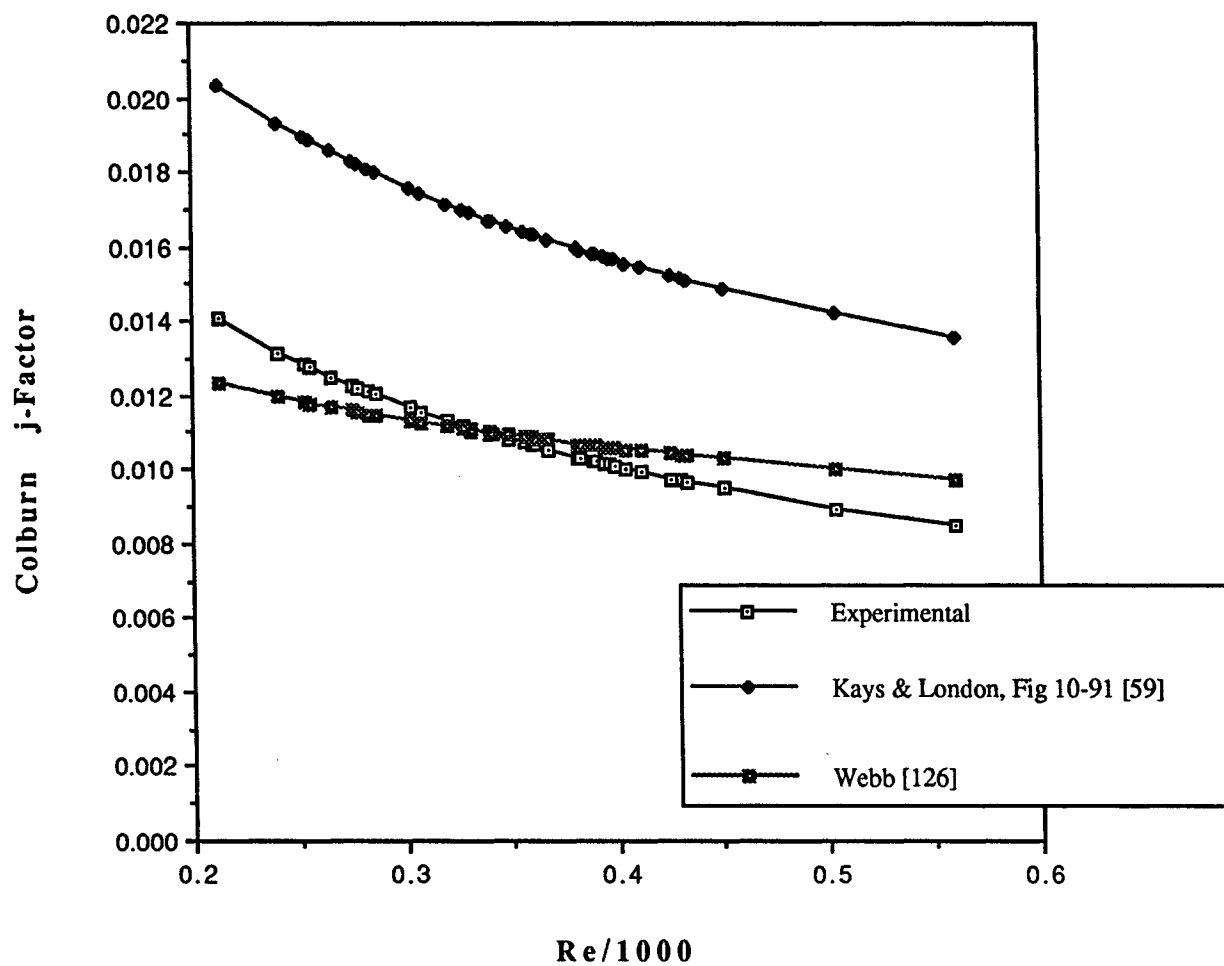


FIGURE 7.38
COMPARISON OF EXPERIMENTAL J-FACTOR DATA
WITH AVAILABLE CORRELATIONS

CHAPTER 8

UNCERTAINTY ANALYSIS

The analysis of experimental uncertainties, or experimental errors as also referred to here, is important in order to gain better understanding and confidence in the values of measured data and computed results. By properly quantifying the experimental uncertainty, results may be interpreted more correctly. The uncertainty analysis principles and methodologies documented in this Chapter are based on published documents by Moffat [76], Taylor [116] and ASHRAE [13].

Uncertainties in experimental results are introduced in measured variables, such as the temperature, pressure, and refrigerant flow rate. Further, these uncertainties are propagated in the calculation of derived quantities, such as the air flow rate, heat transfer rate, effectiveness and NTU, overall thermal resistance or conductance, and individual heat transfer resistances and coefficients.

8.1. General

8.1.1. Definitions

The error associated with a particular variable may be considered to be the difference between the true value of a quantity measured and the observed value. All experimental errors may be categorized as one of two types:

1. Systematic Error - a persistent error that cannot be considered entirely due to chance and that may be corrected through calibration. A measurement with small systematic errors is often said to possess high accuracy.
2. Random Error - an error that results in readings taking random values on either side of some mean value. A measurement with small random errors is said to possess high precision.

Experimental uncertainty may be defined as the estimated value for the error or the error that would result if it could be measured by calibration. Uncertainty may consist of both accuracy and precision errors, although precision (random) error may be only treated by statistical methods. In this Chapter and in referenced documentation, uncertainty is analyzed like a precision error. The propagation of uncertainty is the level to which the uncertainties in individual parameters affect the uncertainty in the final result.

8.1.2. Expressing Uncertainty in a Single Variable

The experimental uncertainty associated with a particular measured or computed variable may be described by specifying the expected value for the quantity followed by either the absolute uncertainty, the fractional uncertainty or both. Specifically stated:

$$z = x \pm w; e \% \quad (8.1.1)$$

where:

- z = variable (e.g., temperature, heat transfer rate, etc.)
- x = best expected value for the variable z
- w = absolute uncertainty, independent of the magnitude of the variable and with the same units as x and z
- e = fractional error, stated as a percentage or a fraction ($e/100$) of the magnitude of the variable.

Fractional error is sometimes called the confidence level, or the probability that the best expected value is within the specified uncertainty range. When expressed independently of one another, the absolute and fractional uncertainties are typically in the two forms that follow:

$$w_z = \pm a \quad (8.1.2)$$

$$e_z = \pm b \% \quad (8.1.3)$$

where w_z and e_z are the absolute and fractional uncertainties, respectively, associated with the measured or computed quantity z , and a and b are the respective magnitudes of each uncertainty.

8.1.3. Combining Uncertainties of Several Variables

For mathematical operations involving two or more variables and the absolute uncertainty associated with each variable, the rule for combining the individual uncertainties in order to determine the overall uncertainty in the final result may be summarized as follows:

$$w_r = \left[\left(\frac{\partial R}{\partial z_1} w_1 \right)^2 + \left(\frac{\partial R}{\partial z_2} w_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial z_n} w_n \right)^2 \right]^{1/2} \quad (8.1.4)$$

where: w_r = overall uncertainty in the final result
 w_n = uncertainty in the nth variable
 R = function containing the independent variables $z_1, z_2, \dots, z_n$
 $= R(z_1, z_2, \dots, z_n)$

In the most detrimental way, the uncertainty in a result may be approximated by summing the absolute values of the individual uncertainties. However, since the concept of uncertainty involves random deviation of a variable about an average value, it is highly improbable that the magnitude and direction of all uncertainties might simultaneously be the same. Therefore, in most instances, Equation (8.1.4) is considered the "recommended procedure" for evaluating the overall uncertainty.

8.2. Uncertainty in Data Analysis

This section discusses the individual uncertainties in specifically measured or computed test variables and the propagation of these uncertainties in further data reduction techniques. It is divided into two parts. The first portion considers the experimental data collected with single phase refrigerant flowing inside of the condenser. The second portion deals with the experimental data collected with two phase refrigerant flowing inside of the condenser. Some facets of the discussion are identical and independent of the refrigerant phase, as indicated.

8.2.1. Uncertainty in Single Phase Data Analysis

The analysis of single phase data is undoubtedly the most simple and straightforward in comparison with the two phase data. In addition, for all experimental data collected for a wide range of air and refrigerant flow rates, the refrigerant acted as the minimum heat capacity fluid.

8.2.1.1. Temperature Difference

Inlet and outlet air temperatures at the condenser test section are measured by two independent and redundant methods both upstream and downstream of the condenser, i.e., an RTD averaging sensor and a thermocouple grid. In addition, inlet and exit refrigerant temperatures are measured by RTD immersion probes. According to factory calibration data available from the manufacturers, the calibrated accuracy associated with each of the temperature measurement devices is relatively the same and equivalent to approximately ± 1.5 °F (0.83 °C) over the temperature range of 32 to 200 °F (0 to 93.3 °C). For any temperature difference ΔT , the corresponding absolute uncertainty may be determined by combining the individual uncertainties in each temperature measurement in accordance with Equation (8.2.1) below.

$$w_{\Delta T} = [(1.5)^2 + (1.5)^2]^{1/2} = \pm 2.12 \text{ °F (1.18 °C)} \quad (8.2.1)$$

The fractional error for the temperature difference may be expressed as:

$$e_{\Delta T} = w_{\Delta T} / \Delta T \quad (8.2.2)$$

Obviously, the magnitude of the measured temperature difference has a significant affect on the size of the fractional error.

The absolute uncertainty associated with the bulk mean fluid temperature and the temperature difference may be computed in the same manner. This is because the rules for combining uncertainties for the addition or subtraction of individual variables are identical. Thus, the absolute uncertainty corresponding to the bulk mean fluid temperatures of the refrigerant and air streams is also 2.12 °F (1.18 °C).

The fractional uncertainty is highly dependent on the magnitude of average temperature.

$$e_T = w_T/T_m \quad (8.2.3)$$

8.2.1.2. Air Flow Rate

As discussed in Chapter 5, the determination of air flow rate is in accordance with ANSI/ASHRAE Standard 41.2-1987 and 51-1985. An analytically developed error analysis for the supply fan air flow rate is detailed in the appendix of ANSI/ASHRAE Standard 51-1985 [5] as well as by Bohanon [20]. The following relationship is specified for determining the fractional error in the air volumetric flow rate:

$$e_Q = \left[e_C^2 + e_A^2 + e_N^2 + (e_f/2)^2 + (e_p/2)^2 \right]^{1/2} \quad (8.2.4)$$

where:

- e_C = fractional error in nozzle discharge coefficient = 0.012
- e_A = fractional error in area of nozzle = 0.005
- e_N = fractional variation in fan speed = 0.01
- e_f = fractional error in nozzle differential pressure = 0.01
- e_p = fractional error in nozzle inlet static pressure = 0.01

The numerical values for the fractional errors e_C , e_A , and e_N have been previously generated by Spitler [107]. The fractional errors in pressure measurements, e_f and e_p , are based on the calibrated accuracy of the Setra electronic pressure transducers, as discussed in Section 3.1.7.

Based on Equation (8.2.4) and the stated values for the five fractional errors, the total resulting or uncertainty associated with the volumetric air flow rate computation is ± 1.8 percent. From Section 5.9, the maximum uncertainty in the specific volume of air is about ± 1.0 percent. Based on these two uncertainties, the resulting fluctuation in the air mass flow rate is ± 2.0 percent.

8.2.1.3. Refrigerant Flow Rate

The volumetric flow rate of refrigerant is directly proportional to the voltage output signal from the flowmeter current loop. However, an error or uncertainty exists in the manufacturer's calibration data (± 0.5 percent) and the voltage drop across the precision resistor (± 1.5 percent). The cumulative fractional error may be expressed as a combination of these two individual errors.

$$e_Q = \left[e_I^2 + e_v^2 \right]^{1/2} \quad (8.2.5)$$

where: e_I = fractional error in flowmeter calibration data
(based on current output) = 0.005
 e_v = fractional error in voltage drop across precision
resistor = 0.015

Based on the above known uncertainties, the cumulative fractional error in the refrigerant volumetric flow rate computed from Equation (8.2.5) is ± 0.016 or ± 1.6 percent. From Section 5.6.1, the uncertainty in the saturated liquid density of R-12 corresponding to a ± 1.5 °F (0.83 °C) variation in the measured temperature averages 0.4 percent for the refrigerant temperature range of 118 to 182 °F (47.8 to 83.3 °C). Thus, the resulting error associated with the refrigerant mass flow rate is slightly less than ± 1.7 percent.

8.2.1.4. Heat Transfer Rate

For single phase fluids flowing at relatively constant pressure, the heat transfer rate may be computed by Equation (8.2.6).

$$Q = \dot{m} c_p \Delta T \quad (8.2.6)$$

The uncertainty corresponding to the heat transfer rate may be estimated by the following equation below.

$$e_Q = \left[e_m^2 + e_{c_p}^2 + e_{\Delta T}^2 \right]^{1/2} \quad (8.2.7)$$

In Equation (8.2.7), the individual fractional errors are defined as:

$$\begin{aligned} e_m &= \text{fractional error in mass flow rate} \\ e_{c_p} &= \text{fractional error in constant pressure, specific heat} \\ e_{\Delta T} &= \text{fractional error in temperature difference} \\ &= [(e_{T_m})^2 + (e_{T_{out}})^2]^{1/2} \end{aligned}$$

Based on air-side experimental data, e_m is given in Section 8.2.1.2 as 0.020 and e_{c_p} may be estimated from Section 5.6.2 as 0.012 (i.e., 0.243 ± 0.003 Btu/lb_m °F). The problems associated with fractional uncertainty in temperature difference have also been previously introduced in Section 8.2.1.2. The fractional error in the heat transfer rate is also highly dependent on the uncertainty in the temperature difference. In terms of known and unknown quantities, Equation (8.2.7) may be rewritten as follows:

$$e_Q = [(0.020)^2 + (0.012)^2 + (2.12/\Delta T)^2]^{1/2} \quad (8.2.8)$$

Thus, for a 10 °F (5.6 °C) temperature difference, the resulting fractional error in the air-side heat transfer rate is ± 21.3 percent. Similarly, for 15 °F and 20 °F (8.3 and 11.1 °C) temperature differences, the fractional uncertainties are ± 14.3 and ± 10.8 percent, respectively. The contributions by the air mass flow rate and the specific heat to the total fractional uncertainty are negligible compared to the contribution by the temperature difference.

Based on refrigerant-side experimental data, e_m is given as 0.017 in Section 8.2.1.3. Utilizing information from Section 5.6.1., the variation in the specific heat of saturated liquid R-12 corresponding to an absolute uncertainty of ± 2.12 °F in the bulk mean fluid temperature averages ± 0.8 percent in the refrigerant temperature range of 118 to 182 °F (47.8 to 83.3 °C). Thus, the cumulative fractional error associated with the refrigerant-side heat transfer rate may be expressed by Equation (8.2.9).

$$e_Q = [(0.017)^2 + (0.008)^2 + (2.12/\Delta T)^2]^{1/2} \quad (8.2.9)$$

The fractional uncertainty is primarily a function of the refrigerant temperature difference. The range of temperature differences encompassing all of the single phase refrigerant flow testing is approximately 27 to 102 °F (15.0 to 56.7 °C). From Equation (8.2.9), this range of ΔT translates into a range of fractional error of 0.028 to 0.081. The error value increases as the temperature difference decreases.

8.2.1.5. Fluid Heat Capacity Rates

Based on previously introduced information, the fractional uncertainties related to the products $(\dot{m}c_p)_{\text{air}}$ and $(\dot{m}c_p)_{\text{ref}}$ and subsequently, the heat capacity ratio C_r , may be determined independently. From Equation (8.2.8), the fractional error associated with the air heat capacity rate $(\dot{m}c_p)_{\text{air}}$ alone is ± 2.3 percent. Similarly, from Equation (8.2.9), the fractional uncertainty associated with only the refrigerant heat capacity rate $(\dot{m}c_p)_{\text{ref}}$ is ± 1.8 percent.

The resulting absolute uncertainty in the heat capacity ratio may be evaluated from Equation (8.2.10).

$$w_C = C_r [(0.023)^2 + (0.018)^2]^{1/2} = 0.0292 C_r \quad (8.2.10)$$

Further, the fractional uncertainty is equivalent to the following:

$$e_C = w_C/C_r = 0.0292 \text{ or } \pm 2.9 \text{ percent} \quad (8.2.11)$$

8.2.1.6. Uncertainty in Effectiveness and NTU

The absolute uncertainty in the effectiveness computation may be determined from Equation (8.2.12).

$$w_\epsilon = \left(\frac{\Delta T_{\min}}{\Delta T_{\max}} \right) \left[(2.12/\Delta T_{\min})^2 + (2.12/\Delta T_{\max})^2 \right]^{1/2} \quad (8.2.12)$$

where $\Delta T_{\min}$ is equal to the temperature difference for the fluid with the minimum heat capacity $(\dot{m}c_p)_{\min}$ and $\Delta T_{\max}$ is equal to the maximum possible

temperature difference in the two fluid, crossflow heat exchanger. Further, the fractional uncertainty in the effectiveness may be stated as:

$$e_{\epsilon} = \left[(2.12/\Delta T_{\min})^2 + (2.12/\Delta T_{\max})^2 \right]^{1/2} \quad (8.2.13)$$

For all single phase experiments with liquid refrigerant flowing inside the condenser tubes, the minimum heat capacity fluid is the refrigerant and the calculated values for effectiveness range from approximately 0.80 to 0.98, with 84 percent of the data falling within the effectiveness range of 0.89 to 0.98. In addition, the effectiveness increases as the magnitudes of both $\Delta T_{\min}$ and $\Delta T_{\max}$ increase. For all testing, the range of refrigerant temperature differences is 27 to 102 °F (15.0 to 56.7 °C). Therefore, for single phase testing, the fractional uncertainty associated with effectiveness must be considered on a case by case basis since the refrigerant temperatures consistently vary.

In order to identify an approximate order of magnitude related to the effectiveness error, for a 27 °F (15.0 °C) refrigerant temperature difference and a corresponding effectiveness of 0.80, the resulting fractional error as computed by Equation (8.2.13) is ± 10.2 percent. Similarly, for a 102 °F (56.7 °C) temperature difference and a corresponding effectiveness of 0.97, the resulting fractional error in the effectiveness is ± 2.9 percent. The average for the fractional uncertainty is ± 5.6 percent.

In accordance with the ASHRAE guidelines and Section 8.1.3, the resulting fractional error in the number of transfer units (NTU) must be determined by differentiating the appropriate effectiveness-NTU relationship presented in Chapter 5 with respect to each of the independent variables, C_r and ϵ . Ideally, NTU must also be evaluated on a case by case basis since the heat capacity ratio and effectiveness vary as the refrigerant flow conditions are changed. The results indicate that the error in NTU is highly sensitive to the error in effectiveness and negligibly affected by the error in the heat capacity ratio. For example, the range of fractional uncertainty associated with the NTU is ± 30 to ± 74 percent. The average is about ± 38.7 percent.

8.2.1.7. UA Value

The overall thermal conductance of the heat exchanger (UA) is directly a function of the NTU and $C_{\min}$, the fluid with the minimum heat capacity. The fractional uncertainty in the UA values may be determined from Equation (8.2.14).

$$e_{UA} = \left[e_{NTU}^2 + e_{C_{\min}}^2 \right]^{1/2} \quad (8.2.14)$$

Since the fractional uncertainty in NTU is large compared to the uncertainty in $C_{\min}$, the error in the minimum heat capacity may essentially be neglected. Thus, for the range of experimental test conditions, the fractional uncertainty in UA averages approximately ± 38.7 percent.

8.2.1.8. Reynolds Number

The air-side and refrigerant-side Reynolds numbers are generally defined as

$$Re = \frac{GD_h}{\mu} \quad (8.2.15)$$

where G is equal to the mass flow rate per unit area (W/A). The uncertainty associated with the air-side and refrigerant-side heat transfer surface areas is roughly ± 2.0 percent. The uncertainties in the air and refrigerant mass flow rates are previously given as ± 2.0 and ± 1.7 percent, respectively. In accordance with Section 5.6.2, the uncertainty in the absolute viscosity is negligible for air. Per Section 5.6.1, the variation in absolute viscosity for refrigerant R-12 corresponding to a bulk mean fluid temperature uncertainty of 2.12°F (1.18°C) averages approximately ± 1.1 percent over the refrigerant temperature range of 118 to 182°F (47.8 to 83.3°C).

Based on the itemized uncertainties, the overall fractional uncertainty in the air-side Reynolds number is ± 3.5 percent; whereas, the fractional uncertainty in the refrigerant-side Reynolds number is ± 3.3 percent.

8.2.1.9. Prandtl Number

For the air-side and the refrigerant-side, the Prandtl number is defined in terms of the fluid transport properties as follows:

$$Pr = \frac{c_p \mu}{k} \quad (8.2.16)$$

For the air-side, the uncertainty in thermal conductivity and absolute viscosity are negligible, per Section 5.6.2. Therefore, the uncertainty in the Prandtl number is essentially the same as the error in the specific heat. From Section 8.2.1.4, the uncertainty in specific heat is ± 1.2 percent.

For the refrigerant-side, the uncertainty in specific heat and absolute viscosity have been determined previously as ± 0.8 and ± 1.1 percent, respectively. From Section 5.6.1, the fractional uncertainty in thermal conductivity corresponding to a 2.12°F (1.18°C) variation in the bulk mean fluid temperature averages approximately ± 0.8 percent over the refrigerant temperature range of 118 to 182°F (47.8 to 83.3°C). Thus, the overall fractional uncertainty in the refrigerant-side Prandtl number is ± 1.7 percent.

8.2.1.10. Individual Heat Transfer Resistances

Neglecting the fouling resistances and assuming that the wall resistance is negligible in comparison with the refrigerant-side and air-side resistances, then the overall resistance balance on the air-cooled condenser may be written as follows:

$$\frac{1}{UA} = R_{\text{air}} + R_{\text{ref}} = \frac{1}{(\eta h A)_o} + \frac{1}{(\eta h A)_i} \quad (8.2.17)$$

For single phase refrigerant flow inside the condenser tubes, the refrigerant-side or inside convective heat transfer coefficient may be expressed in the following form.

$$h = C_o Re^a Pr^b \left(\frac{k}{D_h} \right) \quad (8.2.18)$$

The fractional uncertainty in the refrigerant-side heat transfer coefficient may be expressed by Equation (8.2.19).

$$e_h = \left[a^2 (w_{Re}/Re)^2 + b^2 (w_{Pr}/Pr)^2 + (w_k/k)^2 + (w_D/D_h)^2 \right]^{1/2} \quad (8.2.19)$$

The fractional uncertainty in the refrigerant-side thermal conductivity and hydraulic diameter are ± 0.8 and ± 2.0 percent, respectively. In addition, the overall fractional uncertainty in the inside convective heat transfer coefficient is dependent on the exponents and associated uncertainties for the two dimensionless parameters. Specifically, for a Reynolds number exponent of 0.9 and a Prandtl number exponent of 0.4, the corresponding fractional uncertainty is approximately ± 3.7 percent. Similarly, for a Reynolds number exponent of 0.8, the overall fractional uncertainty for the inside heat transfer coefficient is reduced to ± 3.5 percent. Assuming that the fin and surface efficiency are evaluated correctly and a ± 2.0 percent error in the refrigerant-side heat transfer area, then the associated fractional uncertainty in the refrigerant-side thermal resistance is approximately ± 4.0 percent.

The unknown air-side thermal resistance is equivalent to

$$R_{air} = \frac{1}{(\eta h A)_o} = \frac{1}{UA} - \frac{1}{(\eta h A)_i} \quad (8.2.20)$$

The uncertainty associated with the overall thermal resistance ($1/UA$) is large in comparison with the uncertainty associated with the refrigerant-side resistance. Therefore, the resulting error in the air-side resistance and the convective heat transfer coefficient is approximately ± 38.0 percent.

8.2.2. Uncertainty in Two Phase Data Analysis

As indicated previously, during experimental testing with two phase refrigerant flow inside of the condenser tubes, test conditions were adjusted so that the refrigerant desuperheating and subcooling regions were minimized. In addition, the assumption of a uniform air-side heat transfer coefficient was incorporated by neglecting the single phase refrigerant flow regions and considering the two phase flow region alone. In contrast to single phase refrigerant flow testing, the air acted as the minimum heat capacity fluid during experiments with two phase refrigerant flow.

For two phase refrigerant flow experiments, the magnitude of the uncertainty in the fluid temperatures and flow rates are identical to those identified previously in Sections 8.2.1.1 thru 8.2.1.3.

8.2.2.1. Effectiveness and NTU

For two phase refrigerant flow experiments, the absolute and fractional uncertainties associated with the heat exchanger effectiveness may each be computed by the same formulas shown in Section 8.1.6. These formulas are also repeated here.

$$w_{\epsilon} = \left(\frac{\Delta T_{\min}}{\Delta T_{\max}} \right) \left[(2.12/\Delta T_{\min})^2 + (2.12/\Delta T_{\max})^2 \right]^{1/2} \quad (8.2.21)$$

$$e_{\epsilon} = \left[(2.12/\Delta T_{\min})^2 + (2.12/\Delta T_{\max})^2 \right]^{1/2} \quad (8.2.22)$$

where $\Delta T_{\min}$ is equal to the temperature difference for the fluid with the minimum heat capacity $(\dot{m}c_p)_{\min}$ and $\Delta T_{\max}$ is equal to the maximum possible temperature difference in the two fluid, crossflow heat exchanger. For all two refrigerant flow experiments, the air is the minimum heat capacity fluid and the maximum attainable temperature difference is equal to the refrigerant condensing temperature minus the air inlet temperature.

Based on the two phase data, the range of calculated values for the effectiveness is 0.49 to 0.68. In addition, the heat exchanger effectiveness increases as the magnitudes of both $\Delta T_{\min}$ and $\Delta T_{\max}$ increase. For all testing, the range of air temperature differences is approximately 20 to 66 °F (11.1 to 36.7 °C). Thus, for two phase testing as in single phase testing, the fractional uncertainty associated with effectiveness must be considered on a case by case basis since the air temperatures consistently vary.

Utilizing the results of data reduction and Equation (8.2.22), the fractional uncertainty for the effectiveness ranges from ± 3.9 percent at an air temperature difference of 66 °F (36.7 °C) to ± 11.9 percent at an air temperature difference of 20 °F (11.1 °C). For the 30 two phase data points, the average fractional uncertainty in effectiveness is ± 6.2 percent.

In accordance with Equation 5.20, the following equation may be implemented to determine the resulting absolute uncertainty in the number of transfer units (NTU) on a case by case basis. The equation verifies that the uncertainty in NTU is only dependent on the uncertainty in the effectiveness.

$$w_{\text{NTU}} = \left[\left(\frac{\partial \text{NTU}}{\partial \epsilon} w_{\epsilon} \right)^2 \right]^{1/2} = \frac{w_{\epsilon}}{1-\epsilon} \quad (8.2.23)$$

Further, the fractional uncertainty in NTU may be evaluated by Equation (8.2.24).

$$c_{\text{NTU}} = w_{\text{NTU}}/\text{NTU} \quad (8.2.24)$$

From Equations (8.2.23) and (8.2.24), the overall fractional uncertainty in the NTU ranges from ± 6.2 percent to ± 12.4 percent with 80 percent of the values less than ± 10.0 percent. The average fractional uncertainty in NTU for the 30 two phase data points is approximately ± 9.0 percent.

8.2.2.2. UA Value

The fractional uncertainty in the overall thermal conductance for the two phase data is directly related to the NTU and the minimum heat capacity fluid, which is the air stream. From Section 8.2.1.5, the fractional uncertainty in the air heat capacity rate $(\dot{m}c_p)_{\text{air}}$ is ± 2.3 percent. Thus, based on the form of Equation (8.2.14) derived previously, the average fractional error in the UA values is approximately ± 9.3 percent.

8.2.2.3. Individual Heat Transfer Resistances

For two phase experiments, Wilson Plots were generated of the form:

$$\frac{1}{UA} = \frac{1}{V_{\text{air}}^a} + (R_{\text{ref}} + R_{\text{wall}}) \quad (8.2.25)$$

Neglecting the wall resistance, the refrigerant-side resistance is equivalent to

$$R_{\text{ref}} = \frac{1}{UA} - \frac{1}{V_{\text{air}}^a} \quad (8.2.26)$$

In Equations (8.2.25) and (8.2.26), the independent flow variable V_{air} is equal to the air velocity through the minimum free flow area of the condenser coil. The fractional uncertainty in the air velocity by considering the following definition relating the velocity to the air-side Reynolds number:

$$V_{\text{air}} = \frac{\mu Re}{\rho D_h} \quad (8.2.27)$$

The fractional uncertainty in the air velocity may be expressed in terms of the individual fractional uncertainties as follows:

$$e_v = \left[(w_{Re}/Re)^2 + (w_{\mu}/\mu)^2 + (w_{\rho}/\rho)^2 + (w_D/D_h)^2 \right]^{1/2} \quad (8.2.28)$$

According to Sections 8.2.1.8 and 5.6.2, the individual fractional uncertainties appearing in Equation (8.2.28) may be identified as follows:

$$\begin{array}{ll} w_{Re}/Re = 0.035 & w_{\mu}/\mu = 0.011 \\ w_{\rho}/\rho = 0.010 & w_{D}/D_h = 0.020 \end{array}$$

By substitution, the resulting fractional uncertainty in the air velocity through the minimum free flow area is ± 4.3 percent. Therefore, based on Equation (8.2.28) and a velocity function exponent of 0.6, the fractional uncertainty associated with the refrigerant-side resistance is about ± 9.7 percent. Similarly, for a velocity function exponent of 0.5, the fractional uncertainty in the refrigerant-side resistance is approximately ± 9.5 percent.

The air-side resistance is related to the overall thermal resistance and the refrigerant-side resistance by the following equation:

$$R_{air} = \frac{1}{UA} - R_{ref} = \frac{1}{(\eta h A)_o} \quad (8.2.29)$$

Thus, the resulting fractional uncertainty in the air-side resistance is about ± 13.4 percent. Based on an error of ± 2.0 percent in the air-side heat transfer area, the resulting fractional uncertainty in the convective heat transfer coefficient is ± 13.5 percent.

8.2.3. Comparison of Single Phase and Two Phase Uncertainties

Based on Sections 8.2.1 and 8.2.2, there is a significant difference between the overall fractional uncertainties associated with the two phase experimental data in comparison with the single phase experimental data. This may be explained by the larger temperature differences observed during the two phase experiments and the resulting insensitivity of the effectiveness and NTU computations to these large temperature differences.

CHAPTER 9

CONCLUSIONS AND RECOMMENDATIONS

9.1. Summary of Results

A full condenser experimental facility has been designed and constructed for the Air Conditioning and Refrigeration Center (ACRC) at the University of Illinois at Urbana-Champaign to evaluate the steady-state and transient, heat transfer and pressure drop characteristics of typical fin-and-tube condensers employed in mobile air conditioning and household refrigerator/freezer applications. The apparatus has been designed and constructed to allow production of a wide range of air flow rates and inlet air temperatures, as well as a wide range of refrigerant flow rates and inlet refrigerant temperatures, at the test condenser.

A comprehensive review of the literature related to empirically-determined heat transfer and friction characteristics of various compact heat exchanger surfaces has been included in this presentation. In addition, separate sections have been dedicated to the discussion of the original and modified Wilson Plot techniques, which are often utilized to evaluate the individual heat transfer resistances in condensers and evaporators.

Baseline testing with refrigerant R-12 has been completed with one test condenser. The modified Wilson Plot techniques were utilized to evaluate the air-side heat transfer coefficients and Colburn j-factors in an automotive, finned tube condenser. Experimental data was collected based on either single phase refrigerant flow through the condenser tubes and or two phase refrigerant flow through the condenser tubes and air in crossflow over the finned tube surface. Some of the results gathered from conducting these experiments and from the subsequent data analysis may be summarized as follows on the next few pages.

1. It has been demonstrated that the experimental facility is capable of generating the test conditions for which it was designed. Operating conditions sustainable by the air-side include condenser flow rates from 10 to 4500 cfm (0.0047 to 2.124 m³/sec) and inlet air temperatures from 30 to 150 °F (-1.1 to 65.6 °C). For the refrigerant-side, the apparatus may produce condenser flow rates from 0 to 2 gpm (0 to 0.126 liters/sec) and inlet temperatures that possess 0 to 100 °F (0 to 55.6 °C) of superheat and is capable of withstanding working pressures up to 500 psia (3447 kPa).
2. As a result of baseline testing with refrigerant CFC-12, accuracy and repeatability in experiments and calibration of instrumentation has been validated by comparison of the air-side and refrigerant-side energy balances at the test condenser. For heat transfer rates of approximately 6000 to 55,000 Btu/hr (0.176 to 1.611 kW), the average error in the energy balances ranges from 1 to 15 percent. The accuracy is best at the highest heat transfer and air flow rates due to less pronounced air flow nonuniformities.
3. A dimensionless Colburn modulus or j-factor versus Reynolds number correlation has been developed that accurately correlates all of the experimental data within ± 11.0 percent. This empirical correlation has been compared with Webb's wavy fin correlation [126], and ninety percent of the data agrees within ± 7.9 percent over the range of air-side Reynolds number from approximately 200 to 600.

4. The Wilson Plot concept has been analyzed from a critical approach and modified techniques have been utilized successfully to evaluate the air-side heat transfer resistances of the condenser and to generate a j-factor correlation. Both single phase and two phase refrigerant flow tests were performed to provide alternative procedures for determining the outside resistances to heat transfer. For testing with single phase fluids on both sides of the heat exchanger, the Wilson Plot procedure was utilized to eliminate the annular flow or refrigerant-side resistance. A Reynolds number exponent of 0.9 appears to correlate the heat transfer data most accurately for these experiments. For testing with two phase refrigerant flow inside the condenser tubes, the Wilson Plot technique was incorporated to eliminate the air-side resistance. A velocity function exponent of 0.5 seems to minimize the error between the correlation and the experimental data.
5. For single phase fluids on both sides of the condenser, air-side heat transfer resistances have been obtained utilizing both the modified and original Wilson Plot correlating equations, i.e., Equations (4.4.4) and (4.5.6), respectively, and comparison of the results yields less than a two percent error between the individual air-side heat transfer resistances. This fact enables the variation of refrigerant transport properties with temperature to be neglected, so that the original form of the Wilson Plot function may be implemented to correlate the single phase data, i.e.,

$$\frac{1}{UA} = \frac{1}{C_1 V^a} + C_2$$

6. For testing with single phase fluids on both sides of the condenser, refrigerant-side convective heat transfer coefficients have been determined from the modified Wilson Plots and compared with values computed from the single phase correlations of Dittus-Boelter, Petukhov-Popov and Gnielinski. Results indicate that the Dittus-Boelter correlation tends to predict values for the heat transfer coefficient that are 20 to 30 percent higher than the other two correlations and the experimental refrigerant-side heat transfer coefficients compare most favorably with the Gnielinski and Petukhov-Popov correlations.
7. A comprehensive test plan has been established for future steady state, baseline testing with the chlorofluorocarbon refrigerant CFC-12 and subsequent experiments with the ozone-safe, hydrofluorocarbon refrigerant HFC-134a.

9.2. Suggested Improvements

Uncertainties present in the experimental apparatus warrant that modifications also be made to the system in the near future. Some of these suggested improvements for both the apparatus air-side and refrigerant-side are listed below. In particular, baseline testing revealed that there was significant stratification in the air duct at low air flow rates. Items 2 and 3 below are recommendations that may lead to reduction of this stratification or nonuniformity.

1. Thermocouples (Type T, 30 gage wire), Omega RTD's and the Setra differential pressure sensor (at the condenser test section only) should each be individually laboratory calibrated to reduce the manufacturer's standard calibration accuracy.

2. Air blenders, turbulators or equivalent air mixing hardware should be installed in the duct system to reduce the air velocity and temperature nonuniformities that exist both upstream and downstream of the condenser test section.
3. A branch air duct of smaller cross sectional flow area should be designed for installation downstream of the electric duct heater to accommodate the eventual testing with household refrigerator/freezer condenser coils. The existing 24" x 24" sheet metal duct is oversized for the low air flow rates associated with these condenser coils.
4. Design and installation of a refrigerant aftercondenser downstream of the condenser should be examined for any future two phase flow testing due to the inadequate amount of subcooling observed at the condenser exit for the condition of air flow and high refrigerant heater power input.
5. The condensing pressure should be considered as another possible test variable; however, a method for regulating this pressure accurately will then be required since this cannot be accomplished with the refrigerant heater and the circulation pumps. Possible solutions include installation of a needle-type, throttling valve or a nitrogen-based, bladder tank downstream of the condenser.

9.3. Future Work

Finally, the experimental facility described in this document may accommodate the future work necessary with the air-cooled condensers of mobile air conditioning and household refrigerator/freezer systems and alternative refrigerants including the following:

1. Analysis of the air pressure drop performance of different condenser geometry types, including the determination of the j-factors, so that j/f versus air-side Reynolds number information may be generated. This is important because a desirable increase in air-side heat transfer coefficient may sometimes be offset by an undesirable increase in the air pressure drop.
2. Investigation of techniques to optimize refrigerant circuiting and to reduce condenser weight and volume, thereby lowering refrigerant charge. In particular, for mobile applications, space and weight design allowances are restricted since these two parameters are directly influence the overall cost and marketability of a product.
3. Study of the effects of air flow maldistribution on the heat transfer and friction characteristics of the condenser.
4. Analysis of transient condenser performance, particularly automotive condensers during "pull-down" conditions when rapid cooling of the passenger compartment is necessary.

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APPENDIX A
FULL CONDENSER TEST FACILITY
START-UP, OPERATING AND MAINTENANCE PROCEDURES

The following sections contain the full condenser test apparatus start-up, operating and maintenance procedures, including instructions for the refrigerant loop, air loop, water loop, and data acquisition system. Prior to starting up the system, it is recommended that one read and understand the operating instructions comprehensively. As a reminder, safety glasses should be worn at all times when operating or in the vicinity of the test apparatus. In addition, during operation the apparatus should not be left unattended in any instance so that any unforeseen problems may command immediate attention.

A.1. Air-Side and Refrigerant-Side Loops

1. Check transfer switch position. The transfer switch should be set in the "DOWN" position so that electrical panels associated with the ACRC project have power supply.
2. Assure that necessary circuit breakers in the electrical panels and disconnect switches located near major equipment are in the "ON" position. Typically, power shall be required for the following components (equipment with disconnects denoted by asterisk *)
 - refrigerant heater, including evaporating and superheating sections (*)
 - one refrigerant circulation pump
 - supply air fan (*)
 - electric duct heater (*)
 - heat pump and water circulation pumps, if needed (*)
 - 24 VDC power supplies. Two (2) power supplies furnish excitation voltage to the five (5) pressure transducers, flowmeter, and air-side differential and static pressure sensors.

Pilot lights are provided on the individual control panels of the supply fan variable speed drive, heat pump, and water circulation pump variable speed drive to verify power.

3. Assure that all refrigerant shutoff valves are in the open position, excluding the valve at the inlet to the flowmeter and the one (1) valve on the inlet side of the circulation pump not in service.

4. For single phase refrigerant testing, it is necessary to attach one of the refrigerant, 30-pound capacity refillable cannisters to the loop at the Schrader valve nearest the condenser test section and at the highest point of the refrigerant loop. This cannister acts as an expansion tank, as described herein. In order to continually maintain single phase refrigerant at the condenser inlet during testing, the refrigerant within the cannister must be maintained at a pressure higher than the saturation pressure associated with the refrigerant temperature at the condenser inlet. It is important that the cannister be only partially filled when connecting it to the apparatus in order to allow natural expansion of the liquid when it is heated. The cannister may be pressurized by wrapping it with Minco ribbon-type electrical heaters. The refrigerant pressure within the cannister may be approximated by the measurement indicated by the the condenser inlet pressure transducer. Assure than cannister is not overpressurized. Its maximum working pressure is 500 psia.

For two phase refrigerant testing, the cannister is not necessary and should be removed if still connected. Prior to startup, refrigerant level in the receiver should be at approximately the half way mark.

5. Assure that the two (2) air duct shutoff dampers are in their proper positions to allow full air flow towards the condenser test section and no air flow towards the ASHRAE-related research project.
6. Check that the appropriate nozzle(s) are installed in the air flow measurement chamber and record throat diameter(s). Six (6) different nozzles are available with their throat diameters ranging in size from 1 inch to 10 inches. The nozzle size(s) in use has a noticeable effect on the accuracy of the air flow measurement. Thus, care must be exercised in selecting the proper nozzle(s) for the experiment to be conducted. Consult nozzle test calibration sheet for a guide to selection.
7. Start computer and data acquisition system in accordance with its own start-up and operating instructions given in Section A.3.
8. Energize supply fan via start/stop pushbuttons. Two start/stop pushbuttons exist. The main pushbutton is located on the remote variable speed drive controller inside the control room. The secondary pushbutton is located on the west wall inside the control room near the electrical panels. If the main pushbutton fails to energize the fan, than check the secondary pushbutton. Fan speed may be adjusted by the potentiometer dial on the remote controller. Air volume and mass flow rates may be visually observed on data acquisition output screen.

9. Start one of the refrigerant circulation pumps. Pump speed may be adjusted via the appropriate potentiometer dial located above the two pumps. In general, the upper pump is designed to handle refrigerant flow rates typical of household refrigerator/freezer systems; whereas, the lower pump is designed to handle refrigerant flow rates typical of mobile air conditioning systems. Circulation pumps should not be operated concurrently.
10. After visually inspecting the sight flow indicators for refrigerant flow, slowly set the shutoff valve at the inlet to the flowmeter to the fully open position. The shutoff valve in the flowmeter bypass loop may then be closed.
11. Energize the refrigerant heater. Refrigerant heater output may be regulated via the two (2) potentiometer dials inside the control room. A separate control dial is provided for the superheating and evaporating portion of the refrigerant heater. Only one of these dials should be adjusted at the same time. In addition to the two dials, switches are provided for the individual rows of the refrigerant heater and may be either turned "ON" or "OFF" to limit total power output. These switches, 15 in all, are located on the rear wall of the heater enclosure. During start-up, gradually increase the heater output dials on an incremental basis and continually monitor refrigerant temperatures and pressures, in particular those at the heater exit and condenser inlet, in order to detect any abnormalities and potential overpressurization in the system, as early as possible.

For most experimental testing (e.g., single phase or two phase Wilson Plot studies), the heat input is fixed, and under normal operating conditions, the maximum system pressure will be encountered at the minimum refrigerant flow rate. Therefore, it is recommended that the heat input be adjusted with the refrigerant flow at the anticipated minimum test condition. If the minimum flow condition is not accurately known, then the refrigerant flow and heat input must continually be adjusted until a "safe operating condition" is maintained on the refrigerant-side. This "safe operating condition" may be defined as that state where the refrigerant pressure and temperature at the heater exit or condenser inlet remain relatively stable at a maximum value within operating pressure of the apparatus. Once a safe operating condition has been met and the refrigerant flow rate and heat input no longer require adjustment, the heat input shall remain fixed for the remainder of such testing.

A thermocouple is mounted on one of the tube bends of the refrigerant heater, immediately downstream of the evaporating portion and prior to the superheating portion.

For single phase testing, energizing the evaporator section heaters is normally not necessary. Typically, introduction of heat via the superheat section heaters is adequate.

For two phase testing, the thermocouple measurement at the evaporator exit should be continually monitored so that the amount of superheating that occurs is insignificant (i.e., $T_R < T_{sat}$).

12. Energize heat pump in cooling or heating mode, if necessary, and in accordance with its own start-up and operating procedures in Section A.2.
13. Energize the duct heater, if necessary. Input the desired setpoint for the condenser inlet air temperature (in °F) in accordance with the instructions in Section A.4. The air mass flow rate is input into the duct heater control algorithm directly from the communications program. SCR's on duct heater will cycle to maintain the specified temperature based on measurements by the RTD averaging probe near the condenser inlet and the temperature sensor upstream of the air flow measurement chamber.

In general, the duct heater tends to operate most effectively (i.e., accurately maintain the desired setpoint temperature) when its input control signal is more than 500 mV. The duct heater control signal may be monitored in Location 68, Mode *6, of the data logger. For a given airflow, the input signal is influenced by the temperature difference between the inlet and the user-specified setpoint. This normally occurs when its inlet air temperature is more than 15 °F less than the specified setpoint, but also within the electric heating capacity of the duct heater. If the air inlet temperature to the supply fan is higher than or 10 to 15 °F less than the desired condenser inlet air temperature, then it will normally be necessary to energize the heat pump loop in cooling mode, thereby circulating chilled water through the coil immediately downstream of the supply fan and sufficiently lowering the inlet air temperature to the duct heater. On the other hand, in order to achieve condenser inlet air temperatures more than 40 °F higher than room temperature, it may be necessary to reverse the heat pump and operate the water loop in heating mode thereby increasing the inlet air temperature to the duct heater to within a limit where it possesses the capacity to maintain the high condenser inlet air temperature.

14. For steady-state, Wilson Plot data collection with single phase refrigerant flow, it is necessary to maintain the same refrigerant heat input, air flow rate, and air-side bulk mean fluid temperature throughout testing while varying the refrigerant flow rate from a minimum to a maximum. Adjust the appropriate pump dial accordingly to increase or decrease the refrigerant flow rate.

For steady-state, Wilson Plot data collection with two phase refrigerant flow, it is required maintain the refrigerant-side conditions fixed throughout testing (i.e., mass flow rate, bulk mean fluid temperature and heat input) while varying the air flow rate from a minimum to a maximum. Adjust the fan speed potentiometer accordingly to vary air flow rate.

For two phase testing, the heat input should be set to continually achieve a full condensation, two phase flow process in the condenser with minimal regions of desuperheating and subcooling. Condenser temperatures may be visually monitored via the appropriate adjustment to the data acquisition system.

For steady-state condenser performance testing when it is desired to simulate the typical conditions in mobile air conditioning or household refrigerator/freezer systems, operating the system requires adjusting the fluid mass flow rates, and heat inputs in order to arrive at the desired operating condition.

15. Allow system to reach and operate at steady-state prior to changing any operating parameters. For each steady-state data point, collect at least 20 to 30 readings in order to obtain a representative average. Upon initial start-up, approximately 60 minutes is required to achieve the first steady-state condition. After operating parameters are adjusted, it takes about 15 to 30 minutes thereafter to attain another steady-state data point.

A.2. Heat Pump and Water Loop

1. Assure that heat pump and water circulation pump disconnect switches are in the "ON" position. Power supply and correct mode of operation should be verified by indication lights on control panel. "REV" light should be energized in cooling mode. If indication lights fail, check unit fusing in control transformer box below main panel.
2. Reset desired storage tank water temperature setpoint on Penn aquastat, if required. In cooling mode, the temperature controller is wired to energize the heat pump if the water temperature rises above its setpoint. In heating mode, it is wired to energize the heat pump if the water temperature falls below the setpoint. With no glycol in the system and in order to avoid possible freeze-up of the heat exchanger, the recommended minimum thermostat setpoint is 40 °F. Tank water temperature may be monitored by RTD measurement displayed on output.
3. Check that all valves, other than drain valves, are fully open.
4. Start city water flow (condenser-side) by manually opening inlet and exit globe valves. Open the discharge valve first and then the inlet valve. Verify water flow by pressure gages on inlet and exit water sides.
5. Start constant speed circulation pump and closed loop water flow (evaporator-side) via start pushbutton on east wall outside control room. Visually inspect for water flow by rotameter on exit side of pump.

6. Energize heat pump by placing "COMPR" switch in the "ON" position. Compressor should only be energized after verifying positive water flow on both sides of the heat exchanger. "SLOW" light indicating compressor speed should energize and remain energized during entire operation. Compressor should cycle to maintain desired tank water temperature. Since compressor speed light remains energized, cycling should be verified audibly.
7. Immediately, begin throttling city water flow to maintain a compressor refrigerant suction pressure of 50 to 80 psia. Proper flow is best accomplished by setting the inlet valve fully open and adjusting the discharge valve to achieve correct operating pressure. During normal cooling operation, the corresponding compressor discharge pressure should remain less than 250 psia. City water pressure indicated by dial gages should be approximately 40 psia.
8. Activate variable speed, chilled water pump via start pushbutton and potentiometer inside control room. Controller dial may be set at position "8" for satisfactory operation. Flow may be verified by rotameter at pump discharge. If pump fails to start, check start/stop pushbutton on east wall outside control room.
9. If refrigerant low pressure or high pressure cutout occurs during operation and the heat pump stops, manually reset unit via pushbuttons on panel face. After checking refrigerant pressure gages and resetting pressure cutouts, heat pump should automatically restart after approximately two minutes. If pressure cutout occurs again, check user's manual or consult manufacturer for further details. Factory setpoints for pressure cutouts are 50 and 350 psia for high and low sides, respectively. These settings may be altered manually if water/glycol solution is used.

A.3. Data Acquisition System

The data acquisition system consists of three multiplexers, a single Campbell Scientific 21X micrologger, a MacPlus computer and an RS232 optically isolated interface. The data logger program is written in Table 1 of the data logger. Instantaneous data measurements are displayed in Mode 6. Mode 0 is employed to compile and execute the data logger program. All modes are easily accessed by pressing "*" and the mode number.

1. Initialize the data logger sampling rate by manually encoding the following sequence:

*1	Mode No.
A	Advance
10	Execution Interval (sec)
A	Advance
0	Compile Program

The data logger is now set to sample measurements at 10 second intervals. For slower or faster sampling rates, "10" may be replaced by any time interval, in seconds. However, it has been determined that 10 seconds is the optimum sampling rate for the entire data acquisition assembly.

2. Assure that discharge temperature setpoint for electric duct heater is properly set for automated control by the data acquisition system. The setpoint may be specified directly from the computer keyboard via the communications software or by inputting the following key sequence into the data logger.

*1	Mode No.
A	Advance
44	Location No.
80	Setpoint Temperature (°F)
A	Advance
0	Compile Program

3. In this case, the discharge air setpoint for the duct heater is 80 °F. For other setpoint temperatures, simply replace "80" by the desired value. The duct heater operation is described in more detail in Section A.1.
4. Turn on the MacPlus computer and retrieve the data acquisition program entitled "DataMaster V1.1". A compiled version of the program should already exist.
5. Run the program.

6. The data acquisition program, written in TrueBasic®, is designed to be a user-friendly, interactive program. Shortly after the program is executed, the user shall be initially faced with the following options appearing on the screen.

"[1] Change Test Parameters"
"[2] Change Screen Output"
"[*] Transient Data Collection"
"[4] Steady state Data Collection"
"[5] End DataMaster Session"

The bracketed number indicates the key to hit in order to apply the option. Key options "1" and "2" should be executed during start-up of any testing and are described in more detail below. The asterisk shown for "Transient Data Collection" indicates that this option is currently under development for future transient testing. If at any time during responding to this initial input prompt screen, a bad item is inadvertently specified, then the following item will display to notify the user.

"INVALID INPUT. HIT ANY KEY AND REENTER OPTION"

- 6a. If a "1" is entered at the initial options screen, the following input prompts shall be enforced and require a response from the user.

"DUCT NOZZLE #1 SIZE (INCHES)"

Requires specification of upper nozzle throat diameter installed in air flow measurement chamber for air flow rate computation. Nozzle sizes of 1-inch, 2-inches, 3-inches, 4-inches, 7-inches and 10-inches are available. For a 10-inch nozzle installed, enter "0" at this prompt. If no nozzle throat diameter is specified, the default value is 10 inches.

"DUCT NOZZLE #2 SIZE (INCHES)"

Enter lower nozzle throat diameter installed in air flow measurement chamber. If nozzle is not open, then specify "0". The default value is 4 inches.

"[1] PHASE OR [2] PHASE TESTING"

Specify whether single phase or two phase refrigerant flow testing is being performed. The default setting is "1" for single phase testing.

"[I]P OR [S]I UNITS"

Indicate desired units for screen output. Enter "I" for British unit system; "S" for System Internationale. The default setting is "I".

- 6b. If a "2" is entered at the initial options screen, the following input prompts related to the screen output shall be enforced and therefore, require a response from the user.

"[S]CHEMATIC OR [R]AW DATA"

This request is another screen display option. Input "S" for data and schematic overlay. Enter "R" for data display only. The default setting is "S".

- 6c. After utilizing key options "1" and "2" consecutively or accepting the default settings, enter "4" for steady state data collection and experimental testing. Otherwise, enter "3" for transient data collection and experimental testing. At this time, since the transient data collection software is still under development, a "4" may only be entered at this prompt. In the meantime, transient data collection may be performed with the key option "4".

7. After successful completion of the steps above, the output data screen shall then appear and shall continually update at 60 second intervals. In addition, the following options menu is displayed on the output screen. On the raw data display, this menu appears in the lower left side of the screen; whereas, on the schematic data display, this menu appears in the upper right corner of the screen.

"SAVE TOGGLE-----[S]"

"INLET AIR TEMP SET-----[A]"

"TERMINATE TESTING-----[T]"

"NEXT OPTIONS PAGE-----[N]"

- 7a. The save toggle "S" permits the user to save only the steady state during experimental testing. Simply key input an "S" at any time during testing to save data to the computer hard disk. When data is being saved, the "X" over the "DISK SAVE" icon shall disappear.
- 7b. Enter "A" to reach the following input prompt that allows changing the inlet air temperature (in deg F) setpoint value used for the input control signal to the electric duct heater.

"INLET TEMP VALUE:"

This option is generally unused, except during system startup and shutdown to turn the duct heater on and off.

- 7c. Enter "T" to receive the following input prompt that allows termination of experimental testing

"END THIS DATA SESSION? [Y/N]"

If a "Y" is entered for yes, the data acquisition program ends and the saved file(s) may be retrieved from the computer for data analysis. If a "N" is entered for no, then the initial options screen, described in is restored.

8. If an "N" is key input at the options screen of Step #5, different menu with more options is displayed.

"STEADY-STATE PLOTS-----[S]"

"SET UPDATE INTERVAL---[I]"

"NEXT OPTIONS PAGE-----[N]"

- 8a. Enter an "S" to visualize the steady state plots. However, the software for this option is currently under development.

- 8b. Enter an "I" to reset the time interval by which output data is updated on the computer screen. A 30-second or 60-second updating interval option is available, as given by the input prompt below. The default setting is 60 seconds.

"[T]HIRTY OR [S]IXTY SECOND INTERVAL "

- 8c. Finally, entering an "N" returns to the previous options menu.

9. The communications software developed also. includes several alarms and warning .messages to notify the user of potential, unsafe operating conditions with the apparatus. As identified in the warning status dialogue box on the output data screen, these warning messages may be identified and described as follows:

"SUPERHEATING IN EVAPORATOR"

This alarm occurs when the refrigerant temperature at the exit of the evaporating section of the refrigerant heater, as measured by a thermocouple mounted on the refrigerant heater, and the refrigerant pressure, equal to average of the refrigerant heater inlet and exit pressure transducer measurements, indicates any degree of superheat in the evaporating portion of the heater. The warning message should be acknowledged by decreasing the heat input to the refrigerant.

"2 PHASE FLOW PAST CONDENSER"

This warning message indicates that based on the refrigerant temperature and pressure condition at the condenser exit, there is no subcooling within the test coil or inadequate heat rejection by the coil. This condition should be monitored closely so that two phase flow patterns do not propagate throughout the rest of the apparatus. This error message may be corrected by either increasing the refrigerant flow rate, the air flow rate or decreasing the heat input.

"SYSTEM OVERHEATING-STOP NOW"

This error occurs when there is too much heat being added to the refrigerant and the measured temperature rises above 250 °F. This abnormal rise in temperature may be due to low refrigerant flow rates or no airflow across the condenser, and subsequently, no heat rejection.

"TURN OFF HEAT - LOW AIRFLOW!"

This alarm results when the air flow rate suddenly terminates as a result of a blown fuse, etc. or decreases below 200 lb_m/hr at any time during operation. In order to avoid overheating or overpressurization of the refrigerant-side, the heat input should be discontinued.

"PRESSURE > 500 PSI - STOP NOW"

Occurs when the refrigerant pressure, as measured by any one of the five pressure transducers, exceeds the system maximum pressure rating of 500 psi.

"STOP HEATING - NO R12 FLOW"

This warning occurs when the flowmeter output indicates no refrigerant flow through the loop. This may be a result of flowmeter damage or blockage, bad wiring, or circulation pump problems.

"__ WARNINGS - STOP NOW"

This warning message alerts the user of multiple warning messages, such as overheating, overpressurization, and no R12 flow circulation, all at the same time.

A.4. System Shutdown

Prior to shutting down the system, one must consider the fact that if the system is originally at a high pressure and the pressure is suddenly decreased, the sudden depressurization may result in instabilities in the system.

1. Reopen the flowmeter bypass valve and close the inlet valve to the flowmeter.
2. Decrease heat input to the refrigerant by setting superheater and evaporator controls to zero. Also, shutoff disconnect switches.
3. De-energize duct heater by entering a "0" (meaning 0 °F) for the air inlet temperature setpoint in the communications program and switching the duct heater disconnect or circuit breaker to the "OFF" position.
4. Allow air flow and refrigerant flow circulation to continue in the apparatus until the refrigerant pressures and temperatures in the system return to or approach reasonable levels (e.g., near to or at ambient conditions).

A.5. System Maintenance

1. Plastic gears are utilized in each refrigerant circulation pump. These gears should be replaced for every 2000 hours of operation. Service kits for this purpose are available from Micropump and Cole-Parmer.
2. Air filters at the entrance to the supply fan chamber should be replaced every 3 to 6 months, depending upon the amount of time the air system has been in operation. Dirt and dust build-up should be monitored visually since it increases air flow resistance. Filters are Type DP4-40 extended surface air filters, as manufactured by Airguard Industries and with nominal dimensions of 16" x 20" x 4". Filters are typically available at U of I, Operations & Maintenance Department stores.
3. Drain water loop by connecting 3/4 inch nominal diameter garden hose to system low point and routing hose to lower level, Mechanical Engineering Lab floor drain.

5. Fan bearings should be greased every 6 to 12 months, depending upon the amount of usage.
6. Constant flow, water circulation pump should have bearings lubricated regularly in accordance with nameplate instructions. This lubrication is especially important when pump has been idle for long periods of time.
7. Water loop may be flushed with chemical cleaning agents, Drewsperse 739 and Drewsperse 794, to eliminate suspended particles and debris. These antifoulants may be obtained from the Refrigeration Group at the U of I, Operations & Maintenance Department. Consult UIUC Building Standards, Section 15457, and Material Safety Data Sheets supplied by Ashland Oil Company, Drew Chemical Division, for correct procedures on handling and disposal of chemicals.

APPENDIX B

PARAMETER ESTIMATION TECHNIQUE AND COMPUTER PROGRAM

B.1. Regression Analysis

In general, regression analysis applies to the process of determining constants and coefficients (called parameters) in equations that represent a dependent variable in terms of one or more independent variables. When the parameters fail to appear in linear form in the equation, evaluating their values normally consists of an optimization process. The task is to determine the optimum values of the unknown parameters subject to certain constraints such that a user-specified objective function is minimized. The method of nonlinear least squares (NLLS) is an example of regression analysis and a parameter estimation technique. Stoecker [109] furnishes a more detailed discussion of both linear and nonlinear regression techniques.

B.2. Application of Nonlinear Regression Methods to Wilson Plots

The forms of the functional relationships generated for the modified Wilson Plots make them ideal candidates for a parameter estimation study. From Chapter 4, the general form of the two correlating equations applicable to experiments with either single phase and two phase refrigerant flow are as follows:

$$\frac{1}{UA} = \frac{1}{C_1 \left[Re^a Pr^b \left(\frac{k}{D_h} \right) A_i \right]} + C_2 \quad (B.1)$$

$$\frac{1}{UA} = \frac{1}{C_1 V^a} + C_2 \quad (B.2)$$

Equation (B.1) is associated with the single phase refrigerant flow testing; whereas, Equation (B.2) is associated with two phase refrigerant flow testing. The dependent variable in each of these expressions is the overall thermal conductance, $1/UA$. However, the independent variables differ in each case.

For the single phase refrigerant flow function, Equation (B.1), the independent variables are based on the refrigerant-side properties in accordance with the experimental test procedures, and these variables may be identified as follows:

$$\begin{aligned} \text{Re} &= \rho V D_h / \mu && \text{(Reynolds Number)} \\ \text{Pr} &= c_p \mu / k && \text{(Prandtl Number)} \\ (k/D_h) A_i &&& \text{(arbitrary variable)} \end{aligned}$$

For the two phase refrigerant flow function, Equation (B.2), the independent variable is the velocity term, which is equivalent to the air velocity through the minimum free flow area in accordance with the experimental procedures.

The unknown parameters in Equations (B.1) and (B.2) are as follows:

- a = Reynolds number or velocity function exponent
- b = Prandtl number exponent
- C₁ = inverse of slope of modified Wilson Plot
- C₂ = y-intercept of modified Wilson Plot

Neglecting the tube wall resistance, the Wilson Plot constant C₂ is also essentially equivalent to the air-side thermal resistance for single phase refrigerant flow experiments and is equivalent to the refrigerant-side thermal resistance for two phase flow experiments.

Although the form of the two correlating equations is assumed known prior to proceeding with the analysis, the degree of influence of each of the parameters is not accurately known. However, experimental data exists that may allow proper evaluation of the unknown parameters by incorporation of a nonlinear regression procedure.

B.3. Computer Program

Separate computer programs, each utilizing the method of nonlinear least squares were developed to evaluate the parameters in the modified Wilson plot functions, i.e., Equations (B.1) and (B.2). The nonlinear regression procedure also incorporates the Marquardt damping variable "lambda" which adds stability by making the program more tolerant of poor initial guesses for the unknown parameters. The source code was written in TrueBasic™.

The computer program developed for the analysis of experimental data based on single phase refrigerant flow is presented at the end of this Section. The source code written for two phase refrigerant flow data analysis is not contained herein. However, the program shown may easily be modified to accommodate the two phase analysis.

The main portion of the computer program shown is composed of five (5) separate subroutines and a single "FOR-NEXT" loop. Each subroutine is responsible for a certain functional task and the structured loop iterates the nonlinear regression routine. Although the program listing contains several comment statements that describe the importance of certain sections of the program, the following information contained herein is meant to supplement and further enhance an understanding of the program.

The five (5) subroutines names along with their associated responsibilities may be further described as follows:

1) SUB inputdat

This subroutine, the fundamental starting point of the program, loads the experimental data. It contains the input prompt "NAME OF DATA FILE TO RETRIEVE?", which requires a response that must be the name of one of eight separate TrueBASIC files containing the available single phase experimental data. For example, specification of "4/19/91-DATA" at the prompt loads the file of the same name containing experimental data from testing on 4/19/91. Each of the eight (8) data files contains a certain number of data points ("NOBS") corresponding to the date of testing and as listed in the Table 6.2. Four variables are associated with each data point,

namely, the condenser refrigerant inlet and exit temperatures ("Tci" & "Tco"), the refrigerant mass flow rate ("mdotr") and the 1/UA value ("UAinv") obtained from the appropriate effectiveness-NTU relation for a single pass, crossflow heat exchanger.

2) SUB geom

This subroutine is dedicated to specification of the condenser geometry data (e.g., tube diameter, tube length, number of circuits, etc.) and computation of the hydraulic diameter and inside heat transfer surface and cross-sectional flow areas for use in other parts of the program.

3) SUB thermo

This subroutine utilizes the data specified by the first subroutine to evaluate and return the array of values for the three independent variables; namely, the Reynolds number, Prandtl number and arbitrary ratio (kA_i/D_h). In preceding steps, the subroutine calculates the saturated liquid density, specific heat, thermal conductivity, and absolute viscosity of single-phase refrigerant R12 based on the bulk mean fluid temperature in the condenser. The thermodynamic properties are computed based on the curve-fit, polynomial expressions derived using ASHRAE data and summarized in Section 5.6.1.

4) SUB initLS

This subroutine identifies and initializes the variables specific to the nonlinear least squares routine and renames the previously evaluated dependent and independent variables to their generalized NLLS routine names, that is, the "y" and "x" arrays, respectively. In addition, this portion contains the second interactive command "INPUT INITIAL GUESSES FOR UNKNOWN PARAMETERS, a ?". The user is required to input initial, "educated" guesses for each unknown parameter depending on the value for "nA" at the start of the program. The term "educated" is emphasized since a poor initial guess may cause divergence from a solution to occur. A good initial

guess must be within the "circle-of-convergence" in order to obtain interpretable results.

5) SUB output

This final subroutine is designed to transfer the final results to a user-specified output file for printing. Experimental $1/UA$ values computed by the effectiveness-NTU relations ("y") and $1/UA$ values calculated by the newly defined function with the optimized parameters ("yHat") are printed as well as the associated percent error and sum of the residuals. These equations used for a basis of comparison are defined as listed below:

$$\% \text{ Error} = [(1/UA)_{\text{act}} - (1/UA)_{\text{NLLS}}] / (1/UA)_{\text{act}} \times 100 \quad (15)$$

$$\Sigma(\text{residuals}) = \Sigma [(1/UA)_{\text{act}} - (1/UA)_{\text{NLLS}}]^2 \times 10^{10} \quad (16)$$

A description of the "FOR-NEXT" loop may also be summarized as follows:

FOR-NEXT Loop

This loop calls the nonlinear least squares subroutine ("SUB NLLS") to evaluate the number of unknown parameters ("nA") for their most optimum values. The generalized procedure of the "NLLS" subroutine closely parallels Newton-Raphson sequence of solving nonlinear equations. The damping factor or Marquardt parameter "lambda" is included to reduce the instabilities caused by poor initial guesses for the unknown array of "a()" terms.

The nonlinear regression computer source code containing the aforementioned subroutines and the "FOR-NEXT" loop begins on the next page.

APPENDIX B Program Listing

(Available Upon Request)

APPENDIX C

(Available Upon Request)