Design and Construction of a Condensation Heat Transfer Experimental Facility for Use with Microchannel Condenser Tubing

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DESIGN AND CONSTRUCTION OF A CONDENSATION HEAT TRANSFER EXPERIMENTAL FACILITY FOR USE WITH MICROCHANNEL CONDENSER TUBING

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University of Illinois at Urbana-Champaign, 1994
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ABSTRACT

Microchannel tubing is a relatively new heat-exchanger technology which involves internal heat transfer enhancements. Although microchannel technology offers tremendous design flexibility, it is not well understood. As part of a project that also includes full-condenser modeling and single-port flow modeling, a single-tube experimental facility has been built to evaluate the heat transfer performance characteristics of microchannel tubes using pure refrigerant 134a with air in crossflow. The facility achieves energy balance between air and refrigerant heat transfer rates within ±3%, but two-phase data are presently not attainable.

A brief review of heat transfer coefficient correlations is given to illustrate a few typical results. The design and construction of the facility is explained in detail, emphasizing the air loop. Subcooled, superheated, and two-phase data are collected for various microchannel port geometries, and data reduction techniques and theory are explained. Although energy balance data are presented in this paper, only proprietary heat transfer coefficients have been obtained thus far, and the facility is presently taking data for the publication of non-proprietary heat transfer coefficients.
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<tr>
<td>A</td>
<td>area</td>
</tr>
<tr>
<td>A_c</td>
<td>minimum free flow area</td>
</tr>
<tr>
<td>C</td>
<td>fluid heat capacity</td>
</tr>
<tr>
<td>C_d</td>
<td>discharge coefficient</td>
</tr>
<tr>
<td>C_l</td>
<td>Confidence Interval</td>
</tr>
<tr>
<td>C_r</td>
<td>capacity rate ratio</td>
</tr>
<tr>
<td>c_p</td>
<td>specific heat at constant pressure</td>
</tr>
<tr>
<td>c_v</td>
<td>specific heat at constant volume</td>
</tr>
<tr>
<td>D</td>
<td>tube diameter</td>
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<tr>
<td>f</td>
<td>fin height, Darcy friction factor</td>
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<tr>
<td>F</td>
<td>constant</td>
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<tr>
<td>F_a</td>
<td>expansion factor</td>
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</tr>
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<td>G</td>
<td>Mass flow rate</td>
</tr>
<tr>
<td>h</td>
<td>mass intensive enthalpy</td>
</tr>
<tr>
<td>h_bar</td>
<td>heat transfer coefficient</td>
</tr>
<tr>
<td>d_h</td>
<td>enthalpy difference</td>
</tr>
<tr>
<td>h_p</td>
<td>horsepower</td>
</tr>
<tr>
<td>H_X</td>
<td>heat transfer</td>
</tr>
<tr>
<td>i</td>
<td>measurement delimiter</td>
</tr>
<tr>
<td>j</td>
<td>Colburn j Factor, StPr^{2/3}</td>
</tr>
<tr>
<td>k</td>
<td>thermal conductivity</td>
</tr>
<tr>
<td>L</td>
<td>tube length</td>
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\( \dot{m} \) mass flow rate

\( N \) total number of data points

\( \text{NTU} \) number of transfer units for a heat exchanger

\( \text{Nu} \) Dimensionless Nusselt number, \( \bar{h} L / k \)

\( p \) pressure

\( P \) perimeter length

\( \text{PID} \) proportional-integral-derivative

\( \text{Pr} \) Dimensionless Prandtl number, \( v / \alpha \)

\( \dot{q} \) heat transfer

\( Q \) total heat transfer

\( R \) thermal resistance, result if used in statistics

\( \text{Re} \) Dimensionless Reynold's number, \( \rho V D / \mu \)

\( \text{St} \) Dimensionless Stanton number, \( h / \rho V c_p \)

\( T \) temperature

\( U_A_{\text{tot}} \) overall heat transfer coefficient

\( v \) mass intensive volume

\( V \) velocity

\( w \) tube width

\( W \) fin width

\( x \) vapor quality

\( X \) measured uncertainty

\( X_{\text{lt}} \) Lockhart-Martinelli parameter

\( Y \) gas expansion factor

\( Z \) unity
Greek Symbols

α  thermal diffusivity
β  diameter ratio
γ  ratio of ideal-gas specific heats, \(c_p/c_v\)
ε  heat exchanger effectiveness
η₀  fin efficiency
μ  dynamic viscosity
ν  kinematic viscosity
ρ  density
Σ  summation
τ  time constant

Subscripts

1  upstream
2  throat
act  actual
air  air based
c  characteristic
ds  downstream
e  equivalent
f  fin
h  hydraulic
i  inner
in  value at entrance to tube
L  liquid
Lg two-phase value
m median value
max maximum
min minimum
out value at exit of tube
p primary
r reduced
rad radiation
ref refrigerant based
sat saturation value
tot total
ts test section
us upstream
v vapor
wet wetted
1. INTRODUCTION

Many mobile air conditioning condensers are of serpentine construction, but condensers made with microchannel tubes are challenging this tradition\(^1\). There are many promising attributes of microchannel condensers, but their performance is not yet fully understood, and investigation into the physics of this technology is required in order to fully develop its benefits.

This document will describe the design and construction of a single-tube experimental facility which can be used to learn more about microchannel heat exchangers. The experimental facility is one part of an investigation which also involves full condenser modeling and single-port flow modeling.

1.1 Condenser and Tube Description

In a condenser made with microchannel tubes, the tubes are brazed to headers with air-side heat transfer enhancements nestled in between the tubes. Figure 1.1 is a schematic of a single microchannel tube with a full set of louvered fins on both top and bottom. The refrigerant is circuited using baffles inside the headers, involving more than one tube in each pass, and there are typically three passes total. The microchannel condenser design offers the heat exchanger designer tremendous design flexibility to vary the number of passes, alter the number of tubes per pass, and also alter the port geometry.

There are four, distinctive port geometries available, and Figure 1.2 illustrates these. Recent technology advancements made possible the fabrication of such geometries through both the processes of extrusion and welding. Extruded tubes are extruded in a single step to their final geometry. In the case of the welded tube, the internal webs are fabricated separately and subsequently brazed inside the hollow, seam-welded tube.

1.2 Objectives

A single-tube test facility is needed to produce accurate and precise data that will be used to generate the performance characteristics of microchannel tubes with different port geometries. A single-tube facility allows the researcher to control

\(^1\) Microchannel heat exchanger technology is the subject of United States and foreign patents applied for and issued to Modine Manufacturing Company, Racine, Wisconsin, USA. Current United States patents include 4,615,385, 4,688,311 and 4,998,580.
Figure 1.1: Microchannel Tube Schematic

- Louvered Fins
- Refrigerant Flow
- Microchannel Ports
- Air Flow
Figure 1.2: Microchannel Tube Port Geometries
parameters that cannot be controlled in a full-condenser study (i.e., flow conditions, refrigerant quality, and flow rate at the tube entrance), permitting repeatability and isolation of the parameters of interest.

One purpose of the microchannel facility is to generate data for the development of heat transfer coefficient correlations for microchannel tubing. Additionally, the facility offers a tremendous opportunity to verify existing correlations and compare experimental heat transfer results with theoretical calculations of flow regimes in tubes with different port geometries.

The facility was designed for use with non-chlorofluorocarbon (CFC) refrigerant 1,1,1,2-tetrafluoroethane (R-134a) as the working fluid. Although the facility is presently used for microchannel studies, it was designed to accommodate single-tube studies for any type of tube.

A brief discussion of horizontal tube condensation heat transfer coefficients is given to illustrates a few, typical results and is followed by a detailed explanation of the test facility. The test facility consists of a refrigerant and air side which intersect at the test section. In this document, air-loop design and construction is emphasized, and refrigerant-side components are briefly discussed for completeness. The reader is referred to Luhrs (1994) for a complete discussion of refrigerant-side components (Luhrs, 1994). The discussion of the test facility is followed by a brief explanation of data reduction theory and algorithms.
2. LITERATURE REVIEW

2.1 Introduction

Condensation heat transfer is the dominant heat transfer mode in the condenser of a vapor-compression cycle. Condensation is a form of convection characterized by large heat transfer rates with small temperature differences, and condensation heat transfer coefficients are typically higher than those for convection without phase-change (Incropera & DeWitt, 1990).

The refrigeration and air-conditioning industry is challenged with providing environmentally friendly systems with low charge, low cost, low weight, and low power requirements. For these reasons, non-CFC refrigerants, various heat transfer enhancements, and other new technologies are under continuous investigation. Goodremote (1988) claims that condensers made with microchannel condenser tubes (Goodremote, 1988).

The present section briefly summarizes a few typical condensation heat transfer correlations for the purpose of showing their development. Since microchannel tubes are a relatively new technology, their heat transfer performance is not well-known, and the available literature specific to microchannel heat transfer is limited. The test facility (described in detail in Chapter 3) generates data which are used in the development of such correlations. Therefore, understanding previous work is of paramount importance in ensuring accuracy and avoiding duplication of effort. Only a small sample of general heat transfer correlations are presented, and the reader is referred to Luhrs (1994) and Dobson (1994) for a more comprehensive and thorough discussion (Luhrs, 1994), (Dobson, 1994).

2.2 Horizontal Tube Condensation Heat Transfer Coefficients

The first heat transfer coefficient correlations developed were limited in application and relied on many assumptions. Today, work continues in an effort to develop general theoretical and experimental correlations that do not rely on many assumptions.

For refrigeration systems, initial studies were done using R-12 and R-22 since these were the most widely used refrigerants in industry. Early two-phase flow studies examined individual phases separately, and Akers et. al. derived an expression for the mass flow of the refrigerant as a combination of the two phases:
where $G$ is mass flux, $\rho$ is density, and the subscripts "L" and "v" refer to liquid and vapor, respectively. This was subsequently used to determine the Nusselt number correlation:

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

where Re is Reynold's Number, the ratio of inertial and viscous forces, and Pr is the Prandtl Number which compares momentum and thermal diffusivities (Akers, 1959).

Azer, Abis, and Swearingen (1971) used empirical data to determine a correlation for the condensation of R-12 (Azer, 1971). The result utilizes the Lockhart-Martinelli (L-M) (1949) parameter to yield

\[
Nu = 0.039 \text{Pr}_{L}^{0.337} \text{Re}_{L}^{0.90} \frac{x^{0.90}}{(4.67 - x)} \left( \frac{\mu_{V}}{\mu_{L}} \right) \left( \frac{\rho_{L}}{\rho_{V}} \right)^{1/2}
\]  

where $x$ is quality and $\mu$ is dynamic viscosity. This expression is not in close agreement (±30%) with that developed by Bae, et. al. in 1971. Bae used the momentum/heat transfer analogy to generate an annular flow model whose results agree within 10% of the experimental results he obtained (Bae, 1971). Bae's correlation relies on many assumptions, making it applicable only to a small range of conditions.

A vertical tube was used by Cavallini and Zecchin (1974) to investigate condensation at high velocities. Under these conditions, surface tension effects dominate over buoyancy forces, making the correlation valid for tubes of all orientations (including horizontal). Similar to Akers, Cavallini and Zecchin used a mean Reynold's number for the refrigerant, treating it as a weighted average of the liquid and vapor phase Reynold's numbers (Cavallini, 1974):

\[
\text{Re}_{m} = \text{Re}_{v} \left( \frac{\mu_{V}}{\mu_{L}} \right) \left( \frac{\rho_{L}}{\rho_{V}} \right)^{1/2} + \text{Re}_{L}
\]  

6
Liquid and vapor properties 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

\[ \text{Nu}_m = 0.05 \, \text{Re}_m^{0.80} \, \text{Pr}_L^{0.33}, \quad (2.5) \]

which correlates with experimental R-12 data by less than 15% error.

Correlations involving annular flow in the turbulent regime were developed which agree well with experimental data from many sources, including the Cavallini and Zecchin study. First, Traviss, Rohsenow, and Baron (1973) utilized the turbulent flow equations derived from analysis of condensation under conditions of turbulent forced convection (Traviss, 1973). They used conservation of momentum and the von Karman universal velocity distribution to give

\[ \text{Nu} = F(X_{tt}) \frac{\text{Pr}_L \, \text{Re}_L^{0.90}}{F_2}, \quad (2.6) \]

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

\[ F(X_{tt}) = 0.15 \left( \frac{1}{X_{tt}} + \frac{2.85}{X_{tt}^{0.476}} \right), \quad (2.7) \]

and the L-M parameter can be found by

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

The correlation in (2.6) depends on the refrigerant flow regime. Traviss, et. al. evaluated expressions for the constant, \( F_2 \), using the Reynold's number of the refrigerant liquid entering the condenser:

\[ F_2 = 0.707 \, \text{Pr}_L \, \text{Re}_L^{1/2} \quad \text{Re}_L < 50, \quad (2.9a) \]
\[ F_2 = 5\text{Pr}_L + 5\ln[1 + \text{Pr}_L(0.09636 \, \text{Re}_L^{0.585} - 1)] \quad 50 \leq \text{Re}_L < 1125, \quad (2.9b) \]
\[ F_2 = 5\text{Pr}_L + 5\ln(1+5\text{Pr}_L) + 2.5\ln(0.00313 \, \text{Re}_L^{0.812}) \quad \text{Re}_L > 1125. \quad (2.9c) \]
Shah (1979) took an expression for the liquid heat transfer coefficient and scaled it to produce a two-phase correlation, proposing that this two-phase coefficient is dependent only on the condensation number and the Froude number. This purely experimental relationship is best for $Re_p>3000$, and is described by

$$\overline{h}_{lg} = \overline{h}_L \left[ (1-x)^{0.80} + \frac{3.8x^{0.76}(1-x)^{0.04}}{Pr^{0.38}} \right], \quad (2.10)$$

where $Pr$ is the reduced pressure of the refrigerant (Shah, 1979).

The correlations described above use expressions for the liquid and vapor Reynolds's numbers. These values are calculated with the quality of the refrigerant at the tube inlet in the following expressions:

$$Re_L = \frac{GD(1-x)}{\mu_L} \quad (2.11)$$

and

$$Re_v = \frac{GDx}{\mu_v}. \quad (2.12)$$

Also, the Prandtl number for the liquid refrigerant, $Pr_L$, is used in the expressions. This non-dimensional number is defined as

$$Pr_L = \frac{c_p L \mu_L}{k_L} = \frac{\nu_L}{\alpha_L} \quad (2.13)$$

where $c_p$ is the specific heat at constant pressure, $k$ is thermal conductivity, $\nu$ is kinematic viscosity, and $\alpha$ is thermal diffusivity.

A two-phase, smooth-tube correlation, $\overline{h}_{lg}$, was developed by Boyko and Krukhilin (1967) which uses the independent liquid and vapor phases in the following expression (Boyko, 1967):
\[ \bar{h}_g = 0.024 \frac{k_L}{D} \text{Re}_{Lo}^{0.80} \text{Pr}_L^{0.43} \left\{ \left[ 1 + \left( \frac{\rho_{lv} - \rho_v}{\rho_v} \right) \right]_\text{in}^{1/2} + \left[ 1 + \left( \frac{\rho_{lv} - \rho_v}{\rho_v} \right) \right]_\text{out}^{1/2} \right\}, \quad (2.14) \]

where,

\[ \text{Re}_{Lo} = \frac{GD}{\mu_L}. \quad (2.15) \]

Luu (1979) developed a geometric correction factor for the above expression that accounts for enhanced tubes with internal fins, where \( W \) is the fin width and \( f \) is the fin height (Luu, 1979):

\[ \bar{h}_{g,f} = \bar{h}_g \left( \frac{f^2}{WD_l} \right)^{-0.22}. \quad (2.16) \]

### 2.3 Small Tube Studies

Limited research literature is available for microchannel tubing, but it has been shown that the reduction in refrigerant charge realized with microchannel tubes is significant (Goodremote, 1988). Also, the pressure drop across the condenser and the overall size are reduced when compared with serpentine and plate-fin models.

Furthermore, experimental testing has shown 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 (Struss, 1989).

Finally, R-12 was used to show 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 (Sugihara, 1990). 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.

### 2.4 Results of Literature Search

Heat exchangers which use microchannel tubes appear to be a promising technology, and literature describing their performance is limited. Therefore, data are
needed to aid in the understanding of the physical mechanisms which drive their performance.
3. DESIGN AND CONSTRUCTION OF THE EXPERIMENTAL FACILITY

3.1 Overview of Test Facility

An experimental facility is needed that is capable of producing accurate, repeatable data for the steady-state performance of microchannel heat exchanger tubes. A single-tube facility has been built and tested, and temperature, pressure, and mass flow rate data are being collected. The data are then used to calculate the following performance parameters:

- refrigerant and air heat transfer coefficients
- single-tube refrigerant pressure drop
- inner-surface roughness.

To emulate conditions experienced by a microchannel tube in service, whose current primary application is mobile air-conditioning systems, refrigerant is used with air in crossflow. A refrigerant loop and an air loop intersect at the test section, and their respective minimum operating capabilities are defined by single-tube requirements calculated as a fraction of full-condenser requirements.

3.1.1 Design Philosophy

Throughout the design and construction of the test facility, the full range of operating requirements for microchannel tubes was considered a minimum set of operating requirements for the experimental facility. That is, a versatile, single-tube facility is desired that can be used for other single-tube experiments in the future, and components were selected to exceed the operating requirements of a microchannel tube. The design of the refrigerant and air loops reflects this philosophy. Replacement of the microchannel tube with another type of tube is feasible with only the fabrication of an appropriate transition section, and the air-side transition can be easily modified in a very short time. The design of these transitions permits versatility while inherently improving data repeatability by conditioning the flow of both refrigerant and air.

The charge inventory of the experimental facility is kept to a minimum by minimizing the lengths of transport tubing (especially those tubes containing liquid) and creatively designing to eliminate high-charge components like bladder/accumulators.
3.1.2 Design Criteria

The minimum performance characteristics of the test facility are defined by single-tube requirements calculated as a fraction of full-condenser requirements provided in the literature (Struss, 1989). Table 3.1 compares the requirements of a full condenser and a single tube. The single tube requirements were used as guidelines for component sizing.

However, final component selection was based on accuracy and cost where accuracy is specified and cost is minimized. To determine the required accuracy of individual components, an accuracy of ±3% was specified for the energy balance between the refrigerant and air heat transfer rates, and the desired accuracy of individual components was then calculated. In many cases, the component accuracy is the best available, far exceeding minimum requirements. In order to obtain accurate data, accurate properties are needed. For this reason, a pure refrigerant must be used that is not contaminated with oil, eliminating the possibility of using a compressor. Therefore, pure refrigerant R-134a is circulated with a positive displacement pump.

While the selection of transducers, flowmeters, and temperature measuring devices is defined by operating characteristics and desired accuracy, the selection of heaters, coolers, and other active devices is dominated by both operating characteristics and response time. That is, heaters and coolers must have enough capacity to change the operating conditions of the facility in a reasonable amount of time. Fifteen minutes was chosen as the maximum time available to reach any obtainable condition from the start-up mode where everything is cold.

3.1.3 General Discussion of Refrigerant and Air Loops

The test facility consists of a refrigerant loop and an air loop which intersect at the test section. Figure 3.1 is a schematic of the facility. The refrigerant loop offers control over refrigerant quality, mass flow rate, and pressure and temperature at the test section inlet. Air mass flow rate and air temperature are the parameters controlled in the air loop. Properties are evaluated for both refrigerant and air before and after test section, and these properties allow for the determination of enthalpies and heat transfer rates which are used to determine heat transfer coefficients.
### 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 lb_m/min 300-500 lb_m/hr 0.0379-0.0631 kg/s 2.27-3.79 kg/min</td>
<td>0.50-0.833 lb_m/min 30-50 lb_m/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 lb_m/min 0.467-1.87 lb_m/s 12-57 kg/min 0.20-0.95 kg/s</td>
<td>0.934-4.11 lb_m/min 0.0156-0.0685 lb_m/s 0.06-1.8 kg/min 0.001-0.030 kg/s</td>
</tr>
</tbody>
</table>

3.1.3.1 Refrigerant Loop

Refrigerant R-134a is circulated through the loop using a positive displacement magnetic gear pump. Flow from the pump immediately enters the Micro-Motion™ mass flow meter which operates on the coriolis principle. The mass flow meter outputs a voltage corresponding to the mass flow. The voltage is received by a Powers™ process.
Figure 3.1: Sub-Assembly Test Facility Schematic
controller which utilizes a proportional-integral-derivative (PID) control algorithm to output a signal to the pump controller, ultimately controlling the pump speed and mass flow rate.

Next, the refrigerant enters the Enthalpy Setting Tank (EST). With a fixed length of coil submersed in the liquid inside the tank, control of the exiting refrigerant state is accomplished by controlling the temperature of the liquid in the tank. Thus, one can control the quality when in the two-phase region and also the temperature of subcooled liquid and superheated vapor. The EST itself rests in a liquid bath contained by another tank, the Guard Tank (GT). By keeping the liquid temperature in the Guard Tank the same as that in the EST, the GT prevents heat transfer from the EST to the surroundings which is important when operating in the two-phase region. The EST is discussed in detail later in this document.

The hot refrigerant is cooled by air in crossflow as it passes through the single microchannel tube test section.

Because the refrigerant pump requires subcooled liquid on the suction side, a heat exchanger is needed to remove heat from the refrigerant which may not always be subcooled at the test section exit. This heat exchanger is referred to as the "after-condenser" and imparts a large amount of subcooling (about 40 °C at 225 psi) on the refrigerant, virtually eliminating the possibility of pump cavitation.

The Pressure Regulating Tank (PRT) controls the refrigerant pressure in the loop by controlling the temperature of a pressure vessel which contains two-phase refrigerant at all times. The pressure vessel is immersed in a liquid bath whose temperature is precisely controlled. The PRT is discussed in detail later in this document.

3.1.3.2 Air Loop

Air is circulated through the air loop by a centrifugal blower controlled by a Powers™ process controller using a PID control algorithm. The controller adjusts the blower motor voltage by controlling on an input signal from a pressure transducer which measures the pressure difference in a venturi flow meter.

The air temperature is controlled by a cooler and heater. The cooler is designed to remove heat from the air which has become hot at the test section, ultimately cooling the air below the temperature desired at the inlet to the test section. A PID-controlled heater is then used to add heat to the air, thus controlling the air temperature at the test section inlet.
After the temperature setting devices, the air flow is conditioned using screens and flow straighteners before passing over the test section.

All air-side components described in this section are discussed in detail later in this document.

### 3.2 Discussion of Major Subsystems

A previous Air Conditioning and Refrigeration Center (ACRC) report contains a more detailed discussion of many subsystems not related to the air loop; this document briefly discusses these subsystems and others for completeness (Luhrs, 1994). A detailed discussion of the air loop and its components follows at the conclusion of this section.

#### 3.2.1 Refrigerant Loop Construction

**3.2.1.1 General Refrigerant Loop Construction**

The refrigerant loop which circulates pure R-134a is constructed mainly of 1/4 in. rigid copper tubing, chosen for its structural integrity and connectivity advantages. Unlike Aluminum or flexible hose lengths, copper tubing can either be safely and easily brazed, soldered, or conveniently connected with compression fittings. Permanent joints in the loop are brazed while compression fittings are used for those joints which may require removal. The total refrigerant charge in the loop is about one pound.

The loop features a pressure relief valve with a burst pressure 425 psi. This valve is connected to an evacuated pressure vessel which would capture the refrigerant in the event of a burst valve.

Brass ball valves are used throughout the loop to separate sections of the loop for easy maintenance and test section removal. R-134a is corrosive to many polymer packing materials but not to Buna-N and often not to Teflon™ (PTFE). The refrigerant valves used are rated at 500 psi (100 psi above maximum stand capability) and are packed with PTFE, and no valve leaks have been detected to this date.

The refrigerant loop is insulated with 1/2 in. thick polyethylene pipe insulation with a 1/4 in. inner diameter. This insulation is not necessary to the operation of the facility, but was installed to improve temperature control and prevent condensation.
3.2.1.2 Refrigerant-Side Data Acquisition

Refrigerant temperatures and pressures are measured for both controlling the loop and data acquisition.

Figure 3.2 illustrates the typical refrigerant loop temperature and pressure taps. The pressure tap consists of a 1/32 in. diameter hole drilled perpendicular to the tube wall. In order to drill the hole, a short section of 1/8 in. inner diameter tube is brazed to the outer wall of the tube concentric with the designated spot for the hole. This tube serves as a guide tube for a section of 1/8 in. outer diameter tube which, in turn, serves as a guide tube for the drill bit. Once the hole is drilled, the guide tubes are brazed together and connected to a pressure transducer using compression fittings. Absolute pressure is measured before the pump, EST, and test section, and a differential pressure is also measured across the test section.

Determination of refrigerant temperature is accomplished using ungrounded, sheathed thermocouple probes installed in the copper tubing using compression fittings as shown in Figure 3.2. A 1/4 in. tee is installed in the refrigerant line and connects with a 1/4 in. sweat to female 1/8 in. nominal pipe thread (NPT) fitting. A 1/8 in. NPT to 1/16 in. compression fitting is then used to seal the thermocouple probe. Special limits of error thermocouple connecting wire is used to carry the signal to the data acquisition which is discussed in detail later in this document.

Finally, refrigerant mass flow is measured as discussed in the next section.

3.2.1.3 Refrigerant Flow: Measurement and Control

A positive displacement, magnetic gear, refrigerant pump is used to provide flow rather than a compressor, because pure R-134a can be used with the pump. The compressor requires oil to be dissolved in the refrigerant, making accurate thermophysical property determination difficult. Moreover, steady-state tests prohibit cycling which is common in most compressors. Property considerations combined with limited availability of variable displacement compressors suggested the use of a pump.

Refrigerant mass flow rate is a fundamental parameter needed to calculate a heat transfer rate for the refrigerant at the test section. To this end, a Micro-Motion Flow Meter and Transmitter are used to measure refrigerant mass flow rate. Operating on the coriolis effect, the transmitter outputs a 4-20 milliamp (mA) signal which is converted to a voltage for the purposes of data acquisition. This conversion is accomplished using a resistor with an extremely small temperature coefficient whose resistance was
Figure 3.2: Refrigerant Side Pressure and Temperature Taps
accurately measured with a Hewlett-Packard 34401A Multimeter. As long as the resistance is known, the small temperature coefficient ensures that the element resistance does not change significantly with temperature, implying that the current to voltage transition remains accurate.

The flow meter is supplied with a National Institute of Standards and Technology (NIST) traceable calibration curve and is accurate within 0.5% percent when operating above twenty percent of the maximum rated flow.

The flow meter provides the feedback for the refrigerant mass flow control loop. The flow meter output is accepted by a PID controller which attempts to control the flow to some preset value. The PID outputs a control signal to the pump controller which uses this signal to vary the power to the pump. The pump speed is altered accordingly, and the control loop is complete. Both the PID controller and pump controller can be readily switched to manual control if so desired.

3.2.1.4 Test Section and Test Section Installation: Refrigerant Transition Sections

To validate comparisons between test sections with microchannel ports of different geometries, the refrigerant flow at the inlet to the test section must be controlled and repeatable. Any variables or interactions that cannot be measured must be controlled. Furthermore, the design of the refrigerant-side test section transitions must provide proper sealing and serviceability without jeopardizing the accuracy of the data.

For these reasons, the transition from 1/4 in. copper tubing to the ovular microchannel tube is accomplished in three steps: (a) expansion of the copper tube to a larger diameter, (b) introduction of a flow conditioning chamber which also accomplishes the change in geometry, and (c) final sealing on the microchannel tube. Figure 3.3 illustrates the transition sections used at both the inlet and outlet of the test section. The refrigerant enters the expansion chamber section and encounters a screen which provides a pressure drop, ensuring that the expansion section fills completely, implying even flow of refrigerant into the contraction section. A Buna-N O-ring seals the expansion section to the contraction section. The contraction section destroys existing boundary layers and provides a smooth transition to the microchannel tube by contracting the flow to the inner dimensions of the tube. Finally, the O-ring plate provides a seal between the contraction section, the tube, and the O-ring plate itself, completing the transition.
The pressure and temperature measurements before the test section are integrated into the design of the refrigerant transition section. These measurements are used for property and refrigerant state determination as well as for control. The pressure and temperature taps are in the large chamber of the contraction section and are sealed using compression fittings whose NPT-side screws directly into the contraction section itself. The threads are sealed with refrigerant-safe epoxy.

3.2.2 Pressure Regulating Tank

Without the use of a compressor which is typically used to control not only flow rate but condensing pressure, an alternate means of controlling pressure is required. Figure 3.4 shows the system used to regulate the loop pressure, the Pressure Regulating Tank (PRT). The principle behind the design of the system is that the temperature and pressure of a saturated liquid are dependent. Two-phase R-134a is contained in a pressure vessel immersed in a propylene glycol bath. A sight glass connected to the top and bottom of the vessel allows the operator to view the liquid level in the pressure vessel from outside the tank. By precisely controlling the temperature of the bath, one also controls the refrigerant pressure inside the pressure vessel. This pressure vessel is then connected to the rest of the loop as shown in Figure 3.4. The pressure vessel is sized so that as refrigerant conditions in the loop change from subcooled to superheat, two-phase refrigerant always remains in the pressure vessel.

Cooling lines carrying 13 °C ethylene glycol remove more heat from the tank than that dissipated by the circulation pumps. The additional heat necessary to keep the tank at a constant temperature is provided by a 2.5 kW immersion heater controlled by a PID controller using a thermistor as feedback.

Because great care has been taken to ensure that the bath is at a constant and uniform temperature, the bath is also used as the reference for thermocouple circuits everywhere in the stand. This setup is discussed in detail in section 3.2.7.2.

3.2.3 Enthalpy Setting Tank

Aptly named, the purpose of the Enthalpy Setting Tank (EST) is to set the phase and temperature of the refrigerant immediately before it enters the test section. Figure 3.5 illustrates the EST configuration. As shown in the figure, there are actually two tanks: an inner and an outer tank both filled with a propylene glycol bath.
Figure 3.4: Pressure Regulating Tank
The cold refrigerant passes through heat exchanger coils in the inner tank which is at an elevated temperature relative to the refrigerant. The temperature of the glycol bath, the refrigerant inlet temperature, and refrigerant mass flow rate affect the refrigerant outlet temperature and quality. By altering the glycol bath temperature in the tank, a wide range of refrigerant conditions are attainable at the entrance to the test section. The inner tank has two circulation pumps whose power is removed by the circulating refrigerant even at the lowest refrigerant mass flow rates. Additional heat is provided to control the tank temperature using a six kilowatt immersion heater with the glycol bath temperature as the feedback for the heater control loop. The EST configuration offers the ability to accurately determine the quality of the exiting refrigerant in the event that it is two-phase. This is accomplished in two steps: (a) measuring the temperature and pressure of the subcooled refrigerant immediately before it enters the tank to determine the enthalpy, and (b) measuring the total power input to the tank from the heater and the circulation pumps. The refrigerant exit quality can then be determined with knowledge from these two steps.

However, in order for this scheme to work properly, heat loss to the surroundings is unacceptable and would represent an error in the measurement of power input to the refrigerant. For this reason, the entire inner tank is itself insulated with neoprene and immersed in a propylene glycol bath contained within a 55 gallon stainless steel drum. The temperature of the glycol in this outer tank is controlled by an immersion heater with a temperature difference between the inner and outer tank as the feedback in the heater control loop. The controller adjusts power to the immersion heater to keep the temperature difference between the tanks equal to zero. With no temperature difference between the inner and outer tanks, there is no impetus for heat flow. Both tanks are well-mixed and well-insulated on the top.

3.2.4 Propylene Glycol Bath Mixing: Submersible Pumps

Submersible pumps are used in the Pressure Regulating Tank, Enthalpy Setting Tank, and Guard Tank to ensure even temperature distribution throughout the glycol baths. There are two pumps in each bath, positioned so that there is fluid flow over the heater and also in the vertical direction to eliminate vertical temperature stratification due to buoyancy forces. All pumps are hermetically sealed with epoxy and are rated at three gallons per minute and a maximum temperature of 160 °F. This temperature corresponds to a saturation pressure of about 315 psi for R-134a. Three-gallon-per-minute pumps were chosen, because a limit switch inside the pump is tied to Class A
Figure 3.5: Enthalpy Setting Tank Scheme
insulation which results in the manufacturer's temperature rating of 160 °F. Submersible pumps with larger flow rates have lower temperature ratings (until one gets higher than 30 gpm). Instead of searching for pumps of different design that would meet the temperature and flow requirements, two smaller pumps are used in each tank.

3.2.5 Power Measurement

As previously discussed, the power added to the Enthalpy Setting Tank by both the submersible pumps and the immersion heater must be measured in order to calculate two-phase refrigerant quality at the test section inlet. A commercially available watt transducer is used to measure the power to the submersible pumps, and a prototype design power measurement board accurately measures the power to the six kilowatt immersion heater.

The submersible pump watt transducer is an Ohio Semitronics® Model GW5 with a stated accuracy of 0.2% of the reading plus or minus 0.04% of full-scale. The watt transducer was calibrated and Figure 3.6 illustrates the results plotted against the manufacturer's specifications. A series of resistors with high power ratings were used with a high voltage power source, and the voltage and current in these resistors was accurately measured with a Hewlett-Packard 34401A Multimeter. Power is calculated as the convolution of voltage and current. One can see that the calibration agrees well with the manufacturer's specification (within about 1%), yielding a slope of about 1587 Watts per volt output.

The prototype power measurement circuit was developed for precision and accuracy in high power applications. There are two main components: an isolation amplifier circuit, and a digital signal processor. Figure 3.7a is a schematic of the isolation amplifier circuit while 3.7b illustrates the digital signal processor.

The isolation amplifier circuit is responsible for measuring the instantaneous current and voltage and accurately scaling down these measurements to something appropriate for the analog to digital (A/D) converter. By helping to eliminate ground loops and "isolating" the downstream circuitry, the isolation amplifier offers intrinsic protection to the expensive circuitry downstream. Current is calculated by measuring the voltage across a low-resistance, low-temperature-coefficient current shunt in series with a high-power line. The resistance of the shunt is well-known, and the voltage drop across the shunt is used to calculate current with Ohm's Law. Voltage is measured directly across two power lines. The isolation amplifiers illustrated in Figure 3.7a were calibrated using a Fluke® 5101B Calibrated Voltage Source whose accuracy for
Submersible Pump Watt Transducer Calibration

Figure 3.6: Submersible Pump Watt Transducer Calibration: Power Through Transducer versus Transducer Output Voltage

<table>
<thead>
<tr>
<th>Power (Watts)</th>
<th>Watt Transducer Output Voltage (Volts)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>100</td>
<td>0.05</td>
</tr>
<tr>
<td>200</td>
<td>0.1</td>
</tr>
<tr>
<td>300</td>
<td>0.15</td>
</tr>
<tr>
<td>400</td>
<td>0.2</td>
</tr>
<tr>
<td>500</td>
<td>0.25</td>
</tr>
</tbody>
</table>

Measured Power $y = m1 \times (M0)$

<table>
<thead>
<tr>
<th>Value</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>m1</td>
<td>1587.3873585</td>
</tr>
<tr>
<td>Chisq</td>
<td>33.9809847</td>
</tr>
<tr>
<td>R</td>
<td>0.99992271292</td>
</tr>
</tbody>
</table>
Figure 3.7a: Power Measurement System - Analog Side
Figure 3.7b: Power Measurement System - Digital Side
providing AC power is calculated as the sum of the following errors: 0.05% of output, 0.005% of full-scale, and 50 μV baseline. This is extremely accurate, and the calibration voltages were measured with the Hewlett-Packard as a check. The results of the calibration for both amplifiers are shown in Figure 3.8. One notices that the residual in both cases is unity, indicating that the curve-fit is "exactly" linear.

Figure 3.7b shows the system which uses signals from the isolation amplifiers to calculate power on a continuous basis. There is an A/D converter for both current and voltage input signals. An anti-aliasing filter insures that the harmonics of 60Hz generated in the transformation of the sine wave to a digital square wave are not higher than the 5760 Hz sampling frequency. 5760 Hz is used as a sampling frequency, because it approaches the upper limit of the interrupt procedure time limited by digital signal processor speed. A phase-locked-loop ensures that current and voltage are sampled exactly at the maximum amplitude of the sinusoidal power wave to achieve maximum accuracy. Convolution and integration using Simpson's Rule are accomplished in the digital signal processor. Two of the four user-selectable outputs readable by a data acquisition system are calculated average power: one averaged over three seconds and the other averaged over fifteen seconds. Additionally, an RS-232 interface allows the operator to collect digital data and also analog outputs directly using a personal computer. The overall accuracy of the power measurement system is designed to be 0.1%.

The power measurement system was calibrated using the same Fluke 5101B that was used to calibrate the isolation amplifiers, and Figure 3.9 illustrates the results. An analog echo chip (1QC11B2101 ANLOG ECHO) was used so that the A/D converters for current and voltage could be calibrated separately. One notices that the A/D converter used for measuring voltage saturates around one volt input, indicating that the chip is faulty.

Presently, the power measurement system is not completely installed in the facility; a new system is being fabricated which has been designed with slight improvements to some design parameters not discussed above.

3.2.6 Electrical and Control Circuit Design

This section attempts to detail the design and philosophy of the wiring and control circuits of the test facility. Figure 3.10 is a block diagram which shows the major control loops of the facility. Basically, there is a controller for the refrigerant pump, blower, and every heater. Appendix A contains more detailed drawings for each individual control
Isolational Amplifier Calibration for Voltage Measurement

\[ y = 0.0020745 - 0.0060437x \quad R=1 \]

Isolation Amplifier Calibration for Current Measurement

\[ y = -0.11702 - 4.9927x \quad R=1 \]

Figure 3.8: Isolation Amplifier Calibrations for Power Measurement System
Output Voltage versus Input Voltage Calibration Using Analog Echo Chip for Current Chip

\[ Y = M_0 + M_1^*X \]
\[ M_0 = -0.056283645277 \]
\[ M_1 = 0.93546644069 \]
\[ R = 0.99998872356 \]

Output Voltage versus Input Voltage Calibration Using Analog Echo Chip for Voltage Chip

\[ Y = M_0 + M_1^*X \]
\[ M_0 = -0.0025112535582 \]
\[ M_1 = 0.84431545616 \]
\[ R = 0.99999057288 \]

Figure 3.9: Power Measurement Board Calibration Using Analog Echo Chip for Current and Voltage Chip
loop. Safety of both the operator and the facility are primary considerations that yield the present design.

3.2.6.1 Control Loop Discussion

One notices from Figure 3.10 that power for every heater passes through three separate relays before reaching the heater. The first of these relays is the control relay which is powered by a signal from the PID controller. The relays act as switches; that is, when the controller sends a signal to the relay, the relay allows power to pass to the next circuit element (depending on the relay configuration). Also for every heater is a high temperature alarm relay which allows power to the next circuit element as long as the process variable to the PID controller has not exceeded some value set by the operator. Finally, in the case of the immersion heaters for the glycol baths, there is a relay which is powered with voltage created by a closed level switch immersed in the bath itself. This ensures that power does not reach the heater in the event that the liquid level has dropped below the top of the heating element. For the air heater, the final alarm relay is powered only when there is sufficient airflow to carry the heated air to the temperature measuring device downstream. Therefore, no airflow indicates no power to the air heater.

The refrigerant pump and blower control loops operate on the same principle. The PID controller accepts a process variable from a flow meter (Micro-Motion™ Flow Meter for refrigerant and pressure drop across a venturi for air) and outputs a control signal to a separate controller which adjusts power to the pump or blower.

3.2.6.2 Proportional-Integral-Derivative (PID) Controllers

Powers™ Process Controllers are used as the PID controllers in the facility. Voltage, current, and thermocouple inputs are possible. Voltage, current, and solid-state-relay (SSR) outputs are possible. One can choose to use proportional (P), proportional-integral (PI), PID control, or on/off control. Wherever possible, the facility uses PID control.

As discussed above, most controllers are used in conjunction with relays, therefore, the SSR output is a common control output. Appendix A contains a listing of controller configurations for the test facility which includes but is not limited to process variable type, output type, and alarm settings. Most alarm relays are themselves mechanical relays in the controller through which power (DC or AC) is jumpered to the
Figure 3.10: Test Facility Control Wiring Schematic
electrical alarm relay which transmits power to the heater (or other application). However, a resistor-capacitor (RC) network across the terminals of the mechanical relay to prevent inductive kickback creates problems when trying to switch AC power. Specifically, the capacitor of the RC network passes enough AC power to power the SSR's at all times (whether the relay is open or closed). Because inductive kickback prevention is the purpose of the snubber circuit, and because our resistive heaters have a very low inductance, these circuits have been removed, solving the problem.

3.2.6.3 Standard Practice: Typical Wiring and Power Arrangements

Safety, functionality, neatness, and uniformity are factors that guided the design of the facility wiring. Power supply mounting design and PID controller boxes are standardized throughout the facility.

For example, safety dictates that all wiring connections are made with approved connectors and shrink-wrapped to avoid accidental contact. Wiring layouts are carefully considered to ensure safe design (i.e., without ground loops, etc.), and breakers and properly sized fuses are used everywhere possible. PID controllers power up in alarm condition to ensure that nothing in the facility is activated without the acknowledgment of the operator. Relays are enclosed by aluminum chassis boxes and mounted on aluminum plates held away from the chassis box with ceramic isolators. This design decreases potential exposure and also provides adequate heat rejection.

Functionality is accomplished not only by choosing the correct size wire while considering overcurrent protection for components, but also by shielding wires from electrical noise which is prevalent in the lab. All wires that can possibly be shielded have been shielded, including AC power wires encased in flexible metal conduit grounded to earth according to the National Fire Protection Agency’s National Electrical Code, Article 345 (Agency, 1993). Any extra length on small AC power cords is coiled in a serpentine bundle so that the individual coils help cancel each other's electric field. Metal boxes containing power supplies and PID controllers are earth-grounded. Ground potentials were checked at different 120 VAC outlets throughout the lab, and alarmingly high potentials prompted design changes for the use of one, common outlet ground. Standardized quick-connect connections are used for all pressure transducers so that they may be easily removed and re-calibrated. For every heater, there is a small neon lamp wired across the heater mounted immediately next to the heater PID controller.
These lamps are an important, positive indication that power is getting through all the relays potentially blocking its path to the heater.

Neatness and uniformity are also accomplished with standardized connectors encased in shrink-wrap. Wiring is bundled together in convenient locations and encased in split-loom wire guard, and separate bundles are gathered with Panduit® cable-ties. As mentioned above, all 120 VAC power in the facility is switched by two, centrally located power strips to which other power strips plug into. This feature is for safety and functionality as well as convenience.

DC power supplies are mounted in aluminum chassis boxes while PID controllers are mounted in larger, steel boxes. Power supply outputs are run to terminal strips (one strip for each output) and sometimes also to banana plug jacks. Each separately mounted power supply or PID controller box is appropriately fused and guarded with a lighted rocker switch. Romex® connectors are used to secure incoming and outgoing wires through the side of each box.

3.2.7 Data Acquisition, Instrumentation, and Zone Boxes

Because the calculation of heat transfer coefficients relies on many separate measurements whose errors propagate rapidly during calculations, individual measurement accuracy is of paramount importance when taking data. This section attempts to detail some of the painstaking efforts undergone to design and build an accurate test facility.

As a general rule, when two temperatures or pressures are needed for one calculation as a difference or delta, one absolute and one delta measurement are taken to reduce the uncertainty when compared with taking two separate absolute measurements. For example, if the operator wishes to know the pressure difference between the upstream and throat pressure taps of a venturi, one absolute pressure measurement is made upstream, and a pressure difference is measured between the upstream tap and the throat tap, allowing the throat pressure to be calculated if necessary. Delta measurements are usually more accurate than absolute measurements, because full-scale for the delta measurement is relatively small, and measurement uncertainty is a percentage of full-scale. In this manner, all pressures are known, and accuracy is optimized.
3.2.7.1 Data Acquisition

Data acquisition is accomplished with a Fluke® 2280 DataLogger. The datalogger scans and records sensor inputs which are connected to boards installed in the rear of the machine. The boards are easily installed and removed, and there are twenty inputs per board with three channels per input: high, low, and common. Isothermal boards are used to record DC voltages and some thermocouple measurements used only for operator information. Different scan groups can be created and saved to the datalogger's memory. A scan group is a group of input channels on the datalogger selected by the operator. The scan interval and scan group(s) to be read are easily adjusted by the operator.

A Macintosh Ilse interfaces with the datalogger through Versa-Term Pro®, a software package intended for this use. Instead of the datalogger storing data to be saved on magnetic tape, the data is saved directly to the PC hard drive, where it is easily accessible through network connections.

Sensor inputs to the datalogger isothermal boards are accomplished with one bundle of twenty individually shielded wires, one wire for each input on one board. For each board, there is a separate bundle of twenty wires. These bundles come from the zone boxes (explained later in section 3.2.7).

3.2.7.2 Instrumentation: Temperature and Pressure Measurements

As previously discussed, refrigerant temperature measurements are accomplished with thermocouple probes installed in the refrigerant line with compression fittings. All other temperature measurements are accomplished with 36 gauge, type-T (copper-constantan) special-limits-of-error thermocouple wire with a rated accuracy of ±5μV or ±0.1 °C plus a percentage of the reading. Each lead is insulated with Teflon™ to prevent the leads from shorting anywhere but at the thermocouple junction. Each thermocouple is calibrated using a commercially-available constant-temperature bath and American Society of Testing and Materials (ASTM) standard, approved mercury thermometers with accuracy greater than that specified for the thermocouples themselves. More than one thermometer is required to calibrate over the usable range of the thermocouple, and stem correction factors are applied to the thermometer measurements. During calibration, thermocouples are protected from electrical noise.
which may be present in the fluid of the constant-temperature bath by isolating them in an earth-grounded stainless steel beaker which contains bath fluid.

Thermocouple measurements are referenced to a thermistor in the well-mixed glycol solution of the Pressure Regulating Tank. The datalogger input board has the capability to read thermocouple voltages and convert them to temperatures by referencing a thermistor measuring the temperature of an aluminum scanning plate built into the board itself. However, this design is based on the assumption that the aluminum scanning board is itself isothermal, and this is not the case. A power supply for the thermistor located on one side of the board creates heat, causing a temperature gradient along the scanning plate, invalidating the stated accuracy. Additionally, the built-in thermistor cannot be independently calibrated, forcing the researcher to rely on dated manufacturer's calibrations. For these reasons, a separate thermocouple reference is needed, and the constant-temperature-bath of the PRT is used, implying that all thermocouple voltage-to-temperature conversions are accomplished outside of the data acquisition system. Although the temperature of the bath may change from test to test to set the pressure of the refrigerant, during any one test the temperature is uniform and constant; the design does not depend on the absolute temperature of the bath, only that the temperature is accurately known. In fact, error in reading a thermocouple depends, in part, on the magnitude of the EMF generated, and error is further reduced by using a reference temperature that is closer to the measurement temperature than is ambient. 36 gauge thermocouple wires are immersed in the bath next to a thermistor and individually coated with Plasti-Dip® so that fluid does not travel inside the insulation of the thermocouple wire through mass diffusion and capillary action. Figure 3.11 illustrates the reference thermocouples and shows their connection in the zone box (described in detail later in section 3.2.7.3).

The thermistor is a variable-resistance temperature measurement device requiring calibration using the ASTM thermometers described above. One can measure the voltage drop across the resistive element and correlate the voltage to a temperature through the calibration curve. However, voltage drop across the element is dependent on supply voltage according to Ohm's Law. Although the power supply is accurately regulated at 5.00 volts, small fluctuations in supply voltage can lead to significant errors in calculated temperature. For this reason, both the supply voltage and the voltage drop across the thermistor are measured and recorded by the datalogger, and the ratio of these voltages is used to calculate the measured temperature. To elegantly accomplish this, a Wheatstone Bridge circuit was designed, and Figure 3.12 shows this circuit. Appendix B contains an Engineering Equation Solver® (EES) program that uses the
Figure 3.11: Zone Box and Reference Thermocouple Configuration
Our Thermists (Omega ON-970 44032) are approx. 30 kohms at room temp. At 100 °F, they are about 18.5 kohms. R3 sized to be approximately the resistance of the thermistor just below our lowest PRT bath operating temperature. Accuracy is limited to the thermometer against which we calibrate the thermistor. Resistor ohm measurements by Hewlett Packard Multimeter.

Circuit Layout - As viewed from front side

Figure 3.12: Thermistor Circuit
calibration data to calculate a curve-fit as a function of the voltage ratio of input to
output. Figure 3.13 is a graph of temperature versus thermistor output voltage
assuming an input voltage of exactly five volts.

Pressure transducers are used throughout the facility to obtain pressure
measurements. A previous document by Luhrs (1994) tabulates their position in the
facility and specifications. Although manufacturer's calibrations are provided, each
transducer is calibrated in-house as a check against damage during shipping. The
calibration curves are provided in Appendix C. The pressure transducers require 24
VDC input and all have 0-5 VDC output. The connections for the transducer excitation
and output are made with a facility-standardized connection illustrated in Appendix C.

3.2.7.3 Zone Boxes

The zone boxes implemented in the facility are basically terminal strips that provide
junctions for thermocouples and pressure transducer outputs to be read by the
datalogger. However, by design, these junctions are isothermal and the box itself is
electrically grounded. Figure 3.11 illustrates that the datalogger measures the voltage
across the two copper (positive) leads of the absolute and reference thermocouples
while the negative leads are connected at a terminal with no output. The voltage read
by the datalogger indicates the temperature difference between the absolute reading
and the reference temperature (which is known by virtue of the thermistor).

The zone boxes offer a permanent, stationary means of making fragile
thermocouple connections while at the same time making connections to the input
boards of the datalogger extremely elegant. Also, the bundles of twenty individually
shielded cables from the zone box to the datalogger offer protection from electric fields.

3.3 Air Loop Design

Circulating air while accurately regulating temperature, conditioning and mixing
flow, and measuring flow rate is a task where attention to detail is extremely important.
The centrifugal blower size has been increased from the original design, but both the old
and new configurations are discussed for completeness. Also, while upgrading the
blower, the portion of the loop immediately before and after the blower was redesigned
based on experience from the operation of the original installation.
Thermistor Calibration Results Assuming 5.0V Input

Figure 3.13: Thermistor Calibration Assuming 5.0V Input
3.3.1 General Discussion

Figure 3.14 illustrates the original ductwork configuration where air exits the blower and passes over a heater and cooler (temperature control is discussed later in section 3.3) mounted in a five-inch, galvanized steel, round duct. In an attempt to mix the stratified air, a converging-diverging (C-D) nozzle was installed (temperature stratification is also discussed later in section 3.3). The air then enters a plenum chamber and subsequently contracts to the geometry of the ductwork at the test section. However, before reaching the test section, the air flows over nine screens and a set of straw flow straighteners. At the test section, the flow is again contracted to the exact geometry of the test section to destroy boundary layers, improve flow repeatability, and reduce air temperature stratification after the test section. After the test section, the air flows through a short section of duct where temperature measurements are made, and then abruptly expands to a plenum chamber. Flow travels out of the plenum into three-inch polyvinylchloride (PVC) pipe, enters a straight section of PVC upstream of the venturi, flows through the venturi, and through another, shorter straight section to the suction side of the blower. The following sections offer greater detail on each of the air loop subsystems mentioned above.

3.3.2 Preliminary Fan Selection, Installation, and Performance

Proper fan selection requires knowledge of both velocity and pressure drop requirements, and the blower size was selected based on estimates of air-side pressure drop and the velocity requirements stated earlier. Knowing that the air loop would contain screens (to ensure a constant velocity profile across the duct), flow straighteners, and knowing the overall proposed dimensions of the ductwork, pressure drop was calculated as a function of flow rate. Appendix D contains the EES program written for this purpose. A graph of air velocity versus test section pressure drop was prepared as part of the input for the overall pressure drop program and is also attached in Appendix D. At 150 cubic feet per minute (cfm), the calculated pressure drop of the proposed air loop was about 2.6 inches of water. In the end, a 150 cfm at five inches of water pressure drop blower was selected. This blower was powered with a one-third horsepower (hP) 90 VDC motor. Figure 3.15 illustrates the old and new blower configurations (the new configuration will be discussed in detail later in section 3.3).
Figure 3.14: Ductwork Assembly
Figure 3.15: Old and New Blower Installation Schematics
Control of the motor was accomplished either with a potentiometer or with a PID controller where the venturi pressure drop was the process variable. An output signal from the PID or potentiometer was sent to a PentaPower® KBSI-240D Signal Isolator which conditioned the signal for a PentaPower® KBMM-125 Motor Controller which rectifies 120 VAC power and regulates motor voltage. A wiring diagram for the controller boards and blower is attached in Appendix D.

A test was conducted to determine if the automatic control scheme using the pressure difference across the venturi was acceptable. The test logged the pressure difference across the venturi for two conditions: (a) manual control, constant output implying constant motor voltage, and (b) PID control with venturi pressure difference as the process variable. Figure 3.16 illustrates some of the data taken for two separate flow rates. One notices that PID control offers almost as much stability as constant output, manual control, and the venturi pressure difference is still used as feedback in the control loop of the blower in the new configuration.

Statistical evaluation of airflow parameters was also investigated to determine if better control is achieved at different airflow rates. Appendix D contains a summary of the statistical evaluation of airflow parameters. Seven individual points (each consisting of 20 averaged data points) are averaged to obtain the mean values listed in the table, and the pressures listed are in Pascals. The dimensionless ratio of 95%-confidence-interval to the average-venturi-pressure-differential indicates the relative airflow control. Lower airflow rates have a higher ratio (95% CI/dPavg) than do higher airflow rates, indicating that airflow variance is a larger percentage of total airflow at small flow rates, and steps were taken (explained later) to improve the control of air mass flow rate.

Pressure oscillations were present in the airflow in the original design causing control instability, and the source was traced to a rapid expansion (32° total included angle) at the inlet to the blower. This expansion was replaced with a small plenum chamber which had a fine screen at the blower inlet to ensure filling of the chamber, and the major pressure oscillations were eliminated.

### 3.3.3 Air Temperature Control

Air temperature at the test section inlet is controlled by cooling and subsequently reheating the air upstream of the test section. The air is heated as it passes over the hot test section, work is added by the blower, and a heat exchanger is needed to remove heat from the air. Full condenser experiments in the literature indicate that a full condenser has a heat duty of about nine kW, corresponding to about 0.3 kW for the test.
Figure 3.16: Test Results of PID versus Manual Control of Blower Motor
section in the facility. A customized GE spine-fin evaporator coil is used with a designed capacity of about one kW, and instantaneous heat duty is controlled by varying the flow rate of cold ethylene glycol (about 55°F) with a gate valve installed in series with the heat exchanger. The lowest achievable air temperature to date is about 12°C or about 54°F (the cold fluid temperature), indicating that the heat exchanger effectiveness is can be nearly unity.

Very simply, the air heater was sized so that it could replace the same heat removed by the cooler if necessary. Since heat is added at the test section and the blower, the heater power required should always be less than the heat duty of the cooler, but a heater with power equal to the heat duty of the cooler was chosen as a safety factor for incorrect cooler design. Thus, a one kW coil heater was chosen, and 120 VAC input is used for wiring convenience in the lab. As explained earlier, a PID controller is used to control heater power via an SSR, and this control is necessary to compensate for minor cooling fluid temperature and flow variations. Although the heating control algorithm is PID, the heater actually cycles on and off with the PID calculating the percentage of time on and off. Because the heating element does not fill the entire ductwork cross-sectional area, the air is not heated uniformly, and proper air mixing becomes a concern; this is discussed in detail later in section 3.3. One way to alleviate this problem is to use a heater whose capacity is equal to but does not exceed the heat required to be added. Conversely, an oversized heater that cycles on and off will cause large local-temperature variations and make proper control and uniformity difficult to achieve. As part of investigating proper heater sizing, an EES program was written to evaluate heater power from 0-1000 W, resistance, and current requirements, and this is included in Appendix D.

3.3.4 Flow Conditioning

For proper fan application/installation, flow repeatability at the test section, and flow measurement in the venturi, the airflow must be conditioned to remove swirl, yield a constant velocity profile, and destroy boundary layer growth. The present section describes the flow conditioning at the aforementioned regions of the air loop.

3.3.4.1 Entrance and Exit to Blower

The entrance and exit conditions of a blower can affect blower performance. For example, inlet swirl and blocked air passages before or after the blower can adversely
affect blower performance. For this reason, care is taken to condition the flow before the blower to remove swirl and also to leave air passages as free as possible. Where a design change was not possible to eliminate adverse system effects, these effects were accounted for in the selection of the new blower.

3.3.4.2 Flow Conditioning Before Test Section

Flow conditioning before the test section was integral in the design of the entire air loop and is accomplished in four ways: screens, flow straighteners, contraction sections, and a transition section immediately preceding the test section.

Screens are used to improve flow uniformity and reduce turbulence levels, and there are nine screens installed in the ductwork between the blower and the test section. After the major ductwork dimensions had been decided, several EES programs were written to design the screen open area and thread diameter, and an optimization program is attached in Appendix D. Basically, this program uses simple gas dynamics to calculate pressure drops across the screens, the walls of the plenum, and other ductwork while varying screen parameters and accounting for changes in velocity from the plenum to the smaller hydraulic-diameter ductwork. The output is interpreted as a pressure ratio of the frictional pressure drop in the ductwork immediately preceding a screen and the pressure drop across the screen itself. A screen design was optimized so that this pressure ratio exceeded a value of ten. Stainless steel screens closely matching the designed screens were procured and installed. Additionally, there are screens after the test section to assist in keeping a constant velocity profile in a portion of ductwork with relatively small hydraulic diameter. The screen immediately preceding the test section is much finer than the rest; even though a penalty is paid in large pressure drop, a 100 by 100 mesh per linear in., 34% open area, 0.0042 in. polypropylene screen is used before the test section to ensure a constant velocity profile of air at the test section inlet. Figure 3.17 is an isometric view of the test section, illustrating the screens before and after the test section, the air-side transition (discussed later), and the thermocouples used to measure temperature before the test section.

Flow straighteners are also used to condition the flow before the test section. Unlike screens, flow straighteners are effective in removing swirl and large eddies present in the airflow (Farell, 1993). Conversely, screens are still needed, because energy losses in the flow straightener are not significant enough to affect the velocity profile. An Iso-Bundle type tube bundle straightener is used, where the diameter of the
Figure 3.17: Schematic of Test Section, Screens, Transition Section, and Thermocouple Measurements
round flow straightening tubes is less than 0.2 times the hydraulic diameter of the ductwork in which the tubes are installed, and the length of the tubes is more than twenty times their diameter as per International Organization for Standardization (ISO) Standard 5167 (Laws, 1993). The screens installed immediately before and after the flow straightening tubes serve to hold the tubes in place while improving the operation of the straightener itself by suppressing the "honeycomb wake shear layer instability responsible for downstream turbulence generation" (Farell, 1993).

Contraction sections are used in the conditioning of the airflow before the test section by destroying developing velocity boundary layers. Of course, for proper operation, the contraction itself must be at an angle greater than the slope of the boundary layer at the leading edge for any attainable friction coefficient, $C_f$. One contraction is designed in the air loop upstream of the test section and after the first plenum. Additionally, Figure 3.18 shows a dimensioned cross-section of the bottom-half of the contraction at the test section. Neoprene insulating material secured with clear packing tape is used between the hot test section and the ductwork. Physically, the tape forms the contraction, with air filling the gap between the contraction angle and the solid neoprene. This design accomplishes the desired contraction while taking advantage of the low thermal conductivity of air and neoprene (heat transfer from the test section through the air-side transition section to the ductwork is not desirable).

3.3.4.3 Flow Conditioning Before Venturi

Flow conditioning before a primary flow measurement device is imperative for proper operation of the device, because swirl and other flow non-uniformities present in the approach flow can affect the accuracy of the meter. ISO 5167 outlines several stringent recommendations for flow meter installation: (a) a required straight length after a single 90° bend of 14 times the pipe diameter (D) upstream and 6*D downstream for a venturi nozzle with a ratio of throat diameter to pipe diameter (d/D), beta, of 0.5, (b) a temperature measurement of diameter 0.03*D must be at least five straight lengths upstream, (c) the axis of the flow measurement device must be aligned with the axis of the ductwork within 1°, and other requirements listed in Chapter 6 of ISO 5167 (5167, 1980). Figure 3.19 illustrates the air loop sections before and after the venturi for both the old and new air loop setups (with the new blower, the venturi section of the air loop was rebuilt as well). In the new setup, a screen is used in conjunction with a honeycomb flow straightener at the start of the 48 in. (16*D) straight section upstream of
Figure 3.18: Cross-Section of Air-Side Transition - Dimensioned
Figure 3.19: Air Loop Sections before and after venturi
the venturi. The straightener is used to reduce swirl, although evidence exists that small swirl intensities (although magnified in the venturi throat) do not significantly affect the accuracy of the venturi flow meter (Barbin, 1981). However, nonaxisymmetric flow can produce large variations in local throat pressure, but the flow straightener installed should alleviate this problem as well (Rapier, 1981 September 9-11). Rotation vanes installed before elbows can improve exit-flow uniformity and reduce separation, but none are used presently (Flaska, 1993). The PVC pipe was cleaned and the diameter was verified as per ISO 5167 Section 6.1.5 (5167, 1980). With the new design illustrated in Figure 3.19, venturi removal is accomplished in about 30 seconds while installation and sealing takes about five minutes.

3.3.5 Air Mass Flow Measurement

For air mass flow measurement, a custom-made, short-form venturi made of PVC is used to measure purely subsonic airflow. The venturi has a classical venturi machined convergent and truncated diffuser nozzle with a total included angle of 15°, and connects to the PVC pipe through flanges with holes that match an ANSI 150# flange. The upstream pipe diameter is three inches, and the ratio of throat diameter to pipe diameter (d/D, beta, β) is 0.5. Theoretically, the diffuser truncation does not affect venturi performance, increasing only irrecoverable pressure loss. The present section outlines the use and operation of the short-form venturi differential pressure device for measuring air mass flow rate.

3.3.5.1 Sealing

For accurate air mass flow rate measurements, the air loop must be completely sealed, because any leakage (in or out) could represent an error in mass flow rate measurement (as well as cause other problems). The air in the loop is basically at atmospheric pressure except for static pressure differences caused by the blower. The sections of ductwork are flanged together, and the flanges are covered with compressible neoprene. As the flanges are bolted together, the neoprene is compressed, forming an airtight seal that was tested with soap solution. At the corners of the flanges, any openings were sealed with GE® Silicone Adhesive. At all places in the loop where sealing is expected to be permanent, silicone is used to accomplish the seal. Where the seals are expected to be frequently disturbed, duct tape is used for areas of simple geometry where joint strength preservation is necessary, and Panduit®
Duct Seal is used everywhere else. The entire loop was checked for leaks, and leak testing is standard procedure following any modifications to the loop.

3.3.5.2 Venturi

3.3.5.2.1 General Venturi Discussion

Figure 3.20 illustrates a classical venturi. Over the last twenty years, the classical venturi has been phased out by metering devices known as "flow tubes" which have the following characteristics:

- extremely short laying length compared with the classical venturi tube
- significant magnification of differential pressure
- relatively small irrecoverable pressure loss, especially when considered as a percentage of the differential (ASME, 1973).

In the present facility, the convergent of the flow tube used was manufactured to be exactly that of the classical venturi, and outlet nozzle is a truncated cone of 15° total included angle (compared with 7° total included angle for the classical venturi) (ASME, 1971). An additional difference between the flow tube used and the classical venturi is the placement of the upstream pressure tap. The tap is located axially with the approach flow in a stagnation chamber behind the convergent. This should provide a pressure measurement free of dynamic pressure effects, but the flow tube does not conform to any hydraulic shape classification found in the literature (ASME, 1973). Furthermore, because the outer diameter of the convergent is not within 1% of the value used for D as per ISO 5167 Section 6.6.1.2, the value of the beta parameter must be questioned (discussed later in section 3.3.5.2) (5167, 1980). The actual throat diameter was checked against the manufacturer's specification by averaging the diameter measurement taken in four places in the plane of the throat pressure tap, and a 0.0005 in. discrepancy was found (too small for concern).

From here forward, the flow tube presently in the test facility is referred to as a venturi.

3.3.5.2.2 Theoretical Venturi Equations

The present section outlines the theoretical equations used to calculate air mass flow rate from a differential pressure reading from the venturi. Firstly, twenty data points (pressure transducer voltages) are taken in immediate succession for the calculation of
Figure 3.20: Figure II-III-26, page 231, Fluid Meters
Their Theory and Application: Diminsional Proportions of
Classical (Herschel) Venturi Tubes With a Rough-Cast, Convergent Inlet Cone
one air mass flow rate (explained later in Chapter 4, Data Reduction). These voltages are averaged, and a differential pressure is obtained through the calibration curve of the pressure transducer. Knowledge of absolute pressure and temperature upstream is also necessary for the determination of thermophysical properties. The following equation is used for the determination of final, corrected mass flow rate:

\[ \dot{m}_{\text{fact}} = C_d \cdot F_a \cdot Y \cdot (\dot{m}_{\text{ideal, incompressible}}) \]  \hspace{1cm} (3.1)

where \( C_d \) is the discharge coefficient, \( F_a \) is the expansion factor for temperature effects in the venturi, and \( Y \) is the gas expansion factor.

Ideal, incompressible mass flow rate is given by the following expression:

\[ \dot{m}_{\text{ideal, incompressible}} = A_2 \sqrt{2 \rho (P_1 - P_2)} \sqrt{\frac{1}{1 - \beta^4}} \]  \hspace{1cm} (3.2)

where \( A_2 \) is the venturi throat cross-sectional diameter, \( \rho \) is the gas density as a function of temperature and pressure, \( P_1 \) is the upstream pressure, \( P_2 \) is the throat pressure, and \( \beta \) is the diameter ratio which for the venturi used equals 0.5.

A detailed explanation of the discharge coefficient is given in the next section.

\( F_a \) is an expansion factor for the venturi to account for thermal expansion effects. The manufacturer provided two data points which are fit to a line and linearly interpolated for the purpose of data reduction. These two points were calculated using a commercially available software package, and the equation of the line is as follows:

\[ F_a = 0.99075928 + (3.20069298 \times 10^{-5})T \]  \hspace{1cm} (3.2)

where \( T \) is temperature in Kelvin. Linear interpolation may not accurately represent the physics of the thermal expansion of this complex geometry, but the value of \( F_a \) changes 0.16% over a 50°C temperature difference, making linear interpolation feasible. The points were provided with five significant figures, but all decimal places are retained during calculations.

The coefficient \( Y \) is called the gas expansion factor and is given by
\[ Y = \left( \frac{p_2}{p_1} \right)^{\frac{\gamma}{\gamma-1}} \left( 1 - \frac{p_2}{p_1} \left( \frac{\gamma}{\gamma-1} \right) \right) \left( 1 - \beta^4 \frac{p_2}{p_1} \right)^{\frac{1}{2}} \]  

(3.3)

where \( \gamma \) is the ratio of ideal-gas specific heats \((C_p/C_v)\), and the compressibility factor, \( Z \), is unity (ASME, 1971). Although the literature outlines methods to circumvent the ideal gas assumption, a constant value of \( \gamma \) is used (Sullivan, 1984).

Combined with the discharge coefficient, these parameters yield a value for actual air mass flow rate, and all calculations are performed in Standard International (SI) units.

### 3.3.5.2.3 Venturi Discharge Coefficient

The discharge coefficient is a term used in the calculation of the actual air mass flow rate to account for all effects not already accounted for by \( F_a \) and \( Y \), and is defined as

\[ C_d = \frac{\text{actual flow rate}}{\text{theoretical flow rate}} \]  

(3.4)

where theoretical flow rate may already include \( F_a \) and \( Y \) (Benedict, 1984). It is generally agreed that the discharge coefficient is a function of Reynold's Number (Re) although ISO 5167 treats \( C_d \) as a constant above Re of \( 2 \times 10^5 \) (5167, 1980). However, at or above this Re, \( C_d \) usually is constant; the problem arises when trying to evaluate \( C_d \) at smaller Re, and Appendix E contains a summary of some \( C_d \) versus Re relations found in the literature. One is easily confused when deciphering the meaning of these different relations which are not all in agreement. For example, the manufacturer-suggested relation does not exceed a Re of \( 10^5 \) which is attainable in the present test facility, but the ISO 5167 is only valid for Re greater than \( 10^5 \); extrapolation of the manufacturer-suggested curve suggests agreement with the stated ISO 5167 value and associated uncertainty. Other relations from the literature are in disagreement with both of the aforementioned relations.

As a further complication, \( C_d \) versus Re is also highly dependent on the hydraulic shape of the flow tube, but documented relations for \( C_d \) versus Re for each hydraulic
shape are not abundant (none were found) (Halmi, 1974). For these reasons and others, it was decided to calibrate the venturi in-house.

3.3.5.2.4 Venturi Calibration

The venturi was calibrated with the facility illustrated in Figure 3.21. The facility shown was built specifically for calibrating the venturi used in the facility and was constructed of PVC. One may notice from Figure 3.21 that the scale of the calibration facility is rather large with over fourteen feet (58*D) of straight length before the venturi. The venturi was clamped in place with a Unistrut™ assembly to prevent forces arising from momentum changes from destroying the facility (the force at an elbow for a high flow rate was determined to be about 150 lbf). Although not immediately apparent from the drawing, the section of PVC piping after the venturi reaches a higher elevation than the control valve, ensuring that the entire system is filled with liquid, allowing the valve to control the flow rate. Without this setup, at lower flow rates, the water head is actually controlled by the head in the pipe immediately after the valve, destroying the accurate control attainable with the valve. At the outlet, a header pipe is installed to prevent siphoning effects from affecting flow rate control.

Water is supplied upstream of the valve at a constant pressure of about twenty feet of water (plus or minus about one inch of water), and the facility can supply water at the maximum rate of about one cubic foot per second (1 cfs). At the outlet, water is drained into a large storage tank (30' x 15' x 15') where pressure measurements of the water at the bottom of the tank can be made to determine the head in the tank.

Appendix E contains EES programs used for preliminary calculations and reduction of the calibration data. Pressure differentials in the venturi were measured with a pressure transducer calibrated before and after the series of data points for the venturi calibration were taken. An average of over 200 pressure readings is calculated to determine the venturi pressure difference for one flow rate. Volumetric flow is measured using the storage tank, time is measured using a digital stopwatch, and the temperature of the water before and after each run was measured for accurate density determination. Figures 3.22 and 3.23 illustrate the results of the calibration. Figure 3.22 is a graph of $C_d$ versus $Re$, and one notices that this relation is not in close agreement with the aforementioned relations attached in Appendix E. Specifically, at $Re$ above $10^5$, the calibrated $C_d$ has a constant value of about 0.975 while the literature specifies a value closer to 0.995. Figure 3.23 shows air mass flow rate versus the square root of
Figure 3.21: Venturi Calibration Facility
Venturi Calibration Water Mass Flow Rate versus Square Root of the Product of Density and Change in Pressure

\[ y = m_1 (M_0) \]

<table>
<thead>
<tr>
<th>Value</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>m1</td>
<td>0.0016266639005</td>
</tr>
<tr>
<td>Chisq</td>
<td>0.018424732234</td>
</tr>
<tr>
<td>R</td>
<td>0.99950850961</td>
</tr>
</tbody>
</table>

Figure 3.22: Venturi Calibration Water Mass Flow Rate versus Square Root of the Product of Density and Change in Pressure
Venturi Calibration Discharge Coefficient versus Reynolds Number

Figure 3.23: Venturi Calibration Discharge Coefficient versus Reynolds Number
the product of density and pressure difference across the venturi. As expected, the
graph is fairly linear and was made to verify the discharge coefficient data and identify
bad data points. However, the discharge coefficient graph of Figure 3.22 is much more
sensitive to small parameter variations due to the extremely small scale. The calibration
results are used in the data reduction programs to calculate actual air mass flow rate.

3.3.5.2.5 Pressure Dampers

It was found that pressure differential measurements across the venturi could be
stabilized with the use of pressure dampers, reducing uncertainty by improving the
precision of the measurement. Pressure dampers are installed in the pressure tap lines
from the venturi to the pressure transducer and basically consist of extremely tightly
packed Kleenex™. Data were taken before and after the installation of the pressure
dampers, and the results are shown in Figure 3.24 for various air mass flow rates. The
variance in the measurement is significantly reduced with the installation of the
dampers, and they are a permanent fixture in the test facility. Even with periodic
pressure disturbances of high frequency, time-averaged mean differential pressure
measurement is not affected by the frequency response of the transducer (Husain, 1984).

3.3.5.2.6 Test Section Velocity versus Venturi Pressure Differential

For the purposes of decision making and general test facility knowledge, a graph of
air velocity at the test section versus venturi pressure differential was prepared, and
Figure 3.25 illustrates the result. These results were used to help conclude that a new
blower was necessary for the facility.

3.3.6 Airflow Temperature Measurement

For accurate heat transfer rate determination, air-side energy transfer must be
calculated as a function of air mass flow rate (determined from the venturi) and average
air temperature change across the test section. However, because the test section
temperature inherently changes across its length primarily in the direction of refrigerant
flow, downstream air energy calculation becomes a difficult task.
Pressure Damping:
Venturi Delta Pressure Transducer Voltage versus Time

Figure 3.24: Venturi Delta Pressure Transducer Voltage versus Time
Velocity at test section versus Pressure drop across Venturi

Figure 3.25: Velocity at Test Section versus Delta Pressure across the Venturi
3.3.6.1 Test Section Thermocouple Configuration

Figure 3.26 illustrates the thermocouple configuration before and after the test section. The air temperature before the test section should be constant and uniform, but five absolute temperature measurements are taken upstream and averaged to find the upstream air temperature. One absolute air temperature measurement taken upstream is also used as feedback in the air heater control loop. Temperature measurements are taken downstream immediately after the test section to prevent heat loss to the surrounding ductwork from influencing accuracy (ductwork insulation is discussed in section 3.3.7). Figure 3.26 illustrates nine temperature measurements downstream, each taken as a delta temperature from its upstream counterpart (discussion of thermocouple circuiting can be found in section 2.7.2). These nine measurements are used to calculate air-side heat transfer, and the calculation algorithms are discussed in a section 4.2).

3.3.6.2 Air Temperature Uniformity Upstream of Test Section

Without a device specifically designed for mixing the air after the temperature control devices, the air temperature upstream of the test section was stratified across the direction of flow. This is extremely undesirable, and is a difficult problem to solve without redesign of the air loop. Many potential solutions were attempted, including (a) a diffuser at the inlet to the first plenum to change momentum and encourage mixing, (b) a converging-diverging (C-D) nozzle after the heater (and also before the first plenum) to force the flow to mix in an area reduction, and (c) a small axial fan placed at the exit of the C-D nozzle to further encourage mixing. With each modification to the loop, the temperature stratification at the test section was reduced, and the axial fan had by far the greatest impact to this end. Coincidentally, shortly after the stratification was reduced within thermocouple error, the blower was replaced and the air loop around the blower redesigned. Experience with air mixing was the impetus for placing the temperature control devices before the blower in the new design.
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

Figure 3.26: Air Thermocouple Grid
3.3.7 Ductwork Insulation

Because insulation is expensive, calculations were performed to determine the amount of insulation necessary to limit heat loss at the test section to an acceptable level. Acceptable heat loss was decided to be less than 0.1% of the smallest experimental heat transfer rate expected from the test section. With these criteria, an EES program was developed that could vary insulation thickness, convection heat transfer area as a function of insulation thickness, and other heat transfer parameters in order to calculate heat loss at the test section. The thermal conductivity, \( k \), of the Thermax® Insulation Board used was calculated to be \( k_{\text{THERMAX}} = 0.020 \text{ W/m-K} \) from the manufacturer's specifications of R-value. This value is about 10% lower than values obtained from the literature for insulation board (Incropera, 1990). This program is attached in Appendix F, and the results are illustrated in Figure 3.27. One notices from the graph that the heat loss is extremely small (about 0.15 W at four inches thickness), and this is primarily due to the small heat transfer area resulting from close placement of the upstream and downstream thermocouples. Presently, four inches of insulation cover the ductwork from the front plenum to the rear plenum which includes the test section area.

Materials with low thermal conductivity like insulation can take a long time to reach steady state. The time constant, \( \tau \), of the ductwork insulation was calculated using the following relations:

\[
\tau = \frac{L_c}{\alpha} \tag{3.4}
\]

\[
\alpha = \frac{k}{\rho C_p} \tag{3.5}
\]

where \( L_c \) is a characteristic length, \( \alpha \) is thermal diffusivity, \( \rho \) is density, and \( C_p \) is the specific heat at constant pressure. This calculation results in a time constant of 1.06 hours for four inches of insulation.

3.3.8 Test Section Air-Side Heat Transfer Enhancement and Final Blower Selection and Installation

The test facility was originally designed to accommodate microchannel tube test sections with louvered fin air-side heat transfer enhancements brazed to the tube.
Heat Loss versus Insulation Thickness at the Test Section

Figure 3.27: Heat Loss versus Insulation Thickness at Test Section
While in the final stages of troubleshooting the facility to refine the energy balance, the test sections were made available only without the air-side heat transfer enhancements. Keeping air-side heat transfer resistance low relative to the refrigerant-side increases the accuracy of the calculated refrigerant side heat transfer resistance, increasing the accuracy of the refrigerant heat transfer coefficient determination. Consequently, the important responsibility of reducing air-side heat transfer resistance was placed on the researchers. The design of heat transfer enhancements and the selection of a larger blower were integral, because air-side resistance is dependent on air velocity. The air-side heat transfer enhancements investigated were limited to fins. A blower size was selected based on proposed fin configuration (conversely dependent on blower selection) and estimated air-loop pressure drop after partial re-design of the air loop to reduce the pressure drop.

3.3.8.1 Fin Configuration

The theoretical performance of aluminum (Al) plate fins and accordion style fins in both copper (Cu) and Al were compared with experimental results obtained from the facility using the brazed, louvered fins. Figure 3.28 is a graph of the numerical results of the comparison, and Figure 3.29 is a schematic of the fin configurations investigated. Appendix G contains EES programs written to evaluate heat transfer parameters for each fin type. The plate-fin program automatically varies fin thickness and fin spacing (as a function of fin thickness) to avoid boundary layer interactions in order to minimize air-side resistance. As shown in Figure 3.28, an air-side resistance of 0.018 W/K (the largest resistance with the brazed, louvered fins) was chosen to be the maximum tolerable heat transfer resistance for the new fin configuration based on calculations for accuracy requirements. With a test section velocity greater than 10 m/s practically unobtainable even with pressure drop reductions and a new blower, louvered fins appear to be the only answer. It should be noted that louvered fins were not investigated theoretically, only experimentally. Each fin louver destroys existing velocity boundary layers and a new boundary layer begins at the leading edge of the louver. Because local heat transfer coefficients are smaller inside the boundary layer than outside, using louvers to reduce average boundary layer thickness improves the overall heat transfer coefficient when integrated over the entire surface.

Louvered fin stock was procured and is epoxied to the test section with Devcon™ Aluminum Liquid (F-2) 10710 epoxy which is composed of approximately 80% Al and
Air-Side Heat Transfer Resistance versus Air Velocity at Test Section, Air Temperature=298 K

Figure 3.28: Air Side Heat Transfer Resistance versus Air Velocity at Test Section
Figure 3.29: Fin Schematic

Plate Fins

 Accordian Fins

 Louvered Fins
20% epoxy. This epoxy is used because of its good heat transfer characteristics, and the fins are attached to the tube using a procedure developed in-house which applies pressure evenly to the fins, clamping them to the test section while the epoxy cures, reducing contact resistance (Dart, 1959), (Fenech, 1963), (Nho, 1989).

3.3.8.2 Final Blower Selection

As previously discussed, blower selection is based on both velocity and pressure drop requirements. The original blower was selected based on knowledge of published velocity values and calculated pressure drop while the final blower was selected based on velocity required for the selected fin configuration (as previously discussed), measured pressure drop for the system, and installation effects (Bolton, 1990 March 14-15), (Riera-Ubiergo, 1990 March 14-15), (Moss, 1990 March 14-15). A system curve of the air loop was generated with the original blower to aid in the decision of a new blower, and Figure 3.30 illustrates the results. The static pressure was measured immediately before and after the fan, and an EES program attached in Appendix G converts these measurements to a fan total static pressure. When graphed against volumetric flow rate, a characteristic system curve is obtained which is helpful in choosing a blower. One notices from Figure 3.30 that the pressure drop at 100 cfm was approximately 4.5 inches of water for the original system. This value is rather high, and because pressure drop increases as the square of air velocity, reducing pressure drop is a more attractive means to increase air flow than increasing blower size. In this case, a combination of pressure drop reduction and increased blower capacity was deemed necessary.

A centrifugal, radial blower is used powered with a 230 V, single hP, three-phase electric motor which is controlled with a PID controller outputting to an inverter where the venturi differential pressure is used as feedback in the control loop. The installation of the new blower was used as an opportunity to reduce air-side pressure drop, enhance air mixing after the temperature control devices (discussed earlier), and improve the flow conditioning for both the blower and the venturi. The resulting installation is illustrated in Figure 3.15. One can see from Figure 3.30 that the air loop pressure drop was reduced from 4.5 to about 1.8 inches of water at 100 cfm, and the maximum achievable flow increased from about 105 cfm to 140 cfm (33% increase). The most significant installation difference between the old and new systems is the fact
Figure 3.30: Old and New Fan System Curves

Old and New Air Loop System Curves
that the blower is used a mixing device for air temperature uniformity in the new configuration as illustrated in Figure 3.15.

Calculations support that the air-side heat transfer resistance for the new system should be equal to or less than the resistance achievable with the original blower configuration and brazed, louvered fins. However, data are being taken presently, and these calculations have not been verified at the time of writing.

3.3.9 Energy Balance

As mentioned earlier, the selection of accurate measurement devices reflects the ambitious goal to attain an energy balance between air and refrigerant heat transfer rates within three percent. Although energy balance within three percent has been attained at all air flow rates except the lowest (discussed later) attaining energy balance has been an arduous task, mostly due to the inherent difficulty present in calculating energy added to the air at the test section.

By far the most significant improvement in calculating air-side heat transfer is the method used to measure downstream air temperature (discussed earlier). However, operation of the facility is still not perfect (perfect energy balance at all operating conditions), and radiation heat transfer was investigated as a possible physical interaction. Appendix G contains an EES program developed to calculate the worst possible effect that radiation heat transfer from the hot test section to the thermocouple beads could have on the energy balance. The enthalpies of the upstream and downstream air are calculated with and without radiation effects, and the program result is expressed as a ratio:

\[
\text{hratio} = \frac{(h_{ds} - h_{us})}{(h_{ds,rad} - h_{us,rad})}
\]

where "ds" indicates downstream air, "us" indicates upstream air, and "rad" indicates radiation effects are accounted for. At worst case (slowest air flow rates which have the least convection heat transfer), \(\text{hratio}\) is equal to only 1.010, indicating that radiation effects may be the cause for a one percent difference in energy balance at the most.

An effect that is being investigated presently is the fact that air temperature may be stratified in the vertical direction after the test section, indicating that vertical placement of thermocouples could affect energy balance calculations (obviously undesirable).
These physical interactions are being investigated in an attempt to improve the energy balance within the limits of the facility measuring accuracy.
4. EXPERIMENTAL PROCEDURE AND RESULTS

The immediate goal of the experimental facility is to obtain data for use in the calculation of refrigerant-side heat transfer coefficients for microchannel tubes. These coefficients are a measure of performance, allowing one to compare microchannel tubes with conventional tubes and also different microchannel geometries with each other. To this end, subcooled, superheated, and two-phase refrigerant flows at various inlet temperatures and pressures are investigated on microchannel tubes with different port geometries and similar outer dimensions. This section briefly summarizes (a) the motivation for gathering specific data, (b) data reduction techniques, and (c) and preliminary results.

4.1 Data Collection

4.1.1 Subcooled Refrigerant Inlet Condition

The first data needed for a given test section is data taken with subcooled refrigerant test section inlet conditions while varying both air and refrigerant mass flow rates. Subcooled refrigerant is certainly not the usual inlet condition for any condenser, but the data are needed in the calculation of heat transfer coefficients.

A Wilson Plot analysis is performed to obtain refrigerant-side heat transfer resistance for this analysis. The Wilson Plot is a graphical technique that uses subcooled refrigerant data at different refrigerant mass flow rates each evaluated at several air mass flow rates to find air-side heat transfer resistance. This technique is explained in detail in section 4.2.2. Once all of the data is used collectively to determine refrigerant-side heat transfer resistance, some of the data can be re-used to calculate heat transfer coefficients.

The subcooled data used to calculate refrigerant-side heat transfer coefficients is the data at the highest air mass flow rate. This is because the air-side resistance is minimized at this operating condition, maximizing the ratio of refrigerant-side resistance to the total resistance, thereby maximizing the accuracy of the heat transfer coefficient determination.

The test facility is easily controlled to provide different and steady inlet conditions to the test section, and the operator simply needs to decide the operating points for such subsystems as the Enthalpy Setting Tank, Guard Tank, Pressure Regulating Tank, pump speed, and blower speed. Once the operating points are decided, the PID controllers "take over" and the facility is allowed to reach steady-state (about one-half
hour) before data is collected. Forty subcooled data points are needed for each test section (five refrigerant mass flow rates at each of four air mass flow rates) to fully investigate the parameters of interest. The reader is referred to Luhrs (1994) for a more complete discussion of facility operation data collection methods (Luhrs, 1994).

4.1.2 Superheated Refrigerant Inlet Condition

Superheated refrigerant inlet conditions are the typical operating conditions for a condenser. The heat transfer coefficient correlations generated using this inlet condition may prove to be more useful than those calculated with subcooled inlet conditions.

The superheat data is collected in virtually the same manner as the subcooled data, and the reader is again referred to Luhrs (1994) for a more complete discussion of data collection methods (Luhrs, 1994).

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

Full condensation of superheated vapor to subcooled liquid is attainable in the test section. As expected, the amount of subcooling (if any at all) after condensing from superheat has the highest dependence on refrigerant mass flow rate.

4.1.3 Two-Phase Inlet Conditions

In some situations, mobile (vehicular) air conditioning system can provide its condenser with two-phase refrigerant. For this reason, two-phase refrigerant inlet conditions 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 temperature, air mass flow rate, the refrigerant quality, and the refrigerant mass flow rate.

Two-phase data collection and facility operation remain the same as for single-phase data collection, and the reader is again referred to Luhrs (1994) for a more complete discussion (Luhrs, 1994).
Eighteen data points are collected during two-phase testing of microchannel tubes which consist of two refrigerant inlet qualities, three refrigerant mass flow rates, and three different saturation pressures.

4.2 Data Reduction

Raw data from the test facility consist of voltages converted to temperatures and pressures which are subsequently used in the calculation of useful parameters such as heat transfer coefficients. This section briefly describes the methods used to determine heat transfer rates and coefficients.

4.2.1 Heat Transfer Rates and Energy Balance

4.2.1.1 General Discussion

Heat transfer rates are determined using the convolution of mass flow rate and enthalpy difference for both the air and refrigerant as given by

\[ \dot{q} = \dot{m} \Delta h \]  \hspace{1cm} (4.1)

where \( \dot{q} \) is heat transfer, \( \dot{m} \) is mass, \( \Delta h \) is enthalpy difference across the test section, and an overhead dot symbolizes a rate (Moran, 1988). Refrigerant mass flow rate is measured directly by the refrigerant mass flow meter, whereas air mass flow rate requires evaluation from venturi pressure differential and other parameters (discussed in section 3.3.5). Until recently, text output from the datalogger was parsed and converted to tab-delimited file format using a computer program written in TrueBasic®, the data were conditioned by macros written in the spreadsheet Excel®, and EES programs were used to obtain air and R-134a thermophysical properties, calculate air mass flow rate, and finally calculate heat transfer rates.

A data reduction program is under development which is capable of evaluating thermophysical properties, performing all calculations, and even propagating individual component uncertainty into a final heat transfer coefficient uncertainty. Presently, air mass flow rate is calculated in EES version 3.83 (explained above) using the thermophysical properties contained in this software, and a copy of the code is attached in Appendix H. Thermophysical properties of R-134a are extracted from Refprop4©
The data reduction program uses these values to calculate heat transfer rates, but is not yet capable of calculating heat transfer coefficients or propagating uncertainties. The EES program originally used to calculate air-side heat transfer coefficients is also attached in Appendix H, and some air-side heat transfer area values are not shown for proprietary reasons.

In the calculation air-side Re and Nu, the fin hydraulic diameter is needed, and the following equation is used:

\[ D_h = \frac{4A_c w_{\text{fin}}}{A_{\text{totHX}}} \]  \hspace{1cm} (4.2)

where \( D_h \) is hydraulic diameter, \( A_c \) is minimum free flow area, \( w_{\text{fin}} \) is fin width, and \( A_{\text{totHX}} \) is the total available heat transfer area (Idem, 1990). Reynold's Number is then calculated as

\[ Re = \frac{GD_h}{\mu_{\text{ls}}} \]  \hspace{1cm} (4.3)

where \( G \) is maximum mass velocity (kg/m\(^2\)-s), and \( \mu_{\text{ls}} \) denotes the dynamic viscosity evaluated at the test section temperature and pressure.

4.2.1.2 Energy Balance and Check For Two-Phase Exit Condition

Once heat transfer rates are calculated for both air and refrigerant, they are checked against each other to ensure that they agree within three percent (a rather stringent energy balance check). Energy balance results are discussed in section 4.3.

The possibility exists for two-phase refrigerant at the test section exit. As previously discussed, the two-phase-refrigerant enthalpy cannot be determined from temperature and pressure measurements only. For two-phase exit conditions, air-side energy is used to calculate the refrigerant enthalpy at the exit by subtracting the heat transfer from the refrigerant inlet energy.

4.2.1.3 Data Reduction Program

This section highlights a few capabilities of the data reduction program.
The calculation of air-side energy transfer at the test section is an iterative process. Air temperatures are measured before and after the test section but are not directly applied to enthalpy calculations as in equation 4.1, because the changing temperature of the test section as the refrigerant condenses makes direct application of equation 4.1 difficult. Instead, an overall heat transfer coefficient, $U$, is guessed. With the heat transfer from the refrigerant and the guessed $U$, the effectiveness-number-of-transfer-units ($e$-NTU) method is used to predict downstream air temperatures. The prediction is compared with the measured results, adjusted, and the process is repeated.

Although the test facility operates at steady-state, and, therefore, data should be constant and steady, bad data scans are still possible. One reason may be that voltage spikes in the building supply adversely affect data acquisition equipment. For whatever reason a scan may be bad, the data reduction program automatically calculates averages for the set of scans that compromise a data point and subsequently compares individual scan data with the arithmetic mean and variance, eliminating outliers using Chauvanet's Criterion.

With over 55 measurements, there exists the possibility that one measurement may be in error. For example, a thermocouple not needed for data acquisition or possibly one used for redundancy could be broken. In these cases, a file of data acquisition channels that are "BAD" is used to identify these channels. The data reduction program reads the "BAD" file, and ignores these channels when performing subsequent calculations for data reduction. This is a convenient tool, because it allows the user to modify the operation of the program from outside the program itself.

The data reduction program will be equipped with thermophysical property data for both air and R-134a. Energy balance considerations prompted the immediate dismissal of using EES for R-134a thermophysical properties. EES uses thermophysical property data given in the literature (McLinden, 1989). The Carnihan-Starling-Disantes (CSD) Equation of State is used in EES to predict saturated liquid properties at a given temperature, and then EES makes a pressure correction to obtain property data if the fluid is not saturated. Even with the pressure correction, this method is not the most accurate method but is amenable to mixtures and relatively easy to implement numerically. Because greater accuracy is required than offered with CSD, the Modified Benedict-Webb-Rubin (MBWR) Equation of State is used for the calculation of pure R-134a thermophysical properties (which is used in RefProp4©). MBWR is valid for pure fluids but is numerically more complicated than CSD. Presently, RefProp4© is used to obtain R-134a properties, and EES is used to obtain dry air properties.
The data reduction program propagates individual experimental uncertainties to the final heat transfer coefficient uncertainty using the root-sum square method described in the literature (Moffat, 1988). Although there exist many ways to generate uncertainties depending on the interactions considered, individual uncertainties are given by

\[ \delta R_{xi} = \frac{\delta R}{\delta X_i} \delta X_i \quad (4.4) \]

and are combined using a root-sum-square method given by

\[ \delta R = \left\{ \sum_{i=1}^{N} \left( \frac{\delta R}{\delta X_i} \delta X_i \right)^2 \right\}^{1/2} \quad (4.5) \]

where \( X_i \) is the measurement uncertainty, \( R \) is the result (function), and the partial derivative of \( R \) with respect to \( X_i \) is the sensitivity coefficient (Moffat, 1988).

The partial derivatives are calculated by averaging \( \pm 10\% \) of the estimated uncertainty in a measurement. Individual measurement uncertainty is estimated using manufacturer's specifications for bias and percent of reading, estimations of calibration uncertainty, and data acquisition uncertainties. Interactions such as thermocouples being calibrated at the same time are not considered in the present configuration (Coleman, 1989).

4.2.2 Non-Dimensional Wilson Plot Technique and Heat Transfer Coefficients

After the verification of energy balance, a non-dimensional Wilson Plot Technique (hereafter referred to as Wilson Plot) is used to determine the amount of resistance to heat transfer that is contributed by the air side. The technique was developed by E.E. Wilson for the design and analysis of a heat exchanger. Each set of refrigerant flow rates at a given air flow rate can be used to generate a Wilson Plot which yields air-side heat transfer resistance. The total heat transfer resistance is given by

\[ R_{tot} = \frac{1}{UA} = R_{ref} + R_{wall} + R_{air} \quad (4.6) \]

where the subscripts "tot" and "ref" denote "total" and "refrigerant", respectively. The air-side resistance given in equation 4.6 accounts for contact resistance between the
air-side heat transfer enhancements and the test section. The wall resistance is calculated with classical theory, and with the air-side resistance known from the Wilson Plot and the total resistance known from the data (explained later), the refrigerant-side resistance is calculated with equation 4.6.

In the Wilson Plot, 1/UA is graphed versus refrigerant Re. As the mass flow rate of the refrigerant is increased, the abscissa values decrease. As the refrigerant flow rate approaches infinity, Re\text{ref}^{0.8} approaches zero, corresponding to a refrigerant resistance of zero since the resistance is directly proportional to the refrigerant flow rate. Hence, the value for 1/UA at the y-intercept is composed of all the other resistances for the tube except the refrigerant resistance, allowing for the calculation of air-side resistance.

The total heat transfer resistance is calculated using the ε-NTU method of heat exchanger analysis. The Reynold's number for the refrigerant entering the test section is calculated from the refrigerant properties and is highly dependent on 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 Equation 4.1 and used to evaluate the heat capacities of both the air and refrigerant sides in the following equations:

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

where \( C \) is heat capacity, and \( T \) is temperature (Incropera & DeWitt, 1990). The heat exchanger effectiveness is a ratio of the actual heat transfer to the maximum possible heat transfer, and can be determined by

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

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
\[ C_r = \frac{C_{\text{min}}}{C_{\text{max}}} \]  

(4.10)

where \( C_{\text{max}} \) is the maximum of the two heat capacities. \( C_r \) and \( \varepsilon \) are combined to evaluate the NTU of the condenser tube according to

\[ \varepsilon = 1 - \exp\left\{ \frac{\text{NTU}(0.22)}{C_r} \left[ \exp\left(-C_r \cdot \text{NTU}(0.78)\right) - 1 \right] \right\} . \]  

(4.11)

This correlation applies to crossflow, single pass heat exchangers with both fluids unmixed and has a published accuracy within ±3% (Mason, 1954). However, a series solution generated by Mason can be used to solve for the effectiveness more accurately than the above relation.

The total resistance of the heat exchanger is then determined using the following relation:

\[ \frac{1}{R_{\text{total}}} = \frac{1}{U A_{\text{tot}}} = \frac{1}{\text{NTU} * C_{\text{min}}} . \]  

(4.12)

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

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

(4.13)

where \( \dot{m}_{\text{ref}} \) is the refrigerant mass flow rate, \( P_{\text{wet}} \) 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. With these data, a Wilson plot for each air flow rate can be created to obtain the air-side heat transfer resistance at that air flow rate. Superheat or two-phase inlet-condition data is then collected at air flow rates for which Wilson Plots have been generated and the air-side resistance is known.

A special EES program has been written for the purpose of generating Wilson Plots. According to the theory of the Wilson plot, the slope of the line for each air 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. 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. For a more complete discussion of Wilson Plot generation, the reader is referred to Luhrs (1994) (Luhrs, 1994).

With the ε-NTU method to get total resistance and the Wilson Plot yielding air-side resistance, refrigerant-side heat transfer resistance is known, and the average refrigerant-side heat transfer coefficient is calculated with

\[
R_{\text{ref}} = \frac{R_{f,\text{ref}}}{h_{\text{ref}}(\eta_0A_{f,\text{ref}} + A_{p,\text{ref}})}
\]

(4.14)

where \( \eta_0 \) is fin efficiency. The walls of individual microchannel tube ports actually act as fins, connecting the upper and lower tube walls. \( \eta_0 \) is the efficiency of these "fins", and the wetted areas are separated into (a) the primary area, \( A_{p,\text{ref}} \), which constitutes the inside tube wall surface, and (b) the fin surface area, \( A_{f,\text{ref}} \), which is the surface area of the webbing in contact with the refrigerant.

4.3 Results

4.3.1 Energy Balance

A full set of data (excluding two-phase inlet conditions) has been taken for one test section. Figure 4.1 illustrates the results of the energy balance plot of air-side versus refrigerant-side heat transfer for the Wilson Plot Data (seven refrigerant flow rates at each of six different air flow rates). Since the heat transfer rates should be identical, a linear curve-fit of the data should have unity slope and pass through the origin. As previously discussed, ±3% error is a stringent requirement for the energy balance, and most data fall within this range.

The data that do not satisfy the error bound requirement are the two sets taken at the lowest air flow rate. This could be due to air temperature stratification in the vertical
Energy Balance Using New Upstream Thermocouples and Thermocouples Just Downstream of Test Section

Figure 4.1: Energy Balance for Wilson Plot Data
direction after the test section. This could also be due to R-134a (or air) thermophysical property determinations in which errors become more significant as the absolute magnitude of heat transfer rate decreases. The reader is referred to Luhrs (1994) for a more complete discussion of data trends in the energy balance (Luhrs, 1994).

4.3.2 Wilson Plot

Figure 4.2 shows a Wilson plot generated using all air mass flow rates curve-fit with the same slope. The total heat transfer resistance (1/UA) is plotted against the refrigerant side Reynold's number which is raised to a power. The exponent 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 and fully developed (Stoecker, 1982).

Air-side resistances and refrigerant heat transfer coefficients are calculated but not reported for proprietary reasons.

The facility is operational and collecting data for the publication of non-proprietary heat transfer coefficients. Eventually, data will be presented in the dimensionless forms of Colburn j Factor versus Re and Darcy Friction Factor versus Re as per Kays and London (Kays, 1955). The Colburn j Factor is defined as

\[ j = \text{St} \text{Pr}^{2/3}, \quad \text{and} \quad \text{St} = \frac{\text{NuL}}{\text{Re} \text{Pr}} \]  

where \( \text{St} \) is the Stanton Number (Modified Nusselt Number). The Darcy friction factor is defined as

\[ f = \frac{\Delta P}{\left( \frac{L}{D} \right) \left( \rho \mu_{m}^2 / 2 \right)} = \frac{-(dP/dx)D_h}{\left( \rho \mu_{m}^2 / 2 \right)} \]

where \( L \) is a characteristic length in the direction of flow (Incropera & DeWitt, 1990). Presenting the data in this form permits simple performance comparisons between microchannel heat exchangers and all other types of heat exchangers documented in this way (Kays, 1955).
Figure 4.2: Wilson Plot for All Mass Flow Rates
5. CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

Microchannel heat exchangers are a promising, new technology requiring experimental evaluation, and this document has enumerated the following points:

- microchannel technology offers tremendous design flexibility to the heat exchanger designer, giving the designer the ability to vary port geometry, number of passes, number of tubes per pass, and other parameters
- microchannel technology is not yet well understood, suggesting the need for experimental data
- an experimental facility was designed, built, tested, and is operational, but publishable heat transfer coefficient data are not available at the time of this writing
- the experimental facility is part of a project that also includes full condenser modeling and single-port flow modeling.

5.2 Recommendations

The one aspect of the experimental facility that is not complete is the power measurement system. Without the power measurement system, the two-phase-refrigerant state at the test section inlet cannot be determined, implying that two-phase data cannot be obtained. Support for the power measurement system is installed in the facility, and the device itself needs to be "dropped in" to place when it is procured.

Presently, the refrigerant line immediately preceding the test section is made of copper tubing, but a short section of glass tubing would offer the researcher visual confirmation of flow regime.

With a versatile experimental facility, the final recommendation is that the facility is used for experiments other than single-tube, microchannel condensation studies.
Appendix A

1. Circuit Diagrams
   1. EST Heater Diagram
   2. Guard Tank Heater Diagram
   3. PER Heater Diagram
   4. Refrigerant Pump Control Diagram
   5. Submersible Pump Watt Transducer Diagram

2. Controller Configurations
Enthalpy Setting Tank Heater Wiring Diagram
Guard Tank Heater Wiring Diagram
Pressure Regulating Tank Heater Wiring Diagram
Refrigerant Pump Control Wiring Diagram
Submersible Pump Watt Transducer Wiring Diagram
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</table>
Appendix B

1. Thermistor Calculations
{This program is used to calculate the sum of squared errors between the actual temperature and predicted temperature for the thermistor. By minimizing the sum of squared errors, the best possible curve-fit is obtained for temperature and thermistor voltage ratio output. This version of the program is not configured for minimization. The error in the curve-fit is used in determining the final error in the refrigerant-side heat transfer coefficient}

{Constants}
Ro = 30e3 \{ohms\}
To = 25 + 273.15 \{K\}
R1 = 9.9991e3 \{ohms\}
R2 = 10.0003e3 \{ohms\}
R3 = 29.9903e3 \{ohms\}
sub = R2/(R2+R1)

{Constants}
Ro = 30e3 \{ohms\}
To = 25 + 273.15 \{K\}
R1 = 9.9991e3 \{ohms\}
R2 = 10.0003e3 \{ohms\}
R3 = 29.9903e3 \{ohms\}
sub = R2/(R2+R1)

\{a = 0.0033540, b = 0.000261339, c = 0.00000305700\}

Duplicate i=1,12

{Get data from lookup table}
TCact[i] = lookup(i,1)
Vratioact[i] = lookup(i,2)
TKact[i] = TCact[i]+273.15
term[i] = ln\left(\frac{1}{Vratioact[i]+sub} - 1\right)
TKpred[i] = \frac{1}{a + b*term[i] + c*term[i]^2}
TKpred[i] = TKpred[i]+273.15

{Squared Error Determination}
error[i] = TKpred[i] - TKact[i]
sqerror[i] = error[i]^2

End duplicate i

{Total error determination}
Totalsqerror = sum(sqerror[i], i=1, 12)

{Data from thermistor calibration}

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<th>Vratio</th>
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<td>29.89</td>
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Appendix C

1. Pressure Transducer Calibrations
2. Pressure Transducer Wiring Diagrams
Pressure Transducer Calibrations

Setra Model 270 S/N 321750 Calibration

\[ y = 0.0068104 + 4.0015x \quad R = 1 \]

Port 1 (Volts)

Setra Model 270 S/N 336037 Calibration

\[ y = 0.019354 + 4.0008x \quad R = 1 \]

Port 3 (Volts)

Setra Model 239 S/N 55337 Calibration

\[ y = -0.00041099 + 0.040763x \quad R = 0.9999 \]

Channel 54 (Volts)

Setra Model 239 S/N 351249 Calibration

\[ y = 0.00025842 + 0.9974x \quad R = 1 \]

Channel 52 (Volts)
Pressure Transducer Calibrations
Standardized Test Facility Pressure Transducer Wiring Diagram
for five pin Quick-Connect on XDCR Side

- Out Brown
B

- Excitation Case Ground Black
D

+ Out Brown
A

H

+ Excitation White
e
Appendix D

1. Air Loop P Drop Estimates - EES
2. Wiring Diagram for Old Fan
3. Airflow Statistical Evaluation
4. Heater Sizing Program - EES
5. Screen Calculations - EES
This program is used to estimate the pressure drop through the air loop in order to choose an appropriate blower.

**Air Side Static Pressure Head Estimate**

Function $dP(ReO,V,L,D,rho,flam,fturb)$

$$term = \beta(ReO)^{0.8} \cdot \rho \cdot \frac{V^2}{2}$$

*Prandtl smooth tube approx for $f$ given turbulent flow*

$$dP = \begin{cases} ReD = 2300, & flam \cdot \text{term} \\ ReD = 2300, & fturb \cdot \text{term} \end{cases}$$

**Total Air Volume Flow Rate**

$$Q = Q_{cfm} \cdot \frac{1}{2119} \cdot \frac{1}{min}$$

**Pipe Diameter**

$$D = \text{Din}_{in} \cdot 0.0254 \cdot \frac{m}{in}$$

$$V = \frac{Q_{m^3/s}}{(\pi/4)(D_{m})^2}$$

$$ReD = \frac{V_{m/s} \cdot D_{m}}{\nu_{m^2/s}}$$

*fturb and flam values from Gerhart and Gross, p. 452, and p. 442*

$$\frac{1}{\sqrt{fturb}} = 2.0 \cdot \log_{10}(ReD \cdot \sqrt{fturb}) - 0.8$$

$$flam = \frac{64}{ReO}$$

**Blower to Meter**

$$L[1] = 10 \cdot D$$

$$dP[1] = dP(ReD, V, L[1], D, rho, flam, fturb) \text{ (Pa)}$$

**Meter to Fore Mixing Box**

$$L[2] = 1 \cdot \frac{ft}{in} \cdot 12 \cdot \frac{in}{ft} \cdot 0.0254 \cdot \frac{m}{in}$$

$$dP[2] = dP(ReD, V, L[2], D, rho, flam, fturb) \text{ (Pa)}$$

**Test Section - Data from SAE Paper # 880445**

*Parameter estimation yields $dP_{ts} = C_1 \cdot V^C_2$ where $C_1 = 0.100$ and $C_2 = 1.281$*

$$C_1 = 0.100; \quad C_2 = 1.281$$

$$Ats = .7 \cdot 13.25 \cdot 0.0254 \cdot (m/in.)^2$$

$$Vts = \frac{Q_{m^3/s}}{Ats \cdot m^2}$$

$$dP[3] = C_1 \cdot (Vts \cdot C_2) \cdot \frac{in.H2O}{in} \cdot 249.1 \text{ (Pa/in.H2O)}$$

**Aft Mixing Box to Blower**

$$L[4] = 2 \cdot \frac{ft}{in} \cdot 12 \cdot \frac{in}{ft} \cdot 0.0254 \cdot \frac{m}{in}$$

$$dP[4] = dP(ReD, V, L[4], D, rho, flam, fturb) \text{ (Pa)}$$

**Total $dP$**

$$dP = \sum(dP[i], i=1,4) \div 249.1 \text{ (Pa/in.H2O)}$$

**Air Properties**

$$P = 101.325 \text{ (kPa)}$$

$$T = 50 \text{ (C)}$$

$$\mu = \text{Viscosity}(\text{Air}, T=T) \text{ (N-s/m^2) or (kg/m-s)}$$

$$\rho = \text{Volume}(\text{Air}, P=P, T=T) \text{ (kg/m^3)}$$

$$\nu = \mu / \rho \text{ (kg/m-s/rho kg/m^3)}$$

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## Airflow Statistical Summary

Averages for Seven Points (140 scans)

<table>
<thead>
<tr>
<th>File</th>
<th>mdot(V)</th>
<th>dPavg (Pa)</th>
<th>Sigma dP (Pa)</th>
<th>95% CI (Pa)</th>
<th>sigma/dPavg (%)</th>
<th>95% CI/dPavg (%)</th>
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CI = Confidence Interval  
 dP = Difference of Venturi Pressures  
 Sigma = Standard Deviation
This program assists in the calculation of resistor size in series with the air heater necessary to reduce the air heater power to a specified level. Also, the power through the resistor is calculated.

\[ V = 120 \text{ [V]} \]

\[
\text{Pheatnores} = 1000 \text{ [W]}
\]
\[
\text{Pheatnores} = V \times \text{Inores} \text{ [W]}
\]
\[
\text{Rheatnores} = \frac{V}{\text{Inores}} \text{ [ohms]}
\]

\[
\text{Pheatres} = 250 \text{ [W]}
\]
\[
\text{Pheatres} = V \times \text{Ires} \text{ [W]}
\]
\[
\text{Rtotal} = \frac{V}{\text{Ires}} \text{ [ohms]}
\]
\[
\text{Radditional} = \text{Rtotal} - \text{Rheatnores} \text{ [ohms]}
\]
\[
\text{Presistor} = (\text{Ires}^2) \times \text{Radditional} \text{ [W]}
\]
This program uses gas dynamics calculations to optimize the design of screens in the test facility, varying screen percent open area, thread diameter, and the spacing between screens. The program also accounts for different air velocities in different parts of the loop.

Variable Nomenclature: First Name_Applicability_Units
(applicability codes: d = duct, p = plenum)
(s = screen, f = friction on walls)

Goals to be achieved
(dP_ds_inH2O = .1)
(Re_ps = 85)

rhoair = 1/Volume(Air, T=40°C, P = 101.325(kPa))
mu = Viscosity(Air,T=40°C)

Q_cfm = 120 [cfm] {volume flow rate of air through our tunnel}
delta = .719 {pore area/total area}
dw_in = .009 {in} {thread diameter}
dw_m = dw_in*.0254{m/in}
decaylength_in = 500*dw_in
decaylength_m = decaylength_in*.0254

W_d_in = 13{in.} {Test section width}
H_d_in = 0.75{in.} {Test section height}
A_d_in2 = H_d_in*W_d_in {in^2}
A_d_ft2 = A_d_in2 / 144{in^2/ft^2}

W_p_in = 26{in.} {Plenum width}
H_p_in = 1.5{in.} {Plenum height}
A_p_in2 = H_p_in*W_p_in {in^2}
A_p_ft2 = A_p_in2 / 144{in^2/ft^2}

V_d_fps = Q_cfm/A_d_ft2 / 60{s/min}
V_d_mps = V_d_fps*12{in/ft}*12{in/ft}*0.0254{m/in}
Re_ds = rhoair*V_d_mps*dw_m/delta/mu
R_ds = 6*(1-delta)/(delta^2)*Re_ds^(-1/3)
dP_ds_kPa = .5*R_ds*rhoair*V_d_mps^2/1000
dP_ds_psi = dP_ds_kPa*14.7{psi}/101.325{kPa}
dP_ds_inH2O = dP_ds_psi * 27.68 {inH2O/psi}

V_p_fps = Q_cfm/A_p_ft2 / 60{s/min}
V_p_mps = V_p_fps*12{in/ft}*12{in/ft}*0.0254{m/in}
Re_ps = rhoair*V_p_mps*dw_m/delta/mu
R_ps = 6*(1-delta)/(delta^2)*Re_ps^(-1/3)
dP_ps_kPa = .5*R_ps*rhoair*V_p_mps^2/1000
dP_ps_psi = dP_ps_kPa*14.7{psi}/101.325{kPa}
dP_ps_inH2O = dP_ps_psi * 27.68 {inH2O/psi}
{Screen Optimization (continued)}

{Pressure drop through the ducts leading up to the screens in the test section}
{The length of duct leading up to the screen will be decaylength_in}
{Pressure drop given by Moody chart using hydraulic diameter}
P_{d_in} = 2*H_{d_in} + 2*W_{d_in}
Dh_{d_in} = 4*A_{d_in}/P_{d_in}

Dh_{d_m} = Dh_{d_in}*0.0254
Re_{df} = \rho_{air}V_{d_mps}Dh_{d_m}/\mu

dP_{df_kPa} = fdf*\rho_{air}*\text{decaylength}_m/Dh_{d_m}V_{d_mps}^2/2 / 1000
dP_{df_{psi}} = dP_{df_kPa}*14.7\{psi\}/101.325\{kPa\}
dP_{df_{inH2O}} = dP_{df_{psi}}* 27.68 \{inH2O/psi\}
{f = .04} 1/sqrt(fdf) = -2*log10(2.51/sqrt(fdf)/Re_{df})

{Pressure drop through the plenum sections leading up to the screens in the plenum}
{The length of duct leading up to the screen will be decaylength_in}
{Pressure drop given by Moody chart using hydraulic diameter}
P_{p_in} = 4*H_{p_in} + 4*W_{p_in}
Dh_{p_in} = 4*A_{p_in}/P_{p_in}
Dh_{p_m} = Dh_{p_in}*0.0254
Re_{pf} = \rho_{air}V_{p_mps}Dh_{p_m}/\mu

dP_{pf_kPa} = fpf*\rho_{air}*\text{decaylength}_m/Dh_{p_m}V_{p_mps}^2/2 / 1000
dP_{pf_{psi}} = dP_{pf_kPa}*14.7\{psi\}/101.325\{kPa\}
dP_{pf_{inH2O}} = dP_{pf_{psi}}* 27.68 \{inH2O/psi\}
{f = .04} 1/sqrt(fpf) = -2*log10(2.51/sqrt(fpf)/Re_{pf})

Ratio_d = dP_{df_{inH2O}} / dP_{ds_{inH2O}}
Ratio_p = dP_{pf_{inH2O}} / dP_{ps_{inH2O}}
Appendix E

1. Discharge Coefficient Graphs from the Literature
2. Pre and Post Venturi-Calibration Calculations - EES
ISO International Standard 5167
9.1.5.3 Classical venturi tubes with a machined convergent

50 mm ≤ D ≤ 250 mm

0,4 ≤ β ≤ 0,75

2 x 10^5 ≤ ReD ≤ 2 x 10^6

Under these conditions the value of the discharge coefficient C is:

C = 0,995

Figure 10.65, page 10-68, Flow Measurement Engineering Handbook: Discharge Coefficient for Classical Venturi

Summary of Venturi Discharge Coefficients from Literature
This program contains calculations to assist in the design of a facility used to calibrate a venturi. The normal working fluid is air, but the fluid used in the calibration is water. Pressure drops, flow rates, and Reynolds Numbers are calculated and compared.

\[
\text{Flowair} = 150 \text{ cfm}
\]

\[
\text{Reair} = \text{rhoair} \times \text{Vair} \times d/\text{muair}
\]

\[
\text{rhoair} = 1/\text{Volume}(\text{Air}, T=25, P=101.325) \quad \{\text{kg/m}^3\}
\]

\[
d = 3 \quad \{\text{inches}\}
\]

\[
\text{muair} = \text{Viscosity}(\text{Air}, T=25) \quad \{\text{N-s/m}^2\}
\]

\[
\text{Vair} = \text{Qair}/A \quad \{\text{m/s}\}
\]

\[
\begin{align*}
\text{Qair} &= (\text{Flowair}\{\text{cfm}\})*(1/60)*(1/3.2808)^3 \\ 
A &= (3.14159/4)*(3*2.54/100)^2 \quad \{\text{meters squared}\}
\end{align*}
\]

\[
\text{Rewat} = \text{Reair}
\]

\[
\text{Qwat} = \text{Vwat}\times A \quad \{\text{m}^3/\text{s}\}
\]

\[
\begin{align*}
\text{Vwat} &= \text{Rewat}/\text{rhowat}/d\times\text{muwat} \quad \{\text{m/s}\}
\end{align*}
\]

\[
\begin{align*}
\text{rhowat} &= 1/\text{Volume}(\text{Water}, T=25, P=101.325) \quad \{\text{kg/m}^3\}
\end{align*}
\]

\[
\text{muwat} = \text{Viscosity}(\text{Water}, T=25, P=101.325) \quad \{\text{N-s/m}^2\}
\]

\[
\begin{align*}
\text{Flowwat} &= (\text{Qwat}\{\text{m}^3/\text{s}\})*(3.2808)^3 \quad \{\text{cfs}\}
\end{align*}
\]

\[
\{\text{Solving for the ideal, incompressible pressure drop}\}
\]

\[
\begin{align*}
\text{dPswat} &= (\text{rhowat}/2)*Vthrtwat^2 - (\beta^4)*Vthrtwat^2 \quad \{\text{Pascals difference}\}
\end{align*}
\]

\[
\begin{align*}
\text{Vthrtwat} &= \text{Qwat}/\text{Athrt} \quad \{\text{m/s}\}
\end{align*}
\]

\[
\begin{align*}
\text{Athrt} &= (3.14159/4)*(1.5*2.54/100)^2 \quad \{\text{m}^2\}
\end{align*}
\]

\[
\begin{align*}
\beta &= 0.5 \quad \{\text{dimensionless}\}
\end{align*}
\]

\[
\text{dPsiwat} = \text{dPswat}\times 14.7/101300 \quad \{\text{psid}\}
\]

\[
\{\text{This section of the program calculates the ratio of air pressure drop to water pressure drop for the same Reynolds number, where the conditions of the fluids are given in the property statements.}\}
\]

\[
\begin{align*}
\text{rho1} &= 1/\text{Volume}(\text{Air}, T=25, P=101.325) \quad \{\text{kg/m}^3\}
\end{align*}
\]

\[
\begin{align*}
\text{mu1} &= \text{Viscosity}(\text{Air}, T=25) \quad \{\text{N-s/m}^2\}
\end{align*}
\]

\[
\begin{align*}
\text{rho2} &= 1/\text{Volume}(\text{Water}, T=23, P=101.325) \quad \{\text{kg/m}^3\}
\end{align*}
\]

\[
\begin{align*}
\text{mu2} &= \text{Viscosity}(\text{Water}, T=23, P=101.325) \quad \{\text{N-s/m}^2\}
\end{align*}
\]

\[
\begin{align*}
\text{ratioofinterest} &= \text{rho1}\times\text{mu2}^2/(\text{rho2}\times\text{mu1}^2) \quad \{\text{dimensionless}\}
\end{align*}
\]
This program calculates a discharge coefficient, $C_d$, for venturi calibration using water. Pressure transducer voltages were recorded over time and averaged, and the average flow is determined by calculating the water head in a large storage tank.

Weight of the water in pounds-force calculated from mercury manometer readings that determine water level in tank

$$W_{\text{lbf}} = 4472.78 \times dh_{\text{Hg}} + 1613.2724$$

$$dh_{\text{Hg}} = (h_{12} + h_{22}) - (h_{11} + h_{21})$$  

Calculating volume of water from force

$$\text{Volume} = \frac{W_{\text{lbf}}}{\text{gamma}}$$

$$\text{gamma} = 62.32$$  

for water at $T=24$ C, pounds-force/cubic foot

Calculating actual volumetric flow rate

$$Q_{\text{act}} = \frac{\text{Volume}}{\text{dt}}$$

$$Q_{\text{act SI}} = Q_{\text{act}} \times (\frac{1}{3.2808})^3$$

$$\text{mdot}_{\text{act}} = Q_{\text{act SI}} \times \rho_{\text{water}}$$

Calculating pressure drop from average voltage and pressure transducer calibration equation

$$dP = 0.20683 \times \text{Volts} + 0.033297$$

$$\text{term} = \sqrt{\rho_{\text{water}} \times dP \times 6894.8}$$

Calculating theoretical volumetric flow rate, assumes incompressible and thermal expansion factor=1

$$Q_{\text{theo SI}} = \frac{A_{\text{hrt}} \times ((2 \times dP_{\text{SI}} / \rho_{\text{water}})^{0.5} \times (1 - (1 - \beta)^4)^{0.5})}{1 \times (3.2808)^3}$$

$$A_{\text{hrt}} = \frac{3.14159}{4} \times (1.5 \times 2.5411)$$

$$dP_{\text{SI}} = dP \times 6894.8$$

$$\rho_{\text{water}} = \frac{1}{\text{Volume(Water,T=Twater,P=101.325)}}$$

$$\beta = 0.500$$  

dimensionless venturi diameter ratio

$\beta=0.382$ makes the asymptotical $C_d$ around 0.995

$$\text{OD}_{\text{nozzle}} = 2.891$$  

inches, implies that $\beta=0.518852$

$$Q_{\text{theo Eng}} = Q_{\text{theo SI}} \times (\frac{3.2808}{3})$$

Calculating discharge coefficient as a ratio between actual and theoretical flow rates

$$C_d = \frac{Q_{\text{act}}}{Q_{\text{theo Eng}}}$$

Calculating the Reynolds Number at the throat

$$\text{Re} = \rho_{\text{water}} \times \text{Vel}_{\text{hrt}} \times d / \mu_{\text{water}}$$

$$\text{Vel}_{\text{hrt}} = \frac{Q_{\text{act}} ((3.2808)^3)}{A_{\text{hrt}}}$$

$$d = 1.5 \times 2.54 / 100$$

$$\mu_{\text{water}} = \text{Viscosity(Water,T=Twater,P=101.325)}$$

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Appendix F

1. Test Section Insulation and Heat Loss - EES
This program can, among other things, calculate heat loss as a function of insulation thickness, insulation total heat transfer area calculated as a function of insulation thickness for a given flow length, etc. Radiation is not considered, because the outer surface of the insulation should be very close to ambient temperature.

These data are taken from a paper on microchannel condenser tubes:

$q_{spec} = 4400 \quad [W]$

$\text{numtubes} = 22$

$\text{scalefactor} = 1.5 \quad \{\text{because our test section is longer than the average tube}\}$

$q_{pertube} = q_{spec}/\text{numtubes} \cdot \text{scalefactor} \quad [W]$

$p\%_{allow} = 0.5 \quad \{\text{percentage of heat transfer that we can lose to surroundings}\}$

$q_{allow} = p\%_{allow}/100 \cdot q_{pertube} \quad [W]$  

[different areas are used in the program for different calculations, but none are deleted for convenience]

$A_{airduct} = 8000 \quad [in^2]$

$A_{airpvc} = 1500 \quad [in^2]$

$\{\text{Atotal} = A_{airduct} + A_{airpvc} \} \quad [in^2]$

$A_{temp} = 5472 \quad [in^2]$

$A_{temp} = 0.0254^2 \cdot A_{temp} \quad [m^2]$

$A_{total} = 4 \cdot A_{conv} \cdot 0.0254 \cdot 0.0254 \quad [m^2]$

$x = 10.5 \tan(\alpha) \quad [\text{in}]$

$A_{total} = 0.04790313 \quad \{\text{maximum ductwork heat transfer area around test section, } m^2\}$

[Assume that the outer wall temperature is the same as the inner wall temperature, and that the maximum temperature is either 100 or 150 F, whichever is entered in the program]

$maxairtemp1 = 311 \quad [K] \quad \{\text{Which is } 100 \text{ F}\}$

$maxairtemp2 = 339 \quad [K] \quad \{\text{Which is } 150 \text{ F}\}$

$ambtemp = 298 \quad [K]$

$dT = maxairtemp1 - ambtemp \quad [K]$

$R = dT/q_{allow} \quad [K/W]$

[By fixing $R$ or by fixing $q_{allow}$, one can calculate the other unknown parameter $q_{allow}$ is calculated above, and $R$ can be specified by specifying $L$ below]

[This is the total resistance needed and consists of area, thermal conductivity, and thickness (length)]

$k_{ins} = 0.038 \quad [W/m-K, \text{Taken from L and D Table A.3 for Glass fiber, coated; duct liner}]$

$k_{insmax} = 0.020026 \quad [W/m-K] \quad \{\text{Calculated from Celotex spec sheet - they specify } R = 7.2 \text{ for a one-inch thick board at 75 degrees F}\}$

$R = \frac{L \cdot \text{met}}{(k_{insmax} \cdot \text{Atotal})} \quad [K/W]$

[Rvalue = $L/k$, but the units are really screwed up, and they are ft$^2$hr$^\circ$F/Btu]

$Rvalue = (R^\ast \text{Atotal}[m^2-K/W]) \ast (1000)/3413.3^9/5^3.281^3.281$

$L = 4.0 \quad [\text{inches}]$

$\text{met} = L \ast 2.54/100$

[This equation is added so that a parametric table can be created containing $R$ and $q$]

$q_{allow}p = (1/Rp)^\ast \text{Atotal}$

$\ast dT$
Appendix G

1. Fin Calculations: Plate, Accordion, and Louvered - EES
2. System Curve Calculations - EES
This program can minimize \( R_{\text{air}} \) varying fin thickness, given parameters like fin spacing, air velocity at the test section, and air temperature. Fin spacing is determined by the minimum spacing necessary to avoid boundary layer interactions. This value is a minimum with no factor of safety. Also, the program calculates \( R_{\text{air}} \) for different test section velocities in a parametric table, allowing fin spacing to vary.

{defining parameters for air}

\[
\begin{align*}
T_{\text{air}} & = 298 \text{ K} \\
P_{\text{air}} & = 101.3 \text{ kPa} \\
\rho_{\text{air}} & = \frac{1}{\text{Volume}}(\text{Air}, T = T_{\text{air}}, P = P_{\text{air}}) \text{ kg/m}^3 \\
\text{cp}_{\text{air}} & = \text{SpecHeat}(\text{Air}, T = T_{\text{air}}) \text{ kJ/kg-K} \\
k_{\text{air}} & = \text{Conductivity}(\text{Air}, T = T_{\text{air}}) \text{ W/m-K} \\
\mu_{\text{air}} & = \text{Viscosity}(\text{Air}, T = T_{\text{air}}) \text{ Pa-s} \\
\alpha_{\text{air}} & = \frac{k_{\text{air}}}{\rho_{\text{air}} \cdot (\text{cp}_{\text{air}} \cdot 1000)} \text{ m}^2/\text{s} \\
Pr_{\text{air}} & = \frac{\mu_{\text{air}}}{\alpha_{\text{air}}} \text{ dimensionless}
\end{align*}
\]

\{calculating min fin spacing needed to avoid boundary layer interactions\}

\( V_{\text{ts}} = 5.0 \) \{Velocity at the test section, m/s\}

\[
\begin{align*}
L_{\text{fin}} & = 0.7 \times 2.54/100 \text{ fin height, meters} \\
W_{\text{fin}} & = 0.75 \times 2.54/100 \text{ fin length in flow direction, meters} \\
Rew & = \frac{V_{\text{ts}} \cdot W_{\text{fin}}}{\mu_{\text{air}}} \text{ Reynolds number, dimensionless} \\
\delta & = 5 \cdot W_{\text{fin}}/((Rew)^{0.5}) \text{ boundary layer \( \delta \), m, p.642 Fund. Fluid Mechanics} \\
\text{spacing} & = 2 \cdot \delta \times 1000 \text{ fin spacing necessary to avoid boundary layer interactions, mm} \\
\end{align*}
\]

\{calculating heat transfer parameters\}

\[
\begin{align*}
h_{\text{bar}} & = 0.664 \times (Rew^{0.5}) \times (Pr_{\text{air}}^{0.333}) \times k_{\text{air}}/W_{\text{fin}} \text{ W/m}^2 \cdot \text{K} \\
m & = \frac{(h_{\text{bar}} \times 2/(k_{\text{Al}} \times t_{\text{fin}}))^{0.5}}{2} \\
k_{\text{Al}} & = 280.0 \text{ W/m-K} \\
t_{\text{fin}} & = 0.016 \times 2.54/100 \text{ fin thickness, meters} \\
\text{the above equation can be commented out and the program set to minimize } R_{\text{air}} \text{ while varying } t_{\text{fin}} \\
t_{\text{fineng}} & = t_{\text{fin}} \times 100/2.54 \times 1000 \text{ fin thickness, thousandths of an inch} \\
term & = m \times L_{\text{fin}} \\
\text{Eta}_{\text{fin}} & = \tanh(term)/term \text{ fin efficiency} \\
U_{\text{Afins}} & = h_{\text{bar}} \times s \times \text{Eta}_{\text{fin}} \\
s & = 2 \times L_{\text{fin}}/W_{\text{fin}}
\end{align*}
\]

\{calculating primary and secondary heat exchanger area\}

\[
\begin{align*}
A_{\text{primary}} & = (2 \times W_{\text{tube}} \times L_{\text{tube}}) - A_{\text{totcontact}} \text{ square meters} \\
L_{\text{tube}} & = 24 \times 2.54/100 \text{ meters} \\
W_{\text{tube}} & = 0.75 \times 2.54/100 \text{ meters} \\
A_{\text{onecontact}} & = t_{\text{fin}} \times W_{\text{tube}} \text{ square meters/fin} \\
A_{\text{totcontact}} & = 2 \times (A_{\text{onecontact}} \times \text{finpitch} \times L_{\text{tube}}) \\
\text{finpitch} & = 1/(\text{spacing/1000+fins}) \text{ fins/meter, one side of tube} \\
\end{align*}
\]

\{calculating air-side resistance\}

\[
\begin{align*}
R_{\text{air}} & = 1/(h_{\text{bar}} \times A_{\text{primary}} + h_{\text{bar}} \times \text{Eta}_{\text{fin}} \times A_{\text{secondary}}) \text{ W/K} \{R_{\text{air}}=\text{percenttot} \times R_{\text{tot}} \text{ W/K} \text{ percenttot}=0.20 \text{ dimensionless} \} \\
R_{\text{tot}} & = R_{\text{air}} + R_{\text{ref}} \text{ W/K, } R_{\text{ref}}=0.011 \text{ W/K}
\end{align*}
\]
This program calculates $R_{air}$ for different test section velocities for accordion style fins using fin data from louvered fins without the added area of the louvers. The heat transfer coefficient is calculated using flat plate correlations, although they may not be the best correlations. $R_{air}$ is calculated in a parametric table with different values of test-section velocity.

{defining parameters for air}
$T_{air} = 298 \ \{K\}$
$P_{air} = 101.3 \ \{kPa\}$
$\rho_{air} = 1/\text{Volume}(\text{Air}, T = T_{air}, P = P_{air}) \ \{kg/m^3\}$
$c_{p_{air}} = \text{SpecHeat}(\text{Air}, T = T_{air}) \ \{kJ/kg-K\}$
$k_{air} = \text{Conductivity}(\text{Air}, T = T_{air}) \ \{W/m-K\}$
$\mu_{air} = \text{Viscosity}(\text{Air}, T = T_{air}) \ \{Pa-s\}$
$\nu_{air} = \mu_{air}/\rho_{air} \ \{m^2/s\}$
$\alpha_{air} = k_{air}/(\rho_{air}*(c_{p_{air}}*1000)) \ \{m^2/s\}$
$P_{air} = \nu_{air}/\alpha_{air} \ \{\text{dimensionless}\}$

{calculating minimum fin spacing necessary to avoid boundary layer interactions}
$V_{ts} = 5.0 \ \{\text{Velocity at the test section, m/s}\}$
$R_{ew} = V_{ts}*W_{fin}/\nu_{air} \ \{\text{Reynolds number, dimensionless}\}$
$\delta = 5*W_{fin}/((R_{ew})^{0.5}) \ \{\text{boundary layer thickness, m}, p.642 \text{ Fund. Fluid Mechanics}\}$
$\text{spacing} = 2*\delta*1000 \ \{\text{fin spacing necessary to avoid boundary layer interactions, mm}\}$

{calculating heat transfer parameters}
$h_{bar} = 0.664*(R_{ew}^{0.5})*(P_{air}^{0.333})*k_{air}/W_{fin} \ \{W/m^2-K\}$
$m = (h_{bar}^2/(k_{Al}^*t_{fin}))^{0.5}$
$k_{Al} = 280.0 \ \{W/m-K\}$
$t_{fin} = 0.016*2.54/100 \ \{\text{fin thickness, meters}\}$
{the above equation can be commented out and the program set to minimize $R_{air}$ while varying $t_{fin}$}
$t_{fineng} = t_{fin}*100/2.54*1000 \ \{\text{fin thickness, thousandths of an inch}\}$
$\text{term} = m^*L$
$\text{Etafin} = \tanh(\text{term})/\text{term} \ \{\text{fin efficiency}\}$
$U_{A_{fin}} = h_{bar}*s*\text{Etafin}$
$s = 2*L*W_{fin}$

{calculating areas based on louvered fin data without the extra area added by louvers}
$A_{totHX} = A_{allfins} + A_{prime} \ \{\text{square meters}\}$
$A_{allfins} = 2*(2*(254-1))*A_{1fin} \ \{\text{area of all fins, top and bottom, square meters}\}$
{areas assume that there are 254 base contacts on one side of the tube}
$A_{1fin}$ no longer includes $A_{addlouvers}$
\{Accordian fins (continued)\}

\[ A_{1\text{fin}} = 2^*\left((L*W_{\text{fin}})+(L*t_{\text{fin}})\right) \quad \text{\{square meters\}} \]

\[ L = A_{\text{factor}}*0.303397^*2.54/100 \quad \text{\{L is a characteristic length = fin slant height, meters\}} \]

\[ W_{\text{fin}} = 0.830^*2.54/100 \quad \text{\{fin width, meters\}} \]

\[ t_{\text{fin}} = (0.004^*2.54/100) \quad \text{\{t is the fin thickness, meters, +/- 0.0003\}} \]

\[ A_{\text{add louvers}} = 12^*\left(t_{\text{fin}}^*(0.271415926^*2.54/100)\right) \quad \text{\{additional heat transfer area with louvers, square meters\}} \]

\[ A_{\text{prime}} = 2^*(254-1)^*(0.0905512^*2.54/100)^*w_{\text{tube}} + 2^*A_{\text{stube}} \quad \text{\{square meters\}} \]

\[ w_{\text{tube}} = 0.75^*2.54/100 \quad \text{\{tube width, meters\}} \]

\[ A_{\text{stube}} = l_{\text{tube}}^*h_{\text{tube}} \quad \text{\{square meters\}} \]

\[ l_{\text{tube}} = 23^*2.54/100 \quad \text{\{tube length, meters\}} \]

\[ h_{\text{tube}} = 1.90/1000 \quad \text{\{tube height, meters\}} \]

\[ A_c = 2^*(2^*(254-1))^*(A_{1\text{fin}}) \quad \text{\{minimum free-flow area, square meters\}} \]

\[ A_{1\text{fin}} = 0.000003505 \quad \text{\{free-flow area per fin, square meters\}} \]

\{calculating air-side resistance\}

\[ R_{\text{air}} = 1/\left(h_{\text{bar}}^*A_{\text{prime}} + h_{\text{bar}}^*\eta_{\text{fin}}^*A_{\text{allfins}}\right) \quad \text{\{W/K\}} \]

\[ A_{\text{factor}} = 1.0 \]

\[ \{R_{\text{air}} = \text{percenttot}^*R_{\text{tot}} \quad \text{\{W/K\}} \]

\[ \text{percenttot} = 0.20 \quad \text{\{dimensionless\}} \]

\[ R_{\text{tot}} = R_{\text{air}} + R_{\text{ref}} \quad \text{\{W/K\}} \]

\[ R_{\text{ref}} = 0.011 \quad \text{\{W/K\}} \]

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{This program calculates \( R_{air} \) for louvered fins based on experimental values of velocity at the test section and heat transfer coefficients at the test section.}

{defining parameters for air and defining other known parameters}
\[ T_{air} = 298 \ \text{(K)} \]
\[ P_{air} = 101.3 \ \text{(kPa)} \]
\[ \rho_{air} = 1 \ \text{Volume}(Air,T=T_{air},P=P_{air}) \ \text{(kg/m}^3\) \]
\[ c_{p\text{air}} = \text{SpecHeat}(Air,T=T_{air}) \ \text{(kJ/kg-K)} \]
\[ \mu_{air} = \text{Viscosity}(Air,T=T_{air}) \ \text{(Pa-s)} \]
\[ \nu_{air} = \mu_{air}/\rho_{air} \ \text{(m}^2\text{/s)} \]
\[ \alpha_{air} = k_{air}/(\rho_{air}*(c_{p\text{air}*1000})) \ \text{(m}^2\text{/s)} \]
\[ P_{\text{rair}} = \nu_{air}/\alpha_{air} \ \text{(dimensionless)} \]
\[ k_{air} = \text{Conductivity}(Air,T=T_{avgts}) \ \{\text{thermal conductivity of air @300K, W/mK}\} \]
\[ T_{avgts} = 301.88 \ \text{taken from Res-EES-q,T,Re,mdot, K} \]
\[ k_{AL} = 280 \ \{\text{thermal conductivity of Aluminum, W/mK}\} \]
\{Thermal conductivity of copper equals 710 W/m-K\}

{calculating minimum fin spacing necessary to avoid boundary layer interactions}
\[ V_{ts} = 5.0 \ \{\text{Velocity at the test section, m/s}\} \]
\[ L_{fin} = 0.7*2.54/100 \ \{\text{fin height, meters}\} \]
\[ R_{ew} = V_{ts}*W_{fin}/\nu_{air} \ \{\text{Reynolds number, dimensionless}\} \]
\[ \delta = 5*W_{fin}/((R_{ew})^0.5) \ \{\text{boundary layer thickness, m, p.642 Fund. Fluid Mechanics}\} \]
\[ \text{spacing} = 2*\delta*1000 \ \{\text{fin spacing necessary to avoid boundary layer interactions, mm}\} \]

{using the Colburn \( j \)-factor to back out \( h_{bar} \)}
\[ j = 0.127561 \ \{\text{taken from Data-7Jan, 1994}\} \]
\[ St = j*(P_{rair}^-0.667) \ \{\text{dimensionless}\} \]
\[ Nu = St*P_{rair}/R_{ew} \ \{\text{dimensionless}\} \]
\[ h_{bar} = Nu*k_{air}/W_{fin} \ \{\text{dimensionless}\} \]
\[ h_{bar} = 256.35 \]

{calculating fin efficiency, assuming adiabatic tip}
\[ h_{bar} = 0.664*(R_{ew}^0.5)*(P_{rair}^0.333)*k_{air}/W_{fin} \ \{W/m^2-K\} \ \{\text{flat plate analogy}\} \]
\[ \eta_{fin} = \tanh(\text{term})/(\text{term}) \ \{\text{fin efficiency}\} \]
\[ \text{term} = m^*L \]
\[ m = \sqrt{\text{termtwo}/k_{AL}/t_{fin}} \ \{1/meters\} \]
\[ \text{termtwo} = 2*h_{bar} \]

{calculating areas}
\[ A_{totHX} = A_{allfins} + A_{prime} \ \{\text{square meters}\} \]
\[ A_{allfins} = 2*(2*(254-1))*A_{1fin} \ \{\text{area of all fins, top and bottom, square meters}\} \]
\[ A_{1fin} = 2*((L*W_{fin})+(L*t_{fin}))+A_{addlouvers} \ \{\text{square meters}\} \]

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Louvered fins (continued)

\[ L = \text{Afactor} \times 0.303397 \times 2.54/100 \] \{L is a characteristic length = fin slant height, meters\}

\[ W_{\text{fin}} = 0.830 \times 2.54/100 \] \{fin width, meters\}

\[ t_{\text{fin}} = (0.004 \times 2.54/100) \] \{t is the fin thickness, meters, +/- 0.0003"\}

\[ A_{\text{addlouvers}} = 12 \times (t_{\text{fin}} \times (0.271415926 \times 2.54/100)) \] \{additional heat transfer area with louvers, square meters\}

\[ A_{\text{prime}} = 2 \times (254-1) \times (0.0905512 \times 2.54/100) \times w_{\text{tube}} + 2 \times A_{\text{xstube}} \] \{square meters\}

\[ w_{\text{tube}} = 0.75 \times 2.54/100 \] \{tube width, meters\}

\[ A_{\text{xstube}} = l_{\text{tube}} \times h_{\text{tube}} \] \{square meters\}

\[ l_{\text{tube}} = 23 \times 2.54/100 \] \{tube length, meters\}

\[ h_{\text{tube}} = 1.90/1000 \] \{tube height, meters\}

\[ A_{\text{c}} = 2 \times (2 \times (254-1)) \times (A_{\text{c1fin}}) \] \{minimum free-flow area, square meters\}

\[ A_{\text{c1fin}} = 0.000003505 \] \{free-flow area per fin, square meters\}

{calculating air-side resistance}

\[ R_{\text{air}} = 1/(h_{\text{bar}} \times (A_{\text{prime}} + E_{\text{fin}} / A_{\text{allfins}})) \] \{W/K\}

\[ \text{Afactor} = 1.0 \]

\[ E_{\text{fin}} = (1/(h_{\text{bar}} \times R_{\text{air}} - \text{Afactor} \times A_{\text{prime}})/(A_{\text{allfins}} \times \text{Afactor})) \]

\[ R_{\text{air}} = \text{percenttot} \times R_{\text{tot}} \] \{W/K\}

\[ \text{percenttot} = 0.20 \] \{dimensionless\}

\[ R_{\text{tot}} = R_{\text{air}} + R_{\text{ref}} \] \{W/K\}

\[ R_{\text{ref}} = 0.011 \] \{W/K\}
This worksheet assists in the calculations to generate a system curve of pressure versus volumetric flow rate for the air loop. Given the absolute pressures before and after the fan, the room temperature read off of the barometer, and matching these values with the flow rates calculated from the data using separate EES programs, we can get the total pressure across the fan for a flow rate

\[
\text{ProomHg} = 29.33 \quad \text{[inches Mercury]}
\]

\[
\text{ProomH2O} = \text{ProomHg}/2.036^*27.68 \quad \text{[inches of water]}
\]

\[
\text{ProomSI} = \text{ProomHg}/2.036^*6895 \quad \{\text{Pa}\}
\]

\[
\text{PSinfan} = (\text{PSinmeas} + \text{ProomH2O})^*249.1 \quad \{\text{Pa}\}
\]

\[
\text{PSoutfan} = (\text{PSoutmeas} + \text{ProomH2O})^*249.1 \quad \{\text{Pa}\}
\]

\[
\text{Vinfan} = \text{mdotair}/(\text{rho} \text{infan}^* \text{Ainfan}) \quad \{\text{m/s}\}
\]

\[
\text{mdot units are in kg/s} \implies \text{units of Vinfan are m/s}
\]

\[
\text{Ainfan} = 0.012667686 \quad \{\text{inlet area at fan, m}^2\}
\]

\[
\text{rho} \text{infan} = 1/\text{Volume}(\text{Air, T}=298, P=\text{PSinfan}/1000) \quad \{\text{kg/m}^3, \text{P in kPa}\}
\]

\[
\text{Voutfan} = \text{mdotair}/(\text{rho} \text{outfan}^* \text{Aoutfan}) \quad \{\text{m/s}\}
\]

\[
\text{mdot units are in kg/s} \implies \text{units of Voutfan are m/s}
\]

\[
\text{Aoutfan} = \text{Ainfan}/1.6 \quad \{\text{because the outlet area diffuses immediately to 5'' duct}\}
\]

\[
\text{rho} \text{outfan} = 1/\text{Volume}(\text{Air, T}=298, P=\text{PSoutfan}/1000) \quad \{\text{kg/m}^3, \text{P in kPa}\}
\]

{Calculating the static pressure difference across the fan}

{Equations are taken from ANSI/ASHRAE Standard 51-1985}

{PV is really supposed to be calculated only at the fan outlet as defined in section 9.4.2 of this standard, not at both the inlet and outlet (at least the paragraph is worded as such. I took the definition of PV as applicable to any part of the flow field)}

\[
\text{PttotfanSI} = \text{Ptoutfan} - \text{Ptinfan} \quad \{\text{all pressures in this section are in Pa}\}
\]

\[
\text{Ptoutfan} = \text{PSoutfan} + \text{Pvoutfan} \quad \{\text{Pa}\}
\]

\[
\text{Pvoutfan} = ((\text{mdotair}/\text{Aoutfan})^2)^2/\text{rho} \text{outfan} \quad \{\text{Pa}\}
\]

\[
\text{Ptinfan} = \text{PSinfan} + \text{Pvinfan} \quad \{\text{Pa}\}
\]

{actually Ptinfan should be calculated as given in 9.5.3.3 of standard: which is defined as follows: Pt1 = Ps3 + Pv3 - f*(L1,3/Dh3)*Pv3 where 3 is the measuring point and one is at the fan. Pv is defined as the average velocity pressure}

{In our case, L=0, and the equation reduces to the one used above}

\[
\text{Pvinfan} = ((\text{mdotair}/\text{Ainfan})^2)^2/\text{rho} \text{infan}
\]

\[
\text{PSfan} = \text{PttotfanSI} - \text{Pvoutfan} \quad \{\text{Pa}\}
\]

\[
\{\text{PSfan} = \text{PSoutfan} - \text{Ptinfan} \quad \{\text{Pa}\}\}
\]

\[
\text{Pttotfaninwat} = \text{PttotfanSI}/249.1 \quad \{\text{inches of water}\}
\]

\[
\text{PSfundiff} = \text{PSoutfan} - \text{PSinfan}
\]

{calculating volumetric flow rate, as given by the mass flow rate divided by the density of air at the fan exit}

\[
\text{QfanSI} = \text{mdotair}/\text{rho} \text{outfan} \quad \{\text{m}^3/\text{s}\}
\]

\[
\text{Qfancfm} = \text{QfanSI}*(3.281^3)*60 \quad \{\text{cfm}\}
\]
Calculating radiation effects on temperature measurement in air stream

This program calculates a temperature difference experienced by a spherical thermocouple bead in air that exchanges heat radiatively with a hot surface. This is manifested in the bead, the thermocouple bead temperature

diameter and surface area of thermocouple bead, quiescent and tube temperatures

\[ D = 6.35e-4 \] meters, where 36 AWG wire has \( d = 0.0050 \) inches, and the bead diameter is taken as \( 5d \)

\[ A = 4\times 3.14159 \times (D/2)^2 \] \( m^2 \)

\[ T_{airus} = 25 + 273.15 \] \( K \)

\[ T_{tube} = 55 + 273.15 \] \( K \)

Calculating convection parameters

\[ \text{heat transfer coefficient} = h_{xcus} = \frac{h_{xcus} \times D}{k_{airus}} \] \( \text{dimensionless} \)

\[ k_{airus} = \text{Conductivity(Air, T = Tairus)} \] \( \text{W/m-K} \)

\{I&D, ed. 3, Table 7.9, p. 439, eq. 7.58, correlation for flow over a sphere\}

\{Average \( Nu \), \( 3.5 < \text{ReD} < 7.6e4, 0.71 < \text{Pr} < 380, 1.0 < (\mu/\mu) < 3.2 \} \)

\[ \text{Nuus} = 2 + (0.4 \times \text{ReDus}^{0.5} + 0.06 \times \text{ReDus}^{0.667}) \times \text{Prus}^{0.4} \times (\muus/\mu) \times 0.25 \] \( \text{dimensionless} \)

\[ \text{ReDus} = \frac{\rho_{airus} \times \text{Vair} \times D}{\muus} \] \( \text{dimensionless} \)

\[ \rho_{airus} = \text{Volume(Air, T = Tairus, P=100)} \] \( \text{kg/m}^3 \), \( T \) in \( K \), \( P \) in \( \text{kPa} \)

\[ \muus = \text{Viscosity(Air, T = Tairus)} \] \( \text{N}-\text{sec/m}^2 \)

\[ \text{Prus} = \frac{\muus \times \text{Cpus}}{k_{airus}} \]

\[ \text{Cpus} = \text{SpecHeat(Air, T = Tairus)} \times 1000 \] \( \text{J/kg-K} \)

\[ \text{ratus} = \frac{\muus}{\muus} \] \( \text{dimensionless} \)

(repeating the convection calculations for an approximate downstream air temperature)

\[ \Delta T_{ts} = 15 \] \( \text{temperature increase across test section} \)

\[ T_{airds} = T_{airus} + \Delta T_{ts} \] \( K \)

\[ h_{xcus} = \frac{h_{xcus} \times D}{k_{airds}} \] \( \text{dimensionless} \)

\[ k_{airds} = \text{Conductivity(Air, T = Tairds)} \] \( \text{W/m-K} \)

\{I&D, ed. 3, Table 7.9, p. 439, eq. 7.58, correlation for flow over a sphere\}

\{Average \( Nu \), \( 3.5 < \text{ReD} < 7.6e4, 0.71 < \text{Pr} < 380, 1.0 < (\mu/\mu) < 3.2 \} \)

\[ \text{Nuds} = 2 + (0.4 \times \text{ReDds}^{0.5} + 0.06 \times \text{ReDds}^{0.667}) \times \text{Prds}^{0.4} \times (\muds/\mu) \times 0.25 \] \( \text{dimensionless} \)

\[ \text{ReDds} = \frac{\rho_{airds} \times \text{Vair} \times D}{\muds} \] \( \text{dimensionless} \)

\[ \rho_{airds} = \text{Volume(Air, T = Tairds, P=101.3)} \] \( \text{kg/m}^3 \), \( T \) in \( K \), \( P \) in \( \text{kPa} \)

\[ \muds = \text{Viscosity(Air, T = Tairus)} \] \( \text{N}-\text{sec/m}^2 \)

\[ \text{Prds} = \frac{\muds \times \text{Cpds}}{k_{airds}} \]

\[ \text{Cpds} = \text{SpecHeat(Air, T = Tairus)} \times 1000 \] \( \text{J/kg-K} \)

\[ \text{ratiods} = \frac{\muds}{\muus} \] \( \text{dimensionless} \)

Calculating radiation heat transfer

\[ q_{radus} = A \times \text{Fus} \times \sigma \times \left( T_{beadus}^4 - (T_{tube})^4 \right) \] \( W \)

\[ q_{radds} = A \times \text{Fdds} \times \sigma \times \left( T_{beadds}^4 - (T_{tube})^4 \right) \] \( W \)

\[ \sigma = 5.67e-8 \] \( \text{W/m}^2\text{-K}^4 \)

\[ \text{Fus} = 0.25 \] \( \text{worst case shape factor} \)
\{TC Radiation (continued)\}

Fd s = 0.20 \{worst case shape factor\}

\{Calculating convection heat transfer\}
q\text{convus} = h\text{xus} \cdot A \cdot (T\text{airus} - T\text{beadus}) \{W\}
q\text{convds} = h\text{xds} \cdot A \cdot (T\text{airds} - T\text{beadds}) \{W\}

\{Energy balance on thermocouple bead implies...\}
q\text{convus} = q\text{radus}
q\text{convds} = q\text{radds}

\{difference in actual temperature and that read by thermocouple\}

\Delta T\text{us} = T\text{beadus} - T\text{airus}
\Delta T\text{ds} = T\text{beadds} - T\text{airds}

\{We will use this temperature difference and calculate its effect on the air-side energy balance\}

h\text{s} = \text{Enthalpy}(\text{Air, } T=T\text{airus})
h\text{srad} = \text{Enthalpy}(\text{Air, } T=T\text{beadus})
h\text{ds} = \text{Enthalpy}(\text{Air, } T=T\text{airds})
h\text{dsrad} = \text{Enthalpy}(\text{Air, } T=T\text{beadds})

h\text{diff} = (h\text{ds}-h\text{s})/(h\text{dsrad}-h\text{srad}) \{dimensionless\}
Appendix H

1. Air & Refrigerant Heat Transfer Rate Data Reduction Program - EES
2. Air-Side Heat Transfer Coefficient Calculations - EES
This program calculates heat transfer rates for both air and refrigerant using built-in thermophysical properties. This program is still used for the calculation of air mass flow rate but is no longer used for R134a properties or for the calculation of refrigerant-side heat transfer rate.

{calculating ideal, incompressible mass flow rate}

\[ m_{idinc} = A_{thrt} \times (\sqrt{2 \times \rho_{thrt} \times (p_{bfv} - p_{iv})}) \times (\sqrt{1/(1 - \beta A^4)}) \]  \( \text{kg/s, where pressure is in Pa} \)

\[ p_{bfv} = p_{bfvkPa} \times 1000 \quad \text{where \( p_{bfv} \) is in Pa} \]
\[ p_{iv} = (p_{bfvkPa} - dP_{venkPa}) \times 1000 \]
\[ A_{thrt} = 0.001140091 \quad \text{m}^2 \]
\[ \beta = 0.5 \quad \text{dimensionless} \]
\[ R_{pm} = 8.314/28.97 \quad \text{kJ/kg-K} \]

\( \text{[}\overline{T}_{avg} = \text{average of 3 temperatures before venturi, K}] \)
\[ \rho_{thrt} = 1 \text{Volume}(\text{Air},T=\overline{T}_{avg},P=p_{bfv}/1000) \quad \text{kg/m}^3, \text{where \( P \) is in kPa} \]

\[ \overline{T}_{avg} = T_{avgC} + 273.15 \quad \text{[}\overline{T}_{avg} \text{in K}] \]

{calculating expansion factor}

\[ Y = \text{sqrt}((R^2/(2 \times \gamma)) \times \text{term} \times (1-R^2/(\gamma-1))) \times (1 - (\beta A^4) \times (R^2/(2 \times \gamma))) \]  \( \text{dimensionless} \)
\[ R = p_{iv}/p_{bfv} \quad \text{dimensionless} \]
\[ \gamma = 1.3992 \quad \text{for air = \( cp/cv \), dimensionless} \]
\[ \text{term} = \gamma/(\gamma-1) \quad \text{dimensionless} \]

{calculating venturi discharge coefficient, a function of Reynold's number}

\[ C_d = \text{lookup(lookuprow(2,x),1)} \quad \text{dimensionless} \]
\[ x = \text{log10}(R_{D}) \]
\[ R_{thrt} = \rho_{thrt} \times V_{thrt} \times dhrt \times \mu_{thrt} \]
\[ V_{thrt} = \text{sqrt}(2 \times (p_{bfv} - p_{iv})/\rho_{thrt}(1 - \beta A^4)) \quad \text{m/s} \]
\[ dhrt = 0.0381 \quad \text{m} \]
\[ \mu_{thrt} = \text{Viscosity(\text{Air},T=\overline{T}_{avg})} \quad \text{kg/m-s} \]
\[ R_{D} = \rho_{D} \times V_{pvc} \times D/\mu_{D} \]
\[ \rho_{D} = 1 \text{Volume}(\text{Air},T=\overline{T}_{avg},P=p_{bfv}/1000) \quad \text{kg/m}^3, \text{where \( P \) is in kPa} \]
\[ V_{pvc} = \text{flowSI/Apvc} \quad \text{m/s} \]
\[ Apvc = 0.004560367 \quad \text{m}^2 \]
\[ D = 0.0762 \quad \text{m} \]
\[ \mu_{D} = \text{Viscosity(\text{Air},T=\overline{T}_{avg})} \quad \text{kg/m-s} \]

{calculating expansion factor for venturi temperature effects}

\[ F_a = 0.99075928 + (3.20069298e-5) \times \overline{T}_{avg} \quad \text{dimensionless} \]

{calculating the final, corrected mass flow rate}

\[ m_{dfin} = C_d \times F_a \times Y \times m_{idinc} \quad \text{kg/s} \]

{calculating heat transfer at the test section}

\[ q_{airts} = m_{dfin} \times d_{hrt} \times 1000 \quad \text{W} \]
\[ q_{airpvc} = m_{dfin} \times d_{hpvc} \times 1000 \quad \text{W} \]
\[ d_{hrt} = h_{dsts} - h_{us} \quad \text{kJ/kg} \]
\[ d_{hpvc} = h_{dspvc} - h_{us} \quad \text{kJ/kg} \]
\{heat transfer rate program (continued)\}\\

\[
hdsts = \text{Enthalpy}(\text{Air, } T = T_{ds}) \quad \{\text{kJ/kg}\}
\]

\[
T_{ds} = T_{us} + dT_{tavg}C \quad \{T_{ds} \text{ in } K\}
\]

\[
hdspvc = \text{Enthalpy}(\text{Air, } T = T_{spvc}) \quad \{\text{kJ/kg}\}
\]

\[
T_{spvc} = T_{us} + dT_{pvc}C \quad \{T_{ds} \text{ in } K\}
\]

\[
hus = \text{Enthalpy}(\text{Air, } T = T_{us}) \quad \{\text{kJ/kg}\}
\]

\[
T_{us} = T_{us} + 273.15 \quad \{T_{us} \text{ in } K\}
\]

\{Modifying the heat transfer calculations to account for KE changes\}\\

\[
q_{airpvcmod} = m_{dfin}*dhpvcmod*1000 \quad \{W\}
\]

\[
dhpvcmod = hdspvcmod - hus \quad \{\text{kJ/kg}\}
\]

\{Calculating the velocity at the entrance to the PVC pipe\}\\

\{This uses \(T_{spvc}\) as the temperature recorded in the PVC pipe, and \(dT_{spvc}KE\) as the temperature which accounts for KE changes\}\\

\{\(V_{pvc}\) is calculated above, but this version should be more accurate\}\\

\[
V_{pvcnew} = m_{dfin}/(\rho_{spvc}*A_{pvc}) \quad \{m/s\}
\]

\[
\rho_{spvc} = 1/\text{Volume}(\text{Air, } T = T_{spvc}, P = P_{atm}) \quad \{\text{kg/m}^3\}
\]

\[
(hdspvcmod - hdspvc)*1000 = 1/2*(V_{pvcnew}^2 - V_{tsduct}^2)\]

\[
V_{tsduct} = m_{dfin}/(\rho_{spvc}*A_{tsduct}) \quad \{m/s\}
\]

\[
A_{tsduct} = (24*2.54/100)*(3*2.54/100) \quad \{m^2\}
\]

\[
hdspvcmod = \text{Enthalpy}(\text{Air, } T = T_{spvc}) \quad \{kJ/kg\}
\]

\[
dT_{spvc}KE = T_{spvcmod} - T_{spvc} \quad \{K\}
\]

\{calculating heat flux\}\\

\[
q_{flux} = q_{airts}/A_{prime} \quad \{W/m^2\}
\]

\[
A_{prime} = # \quad \{\text{taken from Res-EES-hcalcs program, square meters}\}
\]

\{calculating volumetric flow rate\}\\

\[
V_{tseng} = V_{tsSI}*3.6*0.62137 \quad \{\text{mph}\}
\]

\[
V_{tsSI} = m_{dfin}/(A_{ducts}*rho_{air}) \quad \{m/s\}
\]

\[
flowSI = V_{thrt}*A_{thrt} \quad \{m^3/s\}
\]

\[
floweng = flowSI*60*((1/0.3048)^3) \quad \{\text{cfm}\}
\]

\{\(A_{ducts} = 0.007821758 \quad \{\text{taken from direct measurements, square meters}\}\}\}

\{\(A_{ducts} = 0.009598768 \quad \{\text{taken from calculations of test section area assuming}=A_{ts},\text{\ m}^2\}\}\}

\{calculating Reynold's Number at the test section\}\\

\[
R_{ts} = G*D_{hfin}/m_{uts} \quad \{\text{dimensionless}\}
\]

\[
G = m_{dfin}/A_{c} \quad \{\text{maximum mass velocity, kg/m}^2\text{-s}\}
\]

\[
A_{c} = # \quad \{\text{minimum free-flow area, taken from Res-EES-hcalcs program, square meters}\}
\]

\[
D_{hfin} = 4*A_{c}*w_{fin}/A_{totHX} \quad \{\text{hydraulic diameter of the fin is the characteristic length, m}\}
\]

\[
w_{fin} = #*2.54/100 \quad \{\text{fin width, meters}\}
\]

\[
A_{totHX} = # \quad \{\text{taken from Res-EES-hcalcs program, square meters}\}
\]

\[
m_{uts} = \text{Viscosity}(\text{Air, } T = T_{avgts}) \quad \{\text{kg/m-s}\}
\]

\[
T_{avgts} = (T_{us} + T_{ds})/2 \quad \{K\}
\]
{heat transfer rate program (continued)}

{calculating Prantl Number at the test section}
Prts = cpair\*muts/kair
    cpair = SpecHeat(Air, T=Tavgts) \{specific heat of air at test section, kJ/kg-K\}
    kair = Conductivity(Air, T=Tavgts) \{thermal conductivity of air at test section, W/m-K\}

{calculating Darcy friction factor at the test section}
fts = -\frac{dpdtst{\Delta}h}{rhots*VtsSI*VtsSI/2} \{Darcy friction factor at test section\}
    rhots = 1/Volume(Air, T=Tavgts, P=pavgtsSI) \{air density at test section, \(T\) in K, \(P\) in kPa\}
    pavgtsSI = (pbftsSI + patsSI)/2 \{average pressure at test section, kPa\}
    pbftsSI = PbftskPa \{pbftsSI in kPa\}
    patsSI = PbftskPa - dPtskPa \{patsSI in kPa\}

{calculating pressure drop across test section, psid}
dPts = pbftsSI - patsSI \{pressure drop across test section, kPa\}
dpdxts = (patsSI - pbftsSI)/wfin \{pressure drop across test section per flow length, kPa/m\}

{assigning values to error functions so I can put them in the parametric table}
a = TusCerr
b = dTsavgCerr
c = dTpvCerr
first = TavgCerr
e = PbftskPaerr
f = dPtskPaerr
second = PbftskPaerr
h = dPvskPaerr
dPref = dPreftskPa
j = dPreftskPaerr
Pin = PrefsftskPa
l = PrefsftskPaerr
n = mdotreferr
Tin = TrefuC + 273.15 \{K\}
fifth = TrefuCerr
dTref = dTrefC
fourth = dTrefCerr
s = dTpvC

{Program to calculate the refrigerant side heat transfer rate for each scan. These values are then used to determine the average heat transfer rate for each data point. Finally, a refrigerant side Reynold's number is calculated}

{Equations to calculate the heat transfer for each scan}
{heat transfer rate program (continued)}

{Determine saturation temperature at loop pressure}
Tsatin = Temperature(R134a,x=.5,P=Pin) \{P in kPa\}
Tsatout = Temperature(R134a,x=.5,P=Pout) \{P in kPa\}
Pout = Pin - dPref \{kPa\}

{Determine degree of subcool}
Tinsubcool = Tsatin - Tin \{K\}
Toutsubcool = Tsatout - Tout \{K\}
Tout = Tin + dTref \{K\}

{Calculate heat loss for the conditions present}
hrefin = Enthalpy(R134a,T=Tin,P=Pin) \{T in K, P in kPa\}
hrefout = Enthalpy(R134a,T=Tout,P=Pout) \{T in K, P in kPa\}
qtemp = mdotref*(hrefin-hrefout) \{kW\}
qref = qtemp*1000 \{W\}

{Calculate Reynold's number for each data point using the geometryfor the total number
of ports}

{Perimeter of microchannel port determined by port geometry}
p = #{m}
muin = Viscosity(R134a,T=Tin,P=Pin)
muout = Viscosity(R134a,T=Tout,P=Pout)
muerror = abs((muin-muout)/muin)*100
Rein = 4*mdotref/(p*muin)
Reout = 4*mdotref/(p*muout)
Reerror = abs((Rein-Reout)/Rein)*100
This program will calculate convective air-side heat transfer coefficients, fin efficiencies, Nusselt numbers, etc. from knowledge of resistances. Units of are W/m²K.

{Calculating areas}
\[ \text{AtotHX} = \text{Aallfins} + \text{Aprime} \]  \hspace{1cm} \{square meters\}
\[ \text{Aallfins} = 2 \times (2 \times (254-1)) \times \text{A1fin} \]  \hspace{1cm} \{area of all fins, top and bottom, square meters\}
\[ \text{A1fin} = 2 \times ((L \times \text{wfin}) + (L \times \text{tfin})) + \text{Aaddlouvers} \]  \hspace{1cm} \{square meters\}
\[ L = \# \times 2.54/100 \]  \hspace{1cm} \{L is a characteristic length = fin slant height, meters\}
\[ \text{wfin} = \# \times 2.54/100 \]  \hspace{1cm} \{fin width, meters\}
\[ \text{tfin} = (\# \times 2.54/100) \]  \hspace{1cm} \{t is the fin thickness, meters, +/- 0.0003\}
\[ \text{Aaddlouvers} = 12 \times (\text{tfin} \times (\# \times 2.54/100)) \]  \hspace{1cm} \{additional heat transfer area with louvers, square meters\}
\[ \text{Aprime} = 2 \times (254-1) \times (\# \times 2.54/100) \times \text{wtube} + 2 \times \text{Axstube} \]  \hspace{1cm} \{square meters\}
\[ \text{wtube} = \# \times 2.54/100 \]  \hspace{1cm} \{tube width, meters\}
\[ \text{Axstube} = \text{ltube} \times \text{htube} \]  \hspace{1cm} \{square meters\}
\[ \text{ltube} = \# \times 2.54/100 \]  \hspace{1cm} \{tube length, meters\}
\[ \text{htube} = \#/1000 \]  \hspace{1cm} \{tube height, meters\}
\[ \text{Ac} = 2 \times (2 \times (254-1)) \times (\text{Ac1fin}) \]  \hspace{1cm} \{minimum free-flow area, square meters\}
\[ \text{Ac1fin} = 0.000003505 \]  \hspace{1cm} \{free-flow area per fin, square meters\}

{Setting known values}
\[ \text{kair} = \text{Conductivity}(\text{Air}, T = \text{Tavgs}) \]  \hspace{1cm} \{thermal conductivity of air @300K, W/mK\}
\[ \text{Tavgs} = 301.88 \]  \hspace{1cm} \{taken from Res-EES-q, T, Re, \text{mdot}, K\}
\[ \text{Rair} = 0.0134720 \]  \hspace{1cm} \{K/W\}
\[ \text{kAL} = 280 \]  \hspace{1cm} \{thermal conductivity of Aluminum, W/mK\}

{Calculating Nusselt number}
\[ \text{Nu} = \text{hair} \times \text{Dh}/\text{kair} \]
\[ \text{Dh} = 4 \times \text{Ac} \times \text{wfin}/\text{AtotHX} \]

{Calculating fin efficiency, assuming adiabatic tip}
\[ \text{term} = \text{m} \times \text{L} \]
\[ \text{etafin} = (\tanh(\text{term}))/\text{term} \]
\[ \text{m} = \sqrt{2 \times \text{hair}/\text{kAL} \times \text{tfin}} \]  \hspace{1cm} \{1/meters\}

{Calculating fin effectiveness}
\[ \text{etanought} = 1 - (\text{Aallfins}/\text{AtotHX}) \times (1 - \text{etafin}) \]

{Calculating hair, W/m²K}
\[ \text{hair} = 1/(\text{Rair} \times (\text{Aallfins} \times \text{etanought} + \text{Aprime})) \]
REFERENCES


Luu, M., 1979, Augmentation of In-Tube Condensation of R-113, Iowa State University, Ames, Ph.D Dissertation Thesis.


