Design and Construction of a Microchannel Condenser Tube Experimental Facility

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DESIGN AND CONSTRUCTION OF A MICROCHANNEL CONDENSER TUBE EXPERIMENTAL FACILITY

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University of Illinois at Urbana-Champaign, 1994
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ABSTRACT

A test facility was built for the purpose of performing heat transfer studies on microchannel heat exchangers. The studies will involve condensation of refrigerant 134a inside the enhanced tubes, although no condensation results are presented in this document. The design and construction of the experimental facility is detailed with a description of each component and its function in the stand. The operation of the facility was verified using an energy balance analysis and the results are presented. The refrigerant and air side heat transfers agree within ±3% at high air flow rates but fall out of this error bound at lower flow rates. Also, a discussion of the method for determining the refrigerant and air side resistances for the tube is given along with the theory for future correlation development. Finally, future modifications to the stand are suggested in order to correct any problems with it, improving the ability of the stand to produce accurate, reliable heat transfer performance data.
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NOMENCLATURE

English Symbols

A area
C fluid heat capacity
Cr capacity rate ratio
cp specific heat under constant pressure
D tube diameter
f fin height
F constant
F1, F2 geometric ratios
G Mass flow rate
h mass intensive enthalpy
h heat transfer coefficient
k thermal conductivity
L tube length
ṁ mass flow rate

NTU number of transfer units for a heat exchanger

Nu Dimensionless Nusselt number, \( \frac{hL}{k} \)
p pressure
P perimeter length
Pr Dimensionless Prandtl number as defined by Eqtn. 2.14
Q total heat transfer
R thermal resistance
Re Dimensionless Reynold's number, \( \frac{pVD}{\mu} \)
Re\text{lo} Reynold's number given by Eqtn. 2.16
T temperature
UA_{tot} total heat transfer coefficient
v mass intensive volume
V velocity of stream
w tube width
x vapor quality
X_{lt} Lockhart-Martinelli parameter as described by Eqtn. 2.8

Greek Symbols
\alpha thermal diffusivity
\varepsilon heat exchanger effectiveness
\eta_o fin efficiency
\mu dynamic viscosity
\nu kinematic viscosity
\rho density

Subscripts
air air based
crit critical value
e equivalent
f fin
i inner
in value at entrance to tube
l liquid
lg two-phase value
m median value
min minimum
max maximum
out value at exit of tube
p primary
r reduced
ref refrigerant based
sat saturation value
v vapor
CHAPTER 1: INTRODUCTION

Condensation heat transfer studies have been conducted since the early 1900's when Nusselt developed a correlation for a thin film condensing inside a tube (Nusselt, 1919). Since then, multiple experiments have been performed which provide more accurate expressions for the heat transfer coefficient of a condensing liquid. With the advent of refrigeration, interest in condensation has increased since a condenser is one of the main components of the basic vapor-compression refrigeration cycle. This document discusses one such investigation into the condensation of refrigerant in a microchannel condenser tube. Figure 1.1 shows a schematic of a full microchannel condenser. The tubes are brazed to the headers with louvered fins in between. The refrigerant is circulated using baffles inside the headers, involving more than one tube in each pass. This reduces the total pressure drop of the condenser due to fewer passes.

These heat exchangers provide the same amount of heat transfer with a much lower refrigerant charge (about 2/3 that of a serpentine condenser) than condensers now commonly used in the industry. This attribute is very beneficial to car manufacturers (where they are currently being used) as any reduction in the space taken up by the components under the hood is greatly needed. Also, a smaller charge requirement translates into a lower cost for installation and less of a threat to the environment should the system ever leak. Because of this, interest in this technology is high. However, little is known about the effects that each tube characteristic (port geometry, tube width, refrigerant inlet quality, etc.) have upon the heat transfer capability of these tubes. More research is required to determine how these parameters can affect this new condenser's performance.

Now, environmental concerns have been introduced into the industry, creating hesitations which have resulted in governmental restrictions. Due to the concern over ozone depletion, greenhouse effect, and other problems associated with the current state of technology, research in refrigeration has grown even more. Now, research focuses on determining the properties and effects of using alternative, non-chlorofluorocarbon (CFC) refrigerants such as 1,1,1,2-tetrafluoroethane (R-134a) as replacements for the commonly used CFC's. Time is being wisely spent on new refrigeration component technology in order to protect our finite environment. This project will follow suit by using ozone-safe R-134a as its working fluid.

The future use of the microchannel condenser technology is predicted to increase substantially over the next decade. The new governmental intervention in the refrigeration and air conditioning industry is also creating an increased interest in this
technology, due to the lower charge and space required by these components. The Montreal Protocol and future legislation like it will create a problem for air conditioner manufacturers. This creates a need for a full understanding of the way these tubes behave since using them could make complying with governmental regulations much more attainable. Though their attributes have proven to be fruitful (Struss, 1989), they are not fully understood. Many variables could be contributing to their performance but little is known about the characteristics which most affect the tubes' behavior.

Due to all of these factors, investigation into the physics of this technology is required in order to fully develop it benefits. This document will describe the design and construction of a test facility which can be used to learn more about these edge-of-technology heat exchangers.

1.1 Tube Description

More information and new breakthroughs in the current state of refrigeration technology is required in order to combat the problems listed above. One such breakthrough has occurred with the advent of a slightly different design for an air conditioning condenser. These condensers, here called microchannel condensers, utilize webs inside tubes with very small hydraulic diameters. Figure 1.2 shows a cross-section of one of these tubes. The tubes are wide and flat, much like serpentine condenser tubes, but are much thinner than the condensers commonly used in car air conditioning today. The tube contains web shaped enhancements on the inside. These webs are believed to further increase the heat transfer characteristics of the tubes. The diagram shows a web with a triangular shape but other port geometries are available. These include square, H, and circular designs. Figure 1.3 gives a schematic of some of the port geometries currently in use. This technology is fairly new because the ability to fabricate the tubes has only recently been developed. Currently, the tubes are extruded, allowing the inserts to be formed along with the tube, or welded, in which case the webs are fabricated separately and brazed inside the tube after insertion. Also, louvered fins are bonded to the top and bottom of each tube in the full condenser. The louverers enhance the fins by reducing the air side resistance of the tubes. In a full condenser, the tubes are stacked upon each other with the fins in between. This conglomerate is then circuitized by two headers, one on each side of the tubes. The headers utilize baffles to direct the flow, creating very few passes (only three on some models). Since the size of the tubes is so small, it would seem that the pressure drop associated with a full condenser would more than offset the added heat transfer benefit.
Figure 1.3: Microchannel Tube Port Geometries

- "H" ports
- Square ports
- Triangular ports
- Circular ports
However, the header arrangement allows for approximately the same pressure drop as seen in plate-fin and serpentine condensers, due to the small number of passes the refrigerant must go through between the entrance and exit.

1.2 Objectives

A single-tube test facility is to be constructed in order to provide information on the effects that varying different tube characteristics have upon the heat transfer performance of microchannel condenser tubes. Utilizing a single-tube facility eliminates the role that the headers play in the performance of these condensers in order to single out each parameter of interest. This provides for a more controlled experimental atmosphere and will prevent any secondary characteristics from influencing the data. In the facility, the ability to vary each parameter of interest must be maintained in order to draw useful conclusions from the data. Precise control of all operating parameters is required.

The test facility consists of a refrigerant and air side. For this report, an emphasis is placed upon the construction of the refrigerant side but a brief discussion of the air side components will be given. This report will describe each component of the refrigerant side in detail. The facility also employs many components which are not standard for such condensation experiments. This project will discuss their theory of operation and their performance upon implementation in the stand. For an in-depth description of the air side components, the reader is referred to a forthcoming Air Conditioning and Refrigeration Center (ACRC) document by D. Andres (Andres, 1994).

No two-phase or superheat refrigerant inlet heat transfer data will be obtained in this project but basic data will be taken which will verify the operation of the facility. The scope of this project encompasses only the construction of the facility. Once the facility is proven to be acceptable, it can then be used for future condensation studies. Initially, microchannel tube heat transfer performance data will collected using the facility, the results of which will be presented in a forthcoming document. The facility was designed for use with refrigerant 134a as the working fluid. However, the facility was also designed and constructed in such a way that it may be used to study condensation in other tubes beside the microchannel variety. This constraint requires a much wider range of applicability from the stand in order to accommodate the other tube geometries and operating conditions.
CHAPTER 2: REVIEW OF EXISTING LITERATURE

2.1 Introduction

The topic of condensation has long been studied by academics and industrialists. In 1916, Nusselt first derived an expression for the heat transfer coefficient of a fluid condensing inside a tube. Since then, great amounts of research have been conducted for the purpose of providing accurate, reliable heat transfer coefficient correlations for condensation. Recently, much of this research has been stemmed from the growing interest in the relationship between the environment and the refrigeration industry. Non-CFC refrigerants, heat exchanger tube enhancements, and other new refrigeration technologies have been investigated due to this increasing concern about our future.

This review attempts to briefly summarize the contributions to date which involve condensation heat transfer correlations. These correlations can be used to compare with those generated from using the test facility discussed in this document. Also, previous experimental facility design was evaluated before the construction of this stand began in order to create a more efficient and accurate test facility for use in this project.

Since microchannel tubes are a relatively new technology, their heat transfer performance has not been studied in depth. This is one of the main reasons for constructing this test facility. The literature that does exist will be discussed, however, in order to demonstrate the need for an accurate experimental facility which can be used to more thoroughly understand these condenser tubes.

2.2 Horizontal Tube Condensation Heat Transfer Coefficients

The derivation of a heat transfer coefficient correlation was originally developed by Nusselt in 1916. This correlation was very limited in its application since it was based upon laminar flow of a thin film in a vertical tube. It carries with it many assumptions which caused interest in determining other, more general correlations, such as those developed by Prandtl. Prandtl's results allowed for more freedom in the type of flow being analyzed than did the Nusselt equation. With the advent of consumer air conditioning and refrigeration, refrigerant flow in horizontal tubes began to be investigated. These projects include experimental and theoretical derivations of condenser tube heat transfer performance.

Initially, refrigerant condensation studies were done using R-12 and R-22 since these are the most widely used refrigerants in industry. This research started heavily in 1959. Akers et. al. looked at two-phase flow by treating both liquid and vapor phases
individually and deriving an expression for the mass flow of the refrigerant as a combination of the two phases:

\[ G_m = G_l + G_v \left( \frac{\rho_l}{\rho_v} \right)^{1/2} . \]  

This was subsequently used to determine the Nusselt number correlation:

\[ Nu = 0.0265Re^{0.80}Pr^{1/3} . \]  

Azer, Abis, and Swearingen (1971) used experimental methods to determine a correlation for the condensation of R-12. The result utilized the Lockhart-Martinelli (1949) parameter to give

\[ Nu = 0.039pq^{337Re} \left( \frac{90}{4.67-x} \right) \left( \frac{\mu_v}{\mu_l} \right) \left( \frac{\rho_l}{\rho_v} \right)^{1/2} . \]  

This expression does not agree very closely (±30%) with that developed by Bae, et. al. in 1971. Bae used the analogy between momentum and heat transfer to generate an annular flow model that agreed to within 10% of the experimental results he obtained. Bae’s correlation carried with it many assumptions which make it applicable only to a small range of conditions and limiting its use.

A vertical tube was used by Cavallini and Zecchin (1974) to investigate condensation at high velocities. Under this condition, the tube orientation does not effect the heat transfer performance much allowing the application of the correlation to horizontal tubes as well. Cavallini and Zecchin determined a mean Reynold's number for the refrigerant much like Akers, treating it as a weighted average of the liquid and vapor phase Reynold's numbers:

\[ Re_m = Re_{v} \left( \frac{\mu_v}{\mu_l} \right) \left( \frac{\rho_l}{\rho_v} \right)^{1/2} + Re_l . \]  

The values for the liquid and vapor are calculated using mean values between the inlet and outlet of the tube being tested. The relation is then used to determine the Nusselt number by
which correlates with experimental R-12 data by less than 15% error.

Later, experiments were conducted which provided expressions generating better agreement among studies done by different researchers, including the Cavallini and Zecchin study. These correlations involve annular flow in the turbulent regime. First, Traviss, Rohsenow, and Baron (1973) utilized the turbulent flow equations derived from an analysis of condensation under conditions of turbulent forced convection. They used conservation of momentum and the von Karman universal velocity distribution to give

\[ \text{Nu}_m = 0.05 \text{Re}_m^{0.80} \text{Pr}_l^{3.3}, \quad (2.5) \]

where the constant \( F(X_{tt}) \) depends upon the Lockhart-Martinelli (L-M) parameter for two-phase, turbulent flow and is defined as

\[ F(X_{tt}) = 0.15 \left( \frac{1}{X_{tt}} + \frac{2.85}{X_{tt}^{4.76}} \right), \quad (2.7) \]

where the L-M parameter can be found by

\[ X_{tt} = \left( \frac{\mu_l}{\mu_v} \right)^{0.1} \left( \frac{1-x}{x} \right)^{0.9} \left( \frac{P_v}{P_l} \right)^{1/2}. \quad (2.8) \]

The correlation given in (2.6) depends upon the flow regime that the refrigerant is in. Traviss, et. al. set up expressions for the constant, \( F_2 \), according to the Reynold's number of the refrigerant liquid entering the condenser:

\[ F_2 = 0.707 \text{Pr}_l \text{Re}_l^{1/2} \quad \text{Re}_l < 50, \quad (2.9a) \]

\[ F_2 = 5\text{Pr}_l + 5\ln[1+\text{Pr}_l(0.09636\text{Re}_l^{5.85} - 1)] \quad 50 < \text{Re}_l < 1125, \quad (2.9b) \]

\[ F_2 = 5\text{Pr}_l + 5\ln(1+5\text{Pr}_l) + 2.5\ln(0.00313\text{Re}_l^{8.12}) \quad \text{Re}_l > 1125. \quad (2.9c) \]
A few years later, Shah (1979) took an expression for the liquid heat transfer coefficient and scaled it to produce a two-phase correlation. He proposed that this two-phase coefficient is dependent only on the condensation number and the Froude number. The single-phase coefficient is described by

\[ \bar{h}_l = 0.023 \text{Re}_l^{80} \text{Pr}_l^{40} \frac{k_l}{D}, \quad (2.10) \]

where \( k_l \) is the condensation number associated with the refrigerant. This correlation was used, in conjunction with data obtained using various refrigerants (R-11, R-12, and R-113) and water, to determine the two-phase expression. This purely experimental relationship is described by

\[ \bar{h}_g = \bar{h}_l \left[ (1 - x)^{80} + \frac{3.8 x^{76} (1 - x)^{04}}{p_r^{38}} \right], \quad (2.11) \]

where \( p_r \) is the reduced pressure of the refrigerant. The correlation corresponds closest with experimental data at \( \text{Re}_l > 3000 \).

The correlations described above use expressions for the liquid and vapor Reynolds numbers. These values are determined utilizing the inlet quality of the refrigerant in the following expressions:

\[ \text{Re}_l = \frac{G D (1 - x)}{\mu_l}, \quad (2.12) \]

and

\[ \text{Re}_v = \frac{G D x}{\mu_v}. \quad (2.13) \]

Also, the Prandtl number for the liquid refrigerant, \( \text{Pr}_l \), is used in the expressions. This non-dimensional number is defined as

\[ \text{Pr}_l = \frac{c_p l \mu_l}{k_l} = \frac{\nu_l}{\alpha_l}. \quad (2.14) \]
Enhanced tubes with internal fins were used by Luu (1979), who developed a heat transfer coefficient correlation which employs a geometric correction factor attributed to the fins. The two-phase, smooth tube correlation, $h_g$, was developed by Boyko and Kruzhilin (1967) by using the independent liquid and vapor phases to produce a mean relationship, much like Cavallini and Zecchin. The Boyko and Kruzhilin expression is given by

$$h = 0.024 \frac{k_l}{D} \text{Re}_{lo}^{0.8} \text{Pr}_{l}^{0.43} \left\{ \left[ \frac{1 + \left( \frac{\rho_l - \rho_v}{\rho_v} \right) x}{\text{in}} \right]^{1/2} + \left[ \frac{1 + \left( \frac{\rho_l - \rho_v}{\rho_v} \right) x}{\text{out}} \right]^{1/2} \right\}, \quad (2.15)$$

where,

$$\text{Re}_{lo} = \frac{GD}{\mu_l} \quad (2.16)$$

This correlation is modified by the geometry of the tube to produce Luu's expression for the heat transfer coefficient of a finned tube:

$$h_{g,f} = h_g \left( \frac{f^2}{wD_l} \right)^{-0.22}, \quad (2.17)$$

where $w$ is the fin width and $f$ is the fin height.

Likewise, Azer and Said (1982) developed a correlation which corrects the smooth tube expression developed by Akers. This correlation does not give excellent agreement with experimental data but can be used as an approximation and may be corrected with further research.

More recently, Kaushik and Azer (1988) used several types of tubes and various fluids to develop a correlation which is applicable to more situations than those given previously. The correlation is based on Akers et. al. work done in 1959 but contains a few modifications which involve geometry concerns and pressure contributions. The Nusselt number expression they developed is given by

$$Nu = 2.078 \text{Re}_e^{0.507} \text{Pr}_{l}^{1/3} \left( \frac{\Delta x D_l}{l} \right)^{0.198} \left( \frac{p}{p_{\text{crit}}} \right)^{-0.14} F_1^{0.874} F_2^{0.814}, \quad (2.18a)$$
with the equivalent Reynolds's number, $Re_e$, described by

$$Re_e = \left[ (1 - x) + x \frac{\rho_l}{\rho_v} \right] Re_i,$$

(2.18b)

where $F_1$ is the ratio of the free flow area to the fin open core flow area, $F_2$ is the ratio of the internal area of a smooth tube having the same inside diameter as the finned tube to the surface area inside the actual finned tube, $\Delta x$ is the change in quality across the tube, and $l$ is the length of the condenser tube. This correlation holds true for $F_1$ values less than 1.4. For values greater than this number, the following expression must be used:

$$Nu = 0.391Re_e^{507}Pr_1^{1/3} \left( \frac{\Delta x D_i}{l} \right)^{198} \left( \frac{P}{\rho_{crit}} \right)^{-14} F_1^{4.742}.$$

(2.19b)

Bonhomme (1991) provided data which were compared with the results obtained by Traviss, Shah, and Cavallini and Zecchin. The experimental results fell within 17-25% of the predicted values for refrigerant side heat transfer coefficients of R-134a in a 0.402 inch diameter tube. The error between the data and the predicted values increased with increasing flow rate and refrigerant inlet quality. Hinde et. al. (1992) conducted experiments involving R-12 and R-134a in an annular counterflow condenser. Empirical data were collected in order to determine purely empirical refrigerant side heat transfer coefficients using Newton's law of cooling. Water was used as the cooling fluid located in the outer annulus of the condenser. The data show an increase in heat transfer coefficient with increasing average refrigerant quality, the greatest increase coming with annular refrigerant flow caused by high refrigerant mass fluxes. The data were consistently lower than the values predicted by the correlations given in equations (2.5), (2.6), and (2.11).

The results of such testing as described above can be compared with experimental data obtained using the microchannel test facility. Also, the effect of the tube enhancements inherent to the microchannel tubing can be investigated, providing insight as to the difference in performance between the conventional round tubes and microchannel tubing.
2.3 Small Tube Studies

Though the microchannel tube technology is a new one, some research has been done utilizing small or microchannel tubes. Goodremote et. al. (1988) showed that the reduction in charge gained by using microchannel tubes is significant. Also, the pressure drop across the condenser and the overall size are reduced when compared with serpentine and plate-fin models.

Also, experimental testing done by Struss et. al. (1989) show that heat transfer performance using a microchannel condenser is equal to or greater than that provided by a serpentine vehicular condenser which requires a much higher refrigerant charge.

Finally, Sugihara and Lukas (1990) further supported the results produced by Struss et. al. Using R-12, they showed that increasing both the refrigerant and air flow rates increases the heat transfer coefficient of the microchannel condenser more drastically than the effect seen on a serpentine condenser. Some of these effects are attributed to the design of the header configuration for the condenser but much of the effect is also due to the characteristics of the tube. This paper details the positive effect that louvered fins have on the heat transfer performance of the heat exchanger. The louvers greatly reduce the air side resistance of the condenser tube. This result is a basis for the use of louvered fins for this project.

These results provide motivation for further experimentation on microchannel tubing. Much is not known about the mechanisms by which these tubes provide such increased heat transfer capabilities. Experimental work can reveal some of these mechanisms and what affects them.

2.4 Refrigerant Condensation Test Facility Design

Many of the correlations given in section 2.2 were determined using empirical results from experimentation. Others were developed from theory (making various assumptions about the characteristics of the refrigerant flow) but were compared to data obtained from experimentation. These facilities were used as models for the test facility constructed for this project.

The stand constructed for this project is similar in design to that used by Dobson (1994) for the study of heat transfer and flow regimes in pure refrigerant condensation. Like Dobson's stand, this facility uses a pump, aftercondenser, and heater in the refrigerant loop. However, the bladder/accumulator used for refrigerant loop pressure
regulation is not used, nor is water used as the cooling fluid. Instead, air is used as a cooling medium in order to more accurately simulate actual vehicular conditions.

The basis for the design of the air loop was derived from a paper written by Gavin (1983). Gavin investigated the evaporation of nonvolatile solute filled drops. In order to collect data, he constructed an extensive wind tunnel facility, much like that described in this report. Included in the wind tunnel are a stilling chamber, screen (velocity regulating) section, a contraction area, and a test section. Much more air flow control was required for his experimentation but his overall design of the facility was used to design the air side of this microchannel tube test facility.

2.5 Results of Literature Search

All of the literature cited in sections 2.2, 2.3, and 2.4 gave ideas and suggestions for the design and construction of the experimental facility discussed in this document. Most experimental set ups used for condensation studies used water instead of air as a cooling medium due to its well-known properties and the higher heat transfer coefficients associated with a liquid. However, air properties have been thoroughly documented. Also, the condenser tubes which are to be tested using this facility contain air-side enhancements which eliminate the need for a liquid cooling fluid.

This literature review was designed to give brief overview of previous condensation studies that have been conducted. These works are used to provide motivation for the design of the stand and to give a sampling of correlations which can be used as a vehicle of comparison for any expressions developed using this test facility. It does not, however, provide an exhaustive list of all previous work done on condensation of refrigerants. For a more comprehensive list of previous condensation experimentation, the reader is directed to the Ph.D thesis written by Dobson (1994).
CHAPTER 3: DESIGN AND CONSTRUCTION OF THE EXPERIMENTAL FACILITY

3.1 Overview of Test Facility

One of the main goals of this project was to construct, debug, and maintain a facility which is to be used to test the steady-state performance of microchannel heat exchanger tubes. The facility must provide data which is both accurate and precise while allowing for strict control of the many process parameters. Operating variables measured on the stand, such as temperature and pressure of refrigerant, will be used to generate heat exchanger tube performance parameters. The parameters for which the facility will be used to determine include (a) the inside heat transfer coefficient for the tube being tested, (b) single-tube pressure drop, and (c) tube surface roughness. These characteristics will be useful in determining the overall effectiveness of the tubes, and provide a comparison to the existing technology.

Currently, this technology is being used in the automotive air conditioning arena. Some small automotive condensers are fabricated utilizing the microchannel tubes and special louvered fins. For this reason, the facility must simulate the actual conditions seen by a condenser while still preventing the occurrence of uncontrollable variable interactions. This requirement will have a great effect on the selection of all facility components.

3.1.1 Design Criteria

Before constructing the test facility, we determined what the requirements of the facility would be. This is a necessary step in order to reduce the number of modifications required later and to avoid wasting resources on components which are unneeded or can be replaced with a more economical and efficient substitute. Elements of the stand must separately and collectively meet the guidelines set out by the design criteria. Determining the final goals of the project will reveal what sort of performance criteria must be met. A test facility which is to be used for evaluating the performance of single heat exchanger tubes has many requirements which must be adhered to, the design must provide for such requirements.

The test facility is to be used for single-tube testing, specifically for microchannel heat exchanger tubing. However, since the one of the goals of the project is to construct a test facility which can be used in future research projects involving single
tube performance experiments, the operating capabilities of the stand must allow for a wide array of operating conditions, both on the refrigerant and air sides.

Secondly, the test facility must be able to physically accept not only microchannel tubes but also a wide array of other tubes (i.e. round). It is not necessary to construct the stand in such a way that any one tube can be removed and another can be installed immediately in its place. Such a feature is not feasible due to the wide variety of tube size and shape possibilities. However, the installation of different tubes of the same general form should be simple and require as little time as possible. Methods for positioning the tubes in line with the rest of the refrigerant loop should be reversible yet still provide a safe, continuous joint. These methods must also provide repeatable data by being able to be used to prescribe the flow of the refrigerant into the test section, including boundary layer destruction and other flow conditioning. The modifications to the stand required to install the tubes should be minimal and cost effective.

Also, one of the benefits of this new technology in heat exchanger tubes is the reduction in charge associated with the use of microchannel tubing. As discussed previously, this provides a benefit to many sectors of industry. In order to keep the philosophy behind the design of the test stand in accordance with the reduced charge nature of these heat exchangers, the final charge of the refrigerant loop should be kept to a minimum. This should be accomplished using as little refrigerant transport tubing as possible and minimizing the collection of refrigerant charge in components such as bladder/accumulators, heat exchangers, and other necessary components. Of course, some high charge components might be required but there use should be kept to a minimum.

Next, the goals for the stand's operating capabilities had to be defined. The facility was to have the capacity to run at the "normal" condenser operating conditions seen during typical car air-conditioning operation. These normal operating conditions were not meant to restrict the facility but to instead set some minimum requirements for the stand's capabilities. The actual capacity of the test facility must offer a much broader range of operating conditions than that seen in actual condenser operation. This would ensure that the stand could be used to test many different kinds of tubes in the future. The specific operating conditions for the microchannel condenser were determined from the literature (Struss, 1989). Table 3.1 lists these parameters. The ranges are typical for full condenser performance and were thus proportioned for the single tube facility by a scaling factor. These values were used in order to size the components which were implemented in the stand.
Table 3.1: Full Condenser and Single Tube Operating Conditions

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Full-Condenser Values</th>
<th>Single Tube Requirements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant Inlet Pressure</td>
<td>230 psi 1.5 MPa</td>
<td>230 psi 1.5 MPa</td>
</tr>
<tr>
<td>Condensing Temperature</td>
<td>55 °C 135 °F</td>
<td>55 °C 135 °F</td>
</tr>
<tr>
<td>Refrigerant Inlet Condition</td>
<td>70 °F (40 °C) superheat</td>
<td>40°F(10°C) subcool to 9 °F (5 °C) superheat</td>
</tr>
<tr>
<td>Refrigerant Mass Flow Rate</td>
<td>5.00-8.33 lbm/min 300-500 lbm/hr 0.0379-0.0631 kg/s 2.27-3.79 kg/min</td>
<td>0.50-0.833 lbm/min 30-50 lbm/hr 0.0038-0.0063 kg/s 0.227-0.379 kg/min</td>
</tr>
<tr>
<td>Refrigerant Pressure Drop</td>
<td>4.0-14 psi 28-96 kPa</td>
<td>1.3-5.6 psi 9.0-39 kPa</td>
</tr>
<tr>
<td>Air Inlet Condition</td>
<td>100 °F 40 °C</td>
<td>100 °F 40 °C</td>
</tr>
<tr>
<td>Air Velocity</td>
<td>405-1622 ft/min 6.75-27.0 ft/s 116-552 m/min 1.93-9.2 m/s</td>
<td>405-1622 ft/min 6.75-27.0 ft/s 116-552 m/min 1.93-9.2 m/s</td>
</tr>
<tr>
<td>Face Area</td>
<td>1.11 ft² 0.1032 m²</td>
<td>0.0358 ft² 0.00333 m²</td>
</tr>
<tr>
<td>Air Volumetric Flow Rate</td>
<td>450-1800 ft³/min 7.5-30 ft³/s 12-57 m³/min 0.20-0.95 m³/s</td>
<td>15-66 ft³/min 0.25-1.10 ft³/s 0.06-1.8 m³/min 0.001-0.030 m³/s</td>
</tr>
<tr>
<td>Air Mass Flow Rate</td>
<td>28.0-112 lbm/min 0.467-1.87 lbm/s 12-57 kg/min 0.20-0.95 kg/s</td>
<td>0.934-4.11 lbm/min 0.0156-0.0685 lbm/s 0.06-1.8 kg/min 0.001-0.030 kg/s</td>
</tr>
</tbody>
</table>

Finally, the required measurement accuracy's for the facility were determined. The individual parameters measured on the stand must lie within their required accuracy to produce an acceptable heat transfer coefficient value. These parameters include measurements of (a) temperature, (b) pressure, (c) mass flow rates, and (d) power inputs. The accuracy of these measurements is largely dictated by the manufacturing capabilities for the instrument of interest. However, the required accuracy for a balance of energy between the air and refrigerant sides and subsequently, the determination of a
refrigerant side heat transfer coefficient, was set. This accuracy was chosen as a ±3% margin of error between the refrigerant and air side heat transfers.

3.1.2 Overview of Refrigerant and Air Loops

Figure 3.1 shows a schematic for the condenser tube test facility. The facility consists of an air and a refrigerant side. The air loop was designed to act as a controlled wind tunnel atmosphere with temperature and air mass flow rate fixed by the operator. The refrigerant loop was designed to act under the conditions of a typical car air conditioning system with operator influence over the (a) condensing temperature, (b) condensing pressure, (c) quality of refrigerant entering the test section, and (d) refrigerant mass flow rate. The individual components for both refrigerant and air sides are listed in their respective places in each loop. The components chosen were required in order to produce steady-state conditions on both the refrigerant and air sides.

3.1.2.1 Air Loop

One of the two main sections of the test facility is the air loop. A fan is used to force the air throughout the wind tunnel's ductwork. Since the air mass flow rate is set using this component, tight control of its operation is required. Also, an accurate determination of the actual mass flow rate is also required. In order to fulfill these two criteria, a venturi flow meter is utilized. The pressure difference between the inlet to the venturi and its throat can be correlated to the mass flow rate of the fluid passing through it. A process controller, which uses a proportional-integral-derivative (PID) control algorithm, regulates the voltage input to the fan motor, thereby keeping the speed of the fan constant.

Once the mass flow rate is set, the air passes over the temperature control devices. These consist of a (a) precoolor and a (b) heater. The precoolor removes heat and fights the operation of the heater. At a steady-state condition, if both heat input and heat removal rates are held constant, the temperature of the air remains constant according to the net amount of heat transfer from the two temperature control devices.

The next section in the air loop is the flow conditioning section. In this section, the air passes through a plenum which serves as a mixing chamber. The plenum is employed in order to remove any temperature stratification due to uneven heating or cooling. Then, the air continues through flow straighteners. Here, conditioners such as
Figure 3.1: Sub-Assembly Test Facility Schematic
straws and screens are used to create a constant velocity profile across the width of the test section.

Next, the air stream reaches the test section. Here, the air temperature and pressure are measured at the inlet and exit of the test section. The inlet temperature should be uniform across the whole duct upon entering the test section. The temperature is then used to determine the enthalpy of the air stream before it passes over the test section. After the tube, temperature and pressure measurements are once again made in order to determine the downstream air enthalpy. The two enthalpies are used to determine the heat transferred to the air stream from the condenser tube.

The air stream develops a thermal gradient as it picks up heat from the test section. This is due to the fact that the refrigerant is cooled as it moves down the length of the condenser tube. Since the refrigerant is at a higher temperature when it enters the tube than when it exits, the air stream has a corresponding thermal gradient perpendicular to the air flow (in the direction of the refrigerant flow). Therefore, a plenum is also utilized downstream of the test section to eliminate this temperature profile. Without the use of an air mixing device, the temperature of the air entering the venturi again will be stratified. This will reduce the accuracy of the mass flow rate, since the mass flow rate is determined from temperatures and pressures measured before the venturi.

3.1.2.2 Refrigerant Loop

The other main section of the stand is the refrigerant loop. The flow of the refrigerant side is controlled by a positive displacement gear pump. A pump is used in order to eliminate the effect of lubrication on the condensing process. A compressor requires oil to be dissolved in the refrigerant. However, an oil/refrigerant mixture will have different heat transfer capabilities than pure refrigerant. Using a pump eliminates the need for lubrication in the refrigerant system. The magnetic positive displacement pump is regulated using a process controller. The mass flow rate of the refrigerant is measured using a Micro-Motion flow meter. The flow rate corresponds to a voltage output from the flow meter. This output is used as the process variable for the pump process controller, which utilizes a PID control algorithm. The mass flow rate of the refrigerant is held constant by this control loop.

After the mass flow rate of the refrigerant is measured with the flow meter, the refrigerant enters the enthalpy setting tank (EST). This device is used to set the state of the refrigerant before it enters the condenser tube. The state set can be (a) the
temperature of refrigerant liquid if a subcooled liquid test is to be run, (b) the quality of refrigerant if a two-phase test is to be performed, or (c) the temperature of the refrigerant vapor if a superheat test is conducted. The condition of the refrigerant, as it exits this component, is determined by the amount of heat added to the copper coils from the glycol bath surrounding them. This heat input can be regulated by the operator or by a control loop involving the temperature of the bath as the process variable. The outer tank acts as an insulator to keep heat from being lost to the surroundings. The EST system will be discussed more in depth later in this document.

Once the refrigerant leaves the EST, it enters the test section. Its temperature and pressure are both recorded and should remain constant at steady-state. Here, heat is given off to the air stream as the refrigerant passes through the tube. This decreases the temperature of the refrigerant (if it enters the test section as a subcooled liquid) or reduces the quality of the refrigerant (if it enters the test section as a superheated vapor or two-phase), possibly cooling it. For the former case, the change in refrigerant temperature as it passes through the condenser tube will reveal the enthalpy loss to the air stream. If the refrigerant exits two-phase, the heat transfer determined from the air side can be used to define the state of the refrigerant at the exit of the test section.

Next, the refrigerant flows through a heat exchanger which is used to fully condense the refrigerant. This process is exceedingly important since it ensures that the pump will be fed only subcooled liquid. If any vapor were to reach the pump, it would severely reduce the life of the pump. The heat exchanger is fed with chilled water at a low temperature from the building's supply.

Finally, the refrigerant, properly condensed, flows back to the pump. Usually, in an air conditioning cycle, a compressor is used to control the flow rate of the working fluid and keep the condensing pressure of the system constant. However, compressors cycle on and off during their operation. This does not offer the pure steady-state condition which is required for the operation of this facility. Instead, the pump is used to control the flow rate and a pressure regulating tank (PRT) is utilized to keep the system pressure constant during testing. The PRT uses a vessel containing two-phase refrigerant. The temperature of the glycol bath surrounding the vessel is held constant by the steady-state operation of a heater and cooler. This, then, not only keeps the pressure of the refrigerant loop constant, but also allows that pressure to be fixed by the operator. The PRT, in combination with the refrigerant pump, eliminates the need for a compressor in the refrigerant loop and offers maximum control of the refrigerant flow.
3.2 Test Facility Design and Implementation

Once the design criteria were determined, the individual components of both the refrigerant and air loops were chosen so as to fulfill these prerequisites. This report details explicitly the design and construction of the refrigerant loop. A forthcoming ACRC report will particularize the design and construction of the air side of the test facility. A brief discussion of the components of the air loop and the motivation behind their design will be made but considerations about their implementation will not be detailed. Components which are utilized in both refrigerant and air loops will be detailed and their relation to the test facility as a whole will be discussed.

3.2.1 Data Acquisition

3.2.1.1 Data Acquisition Device

The data acquisition system installed in the facility is a Fluke 2280 Data Logger. A full description and list of operating commands for this machine is included in Appendix A. The Fluke can scan and record a variety of sensor inputs through the use of interchangeable boards which are easily installed in the back of the machine. Many sensor and other inputs can be monitored using this device. For this project, only isothermal boards used for inputting thermocouples and other DC voltage devices are used. Each input on one of these boards is considered a channel and all of the sensor points on the stand are connected to one of these channels. By this mechanism, the Fluke can record all the pressure, temperatures, mass flow rates, etc. on the stand which are desired by the researcher. These values are then utilized later to determine quantities of interest.

In order to facilitate the use of the Fluke, a computer is used for all interaction between the operator and the data acquisition system. The computer used in the facility is a Macintosh Ilse, with an 80 MB hard drive. Communication between the Fluke and the computer is accomplished using a software package called Versa-Term Pro. Information on and a list of commands for the use of this software are given in Appendix A. The program provides a mechanism by which the Fluke can be programmed and data can be stored. The Fluke itself can store data on a magnetic tape or the data can be written directly to a file on the computer's hard drive, where it can be stored for future retrieval. Data collected using this facility is stored on the computer and gathered later to determine properties and to calculate heat transfers and heat transfer coefficients.
3.2.1.2 Thermocouples

Thermocouples were installed in the test facility for the purpose of measuring temperatures at various points around the stand, including both the air and refrigerant side. All the thermocouple wire used was size 36 gauge, type T (copper-constantan). Each thermocouple came covered with a Teflon™ insulation coating in order to prevent the positive and negative leads from touching anywhere but at the bead. The size of the gauge created problems in working with the wires because they were so fragile. They frequently break and must be replaced.

Before installation, each thermocouple used to obtain temperature data was calibrated using ASTM standard approved mercury thermometers. More than one thermometer was required due to the short temperature ranges associated with more accurate thermometers. Any thermocouple that did not fall within the given accuracy specified by the manufacturer (Omega) for a special limits of error thermocouple (±0.1°C) was replaced with a functioning one. They were calibrated using a constant temperature bath with adequate circulation to ensure a uniform temperature. The thermocouples were also insulated from any electrical noise which might be present in the bath by immersing them in a stainless steel beaker containing bath fluid (a glycol mixture). The beaker itself was then connected to a building ground before any verification testing was conducted, thus providing a path by which the noise could be rejected.

In order to provide temperatures with the highest precision, a separate thermocouple referencing method than that supplied by the data acquisition system is used. The Fluke's input board for thermocouples and other voltage devices uses an aluminum scanning plate. The temperature of this plate is utilized as each thermocouple's reference temperature since they are all connected to it. The Fluke assumes that the aluminum plate is isothermal and uses a thermistor to measure the temperature of the plate at one position on its surface. Unfortunately, the thermistor can not be separately calibrated and therefore its accuracy cannot be verified. Its use as a reference source is therefore suspect. To avoid these problems, we created our own reference source, the state of which we can be sure is isothermal. For each thermocouple being used to measure an absolute temperature in both the air and refrigerant loops, a reference thermocouple is used. These reference thermocouples are submersed in a constant temperature bath (discussed later), the temperature of which is known from a calibrated, high precision thermistor located in the bath in which the thermocouples are immersed. The thermocouples and thermistor are isolated
electrically from each other and the bath by a protective coating but are all suspended in the same area, reducing the possibility of temperature stratification. This referencing method provides more confidence in the temperature measurements made on both the air and refrigerant sides.

3.2.1.3 Zone Boxes

All of the thermocouple wires are connected to an electrically grounded and thermally insulated zone box, which consists of a number of electrical terminals with three inputs each. Figure 3.2 reveals the proper wiring for a zone box terminal being used to measure an absolute temperature. The positive (copper) lead from the thermocouple at the position where the temperature reading is desired is connected to the first terminal plug. The corresponding negative (constantan) lead is connected to plug two of the same terminal. The positive lead from the reference thermocouple is then connected to the third terminal plug. The negative lead is connected to the absolute thermocouple's negative wire at terminal plug two. The voltage difference across the two Cu leads is measured by the Fluke's internal scanner. This voltage corresponds to the temperature difference between the reference bath temperature and the temperature being measured. The Fluke channel to which this voltage is connected is configured as a voltage input channel, not a thermocouple one. All voltage to temperature conversions are accomplished outside of the data acquisition system. Since the temperature of the bath is known very precisely from the thermistor located in it, the desired absolute temperature can be determined using this value and the differential voltage between the reference and absolute thermocouples.

The same type of configuration is used to measure differential temperatures in the facility. The two thermocouples used to determine the temperature difference between two points are wired together in a zone box in the same way as the absolute and reference thermocouples are. The EMF generated between the two positive leads is used to calculate the temperature difference. Since one of the beads is located at the same point that an absolute thermocouple is located, the differential voltage can be used to determine the absolute temperature of the point where the other bead is located. The difference is coupled with the absolute reading at the one point to determine the temperature at the other. This is useful in determining downstream temperatures for various components in both loops.

The absolute temperature measuring method produces less error in the temperature readings. One reason is due to the fact that the error of a thermocouple
Figure 3.2: Zone Box Configuration
reading is in part proportional to the magnitude of the temperature reading being measured. Since the difference between the bath temperature and the absolute temperature being measured produces a much smaller EMF in the wires than that produced if only an absolute were used, the error generated from the reading is correspondingly less. Also, the inherent error incurred using a special limits of error thermocouples is ±5μV or ±0.1°C. This error is coupled with the error from the magnitude of the reading (as has just been discussed) making an accuracy of ±0.1°C impossible. The thermistor, on the other hand, does have an accuracy of ±0.1°C. It requires a power supply for its operation but the added accuracy and precision more than offsets the need for an extra piece of equipment.

The use of a differential measurement to determine absolute downstream temperatures also creates less of an error. If an absolute measurement were made at both positions, the corresponding error associated with the two measurements would be large because the EMF generated would be large. Subtracting the two to get a differential result would only amplify this error. However, the absolute value of a differential reading is small, reducing the error associated with it and propagating this reduction to the error involved in determining the absolute temperature.

The zone boxes are used to route not only the thermocouple outputs, but also the other sensor signals, to the Fluke where they can be recorded. Pressure transducer, flow meter, and other sensor outputs are connected to Fluke channels and scanned in the same way the thermocouples are. The zone box helps protect these signals from electrical interference, ensuring the maximum possible accuracy from them.

3.2.1.4 Pressure Transducers

Pressure transducers are also used on both the refrigerant and air sides. Specifications for the transducers implemented on the stand are given in Table A.10 in Appendix A. The position of each sensor in the facility is also given in the table. Absolute and differential pressure measurements are made at various points in the air loop. The method for determining any absolute downstream pressure, using the upstream and differential pressure across the component, is used in much the same way as with the thermocouples. Consequently, this reduces the error associated with such measurements.

Each pressure transducer is accompanied by a calibration curve and certification. This correlation relates the output voltage of the transducer with the pressure the sensor is measuring. However, the transducer can fall out of calibration during shipping. In
order to eliminate this as a source of inaccuracy, they were recalibrated in our lab. A
deadweight calibration mechanism was used for this purpose. The transducer is
connected to the machine and then the machine is charged with an inert gas (usually
nitrogen). The pressure of the charge is obtained using a mechanism which
counteracts the gas pressure with various masses, the weights of which are precisely
known. The output from the transducer and the pressure required to generate this
output are recorded and the process is repeated over the range of pressures
appropriate for the transducer. A calibration curve, correlating pressure and transducer
output voltage is obtained from this data, and subsequently used to obtain pressure
data once the transducer has been installed in the test facility. This process must be
repeated at regular intervals (every 6 months to 1 year) since the transducers tend to
fall out of calibration with extended use.

3.2.2 Air Side Component Design and Description

This document focuses mainly on the design and construction of the refrigerant
side of a condenser test facility. However, some mention must be made as to the
design and construction of the air side components. A detailed listing of each
component used in the air loop is given in Appendix A.

Figure 3.3 shows the ductwork assembly used in the test facility. The basic design
was derived from the literature (Gavin, 1983). The air flows into a 5in. diameter circular
duct as it exits the fan. It then passes over the precooler and heater, located inside the
5in. duct. Next, it flows through a converging-diverging (C-D) nozzle to reduce the any
temperature stratification created during the previous section. The air then expands as
it flows into the plenum chamber and continues on in the rectangular ductwork section
of the air loop. At the exit of the plenum, the air is sent to the test section through a
contraction in the ductwork from four feet to two feet in width. After passing over the
condenser tube, the air stream enters another plenum chamber and is abruptly
contracted into a section of 3in. PVC pipe. The PVC pipe feed the air into the venturi
flow meter. Upon exiting the venturi, the air stream again enters the fan and traverses
the loop again.

3.2.2.1 Air Side Data Acquisition

An accurate estimation of the air side heat transfer is required to determine
condensation heat transfer coefficients in a refrigerant tube. Measurement of different
Chilled water in

Chilled water out

Air heater

C-D Nozzle

Plenum chamber

Contraction section

Flow conditioning

Test section

48/4

24/1.5

Venturi flow meter

Figure 3.3: Ductwork Assembly
variables which can be used to evaluate air properties is required to achieve this end. These variables are obtained at various positions around the stand, according to the position at which the information desired. Pressure measurements, both absolute and differential, are one group of readings required. An absolute static pressure reading can be made upstream of the test section and used to determine properties of the air stream at that position. Another static pressure tap is located at the entrance to the venturi. This reading is also used to help determine properties which will be used in the calibration curve for the venturi. Differential pressure measurements can be made across the test section to determine the air side pressure drop caused by the condenser tube. A second differential tap is located across the venturi. This measurement can be input into the venturi correlation curve to determine the mass flow rate of the air and also can provide feedback for the control loop which regulates this flow (discussed later).

Temperature measurements are also required to determine the amount of heat transferred on the air side. Three type T thermocouples are used to determine the absolute temperature of the air in the 3in. PVC pipe at the entrance to the venturi. The average of these values is then used to determine properties of the air entering the venturi. These properties are needed for calculating the mass flow rate of air from the pressure differential across the venturi. For the test section, a more extensive measurement grid is required to ascertain air temperatures. A schematic of the thermocouple grid used is shown in Figure 3.4. Since the air temperature is essentially uniform upstream of the test section, only five thermocouples are averaged in order to determine the temperature of the air before it passes over the test section. However, as discussed previously, the air downstream of the condenser tube exhibits a temperature gradient due to the fact that the refrigerant in the tube cools down as it flows down the length of the tube. This occurs if the refrigerant entering the test section is single-phase, condensing (if superheated) or cooling further (if subcooled). It can also occur if the refrigerant enters as a two-phase substance and exits as a subcooled liquid. In order to circumvent this problem, the tube is essentially divided into sections, one for each downstream thermocouple. Heat transfers are then calculated for each section and combined utilizing an area weighted averaging scheme which accounts for this temperature profile. Finally, one of the thermocouples is used as the process variable for the air heater process controller. This thermocouple was chosen as the center point upstream of the test section.
Figure 3.4: Air Thermocouple Grid

LEGEND

A: Absolute T/C

Aref: Reference T/C in PRT bath

U: Upstream T/C for ΔT across TS

D: Downstream T/C for ΔT across TS

Pu: Upstream T/C for ΔT across TS and back duct

Pd: Downstream T/C for ΔT across TS and back duct
3.2.2.2 Temperature Control

One important aspect of the air side operation is the control of the air stream's temperature and thermal uniformity. The temperature of the air stream is controlled by a precooler and heater acting against each other. At steady-state, these two sources add and remove heat in such a way as to hold the air temperature at some constant, preset value. The precooler was designed to remove at least the amount of heat that the test section will input to the air stream. Full condenser experiments conducted previously and detailed in the literature (Struss, et. al., 1989) reveal a total heat transfer capacity of about 9kW for a full condenser. This corresponds to about 0.3kW of capacity per condenser tube. Since this target value represents an average operating condition for a full condenser, a precooler with a cooling capability of about 1kW was chosen in order to make sure the capacity was large enough to cool the air stream down to temperatures below ambient (a design requirement previously set). This was accomplished by customizing a GE spine-fin evaporator to fit directly inside the 5in. circular ductwork at the exit of the blower. The amount of cooling is regulated by controlling the amount of cooling fluid (ethylene glycol at a constant temperature of approximately 55°F) that flows through the heat exchanger. The evaporator fills most of the space inside the duct, which forces most of the air stream to come in direct contact with the heat exchanger. This helps to prevent the formation of any undesirable temperature stratification in the air stream as the cooling load is applied.

In order to create a constant air stream temperature at steady-state conditions, a heater is required to counteract the precooler's removal of heat. The air side heater was chosen with the requirement that it should be able to at least replace the amount of heat removed by the precooler. Since the test section is a condenser tube, it will add heat to the air stream. This will reduce the heater capacity needed to exactly offset the effect of the precooler. However, to ensure that the maximum number of testing conditions can be achieved and that the stand's range of operation is as broad as possible, a 1kW heater was chosen. In order to simplify its implementation, the standard 120VAC input power was used. This input called for a heater with a resistance of about 14.5 Ohms in order to generate the desired 1kW of heat. We chose a 15 Ohm heater and placed it in series with the precooler in the circular section of duct. Its power output is controlled using a Powers Process Controller. The controller uses a proportional-integral-derivative (PID) control algorithm to stabilize the air temperature at the setpoint. With the flow rate of the working fluid in the precooler held constant, the controller uses a thermocouple input from the air on the upstream side of the test section as its process
variable input. A 0 or 5V control signal is then output from the controller for a certain fraction of a time window set by the operator. This fraction of time corresponds to the percent output calculated by the controller utilizing the PID constants generated by the controller during its tuning period. For example, if the time window is 10 seconds and the output calculated by the controller to correct the process variable is 60%, a 5V signal will be output for 6 seconds and a 0V signal will be output for 4 seconds. This signal is sent to a solid state relay (SSR), through which the power line to the heater passes. A 5V signal will activate the control SSR whereas a 0V signal will not activate the SSR and subsequently the heater will receive no electrical power. Two other SSR's are used as alarm relays. These are activated by a 120VAC signal from the wall which is routed through the air heater process controller. As long as the air flow rate is not too low and the air temperature has not exceeded a preselected value, the controller allows the 120VAC signal to pass through it and activate the relays. If the limits of flow rate and temperature are out of range, the controller enters alarm condition, preventing the alarm relays from being activated. If all three of the SSR's are not activated, no power will be sent to the heater. This control scheme effectively causes some percent of the maximum heater output to be supplied to the air stream. At steady-state, this percentage of total heater power is some constant determined by the controller. The same scheme is also used to control other heaters implemented in the test facility.

The heater spans the diameter of the duct but does not heat the air uniformly in the radial direction. This creates the need for some mechanism by which the air can be mixed before it reaches the test section. At the exit of the fan, the air stream moves through a converging-diverging (C-D) nozzle in the 5in. diameter circular duct, serving to mix the stream and help to reduce thermal gradients present in it. Also, a plenum is used after the precooler and heater in order to adequately mix the air stream. The duct after the C-D nozzle, which feeds the air into it, abruptly expands into the large space tended by the plenum chamber. This large space offers adequate room for the air stream to swirl and mix as it encounters the bottom of the chamber, further reducing any thermal gradients which may be formed during its period of heating and cooling in the circular duct.

The galvanized rectangular ductwork, which is bounded by and includes the two plenum chambers, is also insulated to prevent any appreciable heat loss from the walls of the steel. The insulation was chosen with the requirement that the heat loss around the test section should be less than 1% of the minimum heat which could be transferred from the condenser tube. This constraint called for 6in. of insulation in the area where temperature measurements were being made around the tube. The insulation is used
to constrained the accuracy of the heat transfer measurements within the required limit set in the design criteria. Less insulation was used between the upstream plenum and the measurement points upstream of the test section because heat loss in these areas does not appreciably affect any heat transfer measurements made.

3.2.2.3 Flow Conditioning

The flow of the air in the ductwork must be conditioned in order to generate repeatable data and to prevent any other effects, such as boundary layer development or non-uniform air velocities, from influencing the results obtained during testing. Flow conditioning components are used to provide a constant velocity profile along the length of the test section. They also attempt to reduce any boundary layers that are formed as the air flows through straight sections of duct. Eliminating these effects ensures that the heat transfer performance of each section of the condenser tube is affected only by the characteristics of the tube itself, not the air passing over it.

The flow conditioning components which we used include (a) screens, (b) straws, and (c) contractions. Screens were installed in the straight section of duct which precedes the test section. These screens were sized according to the requirement that the pressure drop across them should be ten times that of the duct preceding the screen. They serve as obstacles, slowing the air flow down and allowing it to seep through, creating a constant velocity regime downstream of the screen. Any variations in the air velocity previous to the it are destroyed as the air passes through the reduced flow area provided by the screen.

Contractions are used to destroy boundary layers which develop during the flow of the air through flat sections of ductwork. The boundary layers can reach noticeable thicknesses and can greatly affect the heat transfer capacity of the air and tube over which it passes. Contraction sections destroy their enlargement by constricting the flow directly into the developing boundary layer. A smooth duct contraction is used before the velocity and streamline straightening sections of the ductwork, conditioning the flow upstream of the test section. There is also a contraction of the air directly before the test section, utilizing rigid insulation to feed the air stream into the fins located on the top and bottom of the condenser tube. This measure safeguards against any boundary layer outgrowth as the air exchanges heat with the test section.

Finally, straws are used to straighten the flow of the air stream before it enters the test section. Contractions, sudden or smooth, and obstacles cause ripples and bends in the streamlines of the air stream. This occurrence can affect the heat transfer
capabilities of the air, creating another effect which is unaccounted for when heat transfer coefficients are calculated. The straws force the air to flow through rigid passageways effectively destroying any bends in the streamlines. As the air exits, the streamlines are all parallel in the axial direction of the straws.

3.2.2.4 Air Mass Flow Rate

Control of the air mass flow rate must also be maintained if accurate, repeatable results are to be generated. A venturi and a fan are utilized to produce a constant flow rate of air over the test section. As explained earlier, the venturi is governed by a correlation between the mass flow rate of the air running through it and the pressure drop between the entrance and the throat. A differential pressure transducer is used to determine this pressure difference. The voltage obtained from the transducer is proportional to the actual pressure according to a relation determined by calibration of the sensor. This pressure is then used along with the calibration curve for the venturi, established by the manufacturer, to obtain the mass flow rate of the air. The accuracy of the mass flow rate depends upon the accuracy of the pressure measurement and the accuracy of the venturi calibration. We could not calibrate the venturi in the laboratory so venturi equations derived from the literature had to be used. These correlations take into account the type of venturi being used and provide a means by which the air mass flow rate can be determined from the differential pressure readings.

A blower and motor combination is utilized to drive the air through the ductwork at a constant flow rate. This constant speed is monitored using the venturi. The voltage output from the differential pressure transducer across the venturi is used as the process variable input to another Powers Process Controller, since it is proportional to the actual mass flow rate of the air. The process controller utilizes PID control to keep the voltage output from the transducer (and thus, the air flow rate) at some constant setpoint. It accomplishes this task by continuously varying the voltage signal sent to the motor controller. This will in turn vary the speed of the blower shaft in such a way as to keep the air mass flow rate constant. The output range of the transducer is the same as the output range of the controller signal (or, input range of the motor controller), allowing the setpoint chosen to correspond directly to the voltage output from the differential pressure transducer. However, if a specific mass flow rate is desired, the process of converting the transducer output voltage to a mass flow rate can be reversed and used to determine the differential pressure (and hence, transducer output voltage) required to achieve that specific flow rate. We used the former method when taking data,
determining the resulting mass flow rate from the transducer output after data acquisition was completed.

The fan (blower and motor combination) was sized to produce an air volumetric flow rate which simulates actual driving conditions seen by a car air conditioning condenser (as were given in Table 3.1). This prerequisite dictated that the blower must be able to produce a flow rate of approximately 70cfm. However, this test facility is to be used to determine refrigerant side heat transfer coefficients. Accordingly, the air side heat transfer resistance should be kept as low as possible in order to increase the magnitude of the refrigerant side resistance. Since the refrigerant side heat transfer coefficient is proportional to the refrigerant side resistance, increasing this magnitude of this resistance will increase the accuracy of the coefficients estimated. Therefore, the fan called for must produce at least a maximum of 150cfm, providing for some factor of safety. This value was used to determine that maximum pressure drop in the system which must be reversed by the fan would be about 3in. of water. These specifications and the maximum operating temperature of the air stream were then used to size the fan. The fan chosen delivers 415cfm at 3450rpm and a pressure drop of 3in. of water.

3.2.3 Refrigerant Side Component Design and Description

The bulk of this report deals with the design and construction of the refrigerant side of a single tube condenser test facility. The refrigerant loop incorporates many different constituents. Appendix A contains a list of all of the components implemented in the loop. Some of the components are based upon concepts which have not yet been tested and the implementation of which was to be evaluated during the course of this project. However, the stand must still adhere to the criteria set forth before construction begins. The facility was constructed in such a way as to fulfill as many of these requirements as possible. Table 3.2 shows the present capabilities of the stand. A comparison of Table 3.2 and Table 3.1 reveals how the test facility meets or exceeds almost all of the required operating capabilities set forth during the design stage.
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Stand Capability</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant Inlet Pressure</td>
<td>270 psi 1.85 MPa</td>
</tr>
<tr>
<td>Condensing Temperature</td>
<td>140°F 60°C</td>
</tr>
<tr>
<td>Refrigerant Inlet Condition (superheat or quality)</td>
<td>40°F(10°C) subcool to 9 °F (5 °C) superheat</td>
</tr>
<tr>
<td>Refrigerant Mass Flow Rate</td>
<td>0.265-1.455 lbm/min 15.88-87.30 lbm/hr 0.002-.011 kg/s 0.12-0.66 kg/min</td>
</tr>
<tr>
<td>Refrigerant Pressure Drop</td>
<td>4.35 psi 30 kPa</td>
</tr>
<tr>
<td>Air Inlet Condition</td>
<td>115°F 45°C</td>
</tr>
<tr>
<td>Air Velocity</td>
<td>6928 ft/min 115.5 ft/s 2110 m/min 35.2 m/s</td>
</tr>
<tr>
<td>Face Area (of test section)</td>
<td>0.0599 ft² 0.00557 m²</td>
</tr>
<tr>
<td>Air Volumetric Flow Rate</td>
<td>415 ft³/min 6.92 ft³/s 11.75 m³/min 0.196 m³/s</td>
</tr>
<tr>
<td>Air Mass Flow Rate</td>
<td>25.88 lbm/min 0.431 lbm/s 11.75 kg/min 0.196 kg/s</td>
</tr>
</tbody>
</table>

In some cases, the values given in the table represent the maximum possible value for each parameter. The face area of the test section represents the configuration for the microchannel tubes to be tested in this project. The maximum possible face area is close to the cross sectional area of the test section duct but all of this area is not required for the microchannel tubes. Also, the volumetric flow rate for the air is rated at a loop pressure drop of 3 inches of water. The actual pressure drop in the loop is about...
5in. of water, reducing the velocity of air seen at the test section when compared to the rated value.

3.2.3.1 Refrigerant Loop Construction

The foundation for the refrigerant side of the test facility is the refrigerant loop itself. The refrigerant lines were made from 1/4in. copper tubing, brazed together when possible and joined with compression fittings when brazing was not an option. The amount of tubing used was kept to a minimum in order to reduce the amount of refrigerant charge required. Copper tubing was chosen since it is more rigid than hose or aluminum tubing are and offers better structural integrity. Also, aluminum tubing is much more difficult (and sometimes harmful) to braze than copper tubing. Brazing permanently joins the copper tubing. However, if a section of tubing was to be removed at some point in time (such as the need to remove a test section and replace it with some other tube), compression fittings were used. These keep their mechanical integrity and sealing characteristics even after being subjected to multiple tightening and loosening hysteresis loops. Connective fittings are also much easier to obtain for copper tubing, reducing the difficulty of finding specific joints and reducing the cost of the fittings. Copper tubing is widely used in the refrigeration industry, implemented in such products as air conditioners and refrigerators. The loop construction resulted in a total refrigerant charge of about one pound. This includes the test section, PRT, EST, after condenser, and all the copper line connecting each of these components.

Valves used in the refrigerant loop were brass with compression fittings on either end to allow removal should they malfunction. The valves were chosen according to the packing used in them and their maximum pressure rating. The refrigerant being used for the microchannel tube tests in this project is R134a. This refrigerant is corrosive to many polymer packing materials but has virtually no effect upon Buna-N and in some cases, Teflon™ (PTFE). We used refrigerant valves packed with PTFE and have had no detectable leaks occur from them. The valves are rated to pressures up to 500psi. The maximum condensing pressure required from the design criteria stated previously is 400psi. Therefore, the valves will be able to withstand this operating pressure should it be necessary.

A precautionary measure which was built into the refrigerant loop is a pressure relief valve (PRV). This preventative device will allow refrigerant to be vented to the atmosphere if the pressure in the loop exceeds some value set by the PRV's manufacturer. However, our loop uses a 425psi PRV connected to an evacuated
receptacle cylinder which will capture the refrigerant should it ever be released by the PRV. This measure insures that no refrigerant will escape into the atmosphere even if the valve blows.

The loop was also insulated against heat loss using 1/2in. thick polyethylene pipe insulation. This insulation is suited for 1/4in. copper refrigerant tubing. The insulation was chosen mainly prevent some heat loss from occurring but is not required for the operation of the stand.

3.2.3.2 Refrigerant Side Data Acquisition

As on the air side, many measurements are required on the refrigerant side to determine refrigerant side heat transfer coefficients. Many readings are also used to monitor the operation of the facility as warm-up or testing is being conducted. These measurements can reveal whether or not the stand is operating properly and will generate useful data. Thirdly, some refrigerant side measurements are used as process variables for the control of the individual components in the refrigerant loop.

Pressures are acquired at many areas around the refrigerant loop. The measurement of pressures in the refrigerant line requires that a tap be fashioned at the point of interest. A proper tap requires that a hole be made perpendicular to the direction of the refrigerant flow and that the hole be infinitely small (Benedict, 1977). These two prerequisites serve to prevent any component of the refrigerant velocity from forming in the tube connecting the transducer to the refrigerant stream. If these two requirements are not met, the reading obtained will be some fraction of the stagnation pressure of the refrigerant in the tube (static plus dynamic pressures). In determining properties, only the static pressure is used. Using stagnation pressure to obtain properties will cause an over estimation of the property being determined. Unfortunately, the hole drilled in the refrigerant line for the pressure tap cannot be made infinitely small. The effect of a finite hole size decreases rapidly as the hole size is reduced and a reasonable approximation can be determined according to the size of the tube through which the fluid is flowing (Benedict, 1977). This allows for a manageable drill bit size to be used.

Figure 3.5 shows a schematic for the refrigerant side pressure taps. Holes 1/32in. in diameter are drilled in the wall of the refrigerant tubing. A short, 1/8in. inner diameter (ID) tube is brazed to the outer wall of the 1/4in. refrigerant tubing. This is used a guide tube for the 1/8in. outer diameter (OD) pressure tap tube. The 1/32in. hole is then drilled in the refrigerant tube wall using the center of the guide tube to direct the
Figure 3.5: Refrigerant Side Pressure and Temperature Taps
placement of the hole. The 1/8in. OD tube is then brazed to the 1/8in. ID guide tube and the other end is connected to the pressure transducer. Care must be taken to avoid and remove any burrs on the inside of the refrigerant tubing produced by the drill bit. This will greatly affect the pressure reading obtained from the tap and diminish its accuracy.

An absolute refrigerant pressure measurement tap is made upstream of the test section. The tap is located in the copper tubing which is connected to the condenser tube using the technique discussed above. This reading will help provide the state of the refrigerant before it enters the test section. Absolute readings are also taken before the pump and the enthalpy setting tank (EST). These readings are used more for monitoring than for property determination. They provide information as to whether or not the stand is operating properly. A differential reading is taken across the test section in order to provide single tube pressure drop information. This value is also coupled with the upstream measurement to reveal the absolute downstream pressure. This value is used in determining downstream refrigerant properties. They are also located in the copper tubing before and after the test section since it was not feasible to install taps in the condenser tube itself. However, their location does not provide a completely accurate determination of the tube's pressure drop since there exists a finite length of copper pipe and a connective fitting which contribute to this pressure drop.

Another value used to determine the state of the refrigerant at various positions in the loop is the temperature. Thermocouple readings on the refrigerant side are used in conjunction with pressure values for the purpose of property evaluation. The thermocouples used in the refrigerant loop are ungrounded, sheathed probes. Figure 3.5 also shows a tee made in the copper tubing with the top outletting to an 1/8in. female NPT fitting. The threads of an 1/8in. NPT to 1/16in. compression fitting are covered with refrigerant safe epoxy and then screwed into the copper fitting at the top of the tee. The thermocouple probe can then be inserted into the compression fitting until the tip reaches the middle of the copper tubing. The compression nut is tightened and seals around the probe preventing any leakage. Special limits of error thermocouple connecting wire is used to connect the thermocouple to the proper terminal in a zone box. The probes used are ungrounded because the ground is the stainless steel sheath surrounding the thermocouple wire. This sheath could pick up stray interference from the refrigerant stream and would therefore not offer a proper ground.

An absolute temperature tap was installed upstream of the test section using the technique described above. This location is not the most ideal since the refrigerant must travel through a section of copper tubing and a connective fitting before entering
the test section, possibly losing heat along the way. However, this was the closest place the tap could be made with the current configuration. An estimation of the possible heat loss from this area showed that the heat loss was less than 1/2% of the heat lost to the air stream in the test section. Absolute temperature measurements are also made before the pump and EST using the thermocouple probes. These readings are also used for monitoring the stand's operation.

3.2.3.3 Flow Measurement and Control

The refrigerant mass flow rate is also required to verify the heat exchanger energy balance and produce heat transfer coefficients. The installation of a sensor for the measurement of this variable did not involve any modifications to the refrigerant line as the pressure and temperature sensors do. A Micro-Motion Flow Meter and Transmitter are used to determine the flow rate of the refrigerant. The flow meter's transmitter outputs a 4-20mA signal which is converted to a voltage using a high precision resistor. This voltage is then read by the Fluke Data Acquisition System using a channel on one of the voltage scanner boards. The meter is supplied with a calibration certificate which includes the calibration curve, showing the relationship between the signal output from the transmitter and the actual flow rate of the fluid being tested. The accuracy of the meter is higher than that of many other flow measuring devices and is easily installed into any refrigerant loop, using compression fittings. The meter must be sized according to the (a) maximum pressure in the refrigerant line, (b) maximum mass flow rate expected, (c) maximum temperature of the refrigerant line, and (d) the type of refrigerant to be used. The model we chose safely operates at the maximum values for all of these parameters.

The flow meter is also used to control the flow rate of the refrigerant. In order to gain a steady-state condition on the refrigerant side, the mass flow rate of the refrigerant must be held at some constant, preset value. This value is reached and maintained using a control loop which involves a pump, process controller, and the flow meter. The pump head and motor controller were both sized according to an estimation of the total pressure drop in the refrigerant loop and the maximum desired flow rate at this pressure drop. Conditions of all liquid and all vapor refrigerant flowing through the loop were investigated in order to determine the maximum pressure drop (all vapor) and various expected operating pressure drops in the loop. This information was used, along with flow rate constraints, to properly size the refrigerant pump.
The pump is regulated using a motor controller device. The controller allows manual manipulation of the pump motor by means of (a) a potentiometer or (b) remote regulation from an external signal. We couple the motor controller, set in remote signal operation mode, and a process controller, utilizing PID control, in order to set the refrigerant mass flow rate. This method provides a much more precise and stable regulation of the refrigerant mass flow rate than the potentiometer. The control loop for this mechanism employs the output signal from the mass flow meter in much the same way the air flow rate control loop uses the output from the pressure transducer across the venturi. The meter signal is input to the process controller as its process variable. Hence, either the meter signal can be directly selected or the desired mass flow rate can be chosen and converted to the corresponding meter signal via the correlation given by the meter manufacturer. Once this setpoint is chosen, the process controller will regulate the signal sent to the motor controller, thereby providing a constant refrigerant mass flow rate. During data acquisition, the signal from the mass flow meter is scanned by the Fluke and tallied along with the other information (such as temperatures and pressures) important to the researcher.

3.2.3.4 Test Section

This test facility was constructed primarily to test the microchannel tubes discussed previously. Heat transfer performance data is to be collected in order to determine the effect that various design parameters have upon each tube's heat transfer capacity. During testing, different variables, such as refrigerant inlet condition and condensing pressure, will be varied and their effect on the condenser tube capacity will be evaluated. Since the inlet port geometry and tube width are two of the variables which are to be varied in order to determine their effect on the heat transfer performance of the tube, various size fittings for installing the test section in the refrigerant loop must be obtained. The removal of an old tube and installation of a new one must require as little time and effort as possible while maintaining the integrity of the tube and the accuracy of the data obtained. Also, the flow of refrigerant into the tube must be well conditioned in order to reduce any effects caused by variables other than the one being examined in a particular test. If such interactions occur, the data generated will not be representative of what effect the variable actual has on the tube's performance. They will also reduce the repeatability of such experiments, diminishing the credibility of the data. Finally, the release of any refrigerant into the atmosphere, with the removal or installation of a tube, should not be tolerated. During operation, the entire stand, including the test section,
should remain completely sealed in order to provide a testing environment that will not release possibly harmful substances (refrigerant) into the surroundings.

Figure 3.6 is a schematic of the microchannel tube test section to be used in this project. The test section consists of the microchannel tube and louvered fins bonded to both sides of the tube. The tube itself is 25in. long, with fins covering the middle 23in. The test section spans the width of the wind tunnel ductwork, and is only about 0.075 in. high. Air is allowed to flow only over the fin area due to the placement of the polyurethane insulation. The complete apparatus involves the microchannel tube test section, transition sections, and surrounding ductwork.

Louvered fins are bonded to both sides in order to enhance the heat transfer capability of the tube. This is accomplished by either (a) brazing, using a corrosive resistant aluminum flux, or by (b) using a specialized epoxy which is thermally conductive. The epoxy contains an appreciable amount of aluminum which reduces its resistance to heat transfer. Contact resistance becomes a significant amount of the overall heat transfer resistance if the fins are not in good contact with the tube wall. Brazing provides a smaller contact resistance than the epoxy since the braze material itself is a conductive metal. However, it is more difficult to control the brazing process, creating a problem if the braze alloy does not cover the entire connection surface between the tube and the fins. The epoxy can be spread over the length of fin-tube contact area to ensure a sound joint. Also, brazing requires a furnace and a braze alloy which is applicable to AL-AL joints. Due to the low melting temperature of aluminum, brazing is never a simple task. The epoxy allows for quick and easy fabrication and still provides a low enough contact resistance to keep the total air side resistance low. The tube installed in the stand for this project will have fins epoxied to both sides of the microchannel tube.

The louvers on the fins are used to reduce the air side heat transfer resistance of the test section. These fin supplements increase the area of fin material in contact with the air stream and also tend to create a more turbulent flow regime around the fins, thereby increasing their heat transfer capability. Since the purpose of the facility is to provide a means by which refrigerant side heat transfer coefficients may be obtained, the resistance on the air side must be kept as low as possible in order to increase the accuracy of these values. Louvered fins aid in attaining this goal.
Figure 3.6: Sub-Assembly Test Section
A difficult task in installing the microchannel tube is the connection between the 1/4in. diameter copper tubing and the aluminum tube. There are no existing stock fittings which will connect a round refrigerant tube to a microchannel tube without the use of a header involving a permanent braze joint. Such a method requires a large amount of space for the header and also creates a permanent joint between the loop and the test section, increasing the difficulty associated with changing condenser tubes. Also, brazing copper and aluminum together is a very difficult task due to the lack of commercially available braze alloys which provide a sound joint and the different heat capacities of the two metals (this effects the flow of the braze material). More importantly, it does not allow for any flow conditioning, such as boundary layer destruction, which would help to produce repeatable and reliable data. In order to avoid such obstacles, a transition section was fabricated. The transition section permits the interchange of different test sections as long as the width and height of each is kept relatively constant. It also conditions the flow by providing a rapid contraction which destroys any boundary layer development in the refrigerant stream before it enters the test section. Thirdly, it forms a smooth transition by which the refrigerant can flow from the copper tubing into the heat exchanger tube, reducing entrance effects associated with rapid, uncontrolled contractions.

The transition section is composed of three sections: the (a) expansion section, the (b) contraction section, and (c) the O-ring plate. Figure 3.7 shows these components. The refrigerant enters a 7/8in. copper pipe expansion chamber from the 1/4in. tube before entering the transition section. This section is used to mix the refrigerant before it enters the test section, eliminating any thermal gradients which might have developed in the refrigerant tubing. The expansion chamber is screwed into the expansion section shown in the figure. A screen is placed between the expansion and contraction sections. This aids in slowing the flow in the copper expansion chamber, allowing adequate mixing to occur. The contraction section is used to destroy any boundary layer development through the use of a sharp contraction in the refrigerant line before it reaches the test section. It also physically connects the microchannel tube to the rest of the refrigerant loop. The third component of the transition section is the O-ring plate. This piece is used to mechanically secure the tube to the contraction section and, subsequently, the rest of the refrigerant loop. It also serves to seal the joint between the contraction piece and the O-ring plate itself. For a detailed dimensioned drawing of the transition section, the reader is directed to Figure A.1 in Appendix A.
Figure 3.7: Refrigerant Side Transition Section
Care taken when joining the components of the transition section will prevent leakage of refrigerant from the test section. The operation of installing the test section into the stand must be carried out according to a specific order of steps. First, the O-ring plate is slid onto the tube with the O-ring groove facing outward, followed by the O-ring. In order to prevent any leakage from around the tube, the O-ring groove in the plate and the O-ring itself must be kept clean and debris-free. In the contraction section, there exists a step which prevents the tube from entering the contraction chamber. The tube is inserted into the contraction section until it becomes flush with the step at the end of the tube channel. This step is exactly the height of tube wall thickness, forming a passage for the refrigerant to flow smoothly into the test section without encountering any abrupt transitions. The screws joining the O-ring plate to the contraction should be tightened as much as possible in order to fully compress the O-ring. Finally, the expansion piece is fitted with its O-ring and screen, and then secured to the contraction piece by screws in the flange area of each of the two components. This compresses the O-ring in between the two flanges, sealing the joint from leakage. Some sort of threadtightener is used to prevent the screws from vibrating loose while testing. Vibrations in the facility can cause the screws to become loose and may end up weakening the seal around the tube.

3.2.3.6 Pressure Regulating Tank Scheme

Refrigerant flow and condensing pressure in a commercial air conditioner are controlled through the use of a compressor. However, a compressor system cycles and is therefore subject to transient conditions. Since we want to perform steady-state tests, this option is not applicable in our situation. Instead, we use a pump to control the flow of refrigerant and a Pressure Regulating Tank (PRT) to control the system pressure. Figure 3.8 is a schematic of this component. This component is also used as the constant temperature bath in which the reference thermocouples for each absolute temperature reading are located (as discussed previously).

A one liter vessel is immersed in a 15 gallon stainless steel drum filled with propylene glycol. The one liter size was chosen in order to allow the refrigerant loop to be filled from the supply in the vessel without completely emptying the vessel. It also had to be large enough to allow vaporization of refrigerant in the loop, which forces refrigerant into the 1L vessel without completely filling it. Finally, the size was made as small as possible to achieve these two criteria while keeping the total charge in the loop as low as possible.
Figure 3.8: Pressure Regulating Tank
The vessel immersed in the glycol bath is filled about half way with refrigerant. In order for the PRT to function correctly, the vessel must always contain two-phase refrigerant. A sight-glass which spans the height of the one liter vessel is used to continuously observe the level in it. This ensures that the refrigerant is, in fact, two-phase. To pressurize the loop, the temperature of the glycol bath is increased. This subsequently increases the temperature of the refrigerant in the vessel and, since it is a two-phase substance, increases the pressure as well. The process forces liquid refrigerant out the bottom of the vessel, reducing the liquid level in the vessel. Since more liquid is forced into the loop, the pressure is increased in the loop as well. Further increase in the glycol bath temperature will result in further increase in the pressure of the loop. Once the desired pressure has been reached, the bath (and, consequently, the refrigerant in the vessel) is kept at a constant temperature. This maintains the loop pressure at the desired value regardless of what is occurring in the it. Any changes in the refrigerant temperature or quality are nullified because of the contact between the PRT and the loop.

The vessel is in contact with the refrigerant loop via the line at the bottom of it. A port, connected to the top of the vessel, lies outside the bath. This is where the vessel can be charged with refrigerant. At the point where the line from the bottom of the vessel feeds into the refrigerant loop (just before the gear pump), liquid refrigerant is present. The liquid stream in the loop is in contact with the liquid in the bottom half of the 1L vessel. This allows for the transfer of liquid refrigerant into and out of the vessel, thus holding the system pressure constant.

In order for the PRT to operate correctly, the glycol bath temperature must be kept at a constant, preset value. This is accomplished using a heater and cooling coil system, much the same as in the air stream. The cooling coil provides a cooling load on the bath in an attempt to reduce its temperature. This heat exchanger is supplied with chilled water at approximately 13°C (55°F). Heat input from the immersion heater counter balances the cooling load with a maximum output power of 2.5 kW. It was chosen by constraining the bath to reach a steady-state temperature within 15 minutes and then to maintain this temperature continuously, without large fluctuations. The heater output is controlled using a process controller. Since the thermal mass of the bath is so great, a PID control algorithm is not applicable to adequately control the bath temperature. Instead, an on/off control scheme is used, much like a thermostat. If the bath temperature is below the set point, the heater outputs full power. If the temperature is higher than the chosen set point, no power is provided by the heater. This adequately keeps the bath at a constant temperature. The controller uses a
thermistor immersed in the bath as its process variable. The desired loop pressure is achieved by determining the saturation temperature of the refrigerant in the 1L vessel, which coincides with the desired loop pressure. The thermistor output voltage which corresponds to this temperature is input as the set point for the process controller. This temperature is reached and maintained by the constant heating and cooling loads of the heater and cooling coil, effectively regulating the system pressure.

Since one of the goals of the PRT is to provide a constant temperature bath, effective circulation must occur within the bath to ensure a uniform temperature throughout. The system pressure will remain constant only as long as the temperature in the bath remains constant. In order to achieve this, a circulation pump was installed in the glycol bath. Vigorous pumping of the glycol eliminates vertical or radial temperature stratification.

3.2.3.7 Enthalpy Setting Tank Scheme

The inlet quality of the refrigerant entering a condenser is usually that of a superheated vapor. The condenser is designed to allow subcooled refrigerant to exit the condenser. Refrigerant properties are required in order to determine the heat transfer necessary to achieve this end. In a single tube test such as those which will be conducted using this test stand, the entrance or exit conditions of the refrigerant may be two-phase. At such a state, temperature and pressure are two properties which are no longer independent. Unfortunately, these two properties are the only two which we can measure accurately using conventional sensors. If the refrigerant exists as a two-phase substance when it enters the test section, some other property must be determined in order to completely define the state of the refrigerant. The enthalpy setting tank (EST) provides this information.

Figure 3.9 shows the EST. The system is composed of an inner and outer tank, both made of stainless steel. Initially, steel drums were used because they were much less expensive. Unfortunately, since water was used to fill the two tanks, they rusted at unbelievable rates. Even after being painted with a superior rust-resistant gloss, a 1 inch layer of iron developed at the bottom of the tanks, making circulation of the water impossible. Inhibited propylene glycol was used along with the stainless steel drums to provide a heating environment for the refrigerant coils. Dowfrost HD™ inhibited glycol provides greater resistance to rust formation than does plain tap water. This fluid (which is not considered poisonous) was mixed with water at a dilution ratio of 50% before being used to fill the inner and outer tanks of the EST and the PRT tank.
Figure 3.9: Enthalpy Setting Tank
The function of the EST is to help determine the state of the refrigerant before it enters the test section. This is accomplished through the use of power measurement. After the refrigerant leaves the pump and passes through the mass flow meter, it reaches the EST. If a two-phase or superheat refrigerant inlet quality condition is required at the test section, the refrigerant must be vaporized in the EST. The state of the refrigerant before it enters the EST is evaluated from a thermocouple and pressure transducer, since it is in liquid form. The fluid then enters the copper coil inside the inner tank where it absorbs heat from the glycol bath (which is at an elevated temperature) surrounding it. It exits the tank as two-phase or a superheated vapor, depending upon the amount of heat added by the bath. If this heat added to the refrigerant line is measured, the quality (and hence, the state) of the refrigerant at the exit of the EST can be determined. This state is the state of the stream entering the test section.

The inner and outer tank temperatures are controlled using immersion heaters. The inner tank heater was designed to heat the inner tank bath up to a temperature of 80°C (176°F) in about 15 minutes and hold the temperature there as the refrigerant flows through it. Four kilowatts of power is required to accomplish this, so a six kilowatt heater was chosen to ensure that all desired refrigerant inlet states could be reached. Next, the copper coil in the inner tank, through which the refrigerant flows, was sized. The total length of tube in the tank was chosen to allow total vaporization the subcooled refrigerant to a superheat level of 5°C at the highest refrigerant loop pressure. This required that 50 feet of copper tubing be immersed in the bath. Finally, a 1 kW heater is required in the outer tank. This capacity was attained by requiring that the bath be able to reach its maximum temperature in 15 minutes. The heater must also maintain this temperature assuming an ambient temperature of 20°C (68°F) and a bath temperature of 80°C (176°F). A 2 kW heater was chosen in order to ensure that these criteria can be met.

In order to determine the amount of heat added to the refrigerant line as it passes through the EST, all the heat input into the inner tank must be measured. This heat exchange includes (a) energy input from the heater, (b) heat given off by the circulation pumps, and (c) any exchange between the tank and the ambient atmosphere. The heater and pump powers can be measured with power measurement circuitry, but the ambient heat exchange is much more difficult to quantify. Because of this fact, some sort of insulation scheme must be utilized in order to eliminate any heat exchange between the inner tank and the surroundings. Insulation provides a means for accomplishing this but only to a certain accuracy; an infinite insulation thickness would
be required to completely prevent any heat exchange from occurring. Since the quantities of heat transfer expected from a single tube test are expected to be small, any inaccuracies in determining the refrigerant states will be propagated and amplified when these values are used to determine heat transfer coefficients.

Instead, a jacket tank is used in order to totally eliminate any heat exchange. The inner tank is placed inside the outer tank. The outer tank is also filled with an inhibited propylene glycol bath to level which completely submerges the sides of the inner tank. The tops of both the outer and inner tanks are covered with 8in. of fiberglass insulation to prevent any heat escape from the top of the tanks. The outer tank bath is then held at a temperature equal to that of the inner tank bath through the action of the outer tank heater. If the two baths are at the same temperature, the driving force for heat exchange is eliminated and no heat is lost from the inner tank to the surroundings. Once this situation is established, any energy input from the inner tank heater and circulation pumps must be absorbed by the refrigerant flowing through the coil in the inner tank.

The operation of the enthalpy setting tank depends on the assumption that the outer and inner tank temperatures are the same. In order to fulfill this requirement, a heater is implemented in the outer tank bath and controlled by a process controller. The heater controller receives the temperature difference between the outer and inner tanks as its input process variable. It then regulates the heater output to keep this temperature difference at zero. As discussed previously, PID control of such a large thermal mass is difficult, requiring on/off regulation of the heater's output power (much like the PRT heater control scheme). The inner tank heater is operated by a process controller under the control of the operator. An attempt was made to use a thermocouple in the inner tank bath as a process variable for the controller. This failed because the heater power is too high, making closed loop PID control very difficult due to possible large fluctuations in the process variable caused by the heater cycling on and off. Also, the thermal storage capacity of the tank is so large that the time required for any modification in the heater output to be recognized by the controller is large enough to create an unsteady system. Instead, the percentage output from the controller is set. This fixes the heater output to some predetermined percentage of its maximum. The bath eventually comes to an equilibrium temperature according to the amount of cooling provided by the refrigerant line and the amount of heating supplied by the pumps and heater. This then sets and maintains the temperature of the refrigerant entering the condenser tube (if the refrigerant is single-phase). This control scheme works much more efficient than the closed loop control method. It also suggests that
using the power measured from the heater and pump electrical inputs as the process variable for the inner tank heater controller might well provide the tightest control of the refrigerant's exit quality.

Currently, no power measurement system exists that furnishes the accuracy required for this project but fabrication of such a system is being conducted presently. Without the use of power measurement, no two-phase data can be obtained since the test section entrance state of the refrigerant could not be determined without it. The stand is currently operational for testing single-phase refrigerant inlet conditions.

3.2.3.8 Aftercondenser

As explained previously, the test facility utilizes a pump instead of a compressor to circulate the refrigerant charge through the refrigerant loop. In order to prevent possible damage to the gear pump, the refrigerant at the entrance to the pump must always exist in a subcooled state. The pump can heat up or cavitate, eventually leading to the destruction of the pump, if any vapor exists in the line feeding it. To prevent such malfunctions, an aftercondenser was installed in the refrigerant loop upstream of the pump. In order to size this heat exchanger, we used a worst case scenario for fully condensing the refrigerant before it enters the pump. This situation involves a refrigerant flow rate of 0.013 kg/s (= 100 lbm/hr) and a chilled water supply (an ethylene glycol mixture) at an approximate temperature of 13°C (55°F). A cooling capability of 4kW (13.7 MBtu/hr) is required to fully condense the refrigerant at the extreme case. This constraint led to the selection of an aftercondenser rated to provide a capacity of 8.2 MBtu/hr at a lower flow rate and smaller temperature difference between the refrigerant and cooling fluid. The actual conditions established during the stand's operation provide the means to increase the heat exchanger's capacity. It currently provides more subcooling (up to 40°C) than is necessary but will ensure the safe operation of the pump even with moderate fluctuations in the cooling water supply.
CHAPTER 4: EXPERIMENTAL PROCEDURE AND RESULTS

4.1 Introduction

The experimental facility discussed in the previous chapter was developed for the purpose of conducting heat exchange performance tests on various condenser tubes. Microchannel tubes are the first set of tubes which will be examined. The goal of performing these tests will be to determine the refrigerant side heat transfer coefficient of the tube. Many steps are involved in obtaining such data, including collection, reduction, and presentation. This chapter provides a brief overview of the procedures for collecting various types of data, and then gives methods for presenting the data in a useful, simple manner. The actual results obtained from such tests will be presented in a forthcoming ACRC document.

4.2 Data Collection

In order to determine the effect of various system characteristics on the heat transfer capabilities of a condenser tube, many different tests must be conducted. These tests require choosing which variable is to be investigated and then selecting the operational conditions of interest. For this project, condenser parameters which will be investigated include tube port geometry and width. Also, the effect of different refrigerant inlet conditions will be analyzed, including refrigerant inlet temperature and quality, and condensing pressure. Certain steps must be followed in order to acquire data which can reveal the effect that these parameters have on the condenser tube's performance. Appendix A details the use of the data acquisition device. Likewise, it contains the procedure for starting up the test facility for the purpose of taking data and shutting it down once data acquisition is complete. Therefore, following the procedures detailed in this chapter requires a working knowledge of the stand.

4.2.1 Subcooled Data

The first set of data taken once a new tube has been installed in the facility calls for subcooled refrigerant at the entrance to the test section. An actual condenser would never be used to reduce the temperature of subcooled refrigerant (refrigerant always enters the condenser two-phase or superheated) but this inlet condition is required in order to generate information which will aid in determining the tube's air side resistance (this will be discussed later in this report).
To obtain subcooled data, the test facility must first reach steady state. This includes setting the refrigerant and air mass flow rates. The refrigerant flow rates are chosen according to the researcher's range of interest. In choosing the mass flow rate of the air, the goal associated with the test must be taken into account. For subcooled data, it is imperative to reduce the air side resistance of the tube. This can be accomplished by using a high air mass flow rate. Since the total resistance is composed of both the air and refrigerant side resistances (which are functions of the air and refrigerant heat transfer coefficients), reducing the air side resistance will increase the percentage of the total resistance that the refrigerant side occupies. This will therefore increase the accuracy of the heat transfer coefficient determined from this resistance.

A scan group must be set up before any data can be taken. The scan group should consist of all of the sensor points which the operator requires for the specific test being conducted. For this project, these points include the temperature, pressure, and mass flow rates which are required to determine heat transfer coefficients for the refrigerant side. Setting up a scan group involves programming the Fluke data acquisition device to scan the sensor points of interest. The method by which this and other Fluke operations are accomplished is discussed in the Fluke references detailed in Appendix A. Once the scan group has been set up, the Fluke is programmed to continuously record the channels in the desired scan group twenty times. The data is then stored in a text file and saved on the computer's hard drive, where it can be accessed during data reduction. Average values for each parameter over the twenty scans are used for property determination. Averaging over time gives a more representative value for each parameter.

Taking data involves determining the refrigerant inlet temperature to be investigated and achieving this value under a steady-state operating condition. The loop pressure is chosen and maintained through the use of the PRT. Next, the EST must be used to set the desired temperature of the refrigerant stream entering the test section. Without the use of a power measurement system, the inner tank heater power must be manually set by the operator. The EST inlet and desired exit temperatures, and the refrigerant flow rate are used to calculate the heater power required to reach the correct inlet temperature. This output is then set on the heater controller and the EST is allowed to reach steady-state (approximately 15 minutes). If the temperature of the refrigerant entering the test section has not reached the desired value, the heater output is adjusted accordingly. Small changes in the inner tank heater power cause little change in the bath temperature, so setting the exit temperature is not an exact science.
Once the required temperature is reached and maintained, the inlet state can be defined and data may be taken. The refrigerant inlet subcooling is held constant throughout all subcooled data sets.

Subcooled data acquisition requires setting the air mass flow rate and varying the flow rate of the refrigerant throughout the data set. Data using the first refrigerant flow rate (which was set during warm up of the stand) is taken once the steady-state condition has been reached (after the temperature before the test section remains constant for about 30 minutes). Once this is accomplished, the refrigerant flow rate is increased to the next set point. Accordingly, the EST inner tank heater output is again calculated with the new flow rate and reset until the refrigerant inlet temperature is once again reached. Data is taken again and the process is repeated until the full range of refrigerant flow rates has been tested. For the microchannel tube testing, five refrigerant mass flow rates in the range of 0.005-.011 kg/s will be used.

Once data from all of the refrigerant flow rates has been obtained, the air mass flow rate is changed. A data set can be collected for each of a number of low and high air flow rates and used to determine whether each of the data sets follows an energy balance (discussed later in this report). Four different air flow rates will be investigated using the microchannel tubes. A data set for each air flow rate can be used to generate a Wilson plot and subsequently, a refrigerant side heat transfer coefficient (also explained later). Forty data points for each microchannel tube tested are required to fully investigate the parameters of interest. These consists of five refrigerant mass flow rates for each of four air velocities.

4.2.2 Superheat Data

Once all of the subcooled data has been taken for a particular tube, the refrigerant inlet quality is changed to a superheat condition. This condition is more representative of condenser operation since refrigerant is never subcooled as it enters the condenser in a actual car air conditioning system. The data taken following this procedure will be used to determine heat transfer coefficients for varying degrees of superheat.

The procedure for obtaining superheat data is much the same as that for subcooled data. The stand must reach steady-state before any data can be taken. Also, the condensing pressure must be set using the PRT. This pressure should be chosen as low as possible in order to reduce the amount of heat required to vaporize the refrigerant in the EST and allow a superheat inlet condition to be reached easily. Once this pressure has been selected and attained, the degree of superheat must be
set. This is accomplished by setting the EST heater to a level which will produce refrigerant vapor at the exit of the EST under the current condensing pressure. The amount of heater power required is determined by performing an energy balance on the refrigerant in the coils, using the inlet temperature, mass flow rate, and desired exit temperature of the refrigerant. This percentage is then set on the heater controller and the tank allowed to reach steady-state. Any modifications in the heater output power are then made to achieve the desired refrigerant vapor temperature.

Once conditions on both air and refrigerant sides are steady, data can be taken. The amount of superheat is then varied along a range set by the researcher following the same procedure. After all vapor inlet temperatures of interest have been tested, the refrigerant mass flow rate is incremented and the same inlet temperatures used with the last flow rate are used again with the new one. This process is repeated until desired five flow rates have been investigated. The flow rates chosen should be the same as those used for the subcooled data tests. Finally, the condensing pressure is changed and the process is repeated using the same inlet temperatures and flow rates as with the previous pressure setting.

The total number of data points required to provide all of the necessary two-phase heat transfer information on each microchannel tube is eighteen. This number is the result of two different refrigerant inlet temperatures, three refrigerant mass flow rates, and three condensing pressures. Each point represents a specific refrigerant inlet vapor condition, refrigerant mass flow rate, and condensing pressure. The results can then be used to calculate heat transfer performance characteristics for each tube.

4.2.3 Two-Phase Data

In some situations, an actual car air conditioning system can provide its condenser with two-phase refrigerant. For this reason, a two-phase refrigerant inlet condition will also be investigated in this project. The refrigerant enters the test section two-phase and condenses along the length of the tube. The amount of condensation depends upon the air side conditions and the refrigerant mass flow rate. Full condensation might not occur but is more likely with a two-phase inlet condition than with superheated refrigerant.

Currently, two-phase data can not be obtained because of the lack of a working power measurement system. Without it, the state of the refrigerant entering the test section can not be determined. Hence, a specific quality can not be achieved and
controlled throughout testing. Until the power measurement device is completed and installed, only single phase (subcooled and superheat) data can be taken.

Once power measurement is installed, the effect of refrigerant inlet quality on the performance of a microchannel tube can be investigated. First, the components of the loop must reach steady-state. The condensing pressure is chosen and achieved through the use of the PRT. Once this pressure is reached, the desired refrigerant inlet quality is chosen. The EST is once again used to reach this quality by adding the appropriate amount of heat to the refrigerant line in the inner tank. However, instead of determining the appropriate heater output and regulating it at this value using the heater's process controller, the output from the power measurement device will be used to control the refrigerant quality at the exit of the EST. The state of the refrigerant before and after the EST can be determined from pressure and temperature readings and knowledge of the desired exit quality. The difference between the two enthalpies is the power required to produce refrigerant with the specified quality at the exit. This power can be then used as a set point for the heater process controller. The power measurement device's output is used as the controller's process variable, helping to regulate the signal sent to the heater and controlling the power input to the inner tank to keep the exit quality constant. During data acquisition, this power is measured by the Fluke and stored with all of the other sensor points included in the scan group. The scan group should include all of the sensor points included for the other two types of tests.

After the stand reaches steady-state, the two-phase data can be collected. This process is similar to the procedure used for superheat data acquisition. The saturation pressure is maintained using the PRT and the refrigerant mass flow rate is held constant at the first rate of interest. The inlet quality of the refrigerant is set using the process described above and the Fluke is programmed to take data. Once the first point is taken, the quality is changed and the stand allowed to reach steady-state. Next, the refrigerant flow rate is incremented and the qualities are run through again. After all refrigerant flow rates of interest have been tested, the saturation pressure is incremented and the process is repeated.

For evaluating the microchannel tubes, eighteen data points would be collected during two-phase testing. These points are composed of two refrigerant inlet qualities, three refrigerant mass flow rates, and three different saturation pressures.
4.3 Data Reduction and Results

Once the data on each tube has been collected, it must be reduced to generate useful information about the tube. Heat transfer performance, pressure drop characteristics, and friction factor estimations are examples of the type of parameters which can be obtained using this test facility. For this report, testing was conducted to verify the operation of the stand. Complete condenser tube testing was not conducted for this report. Instead, an energy balance was performed in order to verify the effectiveness of the facility. Also, the theory behind and procedure for determining refrigerant and air side resistances from the subcooled data is given. Methods for determining heat transfer coefficients, friction factors, and other useful tube characteristics will be given and applied to actual data later in the course of the project.

4.3.1 Energy Balance Check

Refrigerant condensation studies create a difficulty for the researcher due to the possibility of two-phase refrigerant states. As stated previously, this fact is a problem due to the lack of accurate sensors which measure quantities other than pressure and temperature. For this project, refrigerant of differing inlet quality (subcooled, two-phase, or superheat) is to be driven through a microchannel condenser tube. The entrance state of the refrigerant is determined using the methods discussed earlier. However, the refrigerant at the exit state may exist as two-phase, creating the need for the ability to measure some property other than temperature and pressure. The state is instead defined by using the air stream. Properties of the air before and after the test section are well known from the pressure and temperature measurements made at these points. Used with the air flow rate, these readings can reveal the total heat transferred to the air side from the condenser tube. This value can then be used to determine properties of the refrigerant exiting the test section, using a first law analysis of the tube.

However, in order to utilize this method of state determination, all heat transferred to the air must be assumed to originate from the test section. In an ideal situation (perfect measurements, no heat loss, etc.), this would be true. In this situation, it is not. Instead, some desired accuracy limit is set, requiring the balance of energy to lie within the specified range. Running subcooled liquid refrigerant through the condenser tube is one way to check whether or not the facility provides this energy balance. The subcooled data discussed in the last section is used for this purpose.
In checking the facility's balance of energy using subcooled data, the heat transferred from the refrigerant side is first determined. This heat transfer is defined as:

\[ Q_{\text{ref}} = \dot{m}_{\text{ref}}(h_{\text{ref,in}} - h_{\text{ref,out}}), \]  

(4.1)

where \( h_{\text{ref,in}} \) and \( h_{\text{ref,out}} \) are the enthalpies of the refrigerant up and downstream of the test section, respectively. These values are determined using the up and downstream temperatures and pressures and the property routines located in the software package Engineering Equation Solver© (EES). EES is a Newton-Rhapson solver with thermophysical properties included in it. This program uses property functions for R-134a given in the literature (McLinden, 1989) to determine refrigerant properties. Subcooled liquid properties are approximated by evaluating the saturated liquid properties at that temperature. The temperatures and pressures for each data point are input into EES, which in turn evaluates the properties of interest. The refrigerant mass flow rate, \( \dot{m}_{\text{ref}} \), is determined using the Micro-Motion flow meter.

Once the refrigerant side heat transfer is determined, the air side heat transfer must be evaluated as well. The energy absorbed by the air as it passes over the test section is described by:

\[ Q_{\text{air}} = \dot{m}_{\text{air}}(h_{\text{air,out}} - h_{\text{air,in}}), \]  

(4.2)

where \( h_{\text{air,in}} \) and \( h_{\text{air,out}} \) are the upstream and downstream enthalpies of the air stream, respectively. EES evaluates these properties assuming that the air is an ideal gas. Thus, only the temperature of the air is required to determine these values. Determination of the mass flow rate of the air, \( \dot{m}_{\text{air}} \), is accomplished using the pressure differential measured across the venturi, and properties of the air around the test section.

Once the air and refrigerant side heat transfers are calculated using the relations given above, they must be compared to determine whether or not the equal energy assumption is valid. Figure 4.1 shows such a comparison. The heat transferred to the air side is plotted against the refrigerant side heat transfer. The data shown is for an air flow rate of approximately 0.040 kg/s. The plot shows that a ±3% error limit on an energy balance is attainable with this test facility. Also, as expected, the data falls on a line. Ideally, the data should lie on a 45° line that passes through the origin. Due to the error in the measurements and other sources, it does not follow this ideal exactly. The line has a slope of one, but at the point where both air and refrigerant heat transfers
Figure 4.1: Energy Balance at an Air Mass Flow Rate of 0.40 kg/s

Figure 4.2: Energy Balance at All Air Mass Flow Rates
should be zero, the line predicts some air side energy exchange. This error is within the limits of error due to the measurements used to determine the heat transfer.

Even though this figure suggests that the stand is functioning correctly, other factors must be investigated. The previous figure was generated from data collected at a specific air flow rate. However, in order to ensure that the test facility will yield useful data at all operating conditions of interest, energy balances must be conducted using other air flow rates. This is accomplished in the same manner as the energy balance shown in Figure 4.1. Subcooled data was taken for a total of six different air mass flow rates. This data was then reduced to determine both air and refrigerant side heat transfers by the same methods as those used to generate Figure 4.1. Figure 4.2 shows the result of this process. The values of the different flow rates used are given in the legend shown in the figure.

Figure 4.2 shows many interesting trends. The air mass flow rate seems to have the greatest effect on the energy balance. At the lower flow rates, the slope of the lines tend to be less than unity. This could be caused by the method of refrigerant property determination. For this plot, EES was used to determine the enthalpy of the refrigerant stream at the entrance and exit of the test section. However, it does not determine these subcooled properties by using an R-134a equation of state for the temperature and pressure range of operation. Instead, it uses the temperature of the refrigerant to evaluate the saturated liquid properties at that temperature. This assumes that the specific internal energy and volume are functions of only temperature, and that the second term on the right side of the following equation,

$$h(T,p) = h_i(T) + v_i(T)[p - p_{sat}(T)], \tag{4.3}$$

is negligible (Moran and Shapiro, 1988). This relation is only a good approximation for the actual enthalpy, which is dependent upon both temperature and pressure. As the refrigerant mass flow rate is increases, the temperature difference (and therefore, the enthalpy difference) across the condenser tube decreases. This means that the error associated with evaluating each enthalpy is a greater percentage of the enthalpy difference, causing an increasing overestimation of the refrigerant side heat transfer with increasing refrigerant flow rate. In order to remedy this problem, a more accurate method for determining the properties of the refrigerant is needed.

Secondly, there is an overestimation of the refrigerant side heat transfer at the lower air flow rates causing a translation of the line to the right. This error is clearly shown in Figure 4.2. At low air flow rates, as the flow rate increases, the error
decreases. However, as the flow rate is further increased to very high levels, this energy balance error again begins to increase, but to much less of a degree. This trend could be due to a number of factors. First, there may be stratification of the downstream temperature in the vertical direction. If the downstream temperature is lower at the point of measurement due to wake effects from the tube, the heat transfer will be underestimated. Also, using such small thermocouples requires extreme care when installing them into the air loop. Any bends in the wire can alter the calibration of the thermocouple. This effect would seem to be small but may influence the heat balance. Thirdly, the thermocouples used in the stand are bare, unshielded beads. These points are subject to radiation effects on both the up and downstream sides of the test section but the maximum effect would be felt by the upstream thermocouples since the temperature difference between them and the tube is greater than the downstream temperature difference. This would cause an overestimation of the upstream enthalpy, effectively reducing the change in enthalpy across the test section. At higher air flow rates, this effect would be reduced because the temperature difference between the air up and downstream if the tube is reduced, diminishing the radiation effects. The air mass flow determination using the venturi seems to be adequate. The differential transducer used to determine the pressure drop in the venturi is calibrated correctly.

Confidence in the test facility's operation is low at low air mass flow rates. An investigation into the effects described above might shed some light on the problem. However, the data of interest for this project will be taken at high air flow rates. Under these conditions, the facility seems to perform well, fulfilling the specified tolerance.

4.3.2 Wilson Plot Analysis

Once the energy balance has been verified, the next step in reducing the data is to generate a Wilson plot. The technique behind this was developed by E.E. Wilson for the design and analysis of any heat exchanger. For a gas-liquid condenser system, this graph can be used to determine the amount resistance to heat transfer that is contributed by the air side. Each subcooled data set collected utilizing this test facility can be used to create a different Wilson plot due to the fact that the air side resistance is held constant in each data set.
4.3.2.1 Wilson Plot Theory

In order to generate a Wilson plot, the data collected during subcooled liquid testing must be used to calculate various important parameters for each tube. Among these parameters is the overall heat transfer resistance of the condenser tube. Appendix contains the code used to determine this and other parameters which are used to create a Wilson plot. In this program, the resistance is calculated using the effectiveness-NTU method of heat exchanger analysis. The Reynold's number for the refrigerant entering the test section is calculated from the refrigerant properties and depends largely upon the port geometry.

The effectiveness of a heat exchanger can be determined using temperature and mass flow data. The refrigerant side heat transfer is determined from Eqtn. 4.1 and used to evaluate the heat capacities of both the air and refrigerant sides by the following equations:

\[ C_{\text{air}} = \frac{Q_{\text{ref}}}{(T_{\text{air, out}} - T_{\text{air, in}})} \]  \hspace{1cm} (4.4)

\[ C_{\text{ref}} = \frac{Q_{\text{ref}}}{(T_{\text{ref, in}} - T_{\text{ref, out}})} \]  \hspace{1cm} (4.5)

Once these values are computed, the effectiveness (which is a ratio of the actual heat transfer to the maximum possible heat transfer) of the condenser can be determined by

\[ \varepsilon = \frac{C_{\text{ref}}(T_{\text{ref, in}} - T_{\text{ref, out}})}{C_{\text{min}}(T_{\text{ref, in}} - T_{\text{air, in}})} \]  \hspace{1cm} (4.6)

where \( C_{\text{min}} \) is the minimum of the air and refrigerant heat capacities.

Next, the capacity rate ratio is evaluated. This quantity is one which is used in an expression for the effectiveness of the heat exchanger. It can be calculated using

\[ C_r = \frac{C_{\text{min}}}{C_{\text{max}}} \]  \hspace{1cm} (4.7)

where \( C_{\text{max}} \) is the maximum of the two heat capacities. This value and the effectiveness determined from (4.3) can be used to evaluate the number of heat exchanger transfer units, NTU, of the condenser tube according to
\[ \varepsilon = 1 - \exp\left\{ \frac{\text{NTU}^{(22)}}{C_r} \left[ \exp\left(-C_r \cdot \text{NTU}^{(78)}\right) - 1 \right] \right\}. \]  

This correlation was obtained from a book by Incropera and DeWitt (1985). It applies to crossflow, single pass heat exchangers with both fluids unmixed. The accuracy of this relation has been established to be within ±3% (Mason, 1954). A series solution generated by Mason can be used to solve for the effectiveness more accurately. Once the NTU value is calculated, the total resistance of the heat exchanger is determined using the following relation:

\[ R_{\text{total}} = \frac{1}{U\text{A}_\text{tot}} = \frac{1}{\text{NTU} \cdot C_{\text{min}}}. \]  

(4.9)

Here, \( U\text{A}_\text{tot} \) is the overall heat transfer coefficient for the tube.

The Reynold's number of the refrigerant must also be calculated in order to generate a Wilson plot. The Reynold's number at the entrance to the test section is used for this purpose and is defined by

\[ R_{\text{ref}} = \frac{4 \cdot \dot{m}_{\text{ref}}}{P_{\text{wet}} \cdot \mu_{\text{ref,in}}}, \]  

(4.10)

where \( \dot{m}_{\text{ref}} \) is the refrigerant mass flow rate, measured by the Micro-Motion flow meter, \( P_{\text{wet}} \) is the wetted perimeter of the tube, determined from the tube size and port geometry, and \( \mu_{\text{ref,in}} \) is the dynamic viscosity of the refrigerant at the inlet. From these equations, the total resistance and Reynold's number can be calculated for each refrigerant mass flow rate. A Wilson plot for each air flow rate can then be created from these two values.

### 4.3.2.2 Wilson Plot

Figure 4.3 shows a schematic of a Wilson plot generated using the data taken at an air flow rate of 0.040 kg/s. The total heat transfer resistance (1/UA) is plotted against the refrigerant side Reynold's number, raised to a power. The power is determined by theoretical analysis, and is taken from a Nusselt number correlation for fluids flowing inside tubes. The analysis assumes that both the air and refrigerant flows are turbulent.
Figure 4.3: Wilson Plot Example Using an Air Mass Flow Rate of 0.40kg/s
and fully developed (Stoecker and Jones, 1982). This assumption is usually accurate for most condensing systems.

The data calculated as outlined above produces a line which reveals important thermal resistance information about the tube. As the mass flow rate of the refrigerant is increased, the abscissa values decrease. At an infinite flow rate, $\text{Re}_{\text{ref}}^{-0.8}$ becomes zero. This point corresponds to a refrigerant resistance of zero, since the resistance is directly proportional to the refrigerant flow rate. Hence, the value for $1/UA$ at this condition is composed of all the other resistances for the tube, excluding the refrigerant resistance. These include the air side, tube wall and contact resistances. The value for all of the other resistances is a constant for a constant air flow rate and inlet temperature. Therefore, once these other contributions are determined, the refrigerant side resistance at each refrigerant flow rate can be determined by subtracting the other contributions from the total. The refrigerant side resistances can then be used to determine the heat transfer coefficient for those conditions.

4.3.2.3 Wilson Plot Data Generation Program

A different plot can be created for each air flow rate, since changing the flow rate will change the air side resistance. However, according to the theory behind the Wilson plot, the slope of the line for each flow rate should be the same value. This is due to the fact that all the data gathered comes from the same tube. For this reason, some method must be used which provides a curve fit with the same slope for each air flow rate. The difference between the fits for different flow rates will then be in the value of the sum of the resistances other than the refrigerant side resistance.

Individual line generation is accomplished using an EES© program. Fluid temperatures, heat transfer, and the Reynold's number of the refrigerant entering the test section for each refrigerant and air mass flow rate point are entered into a lookup table in the program. It then uses these parameters to calculate the values correlated in the Wilson plot. A minimization algorithm in EES is used to search for the best linear fit for all of the data assuming the same slope for each line. It achieves this using a least squares curve fit for each air flow rate, resulting in lines with the same slope but different ordinate intercepts. Figure 4.4 shows the results of this minimization scheme. As the air mass flow rate is increased, the air side contribution to the total resistance decreases. The plot demonstrates this effect. The y-intercept $(R_{\text{tot}}-R_{\text{ref}})$ decreases as the air mass flow rate is increased. Also, the plots shows that as the air flow rate increases, the change in air side resistance from flow rate to flow rate decreases.
Figure 4.4: Wilson Plot Using All Air Mass Flow Rates
Hence, the effect on the air side resistance of changing the air flow rate decreases at higher flow rates. This is due to the fact that the air side resistance is proportional to the inverse of the flow rate.

As discussed earlier, the curve generated is not merely a least squares fit of the data. It is a best fit approximation using the same slope for each air flow rate. Figure 4.5 compares the difference between a least squares curve fit for one set of data and the result of the minimization fit for all of the data for an air mass flow rate of 0.040kg/s. The predicted resistance line consists of the data generated using the curve fit. The plot shows a very close agreement between the two fits. The difference between the two could be caused by a number of factors. First, the equations used in the minimization assume that both stream are fully developed over the heat exchanger's surface. This might not be true on the refrigerant side where the transition section feeding the refrigerant into the test section could cause a length of non-fully developed flow. Also, the refrigerant side Reynold's number at the low flow rates is very close to the laminar-turbulent transition range (about 3000), lying on the lower boundary of the range of applicability for the crossflow effectiveness relation.

4.3.2.4 Further Data Reduction

Once the Wilson plot has been generated and the sum of the resistances other than the refrigerant is evaluated, all other data may be collected and analyzed. The same air side conditions are used as during the subcooled Wilson plot data tests, creating the same air side resistance magnitudes. The total resistance can then be calculated from the data at each point (superheat or two-phase) and the resulting refrigerant side resistance found. The total resistance is described by

\[ R_{\text{total}} = \frac{1}{UA} = R_{\text{air}} + R_{\text{contact}} + R_{\text{tube}} + R_{\text{ref}}, \]  

where \( R_{\text{air}} \) is the air side resistance, \( R_{\text{contact}} \) is the contact resistance associated with the joint between the fins and the outside of the tube, \( R_{\text{tube}} \) is the resistance of the tube wall, and \( R_{\text{ref}} \) is the refrigerant side resistance. The sum of all of the resistances but the refrigerant side is known from the Wilson plot data. The refrigerant side resistance can be calculated by subtracting the sum of the other resistances from the total measured resistance. This can then be used to determine the heat transfer coefficient of the tube on the refrigerant side by
Actual resistance: $\frac{1}{UA} = 0.013977 + 15.31 \text{Re}_{\text{ref}}^{-0.8}$

Predicted resistance: $\frac{1}{UA} = 0.014307 + 15.06 \text{Re}_{\text{ref}}^{-0.8}$

Figure 4.5: Wilson Plot Comparison Between Predicted and Actual Data
The webbing located inside the condenser tube acts as a fin which connects the top and bottom of the tube. Therefore, the wetted areas are separated into (a) the primary area, $A_{p,ref}$, which constitutes the inside tube wall surface, and (b) the fin surface area, $A_{f,ref}$, which is the surface area of the webbing in contact with the refrigerant. These values depend upon the geometry of the webbing and the hydraulic diameter of the tube. Secondly, the fin (webbing) efficiency, $\eta_0$, must be determined and used in equation (4.12). Once these values are calculated, the refrigerant side heat transfer coefficient can be determined for each case tested.

Heat transfer coefficient values are not reported in this document. Data will be taken in the future using the test facility described in this report. The results of these tests will be revealed in an upcoming ACRC document by D. Andres. This report deals only with the facility construction and preliminary data evaluation.
CHAPTER 5: SUMMARY AND RECOMMENDATIONS

5.1 Summary

A test facility has been constructed which can be used for horizontal tube, condensation heat transfer studies. The apparatus was mainly constructed with the purpose of testing microchannel condenser tubes but its range of applicability is not restricted to just these tubes. Other tube sizes and geometries can be installed in the stand and tested with few modifications required. The facility is fully operational under the conditions of interest to the condensation heat transfer studies for which it will be used. A wide range of operating conditions can be achieved with the facility. Also, basic subcooled liquid refrigerant data has been presented which was used to demonstrate some of the data reduction techniques which will be utilized by future projects using the facility. In addition, the data was used for the determination of refrigerant side resistances for one microchannel condenser tube.

The components required to perform condensation heat transfer studies have also been discussed. The theory behind the design and installation of each refrigerant loop element was detailed, while an overview of the air side was given. For a more detailed explanation of the design of the air loop, the reader is directed to a forthcoming document by D. Andres (Andres, 1994). Also, new components which are not standards for such a test facility were discussed. The theory behind their operation, their liabilities, attributes, and implementation were all detailed, providing information about these useful alternatives to the current state of technology.

5.2 Future Considerations

5.2.1 Heat Transfer Correlation Studies

The facility may now be used to generate beneficial heat transfer data on various microchannel tubes under different operating conditions. The results of such testing will also be revealed in a future document by D. Andres (Andres, 1994). Since no two-phase or superheat inlet refrigerant quality data have been presented here, the reader is referred to this report for more information on the performance of these edge-of-technology tubes. The tube parameters which will be varied throughout the test conducted using this facility include:

1) port geometry (triangular, square, H, and circular);
2) tube width;
3) refrigerant inlet quality (superheat, two-phase, subcooled);
4) condensing pressure.

Also, heat transfer correlations will be given which can be compared to those given in Chapter 2 of this report.

Once data has been collected using the microchannel tubes, some modeling can be done in order to verify the test results and provide more "experimental" flexibility than that offered by empirical experimentation. Such modeling can also be used to give information for the purpose of optimizing the design of these condensers, increasing their usefulness. Also, further investigation into header design and material selection might prove to be useful. A valid model created from empirical data could be used to perform such experiments.

An investigation into the possible uses of this technology outside of the car air conditioning arena would provide other industries with the benefits obtained from microchannel condenser use. Possible refrigerator and room air conditioning applications should be studied.

5.2.2 Air Side Modifications

The test facility currently provides the means for testing microchannel condenser tubes. However, there are modifications and additions which can be made that would make the stand able to create more operating conditions and make it easier to operate. On the air side, a new fan configuration might help reduce any temperature stratification in the air upstream of the test section. Currently, the heater and cooling heat exchanger in the duct lie downstream of the fan. This leads to somewhat non-uniform air temperatures after these temperature control devices. Since a fan is an ideal air mixing device, moving the fan downstream of the heater and cooling coil will produce the most uniform temperatures in the air stream as it enters the test section. Also, a larger fan could be installed to achieve higher air flow rates at the test section than are currently available. The air velocities which can be reached using the present fan don't quite reach those desired to simulate all possible car air conditioner situations due to the high pressure drop in the air loop. All of the flow conditioning which was installed in the air loop creates more of a pressure drop than the original fan was designed to accommodate. A larger fan could possibly overcome this high system pressure drop.

Secondly, a more precise means of channeling the air over the condenser tube might be investigated. Currently, rigid insulation board is used to contract the flow from the 24x1.5 inch galvanized duct to an area which permits the air to only pass over the
tube and half of the fin area on both sides of the tube. It is difficult to precisely set the height of the insulation on both the top and bottom of the test section. A more rigid material might be used in order to prevent such uncontrollables and to prevent the shape of the contraction from deforming over the course of removal and addition of numerous tubes.

Likewise, calibration of the venturi flow meter might increase the accuracy of the air side flow rate reading. The calibration curve generated by the manufacturer involves a few assumptions which could be eliminated with proper calibration of the instrument. Such a process is difficult, however, since a calibration facility is expensive to obtain.

The heater presently installed in the air loop does not provide heating to the whole area inside the ductwork. The glowcoil heater lies across the diameter of the circular duct, unable to provide energy to all of the air flowing in the duct. In order to reduce the possibility of temperature stratification due to this and inadequate mixing, a heater coil which more closely resembles the form of the ductwork should be investigated. Such a heater should provide the same amount of heat as the glowcoil, but with a smaller watt density.

5.2.3 Refrigerant Side Modifications

Improvements can also be made on the refrigerant side. A main hindrance to full operation of the facility is the lack of a working power measurement device. Two-phase data cannot be obtained without such a component. Another problem that exists involves an instability in the system at specific refrigerant mass flow rates. There is a range of flow rates over which control of the flow is impossible due to this inherent instability in the loop. The flow meter reading becomes meaningless and unstable at these flow rates, making it impossible for the process controller to keep the flow rate constant. This range varies with system pressure. An investigation into the causes of this and possible solutions would be beneficial.

Also, the transition section used to join the 1/4in. copper refrigerant tubing to the microchannel condenser tube has problems associated with its design. It is an expensive piece to fabricate in bulk but can only be used on tubes with the same width and height if it is to properly seal the loop. This encumbrance limits the types of tubes which can be tested without additional cost. The transition section is also difficult to seal at high pressures. Thirdly, it provides a point of heat loss not accounted for by the measurements taken around the test section. The pressure and temperature measurements on the refrigerant side are obtained before the upstream transition.
section and after the downstream one. Heat loss from these pieces might affect the overall accuracy of these measurements. A more flexible and easier to install joint piece would increase the number tubes which could be easily tested and provide more accurate and reliable heat transfer data.

Finally, the problems with closed loop PID control of the heater in the inner tank of the PID might be reduced with the addition of two smaller heaters which provide the same output as the single 6kW model. This amount of power is required to heat up the tank to operating conditions in a short amount of time but maintaining this temperature does not require that much heat. Two heaters, each with a lesser power output, might be used to heat the tank up. After the tank has reached the set temperature, this temperature could be maintained by just one of the heaters. This would provide for less fluctuation and allow better control of the bath temperature.
APPENDIX A:
OPERATION OF THE TEST FACILITY

A.1 Data Acquisition Device Specifications and Operation

A.1.1 Hardware

The data acquisition device used in this facility is the Fluke 2280 Data Logger made by John Fluke Mfg., Inc. Three important guides exist for use with the Fluke. These operational references are:

1) *The 2280 Series User's Guide*. This manual contains all the information required to operate the data logger once the hardware has been set up. It is to be consulted before performing the many functions capable of being executed on the Fluke. These include: (a) how to program channels and scan groups, (b) the workings of the alarm system, (c) how to establish the output device used, (d) the use of mathematical functions, (e) recording data, and (f) the significance of error messages. This reference is used most often in the day to day operation of the Fluke.

2) *The 2280 Series System Guide*. The system guide details the hardware involved in using the data logger for specific applications. It explains possible applications for the Fluke. It also describes all of the possible installation options and accessories available to the owner and how to install each in the mainframe. When new sensors are implemented in the stand, this guide is consulted to determine the proper scanner and connector required. Currently, all the sensor devices located on the stand output either DC voltage or current signals. The options and accessories presently installed in the system are detailed in Table A.1. Complete specifications for the Fluke are given in this guide, including accuracy's.
Table A.1: Fluke Hardware Used in the Test Facility

<table>
<thead>
<tr>
<th>Option/Accessory</th>
<th>Model Number</th>
<th>Function</th>
<th>Number in Stand</th>
</tr>
</thead>
<tbody>
<tr>
<td>High performance A/D</td>
<td>2280A-161</td>
<td>performs analog-to-digital conversion of input signals; supports up to 5</td>
<td>1</td>
</tr>
<tr>
<td>A/D converter</td>
<td></td>
<td>scanners</td>
<td></td>
</tr>
<tr>
<td>Thermocouple/DC volts</td>
<td>2280A-162</td>
<td>20 channel multiplexer with shielded inputs accepting voltages and</td>
<td>4</td>
</tr>
<tr>
<td>converter</td>
<td></td>
<td>currents</td>
<td></td>
</tr>
<tr>
<td>Isothermal input</td>
<td>2280A-175</td>
<td>20 channel aluminum input board with 3 terminals per channel; 250V</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>maximum input</td>
<td></td>
</tr>
</tbody>
</table>

Many other input scanners, connectors, and output devices are available but not required by this project. For a complete list of possible accessories, the System Guide should be consulted.

3) The 2280 Series Service Manual. This reference provides methods for maintaining the system. Trouble-shooting tips help provide a means for correcting any problems experienced during its use. It goes into great detail about the electronics of each option and accessory, giving diagrams for each. Should the Fluke malfunction, this guide should be checked before any maintenance is performed.

A.1.2 RS-232 Communication Software

We use the Fluke to take and store data from sensors located at various points around the stand. All of the interaction between the operator and the data logger is accomplished through RS-232 communication between a Macintosh Si computer and the Fluke. The interface is controlled by a software package called Versa-Term Pro. This communication package allows the user to type on the keyboard just what would be keyed in on the keypad located on the front of the Fluke. The program allows the data scanned by the Fluke to be stored digitally on the computer's hard drive where it can later be retrieved.

The software commands of interest to the operation of this facility are mainly used to open the communication lines between the computer and data logger. The following
commands are required to take data once both the computer and Fluke have been turned on, and the Versa-Term Pro application is open:

1) "control-x": toggles the Fluke to accept remote control;
2) "shift-4 ($)": allows the computer keyboard to be used for remote control.

A.1.3 Using the Datalogger

All of the commands which govern the operation of the data logger can be used within the software environment. However, when turned on, the Fluke can be programmed only through the keypad on the front panel. In order to configure the data logger to accept remote input from the keyboard, the two commands given in the previous section must be executed.

Once remote control has been established, the Fluke can be used to perform any of the functions detailed in the User's Guide. Many of these procedures must be carried out before data can be recorded. These include setting up the input channels and programming scan groups with the desired channels for data acquisition. Tables A.2, A.3, and A.4 provide a list of the channels currently being used and their respective functions. Also, the zone box from which each channel signal originates is included in the table.
Table A.2: Fluke Channel Designations - Zone Box 1

<table>
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<th>Channel #</th>
<th>Measurement Type</th>
<th>Type of Sensor</th>
<th>Description</th>
<th>Wire Length</th>
</tr>
</thead>
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<tr>
<td>0</td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>10'</td>
</tr>
<tr>
<td>1</td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>10'</td>
</tr>
<tr>
<td>2</td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>10'</td>
</tr>
<tr>
<td>3</td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>10'</td>
</tr>
<tr>
<td>4</td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>10'</td>
</tr>
<tr>
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<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>10'</td>
</tr>
<tr>
<td>6</td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>10'</td>
</tr>
<tr>
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<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>10'</td>
</tr>
<tr>
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<td>PRT</td>
<td>10'</td>
</tr>
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<td>PRT</td>
<td>10'</td>
</tr>
<tr>
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<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>10'</td>
</tr>
<tr>
<td>11</td>
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<td>Tair upstream #1</td>
<td>7'</td>
</tr>
<tr>
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<td>across TS</td>
<td>Welded TC</td>
<td>Tair downstream #1</td>
<td>7'</td>
</tr>
<tr>
<td>12</td>
<td>ΔT air</td>
<td>Welded TC</td>
<td>Tair upstream #2</td>
<td>7'</td>
</tr>
<tr>
<td></td>
<td>across TS</td>
<td>Welded TC</td>
<td>Tair downstream #2</td>
<td>7'</td>
</tr>
<tr>
<td>13</td>
<td>ΔT air</td>
<td>Welded TC</td>
<td>Tair upstream #3</td>
<td>7'</td>
</tr>
<tr>
<td></td>
<td>across TS</td>
<td>Welded TC</td>
<td>Tair downstream #3</td>
<td>7'</td>
</tr>
<tr>
<td>14</td>
<td>ΔT air</td>
<td>Welded TC</td>
<td>Tair upstream #4</td>
<td>7'</td>
</tr>
<tr>
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<td>across TS</td>
<td>Welded TC</td>
<td>Tair downstream #4</td>
<td>7'</td>
</tr>
<tr>
<td></td>
<td>ΔT air</td>
<td>Welded TC</td>
<td>Tair upstream #5</td>
<td>7'</td>
</tr>
<tr>
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<td>--------</td>
<td>-----------</td>
<td>------------------</td>
<td>----</td>
</tr>
<tr>
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<td>across TS</td>
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<td>Tair downstream #5</td>
<td>7'</td>
</tr>
<tr>
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<td>ΔT air</td>
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<td>Tair upstream #6</td>
<td>7'</td>
</tr>
<tr>
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<td>across TS</td>
<td>Welded TC</td>
<td>Tair downstream #6</td>
<td>7'</td>
</tr>
<tr>
<td>18</td>
<td>ΔT air</td>
<td>Welded TC</td>
<td>Tair upstream #7</td>
<td>7'</td>
</tr>
<tr>
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<td>across TS</td>
<td>Welded TC</td>
<td>Tair downstream #7</td>
<td>7'</td>
</tr>
<tr>
<td></td>
<td>ΔT air</td>
<td>Welded TC</td>
<td>Tair upstream #8</td>
<td>7'</td>
</tr>
<tr>
<td></td>
<td>across TS</td>
<td>Welded TC</td>
<td>Tair downstream #8</td>
<td>7'</td>
</tr>
<tr>
<td></td>
<td>ΔT air</td>
<td>Welded TC</td>
<td>Tair upstream #9</td>
<td>7'</td>
</tr>
<tr>
<td></td>
<td>across TS</td>
<td>Welded TC</td>
<td>Tair downstream #9</td>
<td>7'</td>
</tr>
<tr>
<td>Channel #</td>
<td>Measurement Type</td>
<td>Type of Sensor</td>
<td>Description</td>
<td>Wire Length</td>
</tr>
<tr>
<td>-----------</td>
<td>------------------</td>
<td>----------------</td>
<td>-------------</td>
<td>-------------</td>
</tr>
<tr>
<td>20</td>
<td>Tair</td>
<td>Welded TC</td>
<td>absolute Tair upstream #1</td>
<td>8'</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>11'</td>
</tr>
<tr>
<td>21</td>
<td></td>
<td>Welded TC</td>
<td>unusable channel</td>
<td></td>
</tr>
<tr>
<td>22</td>
<td>Tair</td>
<td>Welded TC</td>
<td>absolute Tair upstream #3</td>
<td>8'</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>11'</td>
</tr>
<tr>
<td>23</td>
<td>ΔT air (PVC)</td>
<td>Welded TC</td>
<td>Tair upstream #1</td>
<td>8'</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>Tair downstream in PVC #1</td>
<td>6'</td>
</tr>
<tr>
<td>24</td>
<td>ΔT air (PVC)</td>
<td>Welded TC</td>
<td>Tair upstream #5</td>
<td>8'</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>Tair downstream in PVC #5</td>
<td>6'</td>
</tr>
<tr>
<td>25</td>
<td>ΔT air (PVC)</td>
<td>Welded TC</td>
<td>Tair upstream #9</td>
<td>8'</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>Tair downstream in PVC #9</td>
<td>6'</td>
</tr>
<tr>
<td>26</td>
<td>T ref</td>
<td>Sheathed TC</td>
<td>Tref before TS</td>
<td>6'</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>9'</td>
</tr>
<tr>
<td>27</td>
<td>ΔT ref</td>
<td>Sheathed TC</td>
<td>Tref upstream of TS</td>
<td>9'</td>
</tr>
<tr>
<td></td>
<td>across TS</td>
<td>Sheathed TC</td>
<td>Tref downstream of TS</td>
<td>11'</td>
</tr>
<tr>
<td>28</td>
<td>T water</td>
<td>Welded TC</td>
<td>EST inner tank</td>
<td>8'</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>9'</td>
</tr>
<tr>
<td>29</td>
<td>T water</td>
<td>Welded TC</td>
<td>EST jacket</td>
<td>8'</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>9'</td>
</tr>
<tr>
<td>30</td>
<td>ΔT water</td>
<td>Welded TC</td>
<td>EST inner tank (also to G.H. PID)</td>
<td>8'</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>EST jacket</td>
<td>8'</td>
</tr>
<tr>
<td>31</td>
<td>T water</td>
<td>Welded TC</td>
<td>PRT temperature</td>
<td>10'</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td>10'</td>
</tr>
<tr>
<td>32</td>
<td></td>
<td></td>
<td>unused</td>
<td></td>
</tr>
<tr>
<td>33</td>
<td></td>
<td></td>
<td>unused</td>
<td></td>
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<td>34</td>
<td>ΔT Zone Box</td>
<td>Welded TC</td>
<td>Zone Box 1 top</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>Zone Box 1 bottom</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>---</td>
<td>---</td>
<td>---</td>
<td>---</td>
<td></td>
</tr>
<tr>
<td>35</td>
<td>ΔT Zone Box</td>
<td>Welded TC</td>
<td>Zone Box 2 top</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>Zone Box 2 bottom</td>
<td></td>
</tr>
<tr>
<td>36</td>
<td>ΔT Zone Box</td>
<td>Welded TC</td>
<td>Zone Box 3 top</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>Zone Box 3 bottom</td>
<td></td>
</tr>
<tr>
<td>37</td>
<td>Tair</td>
<td>Welded TC</td>
<td>absolute Tair upstream #5</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td></td>
</tr>
<tr>
<td>38</td>
<td>Tair</td>
<td>Welded TC</td>
<td>absolute Tair upstream #7</td>
<td></td>
</tr>
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<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td></td>
</tr>
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<td>39</td>
<td>Tair</td>
<td>Welded TC</td>
<td>absolute Tair upstream #9</td>
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</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
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Table A.4: Fluke Channel Designations - Zone Box 3

<table>
<thead>
<tr>
<th>Channel #</th>
<th>Measurement Type</th>
<th>Type of Sensor</th>
<th>Description</th>
<th>Wire Length</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>P ref</td>
<td>Setra - 204</td>
<td>ref. pressure before EST</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 356980</td>
<td></td>
<td></td>
</tr>
<tr>
<td>41</td>
<td>Tref</td>
<td>Sheathed TC</td>
<td>absolute Tref outside PRT</td>
<td>5'</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td></td>
</tr>
<tr>
<td>42</td>
<td>P ref</td>
<td>Setra - 204</td>
<td>ref. pressure before TS</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 356981</td>
<td></td>
<td></td>
</tr>
<tr>
<td>43</td>
<td>Power</td>
<td></td>
<td>EST heater power</td>
<td></td>
</tr>
<tr>
<td>44</td>
<td>P ref</td>
<td>Setra - 280E</td>
<td>ref. pressure before pump</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 317237</td>
<td></td>
<td></td>
</tr>
<tr>
<td>45</td>
<td>T air</td>
<td>Welded TC</td>
<td>absolute Tair before venturi</td>
<td>8'</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td></td>
</tr>
<tr>
<td>46</td>
<td>T air</td>
<td>Welded TC</td>
<td>absolute Tair before venturi</td>
<td>8'</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT</td>
<td></td>
</tr>
<tr>
<td>47</td>
<td>Power</td>
<td>Ohio</td>
<td>EST circulation pumps</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Semitronics</td>
<td>power</td>
<td></td>
</tr>
<tr>
<td>48</td>
<td>P air</td>
<td>Setra 270</td>
<td>air pressure before TS</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 321750</td>
<td></td>
<td></td>
</tr>
<tr>
<td>49</td>
<td>Voltage</td>
<td></td>
<td>Fan Voltage</td>
<td></td>
</tr>
<tr>
<td>50</td>
<td>P air</td>
<td>Setra 270</td>
<td>air pressure before Venturi</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 336037</td>
<td></td>
<td></td>
</tr>
<tr>
<td>51</td>
<td>Voltage</td>
<td></td>
<td>Thermistor Reference</td>
<td></td>
</tr>
<tr>
<td>52</td>
<td>ΔP air</td>
<td>Setra 239</td>
<td>air pressure drop across TS</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 351249</td>
<td></td>
<td></td>
</tr>
<tr>
<td>53</td>
<td>Voltage</td>
<td></td>
<td>Thermistor Output</td>
<td>(also to PRT controller)</td>
</tr>
<tr>
<td>54</td>
<td>ΔP air</td>
<td>Setra 239</td>
<td>air pressure drop across venturi</td>
<td>(also to Blower controller)</td>
</tr>
<tr>
<td></td>
<td>Voltage</td>
<td>MicroMotion</td>
<td>ref. flow meter output</td>
<td></td>
</tr>
<tr>
<td>---</td>
<td>---------</td>
<td>-------------</td>
<td>------------------------</td>
<td></td>
</tr>
<tr>
<td>55</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>56</td>
<td>ΔP ref</td>
<td>Setra 228-1</td>
<td>ref. pressure drop across TS</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>S/N 353886</td>
<td></td>
<td></td>
</tr>
<tr>
<td>57</td>
<td>T air</td>
<td>Welded TC</td>
<td>absolute Tair before venturi 8'</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT 4'</td>
<td></td>
</tr>
<tr>
<td>58</td>
<td>T ref</td>
<td>Sheathed TC</td>
<td>absolute Tref before Pump 5'</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT 4'</td>
<td></td>
</tr>
<tr>
<td>59</td>
<td>T ref</td>
<td>Sheathed TC</td>
<td>absolute Tref before EST 6'</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Welded TC</td>
<td>PRT 4'</td>
<td></td>
</tr>
</tbody>
</table>
A.2 Start-Up and Operating Procedures for the Test Facility

Before the stand can be used to take any data, it must be warmed up to steady-state from ambient conditions. The procedure for the loop start-up is given in this section. Also, once the data acquisition is complete, the stand must be shut-down in order to prevent possible damage to the components. This section of Appendix A should be consulted before any data acquisition is executed. A full, working knowledge of the stand is required in order to insure reliable data acquisition and operator safety.

A.2.1 Start-Up Procedure

The following is a step-by-step procedure for getting the facility up to a steady-state condition. These steps should be followed every time the facility is turned on. If data taking has already begun, all steps in this procedure need not be executed. The individual components are grouped according to their function in order to facilitate start-up.

A.2.1.1 Electrical Plugs

The following plugs should be inserted into their appropriate sockets at all times:

**Wall**
- PRT heater plug (240 V, near PRT)
- EST heater plug (240 V, behind stand near PRT)
- Guard tank heater plug (240 V, behind stand near PRT)
- Power Strip 1 (120 V, behind stand near controllers)
- Power Strip 2 (120 V, behind stand near controllers)

**Power Strip 1**
- Fan (Fan controller power)
- Air (Air heater and refrigerant mass flow rate controller power)
- PRT (PRT controller power)
- EST (EST and guard tank controller power)

**Power Strip 2**
- 3 (Power strip 3 cord)
A.2.1.2 Electrical Switches

The following items should be ON at all times. These items receive power from the other power strips that are OFF when not running, but turned ON at startup when Power Strip 1 and 2 are switched on.

- Power Strip 3
- Power Strip 4
- Power Strip 5
- Pressure transducer electrical supply (24 V)
- Thermistor power supply (5 V)
- Power Measurement System power supply

The following should be OFF before startup. Turn them ON in this order.

- PRT Heater wall switch
- Refrigerant Heater wall switch
- Guard Heater wall switch
- Power Strip 1
• Power Strip 2
• Controller electrical boxes (4)
• Ismatec refrigerant pump controller (back of stand)

A.2.1.3 Plumbing

The following should be closed at startup. Move the valves to the fully OPEN position.
• Chilled water supply at wall (blue handle)
• Chilled water supply to Proj 25 test facility (green handle)
• Chilled water return from Proj 25 test facility (green handle)
• Chilled water return at wall (small red dial)

The following should be closed at startup. Fully open them, then turn back 1/2 turn.
• Chilled water supply to PRT (blue dial, near PRT)
• Chilled water supply to After Condenser (blue dial, near PRT)
• Chilled water supply to Air Precooler (blue dial, behind stand near Fluke)

The following refrigerant line valves should be fully open at all times.
• Outside PAT
• Bottom of PAT sight tube
• Inlet to Test Section
• Outlet of Test Section

A.2.1.4 Data Acquisition

Perform the following in this order:
• Turn datalogger key to "program" position. Make sure that the "scan" light is OFF.
• Turn on computer
• Launch the application "VersaTermPro 3.0.2".
• Type "control-x" from the keyboard. The light indicating remote operation on the Fluke should turn ON.
• Type "$" (shift-4) from the keyboard. Datalogger is now ready for remote control from the keyboard.
• The following single-stroke commands may be used at any time.
  "m" for a menu at any time
  "&" to toggle scanning
  "%" to toggle monitoring of a single channel
A.2.2 Shut-Down Procedure

The following lists the procedure to be followed after data acquisition is complete. Each operation must be performed in order to prevent any damage to the facility. Failing to do so can result in possible destruction of the stand's individual components.

A.2.2.1 Data Acquisition

Do the following in this order.
- Stop all scanning or monitoring.
- Type "control-z" to return datalogger to local control.
- Turn off computer.
- Turn off datalogger.

A.2.2.2 Plumbing

Move the following valves to the fully CLOSED position.
- Chilled water supply at wall
- Chilled water supply to Proj 25 test facility
- Chilled water return from Proj 25 test facility
- Chilled water return at wall

Move the following valves to the fully CLOSED position.
- Chilled water supply to PRT
- Chilled water supply to After Condenser
- Chilled water supply to Air Precooler

The refrigerant line valves should be fully open at all times. See startup procedure for details.

A.2.2.3 Electrical Switches

Turn the following OFF in this order. All other switches should remain in the ON position when not operating.
- Ismatec refrigerant pump controller (back of stand)
- PRT wall switch
- Refrigerant Heater wall switch
• Guard Heater wall switch
• Controller electrical boxes (4 of them)
• Power Strip 1
• Power Strip 2

A.2.2.4 Electrical Plugs

All plugs should remain inserted in their appropriate outlets. See Startup Procedure for details.
A.3 Test Facility Parts List

The individual parts located in the test facility are detailed in the following tables. They are split into separate groups according to their function in the stand. Component accuracy's are given where appropriate. This list is not exhaustive but provides an overview of the components which are important to the operation of the stand. This list provides a base for replacing or upgrading any of the components listed.

Table A.5: Refrigerant Flow Control

<table>
<thead>
<tr>
<th>Vendor</th>
<th>Part/Catalog #</th>
<th>Description</th>
<th>Location in Facility</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cole-Parmer</td>
<td>99999-99</td>
<td>Pump head: modified for 425 psi system pressure; Buna-N seals; similar to CP 07002-14</td>
<td>ref. loop: between PRT tap and MM flow-meter.</td>
</tr>
<tr>
<td>Cole-Parmer</td>
<td>L-07617-75</td>
<td>Pump motor and controller: unmounted drive with remote controller; has remote control input capability</td>
<td>N/A</td>
</tr>
<tr>
<td>Powers</td>
<td>535-2114-0BD001</td>
<td>Digital process controller: current proportional output (4-20 mA); 2 alarm relays; DC logic SSR output; remote setpoint capability; autotuning; used for refrigerant flow rate control</td>
<td>control center: refrigerant pump controller box</td>
</tr>
<tr>
<td>MicroMotion (MM)</td>
<td>DS012S100</td>
<td>Mass Flow Meter: 500 psi max line pressure; 50 °F max line temperature; full scale flow 0.0-0.01 kg/s, use with R134a, Stainless steel wetted parts; #4 VCO union; 1/4 in. NPT female connector</td>
<td>ref. loop: between pump and enthalpy setting tank.</td>
</tr>
<tr>
<td>MicroMotion (MM)</td>
<td>RFT97121PRU</td>
<td>Remote flow transmitter: used as signal output from flow meter; 4-20mA</td>
<td>front side of wooden board</td>
</tr>
<tr>
<td>Vendor</td>
<td>Part/Catalog #</td>
<td>Description</td>
<td>Location in Facility</td>
</tr>
<tr>
<td>------------------------</td>
<td>----------------</td>
<td>-----------------------------------------------------------------------------</td>
<td>---------------------------------------------</td>
</tr>
<tr>
<td>Grainger</td>
<td>2C820</td>
<td>High pressure, direct drive blower without motor; delivers 165 cfm at 3450 RPM with 5&quot; H₂O S.P</td>
<td>air loop: between venturi flow meter and air cooling coil</td>
</tr>
<tr>
<td>Bodine Electric</td>
<td>D550</td>
<td>Magnetek 100/50 V, 1/3 HP shunt wound DC motor with 56C frame</td>
<td>N/A</td>
</tr>
<tr>
<td>Bodine Electric</td>
<td>KBSI240D</td>
<td>Signal isolator: accepts a 4-20 mA input control signal; outputs a 0-9 V DC signal to be used with voltage following speed controller</td>
<td>control center: blower controller box</td>
</tr>
<tr>
<td>Bodine Electric</td>
<td>KBMM-125</td>
<td>Variable speed DC motor controller: used with 0-9 V DC input signal voltage following mode</td>
<td>control center: blower controller box</td>
</tr>
<tr>
<td>Powers</td>
<td>1776-2114-0BD001</td>
<td>Digital process controller: 4-20 mA output; 2 alarm relays; DC Logic SSR output; remote setpoint capability; autotuning; used for control of blower input voltage</td>
<td>control center: blower controller box</td>
</tr>
<tr>
<td>Carrier Oehler</td>
<td></td>
<td>Short form venturi flow meter: formed from PVC Barstock 3 in. diameter pipe, β = 0.5 as of quote dated June 6, 1992</td>
<td>air loop: after exit of back plenum and upstream of blower</td>
</tr>
<tr>
<td>Setra</td>
<td>239</td>
<td>Pressure transducer: air pressure drop across venturi; used as process variable input to air flow controller</td>
<td>air loop: at entrance and throat of venturi</td>
</tr>
<tr>
<td>Capitol Plumbing and Heating</td>
<td></td>
<td>2 flanges for venturi; allow connection to 3&quot; PVC pipe</td>
<td>N/A</td>
</tr>
<tr>
<td>Vendor</td>
<td>Part/Catalog #</td>
<td>Description</td>
<td>Location in Facility</td>
</tr>
<tr>
<td>------------------------------</td>
<td>----------------</td>
<td>--------------------------------------------------</td>
<td>---------------------------------------</td>
</tr>
<tr>
<td>General Container Corp.</td>
<td>N455FRC16</td>
<td>55 gallon stainless steel drum; holds glycol mixture; acts as guard tank</td>
<td>contains 20 gallon SS drum</td>
</tr>
<tr>
<td>General Container Corp.</td>
<td>N420FRC</td>
<td>20 gallon stainless steel drum; holds glycol mixture; acts as refrigerant heater bath</td>
<td>inside 55 gallon SS drum</td>
</tr>
<tr>
<td>Omega</td>
<td>MT-260A/240</td>
<td>6 kW, 240 V immersion heater</td>
<td>immersed in inner tank</td>
</tr>
<tr>
<td>Omega</td>
<td>PTH-202</td>
<td>2 kW, 240 V over the side immersion heater</td>
<td>immersed in outer tank</td>
</tr>
<tr>
<td>Edwards Engineering</td>
<td>S1/2 C407</td>
<td>Subcooler-condenser with 4 kW capacity; nom. 1/2 ton unit</td>
<td>between test section and refrigerant pump</td>
</tr>
<tr>
<td>EE Stores Magnecraft</td>
<td>W6125DSX-1</td>
<td>Solid state relay: 25 A nom. safety relay - heaters operational only if have sufficient refrigerant flow</td>
<td>back side of wooden board</td>
</tr>
<tr>
<td>Powers</td>
<td>1776-1114-0BD001</td>
<td>Digital process controller: time proportional output; 2 alarm relays; DC logic SSR output; PV output; remote setpoint capability; autotuning; used for controlling inner tank heater</td>
<td>control center: refrigerant heater controller box</td>
</tr>
<tr>
<td></td>
<td>Serial #: 864660202</td>
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<td></td>
</tr>
<tr>
<td>Powers</td>
<td>535-1114-0BD001</td>
<td>Digital process controller: time proportional output; 2 alarm relays; DC logic SSR output; PV output; remote setpoint capability; autotuning; used for controlling outer tank heater</td>
<td>control center: jacket tank heater controller box</td>
</tr>
<tr>
<td></td>
<td>Serial #: 891540201</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Table A.8: Air Temperature Control

<table>
<thead>
<tr>
<th>Vendor</th>
<th>Part/Catalog #</th>
<th>Description</th>
<th>Location in Facility</th>
</tr>
</thead>
<tbody>
<tr>
<td>General Electric</td>
<td></td>
<td>Evaporator customized to be used as air cooler</td>
<td>air loop: in circular duct between fan and air heater</td>
</tr>
<tr>
<td>Westco Electric</td>
<td>411</td>
<td>1000 W glocoil ceramic heater</td>
<td>air loop: in circular duct between cooler and plenum</td>
</tr>
<tr>
<td>EE Stores Magnecraft</td>
<td>W6125DSX-1</td>
<td>Solid state relay, 25 A nom. safety relay - heaters operational only if fan is operating</td>
<td>back side of wooden board</td>
</tr>
<tr>
<td>Powers</td>
<td>1776-1114-0BD001 Serial #: 864660201</td>
<td>Digital process controller: time proportional output; 2 alarm relays; DC logic SSR output; PV output; remote setpoint capability; autotuning; used for controlling air side heater</td>
<td>control center: air heater controller box</td>
</tr>
</tbody>
</table>

Table A.9: Refrigerant Pressure Control

<table>
<thead>
<tr>
<th>Vendor</th>
<th>Part/Catalog #</th>
<th>Description</th>
<th>Location in Facility</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>General Container Corp.</td>
<td>N415FRC</td>
<td>15 gallon stainless steel drum; used to hold glycol mixture; acts as bath for 1L vessel</td>
<td>in PRT sub-system</td>
<td></td>
</tr>
<tr>
<td>Hoke</td>
<td></td>
<td>1 liter pressure vessel; used to hold two-phase refrigerant in glycol bath</td>
<td>inside 15 gallon SS drum</td>
<td></td>
</tr>
<tr>
<td>Omega</td>
<td>ON-970-44032</td>
<td>30k precision thermistor; used for reading glycol bath temperature</td>
<td>immersed in 15 gallon drum bath</td>
<td>±0.1 °C</td>
</tr>
<tr>
<td>McMaster-Carr</td>
<td>3706K3</td>
<td>refrigerant liquid level sightglass</td>
<td>connected to top and bottom of 1L vessel</td>
<td></td>
</tr>
<tr>
<td>Vendor</td>
<td>Part/Catalog #</td>
<td>Description</td>
<td>Location in Facility</td>
<td>Accuracy</td>
</tr>
<tr>
<td>--------</td>
<td>----------------</td>
<td>-------------</td>
<td>----------------------</td>
<td>----------</td>
</tr>
<tr>
<td>Setra</td>
<td>270</td>
<td>Pressure measurement at the inlet to the test section: 0-20 psia; 0-5 V output</td>
<td>air side: ductwork upstream of test section</td>
<td>&lt; ±0.05% of full scale</td>
</tr>
<tr>
<td>Setra</td>
<td>270</td>
<td>Pressure measurement at the inlet to the venturi: 0-20 psia; 0-5 V output</td>
<td>air side: PVC pipe upstream of venturi</td>
<td>&lt; ±0.05% of full scale</td>
</tr>
<tr>
<td>Setra</td>
<td>239</td>
<td>Differential pressure measurement across test section: 0-5 in. H₂O; 0-5 V output; remote zero adjustment</td>
<td>air side: across venturi</td>
<td>±0.14% of full scale</td>
</tr>
<tr>
<td>Setra</td>
<td>204</td>
<td>Pressure measurement at inlet to test section: 0-500 psi, 0-5 V output</td>
<td>ref. side: between EST and transition section</td>
<td>±0.11% of full scale</td>
</tr>
<tr>
<td>Setra</td>
<td>204</td>
<td>Pressure measurement at inlet to EST: 0-500 psi, 0-5 V output</td>
<td>ref. side: between MM and EST</td>
<td>±0.11% of full scale</td>
</tr>
<tr>
<td>Setra</td>
<td>228-1</td>
<td>Differential pressure measurement across test section: 0-25 psid unidirectional; 0-5 V output; 500 psi maximum line pressure; Buna-N elastomer seals; remote zero adjust.</td>
<td>ref. side: on either side of test section</td>
<td>±0.21% of full scale</td>
</tr>
<tr>
<td>Setra</td>
<td>280E</td>
<td>Pressure at inlet to pump: 0-500 psi; 0-5 V output</td>
<td>ref. side: between after condenser and pump</td>
<td>±0.11% of full scale</td>
</tr>
</tbody>
</table>
A.4 Transition Section Blueprint

The transition section (discussed in 3.2.3.5) is a connective fitting customized for this project. It has the purpose of sealing the test section in the refrigerant loop, and providing a means by which the refrigerant flow entering the test section can be conditioned. The actual contraction piece is made in two halves due to the machining required. It is composed of 304 grade stainless steel in order to allow the two halves to be brazed together, sealing the contraction chamber. This method of joining is the only possibility for preventing leakage as other methods were investigated and found to be ineffective. The O-ring plate and expansion pieces were made out of aluminum for expense and ease of machining but can be made out of stainless steel as well if corrosion poses a problem.

The actual dimensions of the transition section were not given in Figure 3.7 but are included in this appendix for the purposes of possible redesign or refabrication. The schematic included is that which was used to machine the piece by the machine shop. Figure A.1(a) shows the expansion piece. It mainly consists of a flange with a 7/8" female NPT hole in it. This is used to join the contraction piece to the refrigerant loop. Figures A.1(b) and A.1(c) show the contraction piece. As stated before, this piece had to be machined by halves as two mirror images. The tooling would not allow the contraction to be made out of one solid piece. Finally, Figure A1(d) displays a schematic of the O-ring plate. As discussed earlier, this piece seals the back end of the contraction section.

The screw threads tapped for the flanges and for the O-ring plate in the contraction section are close tolerance taps in order to prevent any shifting of the threaded joints during operation. Also, the 1/16" hole in the two halves of the contraction piece are used to lie up the two halves for brazing. These holes should also be drilled with close tolerance as misalignment of the two halves will ruin the flow conditioning.
8-32 screw holes (close tolerance)

Groove for Parker 2-128
O-ring: ID=1.487 w=.103

Figure A.1(a): Expansion Piece
Figure A.1(b): Contraction Piece - side view
Figure A.1(c): Contraction Piece - Top View
O-ring groove: 013

Figure A.1(d): O-ring Plate

0.166 Holes for 8-32 screws (close tolerance)
REFERENCES


