A new methodology for the determination of transport properties of refrigerants

Jian-Yuan Lin
Iowa State University

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A new methodology for the determination of transport properties of refrigerants

Lin, Jian-Yuan, Ph.D.
Iowa State University, 1994
A new methodology for the determination of transport properties of refrigerants

by

Jian-Yuan Lin

A Dissertation Submitted to the Graduate Faculty in Partial Fulfillment of the Requirements for the Degree of DOCTOR OF PHILOSOPHY

Department: Mechanical Engineering
Major: Mechanical Engineering

Approved:
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For the Graduate College

Iowa State University
Ames, Iowa
1994

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**NOMENCLATURE**

A total heat transfer area, \( m^2 \)

\( A_o \) outer insulation surface area, \( m^2 \)

\( A_i \) the i-th sub-section heat transfer area, \( m^2 \)

a thermal diffusivity of fluid, \( k/\rho C_p \)

a,b exponential constant for correlation curve fit

\( a^e,b^e \) exponential constant for measured data curve fit

C coefficient of correlation curve fit

or oil mass fraction

\( C^e \) coefficient of measured data curve fit

CF calibration function

\( C_p \) specific heat, \( kJ/kg \)

D diameter of inside tube, m

d thickness of fluid layer, m

f friction factor

Gr Grashof number

\( \bar{h} \) in-tube average heat transfer coefficient, \( W/m^2 \cdot K \)

\( h_x \) local heat transfer coefficient, \( W/m^2 \cdot K \)
\( \bar{h}_o \) average heat transfer coefficient of heat loss, \( W/m^2 \cdot K \)

\( k \) thermal conductivity, \( W/m \cdot K \)

\( k_s \) thermal conductivity of copper, \( W/m \cdot K \)

\( k_a \) absorption coefficient, \( m^{-1} \)

\( L \) tube length, m or mm

\( l \) the length of cylinders, m

\( \Delta l_i \) the length of i-th section, m or mm

\( \dot{m} \) mass flow rate, kg/s

\( \bar{N}u_D \) Nusselt number based on diameter D

n refractive index

P pressure, Pa or Psia

\( P_c \) critical pressure, Pa or Psia

\( \Delta P \) differential pressure, Pa or Psid

\( P_{vp} \) vapor pressure, Pa or Psia

\( \Delta P_r \) differential reduced pressure, \( (P - P_{vp})/P_c \)

\( Pr \) Prandtl number

\( P_r \) reduced pressure, \( P/P_c \)

\( \dot{Q}, \dot{q} \) heat rate, \( W \)

\( \dot{Q}_t, \dot{Q}_r \) total, radiation heat rate, \( W \)

\( q'' \) heat flux, \( W/m^2 \)

\( q_{net}'' \) net heat flux, \( W/m^2 \)

\( q_{loss}'' \) heat loss flux, \( W/m^2 \)

\( q_{tot}'' \) total heat flux, \( W/m^2 \)
<table>
<thead>
<tr>
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<th>Description</th>
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<td>R</td>
<td>radiation parameter, $r_1 \cdot k_a$, Equation 2.19</td>
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<tr>
<td>$r_i$</td>
<td>radius distance of point i from a line source, m</td>
</tr>
<tr>
<td>$R_a$</td>
<td>Rayleigh number, $Gr \cdot Pr$</td>
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<tr>
<td>$Re_D$</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$r_0$</td>
<td>wire radius, $\mu m$</td>
</tr>
<tr>
<td>SS</td>
<td>sum of square quantity</td>
</tr>
<tr>
<td>$T_r$</td>
<td>the reduced temperature</td>
</tr>
<tr>
<td>$T_s$</td>
<td>the static temperature or outer insulation surface temperature, °C</td>
</tr>
<tr>
<td>$T_a$</td>
<td>ambient temperature, °C</td>
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<tr>
<td>$T_{wi}$</td>
<td>sub-section wall temperature, °C</td>
</tr>
<tr>
<td>$ar{T}_i$</td>
<td>average inlet temperature of the test-section, °C</td>
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<tr>
<td>$ar{T}_o$</td>
<td>average outlet temperature of the test-section, °C</td>
</tr>
<tr>
<td>$ar{T}_0$</td>
<td>equilibrium outlet temperature of the test fluid, °C</td>
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<tr>
<td>$ar{T}_w$</td>
<td>average wall temperature of the test-section, °C</td>
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<td>$ar{T}_f$</td>
<td>average fluid temperature in the test-section, °C</td>
</tr>
<tr>
<td>$ar{T}_s$</td>
<td>average outer insulation surface temperature, °C</td>
</tr>
<tr>
<td>$ar{T}_\infty$</td>
<td>average ambient temperature, °C</td>
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<tr>
<td>$\Delta T_{wf}$</td>
<td>average temperature difference between $\bar{T}_w$ and $\bar{T}_f$, °C</td>
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<tr>
<td>$\Delta T_{id}$</td>
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<td>$\Delta T_{wf_i}$</td>
<td>the i-th sub-section local average temperature difference, $\bar{T}<em>{w_i} - \bar{T}</em>{f_i}$, °C</td>
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<td>$\Delta T^*$</td>
<td>dimensionless temperature group in test-tube, $\bar{T}_o - \bar{T}_i/\bar{T}_w - \bar{T}_f$</td>
</tr>
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\( T_{i,\infty} \) temperature at a radial distance \( i \) from a line source, °C
\( t \) or \( t_i \) time or time \( i \), sec
\( U_{r_{dk}} \) uncertainty of \( k \) by the Dittus-Boelter correlation
\( U_{r_{pk}} \) uncertainty of \( k \) by the Petukhov correlation
\( U_{r_{gk}} \) uncertainty of \( k \) by the Gnielinski correlation
\( V, v \) velocity, m/s
\( \Delta x \) distance between two plates, m

Superscripts
- \( e \) experiment
- \( c \) correlation
- \( * \) non-dimensional quantity

Subscripts
- \( a \) air or absorption
- \( \text{app} \) apparent quantity
- \( bA \) blend A
- \( bB \) blend B
- \( D \) diameter based
- \( f \) fluid
- \( i \) inside or inlet
- \( m \) mixture
- \( \text{net} \) net quantity
- \( o \) outside, outlet, or oil
- \( \text{oil} \) oil or lubricant
- \( \infty \) uniform stream
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>equilibrium quantity</td>
</tr>
<tr>
<td>r</td>
<td>radiation or reduced property or refrigerant</td>
</tr>
<tr>
<td>s</td>
<td>surface</td>
</tr>
<tr>
<td>sat</td>
<td>saturation state</td>
</tr>
<tr>
<td>t or tot</td>
<td>total quantity</td>
</tr>
<tr>
<td>w</td>
<td>wall</td>
</tr>
</tbody>
</table>

**Greek Symbols**

- **\( \alpha \)**: volumetric thermal expansion coefficient, \( K^{-1} \)
- **\( \epsilon \)**: apparent thermal conductivity ratio
- **\( \epsilon_i \)**: emissivity of surface \( i \)
- **\( \epsilon_{1,2} \)**: emissivity of surface 1 to 2
- **\( \lambda \)**: thermal conductivity, \( W/m \cdot K \)
- **\( \lambda_{app} \)**: apparent thermal conductivity, \( W/m \cdot K \)
- **\( \mu \)**: viscosity, \( kg/m \cdot s \)
- **\( \rho \)**: density, \( kg/m^3 \)
- **\( \Gamma_s \)**: Stefan-Boltzmann constant, \( 5.6697 \times 10^{-8}, W/m^2K^4 \)
- **\( \phi \)**: non-dimensional apparent thermal conductivity group, Equation 2.18
- **\( \omega \)**: acentric factor
ACKNOWLEDGEMENTS

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provision of resources.

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Finally, I would like to express my appreciation to my family for their support and understanding. To my wife, Hsiu-Chuan, and my daughter, Hsiu-Yi, they have stayed in Taiwan since I came to study at ISU. We have lived on different sides of the earth since then. At this time, I would like to give my deepest appreciation to them for their sacrifice and understanding. I also would like to thank my father-in-law, mother-in-law, brothers-in-law, and sisters-in-law for their support and encouragement, and care for my wife and daughter throughout this period. In addition, when I decided to come to study in the U.S.A., my mother was seventy-three years old and needed special care. I also knew that I should stay with her as a responsible son. However, I still left her and as a result I couldn't contribute toward her day-to-day care during this period. At this moment, I would like to thank my brothers, my sisters-in-law,
and my wife for taking care of my mother for me. With deepest appreciation, I would like to share my achievements with all of them.
CHAPTER 1. INTRODUCTION

Serving as working fluids, CFC (ChloroFluoroCarbons) refrigerants have been widely used in refrigeration and air-conditioning applications. In the past few decades, CFC refrigerants have been used as standard refrigerants due to their excellent thermal performance, stable chemical properties, and low toxicity. However, since the problems with CFCs were exposed in the early 1970's, CFC refrigerants could no longer be used in these applications. Through a number of steps in usage regulation and prohibition, the CFC refrigerants are to be totally phased-out in the next few years [1, 2]. Since the CFC problems surfaced, people have tried to search for CFC alternatives to take the place of existing refrigerants. So far, several alternatives have been found for some banned refrigerants, such as R-134a for R-12 and R-123 for R-11. Some refrigerant mixtures have also been proposed as potential refrigerant alternatives [3]. Because refrigerant mixtures have some thermal advantages such as energy savings [4], capacity modulation and ozone-safe [5, 6, 7], they have been considered potential CFC refrigerant alternatives. The screening and testing for replacement CFC refrigerants have also been progressing since the late 1980's [8, 9, 10, 11, 12].

However, one of the current topics is the search for alternatives for R-22 and R-502, an azeotropic refrigerant mixture of R-22/R-115 (48.8/51.2% by mass). R-22 and R-502 are widely used as refrigerants in commercial air-conditioners, chillers, and
heat pumps because they have favorable thermophysical properties and well-matched material compatibilities. Although the R-22 ozone depletion potential (ODP) is 0.05 compared to 1.0 for R-11 or R-12 and its global warming potential (GWP) is 0.3 compared to 1.0 for R-11, extended use of this refrigerant will contribute to surplus chlorine in the ozone, and global warming will still be a concern. Therefore, the phase-out of these refrigerants and the search for alternatives is progressing at this moment.

A number of studies focused on searching for R-22 alternatives [13, 14, 15]. Some of the refrigerants and refrigerant mixtures are evaluated by the Air-Conditioning and Refrigeration Institute (ARI) and thought to be potential candidates [8, 16, 17]. Sixteen possible R-22 replacement and six possible R-502 candidates are shown in Tables 1.1 and 1.2.

Table 1.1: Possible R-22 replacement candidates

<table>
<thead>
<tr>
<th>number</th>
<th>refrigerant or refrigerant mixtures</th>
<th>composition (weight %)</th>
<th>trade designation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>R-134a</td>
<td>100</td>
<td>-</td>
</tr>
<tr>
<td>2</td>
<td>R-290</td>
<td>100</td>
<td>-</td>
</tr>
<tr>
<td>3</td>
<td>R-717</td>
<td>100</td>
<td>-</td>
</tr>
<tr>
<td>4</td>
<td>R-32/R-125</td>
<td>50/50</td>
<td>AZ20</td>
</tr>
<tr>
<td>5</td>
<td>R-32/R-125</td>
<td>60/40</td>
<td>-</td>
</tr>
<tr>
<td>6</td>
<td>R-32/R-134a</td>
<td>20/80</td>
<td>-</td>
</tr>
<tr>
<td>7</td>
<td>R-32/R-134a</td>
<td>25/75</td>
<td>-</td>
</tr>
<tr>
<td>8</td>
<td>R-32/R-134a</td>
<td>30/70</td>
<td>-</td>
</tr>
<tr>
<td>9</td>
<td>R-32/R-134a</td>
<td>40/60</td>
<td>-</td>
</tr>
<tr>
<td>10</td>
<td>R-125/R-143a</td>
<td>45/55</td>
<td>AZ50</td>
</tr>
<tr>
<td>11</td>
<td>R-32/R-125/R-134a</td>
<td>10/70/20</td>
<td>Klea 61 (R-407B)</td>
</tr>
<tr>
<td>12</td>
<td>R-32/R-125/R-134a</td>
<td>23/25/52</td>
<td>AC9000</td>
</tr>
<tr>
<td>13</td>
<td>R-32/R-125/R-134a</td>
<td>24/16/60</td>
<td>-</td>
</tr>
<tr>
<td>14</td>
<td>R-32/R-125/R-134a</td>
<td>25/20/55</td>
<td>-</td>
</tr>
<tr>
<td>15</td>
<td>R-32/R-125/R-134a</td>
<td>30/10/60</td>
<td>-</td>
</tr>
<tr>
<td>16</td>
<td>R-32/R-125/R-290/R-134a</td>
<td>20/55/5/20</td>
<td>-</td>
</tr>
</tbody>
</table>
Table 1.2: Possible R-502 replacement candidates

<table>
<thead>
<tr>
<th>number</th>
<th>refrigerant or refrigerant mixtures</th>
<th>composition (weight %)</th>
<th>trade designation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>R-125/R-143a</td>
<td>45/55</td>
<td>AZ50</td>
</tr>
<tr>
<td>2</td>
<td>R-125/R-143a</td>
<td>50/50</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>R-32/R-125/R-134a</td>
<td>10/70/20</td>
<td>-</td>
</tr>
<tr>
<td>4</td>
<td>R-32/R-125/R-134a</td>
<td>20/40/40</td>
<td>Klea 60 (R-407A)</td>
</tr>
<tr>
<td>5</td>
<td>R-32/R-125/R-143a</td>
<td>10/45/55</td>
<td>-</td>
</tr>
<tr>
<td>6</td>
<td>R-125/R-143a/R-134a</td>
<td>44/52/4</td>
<td>HP62 (R-404A)</td>
</tr>
</tbody>
</table>

According to the test results, a mixture of R-32/R-125/R-134a (23/25/52), provided similar cooling and heating capacity with a 3 to 4% reduction in energy efficiency compared with R-22 [17]. Lunger et al. also tested this blend and showed similar results [18]. For R-502 alternatives, Clodic (1994) tested the binary blend of R-125/R-143a (45/55) and ternary blend of R-125/R-143a/R-134a (44/52/4) and then compared the results with R-502 [19]. The test results all showed this ternary blend had power consumption reduced by more than 10% compared to R-502, but the coefficient of performance (COP) and condensing/evaporating temperature of these two blends were nearly the same and considered acceptable. As can be seen, these blends show promise as alternatives, and further tests are being done at this moment.

Another interesting project has been initiated by the U.S. Navy and the Environmental Protection Agency (EPA). The United States Navy currently uses CFC-114 (CCl₂CClF₂) as the working refrigerant in shipboard chiller units. Because CFC-114 performs well in shipboard chillers, which use sea water as the heat sink on the condenser side, CFC-114 has served as a good working fluid in the past. However, CFC-114 is one of the banned refrigerants. With the mandatory phase-out program of CFC's as executed by the Montreal Protocol, it is an urgent task for the U.S. Navy
to find a new refrigerant alternative which is environmentally safe and has acceptable thermal characteristics for replacing CFC-114. Some points of concern for the Navy in switching refrigerant to an acceptable alternative are: 1. safety (non-flammable, non-toxic) in application; and 2. easy to exchange with the present refrigerant (CFC-114) in the field with only minor modifications.

HFC-236ea ($CF_3CFHCF_2H$), a new refrigerant, has already been classified as one of the candidates for CFC-114 alternatives. There are several reasons that this refrigerant shows promise to replace CFC-114. Firstly, the thermophysical properties were investigated and found to be similar to those of CFC-114 [20, 21]. Secondly, according to simulation results [20, 21], the performance of HFC-236ea is similar to the performance of CFC-114. So that it may be a retrofit or drop-in candidate. This latter characteristic is attractive for the shipboard or submarine chiller's refrigerant transition. Other preliminary test results, such as flammability, material compatibility, and toxicity, are also favorable. The EPA, in cooperation with the U.S. Navy, has also shown that HFC-236ea would have other favorable characteristics with respect to naval applications.

**Refrigerant alternatives and properties of interest**

Some of the general steps for refrigerant replacements include property evaluations, heat transfer characteristic studies, a design and/or retrofit study, and field tests. The property evaluations of interest includes thermodynamic properties, solubility and miscibility characteristics, and transport properties. Some of the properties, such as specific heat, viscosity, and thermal conductivity, are important properties in any heat transfer study. The measurement of these latter properties is the
focus of this study.

Today, these properties are known for numerous fluids, however, there are still strong needs for fluids which have not been measured. Those under the classification of new pure refrigerants, refrigerant mixtures, and refrigerant-lubricant mixtures are in particular need. This study focuses on the transport properties of selected alternative refrigerants including one pure new refrigerant (HFC-236ea), two ternary refrigerant mixtures: R-32 (23%)/R-125 (25%)/R-134a (52%) hereafter referred to as blend A and R-125 (44%)/R-134a (4%)/R-143a (52%) hereafter referred to as blend B.

**Lubricants of interest**

Lubricants are required for the proper operation of the compressor in refrigeration systems. However, the use of lubricants will cause changes in properties, such as viscosity, miscibility, and solubility [22, 23, 24], which will eventually affect heat transfer characteristics [25]. Finally, mixing lubricants with refrigerants could affect material compatibilities [26].

The mineral, naphthenic, paraffinic, and synthetic lubricants usually used with CFC refrigerants [27, 28] are not miscible in the HFC refrigerants. Several lubricant alternatives are presently considered as HFC lubricants. These lubricants which include polyol ester lubricants (POEs), alkylbenzenes, and polyalkylene glycols (PAGs) are currently being tested by compressor manufacturers and lubricant suppliers for a wide range of characteristics, such as miscibility, lubricity, and material compatibility. Recent research shows that the POEs and PAGs are more favorable due to good miscibility and stability when subjected to a wide range of operating conditions for
new refrigerants such as R-134a [29, 30]. Some of the possible lubricants which show promise with a wide range of alternative refrigerants as determined by compressor manufacturers are classified as listed in Table 1.3 [31]. As can be seen from Table 1.3, different compressors favor different lubricants. All the lubricants except the last one shown in the table mix well with R-134a and blend B. Because blend A also contains 52% R-134a, these lubricants will probably satisfactory with blend A. It should be noted that researchers are still looking for suitable lubricants for use with blends A and B.

In this study, ICI Emkarate RL-32S lubricant was selected for property experiments with blend A. Various concentrations of lubricant which cover a possible range of concentration circulating in a refrigeration system were mixed with blend A. Measurements were performed in the refrigerant-lubricant mixtures in order to understand the effect of lubricant concentration on properties.

<table>
<thead>
<tr>
<th>polyol ester lubricants</th>
<th>HFC refrigerants</th>
<th>compressor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mobil EAL Arctic 22 CC</td>
<td>R-134a, blend B</td>
<td>Copeland</td>
</tr>
<tr>
<td>ICI Emkarate RL-32S</td>
<td>R-134a, blend B</td>
<td>Bristol</td>
</tr>
<tr>
<td>Castrol Icematic SW-32</td>
<td>R-134a, blend B</td>
<td>Tecumsech</td>
</tr>
<tr>
<td>CPI Solest LT-32</td>
<td>R-23</td>
<td></td>
</tr>
</tbody>
</table>

Objectives

Refrigerant properties, such as thermodynamic properties and transport properties, provide important information as to the acceptability of a refrigerant for use as a working fluid in a refrigeration cycle. Viscosity and thermal conductivity are
especially important in any heat transfer analysis as applied to a refrigeration cycle performance studies. Because HFC-236ea and the two ternary mixtures mentioned above (blends A and B) are new refrigerants, there is very limited property data published for these refrigerants. In addition, these three refrigerants have not been used in commercial refrigeration systems to date. Therefore, the objectives of this research focuses on measuring the transport properties for the alternative refrigerant, HFC-236ea, and the two refrigerant mixtures: blend A and blend B. The transport properties of interest are liquid thermal conductivity and viscosity. Lubricant effects on the transport properties of ternary blend A were also studied.

Methodology

Because of the limitation in predicting transport properties by using theory, experimental measurement is the only direct method of obtaining these two properties. There are a number of ways to measure viscosity such as: The capillary viscometer [32, 33], the vibrating principle viscometer [34], and the torsional oscillation viscometer [35, 36, 37, 38]. For thermal conductivity: The vertical coaxial cylinder method [39, 40], transient hot-wire method [41, 42, 43, 44, 45], and transient hot-strip method [46] are the usual methods employed.

As can be seen from the above list of methods, each property is measured by using a method independent of the other properties. In this study, a new approach is proposed for simultaneously measuring these properties [47]. This approach uses an inline viscometer and a heat transfer test-section to dynamically measure viscosity and thermal conductivity at the same time. Viscosity is measured by a torsional oscillation inline viscometer while thermal conductivity is measured from knowledge
of single-phase heat transfer characteristics of a heated test-section. Meanwhile, by placing a mass flow meter and densimeter in series with the viscometer and test-section, density and specific heat can also be measured.

Research program

The main goal of this study is to develop a method for measuring transport properties of liquid, which are thermal conductivity and viscosity. Then, the method will apply to these property measurements for HFC-236ea and two ternary blends with and without lubricant. Measured data were compared with other theoretical data sources (e.g. REFPROP) [49]. The test facility was calibrated by using the well-known properties of R-22, R-12, and R-113, and the methodology was verified by using the properties of R-114.

Scope

The scope of this study was as follows:

- Design and Construct a test rig for measuring transport properties with emphasis on thermal conductivity based on the knowledge of single-phase heat transfer.

- Install a viscometer in series with a heat transfer test-section for simultaneously measuring viscosity and thermal conductivity.

- Calibrate heat transfer and heat loss characteristics of the test-section by using refrigerants of known properties.
• Verify the accuracy of viscometer with fluids of known viscosity.

• Measure the viscosity and thermal conductivity of CFC-114 and compare with the ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers) standard handbook data.

• Measure the viscosity and thermal conductivity of HFC-236ea and compare with theoretical prediction data (i.e. REFPROP).

• Measure the viscosity and thermal conductivity of two ternary blends (without lubricant) and compare with theoretical prediction data (i.e. REFPROP).

• Measure the viscosity and thermal conductivity of ternary blend A with lubricant.

• Develop prediction equations from measured properties.

In this study, density, $\rho$, and specific heat, $C_p$, of the refrigerants of interest are also measured and discussed.
CHAPTER 2. REVIEW OF EXPERIMENTAL METHODS FOR THE DETERMINATION OF TRANSPORT PROPERTIES

This chapter will review the theory surrounding transport properties and the experimental methods used to determine transport properties. Several traditional methods of measuring transport properties will be reviewed. Existing error sources of each method will also be discussed. Finally, the advantages and disadvantages of each traditional method will also be compared.

Review of theory of transport properties

The elementary model of a molecule assumes that the shape of the molecule is like a rigid sphere of diameter, d. When molecules move randomly at a mean velocity $v$, they collide and transfer momentum or energy assuming velocity or temperature gradients exist. These momentum or energy transfers will cause an exchange of momentum or energy flux as the molecules try to reach an equilibrium status. The momentum or energy flux was found to be proportional to the velocity or temperature gradient. These constants of proportionality are known as transport properties. Viscosity is associated with the coefficient of the velocity gradient while thermal conductivity is proportional to the temperature gradient. These coefficients are also proportional to $vL/3$ according to kinetic theory [50, 51, 52], where $v$ is the average
molecular speed and $L$ is the mean free path. The relations between these properties are usually described by the following equations:

\[
\textit{Momentum flux} = -\frac{\mu}{L} = -\frac{vL}{3mn} \frac{dv}{dz}
\]  
(2.1)

\[
\textit{Energy flux} = -\lambda \frac{dT}{dz} = -\frac{vL}{3} C_v n \frac{dT}{dz}
\]  
(2.2)

where $n$ is the number of molecules in a unit volume and $m$ is the mass of the molecule. The term $nmv$ is often called the momentum density while $C_v nT$ is the energy density. The above equations are basic definitions of transport properties. However, if the average speed is proportional to $(RT/M)^{1/2}$ and the mean free path to $(nd^2)^{-1}$, then the viscosity and thermal conductivity can be rewritten as:

\[
\mu = \frac{m \rho v L}{3} = (\text{constant}) \frac{T^{1/2} M^{1/2}}{d^2}
\]  
(2.3)

\[
\lambda = \frac{c L C_v n}{3} = (\text{constant}) \frac{T^{1/2}}{M^{1/2} d^2}
\]  
(2.4)

Another theoretical equation for thermal conductivity of liquids was proposed by Bridgman in 1923 and later modified by Power et al. [53]. The resulting equation, which has been used for pure substances, is:

\[
\lambda = 2.8 \left( \frac{N}{V} \right)^{2/3} \sigma V_s
\]  
(2.5)

where $N=$ Avogadro's number; $V=$ molar volume; $\sigma=$ Boltzmann's constant; $V_s=$ speed of sound.

The above models are ideal models, however, an interaction force might exist between molecules. As a result, a rigid sphere model is no longer valid and it must be modified. For different molecules, the modification will be different.
Although a number of prediction rules were developed, there is still no universal principle which can be used to cover every group of fluids, especially when applied to mixtures. Baroncini et al. (1980) developed a general correlation for organic liquid thermal conductivity [54]. Although it provided useful information for some existing materials, it still needs further study for new fluids or refrigerant mixtures. Kandlikar et al. (1975) also published a paper to theoretically predict viscosity and thermal conductivity for R-22 and R-12 mixtures [55]. Later, Levy (1981) used a modified Maxwell-Eucken equation to calculate thermal conductivity for two component solution mixtures [56]. Ely at al. (1983) used a corresponding state method to predict the viscosity and thermal conductivity of mixtures [57].

Although the above studies provide useful information, they still have some limitations in actual use, especially for multi-component mixtures. This is due to the fact that the interaction force between molecules is still not fully understood, and some correlations need semi-empirical data from experiments to correct the models. However, one of the mixing rules has been widely used in past studies [94, 96]. This rule is described by the equation shown below:

\[ k_m = k_r (1 - C) + k_o C - 0.72 (k_o - k_r) (1 - C) C \]  
(2.6)

where C represents lubricant concentration.

Lin and Pate (1992) used thermodynamic property data and the speed of sound of the mixture to predict the thermal conductivity of refrigerant mixtures by using Bridgman and Powel's modified Bridgman equation to develop a mixing rule [58]. However, this method needs modified coefficients which requires the calibration and numerous tests of known fluid properties.

Although some theoretical prediction methods and limited experimental data...
for obtaining properties of new pure refrigerant and refrigerant mixtures have been published [60, 61, 62, 63, 64], there are still very limited experimental data for verification, especially for lubricant-refrigerant mixtures. Therefore, experimental methods for obtaining the required data become vital in the current CFC transition.

**Experimental methods for the determination of thermal conductivity**

Thermal conductivity can be measured by experimental methods [65, 66] by using Fourier's law. The methods developed for measuring thermal conductivity can be divided into the following two categories: steady-state methods and unsteady-state methods. The following is a brief review of these methods.

**The steady-state methods**

The steady-state methods differ primarily is the different geometries used in the apparatus. In addition, all steady-state methods require careful experimentation to minimize errors for heat losses and to prevent the onset of any convective motion in the fluid. 1. Horizontal flat-plate method

In this method, the heat passes through a layer of fluid located between two circular horizontal metal plates. With a suitable design along with radiation and convection corrections in the measurement, the thermal conductivity accuracy can be optimized. This method measures the temperature difference across the fluid layer and the heat flow rate with known geometric dimensions. The accuracy of thermal conductivity measurement by using this method depends on the precision of the temperature measurement and the precision of the geometric dimensions of the cell.

2. The vertical coaxial cylinder method
In this method, two coaxial, vertical cylinders form a test cell. Fluid is filled between the annulus of the cylinders, and heat is generated in the center of the inner cylinder. By measuring the heat flow passing through the test fluid and the temperatures at the inner and outer surfaces of the two cylinders, the thermal conductivity can be determined. Again, the experimental uncertainty needs to be refined for improving accuracy.

3. The hot-wire method

The hot-wire method is in principle a special case of the coaxial cylinder method in which the inner cylinder is replaced by a thin wire. The wire not only performs as a heater, but also performs as an internal thermometer. The temperature of the outer surface of the fluid-filled annulus can be evaluated from the dimensions, thermal conductivity, and the outer surface temperature of the outer cylinder. The heat flow generated in the wire is determined from the electric power added to wire. Correction for the heat loss is required for an accurate heat flow estimation. The thermal conductivity is determined from the measured quantities and the dimensions of the cell.

3. Concentric sphere method

In this method, two concentric spheres make up the test cell. The test fluid is filled in the fluid-filled gap. A heater located in the cavity of the inner sphere generates heat that is conducted radially outward to a concentric spherical shell. Temperatures on the inner and outer surfaces of the cylinders are measured by sensors embedded in the surface walls. With known dimensions and temperature measurements, the thermal conductivity can be obtained.
The non-steady methods

The establishment of steady state in an apparatus of considerable mass is a slow process, and measurements may be rather time consuming. It is therefore natural to explore the possibility of employing transient techniques in which the same information may be obtained while the fluid system is subjected for a short interval of time to the influence of a transient temperature field.

1. Continuous line source

A transient heat transfer characteristic of a line source with infinite length and constant energy generation in an extended isotropic fluid can be used to determine the fluid thermal conductivity. From the measurement of the temperatures at a fixed radial distance at two different points in time, the thermal conductivity can be determined by the following working equation:

\[
\lambda = \frac{Q}{4\pi} \frac{\ln(t_2/t_1)}{(T_{2,\infty} - T_{1,\infty})}
\]

(2.7)

where \(T_{1,\infty}\) and \(T_{2,\infty}\) are the measured fluid temperatures at a fixed radial distance from the heat-dissipated line source at two different time points, \(t_1\) and \(t_2\), respectively. A plot of temperature versus the natural logarithm of time should result in a straight line. The unique feature of the solution from this method is that it yields thermal conductivity, \(k\), directly. Recently, the transient hot-wire method has been applied to the measurement of liquid thermal conductivity[10-14]. This method uses a very thin wire as a line source to which heat is applied. The thermal conductivity, \(\lambda\), of the test liquid is measured by measuring heat input, temperature rise of the wire, and the required time. The working equation [65, 66, 67, 68] is given as follows:

\[
\Delta T_{id} = \frac{q}{4\pi \lambda (T_r, \rho_r)} \ln \left[ \frac{4at}{r_0^2 C} \right]
\]

(2.8)
in which $q$ is the heat input per unit length of the wires, $r_0$ the wire radius, and $C$ a numerical constant ($C = e^\gamma$), $\gamma = 0.5772157 \ldots$, Euler's constant). The symbol $a$ represents the thermal diffusivity of the liquid surrounding the wires.

2. Cylindrical, spherical and plane sources

A different approach to that used in the transient hot-wire method is based on a cylindrical, spherical, or plane source taking the place of the line source. In these methods, heat is generated for a short period of time in a relatively large solid body of a simple geometry (cylinder, sphere, plane). After the temperature of this body has been raised by a few degrees above that of the heat receiving body, which is assumed to remain at a constant temperature of the isothermal surroundings, heating is discontinued. Heat is now conducted through the narrow, fluid-filled gap between the two geometrically similar bodies and the change of temperature with time is recorded. The thermal conductivity is then calculated by the working equations which describe the transient heat transfer characteristics for different geometries.

**Error sources**

Despite precautions and sophistication in the design and of the sensors, it is impossible to completely eliminate some sources of error in the measured quantities. There are several error sources which exist in the methods mentioned above, however, two sources of error, namely radiation and convection, are the major contributors affecting the energy transfer through the fluid film.
Radiation effect in parallel plate or cylinder

Thermal radiation in partly transparent fluids affects the heat transfer in two ways; first, the energy is directly absorbed and emitted by the test fluids; second, the energy is absorbed and emitted by the walls or surfaces of the test cell or gap. No matter which way, the radiation heat transfer affects the conduction heat exchange which is used in Fourier's Law for the thermal conductivity measurement methodology. Therefore, the thermal conductivity measurement needs to be corrected for radiation heat transfer. The temperature distribution depends not only on the geometry of the cell system, but also on the temperature of the surfaces surrounding the test fluid.

Leidenfrost analyzed the thermal radiation effects on the thermal conductivity measurement in a vertical and horizontal parallel plate cell [69]. Leidenfrost calculated the ratio of the heat transferred by radiation to the total heat, and found that this ratio was strongly related to emissivity. Table 2.1 was obtained from the calculated results shown in Figure 10 of Leidenfrost's paper [69].

From the radiant heat transfer percentages shown in Table 2.1, it was noted that the radiation heat flow significantly affects the heat conduction which, in turn, affects

<table>
<thead>
<tr>
<th>temperature, K</th>
<th>emissivity, $\epsilon = 0.1$</th>
<th>emissivity, $\epsilon = 1.0$</th>
</tr>
</thead>
<tbody>
<tr>
<td>273</td>
<td>0.1 %</td>
<td>1.0 %</td>
</tr>
<tr>
<td>383</td>
<td>0.2 %</td>
<td>2.0 %</td>
</tr>
<tr>
<td>483</td>
<td>0.5 %</td>
<td>4.1 %</td>
</tr>
<tr>
<td>600</td>
<td>1.0 %</td>
<td>8.8 %</td>
</tr>
</tbody>
</table>

Example for toluene at $\Delta T=5$ K
the accuracy of the thermal conductivity measurement. The higher the temperature and emissivity, the larger the radiant heat exchange. At a temperature of 600 K for toluene, the radiation heat transfer ratio can even reach 8.8% if a black body is used. This result suggests that if no corrections for radiant heat exchange are applied to the measurements, then the values of thermal conductivity will be too high by the same percentages.

As a further step in understanding how radiation affects conduction in a flat plate and concentric cylinder cell, a theoretical analysis is performed below. The situation is one in which the test fluid is completely transparent to radiation of all wavelengths. For this situation, the heat transferred by radiation can be calculated as follows.

1. Parallel flat plate cases

The equations governing the radiation and conduction heat transfer in two parallel flat plates are given as follows:

\[ \dot{Q}_r = \epsilon_{1,2} \Gamma_s A \left( T_1^4 - T_2^4 \right) \]  \hspace{1cm} (2.9)

where \( \epsilon_{1,2} \) is the radiation interchange factor, \( \Gamma_s \) is the Stephan-Boltzmann constant and \( A \) is heat transfer area. For two parallel walls, \( \epsilon_{1,2} \) is given as:

\[ \frac{1}{\epsilon_{1,2}} = \frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1 \]  \hspace{1cm} (2.10)

However, the total heat transferred by conduction, which is the quantity used to calculate thermal conductivity in this apparatus, is based on Fourier’s Law and expressed as:

\[ \dot{Q}_t = \frac{\lambda A (T_1 - T_2)}{\Delta x} \]  \hspace{1cm} (2.11)
where \( \lambda \) is thermal conductivity of test fluid and \( \Delta x \) is the normal distance between emitter and receiver plates or the distance of test cell gap. The radiation ratio to the total heat transfer is then calculated by dividing these two equations, which is:

\[
\frac{\dot{Q}_r}{\dot{Q}_t} = \frac{\Gamma_s \left( T_1^4 - T_2^4 \right) \Delta x}{\lambda \left( T_1 - T_2 \right) \left( \frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1 \right)}
\]

(2.12)

It is of interest to calculate the radiation ratio in order to determine the significance of the radiation heat transfer in relation to the total heat transfer. For this example R-114 was selected as a test fluid. Figure 2.1 plots the radiant heat transfer ratio versus temperature for this case. As shown in Figure 2.1, the radiation heat transfer is quite significant at high temperatures and high emissivities of the plates. This ratio, for example, can even reach +15%. In other words, the thermal conductivity measured under this situation will be 15% higher than the true value. Viskanta (1962) calculated the simultaneous conduction and radiation in an absorbing medium for a parallel flat plate [70]. The calculated results showed that the radiation effects, which depend on the distance between the two plates, was of the same order of magnitude as the conduction.

2. Concentric cylinder cases

The equations which govern the radiation and conduction heat transfer in concentric cylinders are given as follows:

\[
\dot{Q}_r = \epsilon_{1,2} \Gamma_s 2\pi l \left( T_1^4 - T_2^4 \right)
\]

(2.13)

where, again, \( \epsilon_{1,2} \) is the radiation interchange factor, \( \Gamma_s \) is the Stephan-Boltzmann constant, \( l \) is the length of cylinders, and \( r_1, r_2 \) are the radii of the inner and outer
Figure 2.1: Radiant heat transfer percentage in parallel flat plate using R-114 as test fluid
cylinders, respectively. For this case, \( \epsilon_{1,2} \) is given as:

\[
\frac{1}{\epsilon_{1,2}} = \frac{1}{\epsilon_1} + \frac{r_1}{r_2} \left( \frac{1}{\epsilon_2} - 1 \right)
\]  

(2.14)

In this case, the total heat transferred by conduction is expressed as:

\[
\dot{Q}_t = \frac{2\pi \lambda (T_1 - T_2)}{\ln \left( \frac{r_2}{r_1} \right)}
\]

(2.15)

The ratio of radiation to the total heat transfer is then calculated by dividing these two equations, resulting in

\[
\frac{\dot{Q}_r}{\dot{Q}_t} = \frac{\Gamma s r_1 \left( T_1^4 - T_2^4 \right) \ln \left( \frac{r_2}{r_1} \right)}{\lambda (T_1 - T_2) \left[ \frac{1}{\epsilon_1} + \frac{r_1}{r_2} \left( \frac{1}{\epsilon_2} - 1 \right) \right]}
\]

(2.16)

This ratio was calculated by using R-114 as a test fluid. Figure 2.2 shows the percentage of the ratio of radiant heat transfer to total heat transfer versus temperature for R-114 in concentric cylinder case. Compared with the case in parallel flat plate case, mentioned in Figure 2.1, the radiant ratio for both cases is quite similar. Again, it is not surprising that the radiant heat transfer is quite significant at high emissivities and high temperatures.

In summary, it can be stated that errors in thermal conductivity measurements introduced by radiant heat exchange in both parallel flat plate and concentric cylinder cells cannot always be assumed to be negligibly small. In order to achieve optimal accuracy, corrections should be made and the material selected for the test cell walls should be appropriate so that the errors can be minimized.

**Radiation effect on hot-wire cell**

In the transient hot-wire method, the essential relationship is that of the temperature rise of the thin wire immersed in the fluid as a function of time following the
Figure 2.2: Radiant heat transfer percentage in concentric cylinder cell using R-114 as test fluid
stepwise initiation of a heat flux within it. There have been a number of attempts
to carry out an analysis of the process of simultaneous conduction and radiation in
an absorbing fluid for the transient hot-wire instrument [71, 72, 73]. Menashe and
Wakeham analyzed the combined radiation and conduction heat exchange in a tran-
sient hot-wire cell in 1982 [72]. They theoretically calculated the apparent thermal
conductivity measured by this method and drew a conclusion that the apparent
thermal conductivity ratio, $\lambda_{app}/\lambda$, is proportional to the equilibrium temperature,

$$\frac{\lambda_{app}}{\lambda} \propto T_0^3$$  \hspace{1cm} (2.17)

They also calculated the $\lambda_{app}/\lambda$ for n-undercane at 348 K and found that the ap-
parent thermal conductivity of the fluid was as much as 2.5% greater than the true
value.

Akio Saito et al. analyzed the radiation effect on a transient line source measure-
ment by a theoretical approach [73]. Akio Saito et al. analyzed the ratio of apparent
thermal conductivity (radiation involved) to true thermal conductivity and found
it was a function of radiation properties such as emissivity, refractivity, equilibrium
temperature, and geometric parameters, such as the radius of the line source. The
results are tabulated in Table 2.2 (values were cited from Figure 6 of Ref. [73]). In
Table 2.2, the non-dimensional group ($\phi$) of apparent thermal conductivity ratio is
defined as:

$$\phi = \left(\frac{\lambda_{app}}{\lambda} - 1\right) \frac{\pi \lambda}{16 r_1 \Gamma s n^2 T_0^3}$$  \hspace{1cm} (2.18)

$$R = r_1 \cdot k_a$$  \hspace{1cm} (2.19)

where $k_a$ is absorption coefficient, $m^{-1}$. 
Table 2.2: $\phi$ values$^a$ for apparent thermal conductivity ratio

<table>
<thead>
<tr>
<th>$\epsilon$</th>
<th>$R = 0.01$</th>
<th>$R = 0.075$</th>
<th>$R = 0.001$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\epsilon = 0.2$</td>
<td>2.750</td>
<td>2.125</td>
<td>0.750</td>
</tr>
<tr>
<td>$\epsilon = 0.4$</td>
<td>3.188</td>
<td>2.500</td>
<td>1.250</td>
</tr>
<tr>
<td>$\epsilon = 0.6$</td>
<td>3.625</td>
<td>3.000</td>
<td>1.750</td>
</tr>
<tr>
<td>$\epsilon = 0.8$</td>
<td>4.063</td>
<td>3.500</td>
<td>2.313</td>
</tr>
<tr>
<td>$\epsilon = 1.0$</td>
<td>4.500</td>
<td>4.000</td>
<td>2.813</td>
</tr>
</tbody>
</table>

$^a$ values were cited from Figure 6 of Reference [73].

$\phi$ was defined in Equation 2.18.

Table 2.2 shows the $\phi$ values (read from Figure 6 of Ref. [73]) for various $\epsilon$ and $R$. Because $\phi$ is proportional to $\lambda_{app}/\lambda$, an increasing tendency of $\lambda_{app}/\lambda$ with larger $\epsilon$ and $R$ was obtained from Table 2.2. These results, as expected, have the same trends for both the parallel flat plate and the concentric cell arrangements.

In Equation 2.18, the equilibrium temperature, $T_0$, is another function that affects the radiation strength. A plot of $\lambda_{app}/\lambda$ versus $T_0$ is interesting to view. Applying Equation 2.18 to a test condition, the $\lambda_{app}/\lambda$ ratio can be shown. Figures 2.3 through 2.5 show plots of $\lambda_{app}/\lambda$ versus $\lambda$ for various $T_0$ cases at $\epsilon = 0.2$, $\epsilon = 0.6$, and $\epsilon = 1.0$, respectively.

As shown in these figures, the ratio of $\lambda_{app}/\lambda$ generally increases with increasing temperature, $T_0$. In addition, at lower thermal conductivity ranges, this ratio becomes more significant than it is at higher thermal conductivity ranges. Figure 2.6 shows the $\lambda_{app}/\lambda$ versus $\lambda$ at a high temperature of $T_0 = 120^\circ$C for various emissivities. As expected, in this case of higher emissivity, more radiation heat transfer contributes to the conduction heat transfer so that the ratio, $\lambda_{app}/\lambda$, becomes higher. Figure 2.7 shows the $\lambda_{app}/\lambda$ versus temperature at $\epsilon = 1.0$ for different thermal con-
apparent thermal conductivity ratio in transient hot-wire method

Legend

- - - T₀=20 C
- - - T₀=50 C
- - - T₀=120 C

calculated conditions:
R=0.01
wire diameter, d=0.002 cm
emissivity, ε=0.2

Figure 2.3: $\lambda_{app}/\lambda$ ratio versus $\lambda$ for various temperatures at $\varepsilon = 0.2$ case
Figure 2.4: $\lambda_{app}/\lambda$ ratio versus $\lambda$ for various temperatures at $\epsilon = 0.6$ case
Figure 2.5: $\lambda_{app}/\lambda$ ratio versus $\lambda$ for various temperatures at $\varepsilon = 1.0$ case
ductivity ranges. It indicates that the higher the temperature, the higher the ratio, and for lower thermal conductivity fluids, this tendency would be more significant. These results show that in the transient hot-wire method the temperature and emissivity of the wire are two important factors which dominate the thermal conductivity measurement. C. A. Nieto de Castro (1991) [74] stated the contribution of radiative heat transfer to thermal conductivity was found to range between 2 to 50%.

C.A. Nieto de Castro et al. investigated toluene and dimethylphthalate liquid thermal conductivity [75]. They indicated that the radiant effect on the transient hot-wire method is quite significant without any corrections. Gross (1992) [76] measured the thermal conductivities of refrigerants R-134a, R-152a, and R-123 using the hot-wire transient method. Although the accuracy was claimed within ±1.6 ~ 2.0% with proper corrections, the uncertainties were attributed to the ideal mathematical model of a line source in infinite space and the effects of convection and radiation.

Another factor which affects the thermal conductivity measurement error was the wire diameter. From Saito's apparent thermal conductivity ratio equation, Equation 2.18, the apparent thermal conductivity ratios were dominated by the diameter of the thin wire applied. Table 2.3 indicates how the ratio was affected by the wire diameter used. As shown in Table 2.3, the radiation effect is significantly affected by the wire diameter and never completely eliminated although minimized by using a thin wire. However, the experimental uncertainty of the smaller diameter wire would be larger than that for a larger diameter wire. Therefore, an appropriate wire diameter must be selected in order to achieve optimal accuracy by this method.
apparent thermal conductivity ratio in transient hot-wire method

Legend
- emissivity, $e=0.2$
- emissivity, $e=0.6$
- emissivity, $e=1.0$

calculated conditions:
$R=0.01$
wire diameter, $d=0.002$ cm
equilibrium temp., $T_0=120$ C

Figure 2.6: $\lambda_{app}/\lambda$ ratio versus $\lambda$ for various $e$ at high temperature case
radiation effect on thermal conductivity measurement
(transient hot-wire method)

Figure 2.7: $\lambda_{app}/\lambda$ ratio versus temperature for different thermal conductivity ranges
Table 2.3: Ratio of apparent thermal conductivity

<table>
<thead>
<tr>
<th>diameter, μm</th>
<th>$\frac{\lambda_{app}}{\lambda}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.0</td>
<td>1.003</td>
</tr>
<tr>
<td>10.0</td>
<td>1.006</td>
</tr>
<tr>
<td>50.0</td>
<td>1.030</td>
</tr>
<tr>
<td>100.0</td>
<td>1.060</td>
</tr>
</tbody>
</table>

1. $\frac{\lambda_{app}}{\lambda}$ was calculated by Equation 2.18.
2. Cases for tulene with the following parameters:
   - Equilibrium temperature, $T_0$: 360 K
   - $R$: 0.01
   - Thermal conductivity, $\lambda$: 0.1137 $W/m \cdot K$
   - Refractive index, $n$: 1.4961
   - Stefan-Boltzmann constant, $\Gamma$: $5.6697 \times 10^{-8} W/m^2K^4$

Convection effects on thermal conductivity measurement

The design of an experimental apparatus for the determination of the thermal conductivity of a liquid should preclude the possibility of the appearance of free convection [77, 78]. Tsederberg [66] mentioned Kraussold's study of the conditions at which free convection occurs for various liquids located between two concentric cylinders when thermal conductivity was known. Kraussold found that free convective heat transfer arises when $(Gr \times Pr) \geq 1000$. Kraussold indicating that both horizontal and vertical placements of the tube were similar, analyzed the apparent thermal conductivity ratio, $\epsilon$, for concentric cylinders, and developed the following correlation:

$$\frac{\lambda_{app}}{\lambda} = 0.11(Gr \cdot Pr)^{0.29} \quad (2.20)$$

This equation is only valid for $3.8 \leq \log(Gr \cdot Pr) \leq 6.0$. Instead of Kraussold's equation, Equation 2.20, Wakeham et al. [67] used a different equation to estimate
the heat transfer by convection, which for a cylindrical geometry, is:

\[
\frac{\dot{Q}_{\text{conv}}}{\dot{Q}_{\text{cond}}} = \frac{dRa}{720l}
\]

(2.21)

where \(d\) is the thickness of the fluid layer and \(l\) the length of internal cylinder. A quantitative list is shown in Table 2.4.

Table 2.4: An example of convection effects on thermal conductivity in cylindrical geometry

<table>
<thead>
<tr>
<th>Apparent thermal conductivity ratio (based on (l = 120,mm))(^a)</th>
<th>(\lambda_{\text{app}}/\lambda(d = 0.2,mm))</th>
<th>(\lambda_{\text{app}}/\lambda(d = 0.3,mm))</th>
</tr>
</thead>
<tbody>
<tr>
<td>(Ra = Gr \cdot Pr)</td>
<td>(\lambda_{\text{app}}/\lambda(d = 0.2,mm))</td>
<td>(\lambda_{\text{app}}/\lambda(d = 0.3,mm))</td>
</tr>
<tr>
<td>1000</td>
<td>1.003</td>
<td>1.004</td>
</tr>
<tr>
<td>5000</td>
<td>1.013</td>
<td>1.021</td>
</tr>
<tr>
<td>10000</td>
<td>1.027</td>
<td>1.042</td>
</tr>
</tbody>
</table>

\(^a\) cited from Table 6.1 of Reference [67]

From Table 2.4, it is noted that the significance of free convection depends on the Rayleigh number, \(Ra\). For large \(Ra\), the apparent thermal conductivity is quite large. Therefore, to decrease the effect of free convection, control of \(Ra\) is necessary, and the measured thermal conductivity also needs to be corrected even when \(Ra\) is controlled at small values. For a non-steady method, such as the transient hot-wire method, van der Held and van Drunen, and Tye [65] also mentioned and suggested that there might be a corresponding pseudo-steady state, and convection would be expected to occur when the pseudo-steady state temperature reaches a critical value defined by \(Ra \equiv Gr \cdot Pr \geq 1000\) [65]. He derived a criterion for the onset of convection in a non-steady hot-wire cell:

\[
\left( \frac{g\alpha \rho Q r^3}{4\pi \lambda u} \right) T^* \left( e^{0.5 T^*} - 1 \right)^3 \leq 1000
\]

(2.22)
where \( T^* = \frac{4\pi \lambda T_\infty}{Q} \), \( \mu \) dynamic viscosity of liquid, \( a \) wire heat capacity, and \( \alpha \) thermal diffusivity. A plot of \( t^* = \frac{4at}{r_0^2} \) against \( T^*(e^{0.5T^*} - 1) \) can be used to determine the true time for the onset of convection, and the necessary error correction can be made.

**Summary of thermal conductivity measuring methods**

The methods mentioned above for experimental thermal conductivity determination are summarized as: the steady-state method and the unsteady-state method. The main difference between the steady-state and unsteady method is the amount of time consumed. The steady-state method requires more time to reach steady state, which affects the accuracy of measurement because the mathematical model is based on the real steady state conditions, whereas the unsteady-state method allows prompt measurements to be taken. Both methods demand careful experimentation to minimize corrections for heat losses and to prevent the onset of convective motion in the fluids. However, other effects such as time counting deviation, geometric errors, temperature sensor errors, and constant surface temperature controlling scheme, also contribute the overall uncertainty for both methods. Table 2.5 is a brief discussion of those steady-state methods mentioned above, and Table 2.6 describes the advantages and disadvantages for unsteady-state methods.
Table 2.5: Summary of advantages and disadvantages of steady-state methods of thermal conductivity measurements

<table>
<thead>
<tr>
<th>method</th>
<th>horizontal, flat-plate and vertical, coaxial cylinder method</th>
<th>hot-wire method</th>
<th>concentric sphere method</th>
</tr>
</thead>
<tbody>
<tr>
<td>advantages</td>
<td>· static fluid - no flow</td>
<td>· static fluid - no flow</td>
<td>· static fluid - no flow</td>
</tr>
<tr>
<td></td>
<td>· no other properties related</td>
<td>· simplicity and ease construction</td>
<td>· no other properties related</td>
</tr>
<tr>
<td>disadvantages</td>
<td>· difficult in eliminating convection effect</td>
<td>· axial conduction effect of the wire</td>
<td>· difficult in apparatus construction and accurate alignment</td>
</tr>
<tr>
<td></td>
<td>· perturbation along the edges of the heated plates</td>
<td>· precise determination of the wire and tube diameter</td>
<td>· effect of convection</td>
</tr>
<tr>
<td></td>
<td>· difficult in controlling isothermal conditions on the surface of the plates</td>
<td>· accurate coaxial alignment of the wire in the tube</td>
<td>· difficult in real surface temperature measurement</td>
</tr>
<tr>
<td></td>
<td>· radiation effect</td>
<td>· real steady state reaching</td>
<td></td>
</tr>
<tr>
<td></td>
<td>· real steady state reaching</td>
<td>· time consuming</td>
<td></td>
</tr>
<tr>
<td>approximate error range</td>
<td>±1.0 ~ ±15%</td>
<td>±1.0 ~ ±18%</td>
<td>±1.0 ~ ±15%</td>
</tr>
</tbody>
</table>
Table 2.6: Summary of advantages and disadvantages of unsteady-state methods of thermal conductivity measurements

<table>
<thead>
<tr>
<th>method</th>
<th>hot-wire cell - line source</th>
<th>cylindrical, spherical source</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>advantage</strong></td>
<td>• less time consuming</td>
<td>• less time consuming</td>
</tr>
<tr>
<td></td>
<td>• no thermal-resistance layer</td>
<td>• simple construction</td>
</tr>
<tr>
<td><strong>disadvantage</strong></td>
<td>• effect of finite length (originally assume infinitely long)</td>
<td>• effect of heat capacity</td>
</tr>
<tr>
<td></td>
<td>• effect of heat capacity depending on temperature</td>
<td>• effect of thermal boundary layer</td>
</tr>
<tr>
<td></td>
<td>• effect of bounding wall</td>
<td>• onset of convection effect (Ra≥1000)</td>
</tr>
<tr>
<td></td>
<td>• onset of convection effect (Ra≥1000) - when time going</td>
<td>• radiation effect never completely eliminated - correction is difficult</td>
</tr>
<tr>
<td></td>
<td>• radiation effect never completely eliminated - correction is difficult</td>
<td></td>
</tr>
<tr>
<td>approximate error range</td>
<td>±1.0 ~ ±18%</td>
<td>±1.0 ~ ±15%</td>
</tr>
</tbody>
</table>

The uncertainty sources are generally known for all of these methods. Typically, the main uncertainties have been catalogued into two sources: radiation error and convection error. Table 2.7 shows the errors of these main sources.

**Experimental methods for viscosity measurement**

The methods used for the viscosity measurements include the capillary tube [32, 33, 79, 80, 81], falling cylinder [82, 83], rolling ball [84, 85], vibrating-wire method [34], and torsional oscillation method [36, 38], etc. The following is a brief introductory
Table 2.7: Summary of typical uncertainties for experimental thermal conductivity measurements

<table>
<thead>
<tr>
<th>method</th>
<th>radiation</th>
<th>convection</th>
<th>total error</th>
</tr>
</thead>
<tbody>
<tr>
<td>parallel flat plate or concentric cylinder</td>
<td>emissivity and temp. depending - low temp.: $0.1 \sim 1.0%$ ($\varepsilon = 0.1 \sim 1.0$) high temp.: $1.0 \sim 8.8%$ ($\varepsilon = 0.1 \sim 1.0$)</td>
<td>onset $Ra = 1000$ 0.3% ~ 4.2% $Ra:1000 ~ 10000$</td>
<td>$\pm 15%$</td>
</tr>
<tr>
<td>transient hot-wire method</td>
<td>emissivity, wire size, and temp. depending - $\varepsilon$ effect: $1.0% \sim 10%$ wire diameter: $0.3% \sim 6%$ temp. effect: $1% \sim 10%$</td>
<td>less than $1%$</td>
<td>$\pm 18%$</td>
</tr>
</tbody>
</table>

review of these methods.

Capillary viscometer

The capillary viscometer [32, 33] uses the flowing characteristics of a capillary tube with a certain length. The viscosity is related to the following quantities: volumetric flow rate, pressure drop, density of fluid, thermal expansion coefficient of the capillary, kinetic-energy correction factor, and the capillary constant. Of which, the capillary constant is determined by a known viscosity fluid, usually pure water. The viscosity is calculated from a modified Hagen-Poiseuille equation.

Vibrating-wire viscometer

The vibrating-wire viscometer [34] is a viscometer for Newtonian fluids. A circular-section beam of constant length with both ends clamped at fixed points is subjected to a working fluid. The beam is set into oscillation in a single mode
within a plane containing the axis of the beam and perpendicular to it by means of an initial displacement at time $t=0$ in an initially stationary fluid of constant viscosity and density. The viscosity is determined by relationship between the frequency and the magnitude of the oscillation of the wire in a fluid. A correction is required for the specific wire.

**Torsional oscillation viscometer**

This kind of viscometer uses the principle of surface loading: a vibrating surface in contact with a liquid experiences a force which is a function of viscosity [36, 37]. The oscillatory surface may be spherical, cylindrical, or planar. The wave propagation is created outward by the oscillating surface. To eliminate the complexity of counting reflected wave effects, the container size must be large enough so that no disturbance reaches the walls. The viscosity can be found from the relationship between the frequency of the oscillation and the shear rate.

The viscosity measurement in the current study was measured by this kind of viscometer.

**Summary of experimental investigations of liquid viscosity of refrigerants**

Table 2.8 is a summary of previous experimental investigations of liquid refrigerants [81].
Table 2.8: List of previous investigations of liquid refrigerant viscosity measurements

<table>
<thead>
<tr>
<th>Investigator</th>
<th>Year</th>
<th>Refr.</th>
<th>Temp. (K)</th>
<th>Pres.</th>
<th>Method</th>
<th>Acc. (%)</th>
<th>Ref.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Benning</td>
<td>1938</td>
<td>R-11</td>
<td>243-333</td>
<td>sat.</td>
<td>roll. ball</td>
<td>-</td>
<td>[84]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-12</td>
<td>243-332</td>
<td>sat.</td>
<td>roll. ball</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-22</td>
<td>240-318</td>
<td>sat.</td>
<td>roll. ball</td>
<td>-</td>
<td></td>
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<tr>
<td></td>
<td></td>
<td>R-113</td>
<td>243-333</td>
<td>sat.</td>
<td>roll. ball</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-114</td>
<td>242-332</td>
<td>sat.</td>
<td>roll. ball</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>Lilios</td>
<td>1957</td>
<td>R-12</td>
<td>243-332</td>
<td>sat.</td>
<td>capillary</td>
<td>2.2</td>
<td>[85]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-13b1</td>
<td>203-218</td>
<td>sat.</td>
<td>roll. ball</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-113</td>
<td>243-293</td>
<td>sat.</td>
<td>roll. ball</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>Gordon</td>
<td>1969</td>
<td>R-11</td>
<td>245-347</td>
<td>sat.</td>
<td>capillary</td>
<td>-</td>
<td>[79]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-12</td>
<td>245-305</td>
<td>sat.</td>
<td>capillary</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-22</td>
<td>246-312</td>
<td>sat.</td>
<td>capillary</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-13b1</td>
<td>246-301</td>
<td>sat.</td>
<td>capillary</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>Phillips</td>
<td>1970</td>
<td>R-11</td>
<td>209-352</td>
<td>sat.</td>
<td>capillary</td>
<td>-</td>
<td>[80]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-12</td>
<td>202-312</td>
<td>sat.</td>
<td>capillary</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>R-22</td>
<td>201-299</td>
<td>sat.</td>
<td>capillary</td>
<td>-</td>
<td></td>
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<tr>
<td></td>
<td></td>
<td>R-152a</td>
<td>200-318</td>
<td>sat.</td>
<td>capillary</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>Kumagai</td>
<td>1990</td>
<td>11 kinds</td>
<td>273-353</td>
<td>sat.</td>
<td>capillary</td>
<td>±9%</td>
<td>[81]</td>
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<td></td>
<td></td>
<td>Refri.</td>
<td></td>
<td></td>
<td></td>
<td>~ 13%</td>
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<tr>
<td>Okubo</td>
<td>1992</td>
<td>R-123</td>
<td>233-418</td>
<td>20 MPa</td>
<td>capillary</td>
<td>± 1.2</td>
<td>[33]</td>
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<td></td>
<td>R-134a</td>
<td>213-423</td>
<td>30 MPa</td>
<td>capillary</td>
<td>± 1.3</td>
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<tr>
<td>Assael</td>
<td>1992</td>
<td>liquids</td>
<td>270-370</td>
<td>300MPa</td>
<td>vibrating</td>
<td>± 3</td>
<td>[34]</td>
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<td>Diller</td>
<td>1993</td>
<td>R-134a</td>
<td>175-320</td>
<td>sat.</td>
<td>torsional</td>
<td>± 3-8</td>
<td>[35]</td>
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<td></td>
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<td>crystal</td>
<td>c. others</td>
<td></td>
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<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>torsional</td>
<td>± 7-12</td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td>crystal</td>
<td>c. others</td>
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<td></td>
<td>torsional</td>
<td>less than</td>
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<td></td>
<td></td>
<td></td>
<td>crystal</td>
<td>±7-12</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>c. others</td>
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CHAPTER 3. EXPERIMENTAL APPARATUS AND OPERATION

This chapter describes the details and designs of the experimental facility. The construction of the test rig is described and shown. Sensors and test-section calibration are also presented. Finally, the experimental procedures are discussed for proper operation of this test facility.

Experimental facility construction

The experimental test rig was established for the purpose of measuring liquid transport properties of refrigerant, refrigerant mixtures, or refrigerant-lubricant mixtures. This experiment measures single-phase heat transfer coefficients of liquids by a surface temperature method.

Heat transfer test facility

The test-section is a 3/8 inch i.d. by 2 m long smooth copper tube. The measured quantities are tube wall temperature, inlet/outlet fluid temperatures, absolute and differential pressures, viscosity, and the mass flow rate. The apparatus of this test rig is shown in Figure 3.1. A photograph of this test rig is also shown in Figure 3.2.

Eleven T-type thermocouples are installed on the outer wall of the tube at equal distances of 0.2 m, starting from the inlet point and ending at the outlet point, along
Figure 3.1: Schematic diagram of test facility
Figure 3.2: A photograph of test facility
the 2 m long test section. In order to get more average temperature measurements at the inlet and outlet locations, two additional thermocouples were placed 0.1 m from the inlet and the outlet points, on the outer tube wall. Moreover, one thermocouple is placed on the outer insulated wall surface for measuring the temperature there. Also, one thermocouple is used for measuring the room temperature. The thermocouple for measuring room temperature is located in a wooden box which is wrapped with aluminum foil on the outer surfaces to insure the measurement of a radiation and wind flow free temperature. Figure 3.3 shows the details of the test-section configuration. A picture of the test-section is also shown in Figure 3.4. Two RTDs were placed right at the inlet and outlet points of the test section to measure the respective fluid temperatures. All thermocouples and RTDs were calibrated to ±0.05°C.

In order to get more accurate inlet and outlet temperatures, three plastic mixers were installed inside the tube for the purpose of determining a more accurate average fluid temperature. The first one was horizontally placed between the test section and the tube elbow, the second one was located vertically just prior to the tube elbow of the inlet portion, and the third one was installed horizontally immediately after the outlet of the test section.

Two plastic tube connectors which served as heat insulators were placed at both the inlet and outlet of the test-section in order to prevent axial heat loss to the ends of the test-tube. One was placed right before the inlet point of the test-tube, while the other was placed just after the outlet point of the test-tube.

The test section pressure was monitored by an absolute pressure transmitter and a differential pressure transmitter which measured the test section inlet and differential pressures along the test-tube, respectively.
Figure 3.3: Test-section configuration
Figure 3.4: A photograph of test-section
System pressure was controlled by an accumulator which was connected to a nitrogen vessel. Pressure, therefore, was regulated by the nitrogen pressure. System mass flow rate was adjusted by a variable-speed controlled motor-pump assembly. System heating control was achieved by two parallel power connected heat tapes with 675 watts each. The heating capacity was controlled by adjusting the power regulator which was connected to a watt transducer with an accuracy of ± 0.2% of the power readout. The heat tapes were uniformly wrapped along the test section as that a constant heat flux was expected and assumed.

After the heated test section, two condensers were placed between the outlet of the test-section and the inlet of the pumps for removing the heat that was added in the test section. Two pumps were installed in a parallel arrangement with each of the condensers in order to control a wide operating range of mass flow rates. The cooling capacity could also be controlled by adjusting the chiller water flow rate as shown in schematic diagram, Figure 3.1.

All the measurement signals were connected to a HP3457A, switch and control unit, and a HP3488A multimeter. Data acquisition and measuring process control was driven by a 386 PC.

Viscometer

Viscosity is simultaneously measured with other measurements in the test system. In this study, the viscosity is measured by an inline viscometer installed at the inlet portion of the test-section. Viscosity can be dynamically measured at the same time with the other measurements. This viscometer uses a torsional oscillation principle which is the principle of "surface load". A vibrating surface is in contact
with a liquid which experiences a force being a function of the viscosity [36, 37]. This type of viscometer measures the product of viscosity and density. The range of this instrument can vary from a low viscosity range of $0.1 \, cp \cdot g/cm^3$ to a high viscosity range of $500 \, cp \cdot g/cm^3$, which covers most of the pure refrigerants, refrigerant mixtures, and even refrigerant-lubricant mixtures with low lubricant concentrations. The accuracy of the viscosity measurement is ±2%. The fluids is controlled by a bypass valve which regulates flow in/out of a sample cell in which the sensor of the viscometer is mounted. The temperature of the fluid at the sample cell is also measured by a calibrated RTD with an accuracy of ±0.05°C. It should be noted that this kind of viscometer measures the product of kinematic viscosity and density. To get a viscosity readout, density is required. In this test rig, density is directly measured by a densimeter of the vibrating-wire type [48], which is installed in series with the viscometer. A schematic diagram is shown in Figure 3.5. A picture of the viscometer sample cell is shown in Figure 3.6.

**Sensor calibrations**

The physical quantities of direct measurement of this test rig include temperature, mass flow rate, viscosity, pressure, and power input. The temperature measurements include inlet and outlet temperatures, test-tube wall temperatures, insulated wall temperature, and the ambient temperature. The inlet and outlet temperatures are measured by a RTD sensor while the other temperature sensors are thermocouple sensors. All the RTD and thermocouple sensors were calibrated by a standard thermometer with an accuracy up to ±0.05°C.

The other measurements such as mass flow rate, viscosity, pressure, and power
Figure 3.5: Viscometer construction
Figure 3.6: Viscometer sample cell
were calibrated at the factory. The accuracy of each sensor is listed in Table 3.1.

**Test-section heat loss calibration**

An energy balance principle not only can be applied to estimate the heat loss of the test-section for fluids with known specific heats, but it can also be applied in reverse to obtain the heat loss for fluids of unknown specific heats. However, to get accurate specific heat measurements, heat loss calibrations with several fluids with known specific heats are required. It should be noted that the heat loss estimation will be dependent on the specific test-section with its own geometric and insulating characteristics. In other words, the heat loss characteristic is only valid for the test-section being used for calibration and cannot be applied to other test-sections or any situations changed from the original.

The heat loss of the test-section, which is used to calculate the net heat input by measuring the total power input, was obtained from the calibration of previously tested refrigerants which are R-22, R-12, and R-113. The accuracy of the heat loss calibration was tested by R-114. Then, the net heat input of the test-section was used to calculate the specific heat of the test fluids. Figure 3.7 shows the logarithmic

<table>
<thead>
<tr>
<th>sensor</th>
<th>unit</th>
<th>uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>mass flow rate</td>
<td>kg/s</td>
<td>±0.15%</td>
</tr>
<tr>
<td>pressure</td>
<td>psia, psid</td>
<td>±0.2%</td>
</tr>
<tr>
<td>viscosity</td>
<td>cp · g/cm(^3)</td>
<td>±2%</td>
</tr>
<tr>
<td>density</td>
<td>kg/m(^3)</td>
<td>±0.15%</td>
</tr>
<tr>
<td>power</td>
<td>Watts</td>
<td>±0.2%</td>
</tr>
</tbody>
</table>
function of the test-section heat loss associated with logarithmic function of the temperature difference between the average outer insulated surface temperature, \( T_s \), and average ambient temperature, \( T_a \).

The curve fit equation of the heat loss associated with the \( \ln(T_s - T_a) \) is shown below:

\[
\ln(Q_{loss}) = 0.612 + \ln(T_s - T_a)^{1.33} \tag{3.1}
\]

It should be noted that, again, this equation will be only applicable to this particular test-section under the specific construction including the particular insulation. For any other construction, the heat loss must be re-calibrated and the heat loss equation must also be re-developed.

**Accuracy verification of densimeter**

Although the accuracy of densimeter was claimed to be \( \pm 0.15\% \) within 1300 kg/m\(^3\), it was necessary to verify the accuracy with refrigerants of known density. The verified refrigerants used were R-22, R-12, R-113, and R-114, which were also the refrigerants selected to calibrate measured properties by using the current methodology and facility. Figure 3.8 shows the plot of measured density versus the ASHRAE density. As indicated in the deviation band in this figure, deviation was shown to be within \( \pm 2\% \). However, it is necessary to mention that some of the measured densities were higher than 1300 kg/m\(^3\), which is considered the upper level with highest accuracy. Moreover, all the densities measured were subjected to a compressed liquid state rather than a saturation state. As it can be seen, most densities were measured within \( \pm 1\% \).
heat loss calibration: $\ln(Q_{loss})$ vs. $\ln(T_s - T_a)$

$\ln(Q_l) = 0.61201419 + (1.33022535)\ln(T_s - T_a)$

Figure 3.7: Heat loss estimation versus wall-fluid temperature difference
Figure 3.8: Measured densities versus ASHRAE densities
Accuracy verification of viscometer

Viscosity is measured by an inline torsional oscillation viscometer. This type of instrument measures the product of viscosity and density. The accuracy was verified by R-12, R-114, R-113, and pure water. Figure 3.9 shows the plot of the measured viscosity of R-12 versus temperature compared with ASHRAE data. The deviation percentage plot is shown in Figure 3.10. As the plots show, the measured viscosity matches the ASHRAE data within ±2%. The measured viscosity of R-114 and its deviation compared to ASHRAE data are shown in Figures 3.11 and 3.12, respectively, while the measured viscosity of R-113 and its deviation compared to ASHRAE data are shown in Figures 3.13 and 3.14, respectively. The repeatability of the measurements was tested by R-114, which began with the temperature around room temperature and went down to nearly 0° C. Then, the temperature went up to nearly 50° C and went down to room temperature. Finally, the measuring stability of viscosity was verified with pure water. This test measured the viscosity of pure water at room temperature for nearly 17 hours. The measured viscosity of pure water and its deviation with Chemistry Physics Handbook data [86] are shown in Figures 3.15 and 3.16, respectively. As can be seen, the measured accuracy was within ±2%.
R12 viscosity vs. temperature

Legend

- measured data
- ASHRAE data

measured eq. \( \text{viscosity(cp)} = 0.24821 - 0.01885^\circ C \)

ASHRAE eq. \( \text{viscosity(cp)} = 0.25202 - 0.01985^\circ C \)

Figure 3.9: R-12 measured viscosity versus temperature
Figure 3.10: R-12 measured viscosity deviation
Figure 3.11: R-114 measured viscosity versus temperature
Figure 3.12: R-114 measured viscosity deviation
Figure 3.13: R-113 measured viscosity versus temperature

measured eq.: \( \text{viscosity (cp)} = 0.9540 - 0.01321^\circ \text{C} \)

ASHRAE eq.: \( \text{viscosity (cp)} = 0.9531 - 0.01263^\circ \text{C} \)
Figure 3.14: R-113 measured viscosity deviation
Figure 3.15: Pure water measured viscosity versus temperature
Figure 3.16: Pure water measured viscosity deviation for stability test
Experimental operation

This section describes the operation of the test rig and the experimental procedures. In order to obtain optimal test data, the test system should be operated under specific procedures. Based upon operational experiences, these procedures must be followed by the operator in order to achieve success. The test procedures are outlined in the following discussion.

Test system preparation

Before charging the facility with refrigerant, the test system should be evacuated. A system pressure less than 0.4 psia is considered to be ready for evacuation. The chiller can be turned on and the chiller working fluid can be circulated through the heat exchangers while the pressure approaches 0.4 psia. Meanwhile, the refrigerant to be used can be pre-heated in order to increase the pressure which eases transfer into the system. During this time, the data acquisition system is monitoring the status of the system.

Some potential problems can occur if proper preparations are not made. They are:

- Check the material compatibility with refrigerants in all wet parts (seals). Elastimor material, such as pressure relief valve seals, accumulator bladder, and pump diaphragm could be eroded by refrigerants due to incompatibility.

- Leakage of the system could cause a loss of refrigerant or an intake of atmospheric air. This could cause problems if non-condensible gases were to exist in the system, and an adequate system evacuation could be difficult to achieve.
Leaks should be detected and fixed for losses of 2 psia (or more) in a 24 hour period.

- In order to store more refrigerant in the accumulator, chiller glycol must be circulated around the accumulator in order to keep the refrigerant cool and ease charging the accumulator. In addition, the bladder of the accumulator should be pre-charged to a pressure up to 40 psia in order to control the space for receiving the refrigerant. Failure to do this will result in improper functioning of the accumulator.

- Check all the valves to make sure they are appropriately opened or closed.

- Monitor the system status to see if it is in the desired mode (pressures and temperatures).

- A refrigerant can is used as a buffer for overcharging refrigerant and is required by the facility for reclaiming extra refrigerant due to density changes (temperature fluctuations). This can should be properly evacuated when hooked up to the system.

**Viscometer preparation**

This type of viscometer measures the oscillation frequency of a cylinder sensor which is immersed in the refrigerant in a sample cell. There are eight bolts surrounding the flange to connect the transmitter portion and the sample cell. The torque balance of the bolts is very sensitive to the frequency output which converts to digital current output to DAS (Data Acquisition System). The following procedures are required for viscometer preparation:
Carefully lift up the viscometer sensor and carefully clean the sensor surface with a soft tissue to make sure no impurities are deposited on the surface.

Check the gasket material and inspect any damage. Replace it if it is necessary.

Carefully place the sensor into the sample cell while lining up all the bolts.

Tighten all eight bolts with an equal torque of 45 ft - lb.

Eliminate all possible vibration sources in order to avoid any effects on the viscometer frequency.

Adjust potentiometer, a zero voltage adjuster, of the viscometer console to zero output current at 4 mA when sensor is subjected to evacuation (no material). If this is not properly done, an offset could be introduced to the measurement.

Charging refrigerants

The following steps outline the procedures for charging refrigerants:

Make sure system is evacuated at a proper pressure and chiller glycol-water is circulating in the heat exchangers.

Slowly open the valve and let refrigerant flow into the system.

Watch sight-glass to make sure refrigerant is flowing into the system.

Make sure the liquid refrigerant fills up the housings of the refrigerant pumps. Then, turn on the pumps and operate at a low speed.

Release the gas inside the accumulator to allow proper space for the storage of refrigerant.
• Continue charging refrigerant until refrigerant is seen to fully fill the plastic tubes at the highest point of the system. Then, close the refrigerant valve.

• When the refrigerant is properly charged, accumulator glycol could be disconnected. Charge nitrogen pressure into the accumulator to the desired pressure.

• When charging non-azeotropic refrigerant mixtures, because of the influence of composition to the properties, the liquid mixture must be charged to the test system. Vapor charging should be avoided. In addition, to prevent the composition from changing, use of distilled refrigerants should also be avoided.

System operation and data acquisition

Data are taken under steady-state conditions. The following steps control steady-state procedures:

• Apply power to the heat tape and adjust power to desired wattage by variac.

• Apply pressure by adjusting the nitrogen regulation valve.

• Adjust the pump speed controller to control the refrigerant mass flow rate.

• Adjust the controlling valve of the glycol-water to a desired flow rate by watching rotameter.

• Steady-state of the test-section has been achieved when the heat input equals the heat removed. Repeat any of the above steps as required for reaching steady-state.
Steady-state is considered to be reached by monitoring the inlet temperature change of the test-section within ±0.05°C between two scans (approximately one minute). Then, a data point is taken as the average value of twenty consecutive scans.

Another test point may be taken by repeating the above procedures for a different inlet temperature, mass flow rate of refrigerant, or mass flow rate of chiller glycol.

Injecting lubricant and sampling lubricant concentration

Lubricant concentration is an important quantity for understanding lubricant effects on properties. There are a number of methods which could have been adopted in the current study, such as light absorption method [95], vibrating U-tube densimeter sensor and ultrasonic acoustic sensor [96, 97], viscometer lubricant concentration sensor [98], and sampling mass method. In the current study, the last method was adopted over the other methods due to their limited availability.

The injection of lubricant to the test system is achieved by an oil-injection piston with a six inch stoke cylinder. The lubricant concentration sample cell is in line with the injection cylinder. This assembly device is placed at the inlet portion of the test-section. The schematic diagram is shown in Figure 3.17. A picture of the oil-injection assembly and oil-sample device is shown in Figure 3.18. The operation procedures are described as follows.

Oil injection

Evacuate the lower space of the piston and fully close the valve at the oil outlet.
Figure 3.17: Schematic diagram of oil injection and sampling devices
Figure 3.18: A photograph of oil injection and sample cell
after completing evacuation.

- Evacuate the upper space of the piston and fully close the upper valve after finishing the evacuation.

- Suck oil into the cylinder from the lower oil outlet by slowly opening the needle valve and prevent air from leaking into the cylinder.

- Close the lower needle valve when finished charging the oil.

- Place the cylinder up-side-down and evacuate from the cylinder top by hooking up an evacuating hose to remove the residual air remaining or dissolved in the oil.

- Install the cylinder at the position for oil injection.

- Open the valves connected to the test system and apply high pressure nitrogen at the top of the piston.

- Inject oil into the system until the desired amount of oil is reached.

- Circulate refrigerant while the oil is injected and continuously circulate for at least 10 hours to insure the oil is uniformly mixed with the refrigerant. Finally, sample lubricant concentration.

**Lubricant concentration sampling**

- Close the valves of the the sample cell to isolate it from the system.
• Remove the sample cell and weigh the total mass of the cell including the refrigerant and lubricant.

• Slowly open the top valve of the sample cell which will allow the refrigerant to evaporate from the cell. An oil absorber hooked up to the cell can be used when the vacuum pump is applied.

• Weight of the mass of the cell and oil inside the cell is recorded until the weight remains constants.

• Calculate the oil concentration by dividing the net oil weight (oil in the cell + oil in the oil-absorber) by the total weight (refrigerant + oil).
CHAPTER 4. THEORY OF THERMAL CONDUCTIVITY MEASUREMENT

In this chapter, the theory of thermal conductivity measurement will be described. Two methods were developed in this study, which will be referred to as Approach 1: The Nusselt number method; and Approach 2: The Prandtl number method. In the Nusselt number method, the heat transfer characteristics of the test-section and Nusselt number correlations were used to obtain thermal conductivity, while in the Prandtl number method, thermal conductivity was directly related to the Reynolds number, $Re_D$, and the temperature characteristics of the test-tube. Detailed mathematical approaches are described below.

Approach 1: The Nusselt number method

In this method, thermal conductivity is obtained from heat transfer characteristics of a heated test-tube. From experimental measurements, an average heat transfer coefficient for single phase conditions in the test-tube can be measured. Meanwhile, the heat loss of the test-tube can also be measured for a specific test-tube. The average Nusselt number was calculated from the heat transfer measurements, and thermal conductivity was then calculated from the Nusselt number correlations. Details of the measuring principles are described as follows.
Heat transfer measurement

The local heat flux is calculated from the following equation:

$$q'' = h(T_w - T_f)$$

(4.1)

so that the heat transfer coefficient can be written as:

$$h = \frac{q''_{net}}{(T_w - T_f)}$$

(4.2)

where $T_w$, $T_f$ are the average tube wall temperature and fluid temperature, respectively. The quantity of $T_w - T_f$ can be calculated from either the individual average quantity of wall temperature, $T_w$ and fluid temperature, $T_f$ or the quantity of $T_w - T_f$, which is described in a later section. The net heat transfer rate, $q_{net}$ to the fluid can be calculated from the total heat input from power applied $q_{tot}$ and heat loss to the environment, $q_{loss}$, as follows:

$$q_{net} = q_{tot} - q_{loss}$$

(4.3)

The purpose of measuring heat loss from the test-section, $q_{loss}$, is to obtain the net heat input to the test-section, $q_{net}$. The heat loss is transferred to the environment around the test-section by natural convection and radiation. For calculating the heat loss, the following equation is used:

$$q_{loss} = h_o A_o (T_s - T_a)$$

(4.4)

where $T_s$ and $T_a$ are the average outer insulation surface temperatures and the ambient room temperatures, respectively, which can be obtained from experimental measurements. The overall heat transfer coefficient for the heat loss of the test-section, $h_o$, can be determined and calibrated from experimental measurements by
a known fluid's $C_p$. This equation has been already obtained and shown in Chapter 3, Equation 3.1. Once the heat loss characteristics of the test-section are obtained, they can be applied to other unknown fluids' $C_p$ calculation.

By knowing the test-section heat loss characteristics, the $I_{net}'$ can be calculated from an energy balance for a working fluid with an unknown $C_p$. Then, $C_p$ can be obtained as follows:

$$C_p = \frac{\dot{q}_{net}}{\dot{m}(T_o - T_i)}$$

where $T_i$ and $T_o$ are the mean inlet and outlet temperatures, respectively, while $\dot{m}$ is mass flow rate. This specific heat measurement is important for new refrigerant mixtures and blends and for refrigerant-lubricant mixtures when nonideal mixing may occur. In both cases, any theoretical calculation of $C_p$ from the pure fluids (either the pure refrigerant or pure lubricant) will not result in an accurate specific heat calculation.

**Nusselt number correlations**

Heat transfer characteristics are usually explained by the Nusselt number which is defined as $\overline{h}D/k$. A number of single-phase Nusselt number correlations have been published for this situation. Some examples are the Dittus-Boelter correlation [87, 88], the Petukhov and Popov correlation [89], and the Gnielinski correlation [90]. These three correlations are given below:

The Dittus-Boelter correlation (for heating):

$$\overline{N}u_D = 0.023Re_D^{0.8}Pr^{0.4}$$
The Petukhov correlation:

\[ \tilde{N}_uD = \frac{(f/8) Re_D Pr}{1.07 + 12.7 (f/8)^{1/2} (Pr^{2/3} - 1)} \quad (4.7) \]

The Gnielinski correlation:

\[ \tilde{N}_uD = \frac{(f/8) (Re_D - 1000) Pr}{1 + 12.7 (f/8)^{1/2} (Pr^{2/3} - 1)} \quad (4.8) \]

where \( f \) is friction factor. For smooth tube \( f \) are:

- \( f \) in Petukhov correlation,
  \[ f = (1.82 \log_{10} Re_D - 1.64)^{-2} \quad (4.9) \]

- \( f \) in Gnielinski correlation,
  \[ f = (0.79 \ln Re_D - 1.64)^{-2} \quad (4.10) \]

In the above equations, \( Re_D \) and \( Pr \) are defined as:

\[ Re_D = \frac{\rho V D}{\mu} \quad (4.11) \]
\[ Pr = \frac{\mu C_p}{k} \quad (4.12) \]

As mentioned previously, with \( \tilde{h} \) and \( \tilde{m} \) measured, the thermal conductivity can be obtained from the equality by setting the Nusselt number, \( \tilde{h} D/k \), equal to above correlations, Equations 4.6, 4.7, or 4.8.

**Calibration function**

The method used to determine thermal conductivity involves applying the Nusselt number correlations and backing out the thermal conductivity. Therefore, an
accurate Nusselt correlation is necessary in order to obtain an accurate value of thermal conductivity. Although some published Nusselt correlations have claimed good accuracy, the calibrations from the heat transfer data are still needed. This is because the correlations were obtained by curve fitting a large number of data with a wide range of Pr and Re, which might not apply to some specific fluids with a limited range of Pr and Re. Moreover, the thermal conductivity was originally used for obtaining those correlations. Now, it will be obtained in a reversed manner. Therefore, in order to obtain an acceptable thermal conductivity, an accurate Nusselt correlation is required in the specific range of Pr and Re.

The method used for calibrating Nusselt number correlations is one which assumes a calibrated function exists between the values of Nusselt number calculated from correlations and definition. The calibration function will be denoted as CF. With this assumption, a bunch of CF values can be obtained from experimental data. Then, the CF correlations can be obtained by a curve fit for various Pr and Re. The mathematical procedures are interpreted as follows.

The definition of the Nusselt number is:

\[
\tilde{N}_u = \frac{\dot{h}D}{k}
\] (4.13)

The CF, therefore, exists between \(\tilde{N}_u\) and correlations like

\[
CF = \frac{\tilde{N}_{uD_{experiment}}}{\tilde{N}_{uD_{correlation}}} = f(Pr, Re_D)
\] (4.14)

The calibrated Dittus-Boelter correlation (for heating):

\[
\tilde{N}_{uD} = 0.023Re^{0.8}Pr^{0.4}CF(Pr, Re_D)
\] (4.15)
The calibrated Petukhov correlation:

\[ \tilde{N}u_D = \frac{(f/8) \frac{Re}{Pr}}{1.07 + 12.7 (f/8)^{1/2} (Pr^{2/3} - 1)} CF(Pr, Re_D) \]  \hspace{1cm} (4.16)

The calibrated Gnielinski correlation:

\[ \tilde{N}u_D = \frac{(f/8) (Re_D - 1000) Pr}{1 + 12.7 (f/8)^{1/2} (Pr^{2/3} - 1)} CF(Pr, Re_D) \]  \hspace{1cm} (4.17)

Regression of calibration function

Because the Re and Pr are not of the same order of magnitude, a logarithmic transformation function is applied for convenience. Other types of transformations are possible. In the following equations, the superscript, e, denotes curve fit equations obtained from measured data while the superscript, c, denotes curve fit equations obtained from correlations.

\[ \ln \left( \tilde{N}u_D^e \right) = \ln \left( \tilde{N}u_D^c \right) + \ln (CF) \]  \hspace{1cm} (4.18)

or,

\[ \ln (CF) = \ln \left( \tilde{N}u_D^e \right) - \ln \left( \tilde{N}u_D^c \right) \]  \hspace{1cm} (4.19)

Then, a curve fit for \( \ln (CF) \) can be obtained. A linear relationship between \( \ln \left( \tilde{N}u_D/Pr^n \right) \) and \( \ln (Re_D) \) is well known from past knowledge. Therefore, an easy method to correlate these two logarithmic quantities is of the linear form:

\[ \ln \left( \tilde{N}u_D/Pr^n \right) = A + B \cdot \ln (Re_D) \]  \hspace{1cm} (4.20)

where \( A, B \) and \( n \) are the curve fit constants. These constants were obtained differently from the various applied correlations. The calibration function, CF, is therefore
obtained differently from the various correlations and is denoted as:

\[ CF = \frac{\tilde{N}_u^e}{\tilde{N}_u^c} \]  \hspace{1cm} (4.21)

Once the calibration function, \( CF \), is obtained as a function of \( Re_D \) and \( Pr \), the modified Nusselt correlations can be used to determine thermal conductivity as shown in the following sections.

With this assumption, a set of \( CF \) values can be obtained from experimental data and correlation calculations for different fluids. \( CF \) correlations can then be obtained by curve fitting, \( CF \) with \( Pr \) and \( Re_D \). The mathematical procedures are described below.

For in-tube heat transfer in this study, the following curve fit equations are used for experimentally measured Nusselt numbers, \( \tilde{N}_u^e \), and correlation Nusselt numbers, \( \tilde{N}_u^c \), respectively.

\[ \tilde{N}_u^e_D = C^e Re_D^{a^e} Pr^{b^e} \]  \hspace{1cm} (4.22)

\[ \tilde{N}_u^c_D = C Re_D^{a^c} Pr^{b} \]  \hspace{1cm} (4.23)

It should be noted that the Dittus-Boelter equation is already in this form while the other two in-tube equations mentioned earlier must be curve fit into this form for consistent treatment. In addition, the order of magnitude of \( Pr \) and \( Re_D \) is quite different. Again, a logarithmic function is recommended for convenient curve fitting. By dividing Equation 4.22 by Equation 4.23, \( CF \) can be expressed as:

\[ CF = \frac{C^e}{C} Re^{a^e-a} Pr^{b^e-b} \]  \hspace{1cm} (4.24)
Thermal conductivity calculations

After a calibrated \( \dot{q}_{net} \) is obtained, the heat transfer coefficient can be calculated from an average \( T_w - T_f \), which is measured. The Nusselt number will be obtained thereafter. Then, the thermal conductivity can be obtained from the calibrated Nusselt number correlations. However, the calibration function, CF, should be obtained from known property fluids. Once the calibration function, CF, is obtained as a function of \( Re_D \) and \( Pr \), the thermal conductivity can be expressed in general form as follows.

\[
k = \left[ CF \left( Pr, Re_D \right) \left( Nu_{D_{correlation}} \right) \right]^{-1} \overline{h} D \tag{4.25}
\]

where \( \overline{h} \) denotes the average heat transfer coefficient experimentally measured.

For each calibrated Nusselt number correlation, the \( k \) values are expressed as the following:

for Dittus-Boelter correlation:

\[
k_d = \left[ 0.023 Re_D^{0.8} Pr^{0.4} CF \left( Pr, Re_D \right) \right]^{-1} \overline{h} D \tag{4.26}
\]

for Petukhov correlation:

\[
k_p = \frac{1.07 + 12.7 (f/8)^{1/2} (Pr^{2/3} - 1)}{CF \left( Pr, Re_D \right) (f/8) Re_D Pr} \overline{h} D \tag{4.27}
\]

for Gnielinski correlation:

\[
k_g = \frac{1 + 12.7 (f/8)^{1/2} (Pr^{2/3} - 1)}{CF \left( Pr, Re_D \right) (f/8) \left( Re_D - 1000 \right) Pr} \overline{h} D \tag{4.28}
\]

For each of the three calibrated Nusselt correlations, the thermal conductivity equation is of a different form. The final equation for each correlation is presented
below. It is important to note that the Prandtl number used in each theoretical correlation still contains the thermal conductivity, k. Each of the three correlations handles this problem differently.

- The k equation from the Dittus-Boelter correlation

\[
k_d = \left[ \frac{-\bar{h}D}{CeReD\alpha^e (\mu C_p)^{b^e}} \right]^{\frac{1}{b^e}} \quad (4.29)
\]

- The k equation from the Petukhov and Popov correlation

\[
Pr^{2/3} = \frac{Ce(f/8)^{1/2} ReD^{a-a+1} \mu C_p}{12.7Ch^e D} Pr^{b-b} + \frac{1.07}{12.7 (f/8)^{1/2}} - 1 = 0 \quad (4.30)
\]

However, since k is implicit in Pr, k must be solved for by solving for Pr first from the above equation. In other words, k is divided into \(\mu C_p\) to obtain Pr which is then treated as the unknown in the above equation. This equation is not an explicit linear type, but it can be solved by a numerical iteration such as the Newton-Raphson method. Once Pr is determined, the thermal conductivity, k, can be obtained from \(\mu C_p/Pr\).

- The k equation from the Gnielinski correlation

As before, the k must be solved implicitly. The resulting Pr equation is:

\[
Pr^{2/3} = \frac{Ce(f/8)^{1/2} ReD^{a-a} (ReD - 1000) \mu C_p}{12.7Ch^e D} Pr^{b-b} + \frac{1}{12.7 (f/8)^{1/2}} - 1 = 0 \quad (4.31)
\]

Again, Pr in this equation must be determined by a numerical iteration method.
Approach 2: The Prandtl number method

In the previous section, we discussed the Nusselt number method for thermal conductivity calculation and the concept of the relative accuracy of the Nusselt number method by a calibration function. In this section, we will follow similar concepts but bypass the Nusselt number to discuss the Prandtl method for thermal conductivity calculations. Unlike the hot-wire transient method for thermal conductivity measurements, this method has only recently been applied to thermal conductivity measurements. This method measures the Prandtl number, and the thermal conductivity is calculated by knowing the Prandtl number, viscosity and specific heat. The mathematical approach follows.

Prandtl number and other related parameters

Based on past studies, the Nusselt number, $\bar{N}u_D^p$ in a turbulent channel flow was known as function of the Reynolds number, $Re_D$, and the Prandtl number, $Pr$. From the definition of Nusselt number, $\bar{N}u_D^p = \bar{h}D/k$, like the Dittus-Boelter correlation, the following equation is rewritten to represent heat transfer characteristics for a turbulent flow heated from the tube wall:

$$\frac{\bar{h}D}{k} = C \cdot Re_D^n \cdot Pr^n$$

(4.32)

where $\bar{h}$ is an average heat transfer coefficient which equals $\dot{Q}_{net}/A\Delta T_{wf}$. By introducing $\dot{m}$, $C_p$, and the temperature difference between the inlet and outlet, the above equation can be rearranged as:

$$\frac{\dot{m}C_p (T_o - T_i) D}{kA (T_w - T_f)} = C \cdot Re_D^n \cdot Pr^n$$

(4.33)
After substituting the definition of Prandtl number, this equation can also be rewritten as:

\[
P_r = \left[ \frac{D}{4CL} Re D^{1-m} \left( \frac{T_o - T_i}{T_w - T_f} \right) \right]^{\frac{1}{n-1}}
\]

(4.34)

In other words, the Prandtl number is expressed as a function of \( Re D \) and a dimensionless temperature group, \( \left( T_o - T_i \right) / \left( T_w - T_f \right) \). If this dimensionless temperature group is defined as \( \Delta T^* \), the relationship is expressed as:

\[
P_r = f \left( Re D, \Delta T^* \right)
\]

(4.35)

\[
\Delta T^* = \frac{T_o - T_i}{T_w - T_f}
\]

(4.36)

This is the relation for the measured \( P_r \). It should be noted that all the quantities on the right hand side of the equation can be obtained from experimental measurements. These involve \( \dot{m}, \mu, T_w, T_f, T_i, \) and \( T_o \). In other words, with the measurement of these quantities, the Prandtl number can be obtained. However, for fluids with known properties, the \( P_r \) can also be calculated from \( \mu C_p / k \). Therefore, the thermal conductivity can then be calculated from \( P_r \). This method is simpler than the Nusselt number calibration method.

**Prandtl number expression**

The \( P_r \) relationship has been shown in Equation 4.35, which related it to \( Re D \) and \( \Delta T^* \). An expression for \( P_r \) in terms of \( Re D \) and \( \Delta T^* \) is required for easy use, and it requires calibration. This calibration for \( P_r \) is very important to the accuracy of the thermal conductivity measurement. The expression of \( P_r \) with \( Re D \) and \( \Delta T^* \) in a closed form is very difficult to obtain because it requires a number of test data.
However, one of the represented approaches for this function is to directly express Pr into a function of $Re_D$ and $\Delta T^*$ in a polynomial form. An example of a full term 3-degree polynomial function considered adequately is given as:

$$Pr = a_0 + a_1 Re_D + a_2 \Delta T^* + a_3 Re_D^2 + a_4 (Re_D \cdot \Delta T^*) + a_5 \Delta T^{*2} + a_6 Re_D^3 + a_7 (Re_D^2 \cdot \Delta T^*) + a_8 (Re_D \cdot \Delta T^{*2}) + a_9 (\Delta T^{*3})$$

(4.37)

where the coefficients, $a_i$, can be fitted from measured data from fluids of known properties.

Uncertainty analysis

This section describes the uncertainties in the thermal conductivity measurements by the methods of Approach 1 and Approach 2. A propagation-of-error method [91] is used to estimate the experimental uncertainties associated with each experimental measurement. The sensor and equipment uncertainties are listed in Table 4.1. The derivation of all related uncertainty equations is also presented in Appendix B, which includes the thermal conductivity uncertainty by Approach 1 (three correlations) and Approach 2, heat exchange rate uncertainty, $Re_D$ uncertainty, friction factor ($f$) uncertainty, heat transfer coefficient uncertainty, $\Delta T^*$ uncertainty, and Pr uncertainty.
Sensor and geometry uncertainty

Based upon the uncertainties in the measured data (both sensor and geometry related), which is used to determine thermal conductivity, the uncertainty in \( k \) calculated from the three different correlations can be determined. Therefore, the uncertainty analysis for various ranges of \( Pr \) and \( Re_D \) was performed.

Sensor and geometry uncertainties are listed in Table 4.1.

Table 4.1: Sensor and geometry uncertainty

<table>
<thead>
<tr>
<th>sources</th>
<th>uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>length, ( L )</td>
<td>0.1 mm</td>
</tr>
<tr>
<td>diameter, ( D, D_0 )</td>
<td>0.1 mm</td>
</tr>
<tr>
<td>mass flow rate, ( m )</td>
<td>0.15% in ( kg/s )</td>
</tr>
<tr>
<td>temperature, ( T_i, T_o, T_w )</td>
<td>0.05°C</td>
</tr>
<tr>
<td>viscosity, ( \mu )</td>
<td>2.0%</td>
</tr>
<tr>
<td>specific heat capacity, ( C_p )</td>
<td>2.0%</td>
</tr>
</tbody>
</table>

Uncertainty estimation and comparison between Approaches 1 and 2

Based upon the existing sensor and equipment uncertainties, the thermal conductivity uncertainties were estimated and compared. In Approach 1, the thermal conductivity calculated from the three different correlations was estimated for a typical operating condition. The results are tabulated in Table 4.2. Table 4.2 shows the uncertainty, presented as the percentage of measured thermal conductivity, by Approach 1 for the three Nusselt number correlations. The uncertainty analysis was based upon sensor and geometry uncertainty listed in Table 4.1 for a particular operating condition listed at the bottom of Table 4.2. The uncertainty varies not only with the correlation used but also with various \( Re_D \) and \( Pr \).
Table 4.2: Thermal conductivity uncertainty percentages for the three Nusselt number correlations

<table>
<thead>
<tr>
<th>case</th>
<th>$Re_D$</th>
<th>$Pr$</th>
<th>$Ur_{dk}%$</th>
<th>$Ur_{pk}%$</th>
<th>$Ur_{gk}%$</th>
</tr>
</thead>
<tbody>
<tr>
<td>at</td>
<td>10000.00</td>
<td>1.0</td>
<td>6.75</td>
<td>6.43</td>
<td>6.35</td>
</tr>
<tr>
<td>lower</td>
<td>20000.00</td>
<td>1.0</td>
<td>6.79</td>
<td>7.42</td>
<td>6.97</td>
</tr>
<tr>
<td>Pr</td>
<td>40000.00</td>
<td>1.0</td>
<td>6.92</td>
<td>8.61</td>
<td>7.81</td>
</tr>
<tr>
<td></td>
<td>80000.00</td>
<td>1.0</td>
<td>7.29</td>
<td>10.29</td>
<td>9.13</td>
</tr>
<tr>
<td></td>
<td>160000.00</td>
<td>1.0</td>
<td>8.29</td>
<td>13.33</td>
<td>11.67</td>
</tr>
<tr>
<td>at</td>
<td>80000.00</td>
<td>1.0</td>
<td>7.29</td>
<td>10.29</td>
<td>9.13</td>
</tr>
<tr>
<td>middle</td>
<td>80000.00</td>
<td>6.0</td>
<td>6.84</td>
<td>5.97</td>
<td>5.79</td>
</tr>
<tr>
<td>$Re_D$</td>
<td>80000.00</td>
<td>11.0</td>
<td>6.80</td>
<td>5.46</td>
<td>5.36</td>
</tr>
<tr>
<td></td>
<td>80000.00</td>
<td>16.0</td>
<td>6.78</td>
<td>5.25</td>
<td>5.18</td>
</tr>
<tr>
<td>at</td>
<td>10000.00</td>
<td>1.0</td>
<td>6.75</td>
<td>6.43</td>
<td>6.35</td>
</tr>
<tr>
<td>lower</td>
<td>10000.00</td>
<td>6.0</td>
<td>6.73</td>
<td>5.08</td>
<td>5.24</td>
</tr>
<tr>
<td>$Re_D$</td>
<td>10000.00</td>
<td>11.0</td>
<td>6.73</td>
<td>4.90</td>
<td>5.09</td>
</tr>
<tr>
<td></td>
<td>10000.00</td>
<td>16.0</td>
<td>6.73</td>
<td>4.82</td>
<td>5.01</td>
</tr>
</tbody>
</table>

operation conditions:

- viscosity, $\mu$: $2.0 \cdot 10^{-4} \text{Pa} \cdot \text{s}$
- specific heat, $C_p$: $1.0 \text{kJ/kg} \cdot \text{K}$
- inlet temperature, $T_i$: $10^\circ \text{C}$
- heat input rate, $\dot{q}$: $1 \text{kW}$

From the uncertainty analysis shown in Table 4.2, we learn that:

1. At lower $Pr$, the higher the $Re_D$, the larger the uncertainty. However, the uncertainty of $k$ through the Dittus-Boelter correlation seems less affected by $Re_D$ than the other two correlations.

2. At middle $Re_D$, $Pr$ significantly affects the $k$ uncertainty for the Petukhov and Popov correlation and the Gnielinski correlation.

3. At lower $Re_D$, the uncertainty in $k$ from these three correlations is not significant for a change in $Pr$. 
4. From an overview of the uncertainty in k from these three correlations, the Dittus-Boelter correlation seems to be more stable than the other two correlations, even if these two correlations have smaller uncertainty values at some particular operating conditions.

Because CF is obtained from two Nusselt number ratios as mentioned earlier, the only uncertainty source of CF is from the Nusselt number curve fit, which will be dependent on the curve fit method used and how the data are fit.

It is interesting to compare the uncertainty differences between Approaches 1 and 2. From an overview, the variables involved in Approach 1, regardless of the correlation, are more complicated than those in Approach 2. Therefore, it implies that the uncertainty of Approach 2 seems to be less than that in Approach 1. This point can be verified later. The thermal conductivity uncertainty percentages were calculated at some typical conditions of operation for both approaches. The following figures plot the uncertainty percentage versus $T_w - T_f$. Figure 4.1 compares the results at $T_i = 5^\circ C$ and $\Delta T = 10^\circ C$, and Figure 4.2 compares the results at $T_i = 5^\circ C$ and $\Delta T = 20^\circ C$. From these two figures, we can see the uncertainty of Approach 1 seems twice as large as the uncertainty in Approach 2. They show the same trends, and $T_i$ and $\Delta T$ don’t seem to be affected significantly. For other operating conditions, Figure 4.3 compares the results at $T_i = 10^\circ C$ and $\Delta T = 5^\circ C$, and Figure 4.4 compares the results at $T_i = 10^\circ C$ and $\Delta T = 20^\circ C$. Again, from these two figures, the trends are exactly the same as with the first two conditions. Therefore, we can draw conclusions that:

- $\Delta T_{wf}$ significantly affects the uncertainty of the measured thermal conductivity in both approaches
• $T_i$ and $\Delta T$ have less effect on the uncertainty of the measured thermal conductivity in both approaches

• Approach 2 has less uncertainty than Approach 1

Summary of approaches

Two methods were discussed for the calculation of thermal conductivity: The Nusselt number calibration method and the Prandtl number method. Based upon the uncertainty in the Nusselt number calibration method, the uncertainty is induced by measured $\bar{h}$, $\dot{m}$, $\mu$, $C_p$, and CF curve fittings. However, in the Prandtl number method, the uncertainty sources are only the $Re_D$ which is $\dot{m}$ and $\mu$, the $\Delta T^*$, and the Pr curve fits. The accuracy of the thermal conductivity measurement can be calculated from the uncertainty of each source. However, there are less variables used for the curve fits in the Prandtl number method than those in the Nusselt number method. Therefore, the Prandtl number method is thought to be simpler and more convenient for obtaining thermal conductivity values.

Based upon the uncertainty analysis, the uncertainty of Approach 2 is only half of that for Approach 1. Approach 2 is therefore believed to be more accurate than Approach 1.
Figure 4.1: The uncertainty estimation at $T_i = 5^\circ\text{C}$ and $\Delta T = 10^\circ\text{C}$
Figure 4.2: The uncertainty estimation at $T_i = 5^\circ C$ and $\Delta T = 20^\circ C$
uncertainty comparison between approach 1 & 2

Figure 4.3: The uncertainty estimation at $T_i = 10^\circ C$ and $\Delta T = 5^\circ C$
uncertainty comparison between approach 1 & 2

Figure 4.4: The uncertainty estimation at $T_i = 10^\circ$C and $\Delta T = 20^\circ$C
CHAPTER 5. EXPERIMENTAL DATA CALCULATIONS

In this chapter, the experimental analysis of the raw data is presented. The heat transfer coefficient and parameter calculation are described. Additionally, the thermal conductivity calculations from experimental data is also described. In the data analysis, each measured quantity was obtained by the average of twenty scans, and a FORTRAN program was then used to reduce the data. Appendix F contains a copy of the data analysis program. For regression of multiple variables, a SAS [92] program was also used for aiding in correlation development.

Data reduction

The data analysis procedure for the calibration refrigerants (R-22, R-12, R-113, and/or R-114) and measured refrigerant (R-236ea) are different from some treatments. However, for the raw data reduction, they follow the same procedure.

Inner wall temperature corrections

In this test-tube, the surface temperature is not measured on the actual inner tube wall surface because of the difficulty of installing thermocouple beads there. Instead of installing on the inner surface, all thermocouple beads for measuring wall surface temperature were buried half way into the wall thickness. In order to obtain
a more realistic surface temperature, a correction is required to obtain the real inner wall surface temperature. A correction equation was approached by a heat conduction model and described as follows:

$$\dot{q}_{net} = \frac{2\pi L k_s (T_{r,2} - T_{r,1})}{\ln(r_2/r_1)}$$

where $\dot{q}_{net}$ is net heat input rate, $L$ is tube length (2 m in current test-tube), $r_2$ is the radius of the point where thermocouple beads are buried, $r_1$ is the inside radius of the tube, $k_s$ is the thermal conductivity of solid tube ($k_s = 401 \text{ W/m \cdot K}$ for pure copper), and $T_{r,1}, T_{r,2}$ denote the temperatures at $r_1$ and $r_2$, respectively. A correction temperature is then expressed as:

$$\Delta T_c = \frac{\dot{q}_{net} \ln(r_2/r_1)}{2\pi L k_s}$$

Therefore, the real inner wall temperature, $T_{r,1}$ is corrected by:

$$T_{r,1} = T_{r,2} - \Delta T_c$$

$T_{r,2}$ in this case is the measured temperature of thermocouple.

**Average quantity calculations**

Average quantities include timing average which indicates the average of measuring quantities in a certain scanning time interval, and location average which indicates the average of measuring quantities in different positions. These average measuring quantities are calculated as follows.
Raw data average

In order to obtain a more average representation of experimental data, twenty scans were taken for a steady-state situation, and these twenty values were then averaged for each representative quantity. These quantities include the following variables: $T_i$, $T_o$, $T_{w_i}$, $T_s$, $T_a$, $\dot{m}$, $Q_{tot}$, and $\mu$. Therefore:

$$\text{average quantity} = \frac{1}{n} \sum_{i=1}^{n} (\text{quantity})_i$$  \hspace{1cm} (5.4)

where $n$ stands for the total number of scanning times ($n=20$), and $i$ represents the $i$-th scan.

Average $\Delta T_{wf}$ calculations

The average fluid temperature of the test-tube, $\bar{T}_f$, was therefore taken as the average temperature of the inlet and outlet temperature, $T_i$ and $T_o$, of the test-tube. However, in the average $\Delta \bar{T}_{wf}$ calculation, which is $\bar{T}_w - \bar{T}_f$, the finite control volume method was used which considers the local heat transfer characteristics (thermal entrance effect) rather than the all-points average method, Equation 5.5. The mathematical derivation is presented as follows. For the tube wall and fluid temperature, the averages are calculated as the following:

$$\bar{T}_w = \left(\frac{1}{13}\right) (T_{w1} + T_{w2} + \cdots + T_{w13})$$  \hspace{1cm} (5.5)

$$\bar{T}_f = \left(\frac{1}{2}\right) (T_i + T_o)$$  \hspace{1cm} (5.6)

Equations 5.5 and 5.6 are used for obtaining an average wall temperature which assumes a linear wall temperature distribution. It is applicable for fully developed
regions subjected to a constant heat flux. However, the entrance effect, in some situations, is significant [88, 51]. In those cases, the wall temperature distribution in that region is no longer linear. Therefore, if a linear average of the wall temperature is taken for the average wall temperature, that will affect the accuracy of the heat transfer coefficient calculation. Another approach, the finite control volume method mentioned above, is based on an energy balance which provides a better representation than the linear average method. This method is applied under the assumption of a constant heat flux situation. In this situation, the amount of heat added to a local section is everywhere equal as long as the surface area is equal. That means that an equal amount of heat is obtained for an equal length of heated section in a constant cross section area tube. An energy balance equation for a heated section can be expressed as:

\[ \dot{Q} = \dot{q}_1 + \dot{q}_2 + \cdots + \dot{q}_i + \cdots + \dot{q}_n \]  

(5.7)

where the subscript \( n \) represents the number of \( n \) sections and \( i \) represents the \( i \)-th sub-section.

Each \( \dot{q}_i \) would be the same due to the constant heat flux, if each sub-section has an equal surface area. The above equation can also be written as:

\[ \dot{Q} = q'' \left( A_1 + A_2 + \cdots + A_i + \cdots + A_n \right) \]  

(5.8)

However, since \( \dot{Q} \) is equal to \( hA\Delta T_{wf} \), the above equation can be rewritten as:

\[ \Delta T \int_0^l h_x dA = q'' \left( A_1 + A_2 + \cdots + A_i + \cdots + A_n \right) \]  

(5.9)

or rewritten into the finite control volume form,

\[ \Delta T_{wf} \sum_{i=1}^n h_i A_i = q'' \left( A_1 + A_2 + \cdots + A_i + \cdots + A_n \right) \]  

(5.10)
Based upon the assumption of a constant heat flux in each sub-section, \( h_i \) is equal to \( q''/\Delta T_{wf_i} \). The \( \Delta T_{wf_i} \) is a locally averaged quantity, which is \( T_{w_i} - T_{f_i} \). Substituting \( h_i \) into above equation,

\[
\Delta T_{wf} \left[ \sum_{i=1}^{n} \frac{q'' A_i}{\Delta T_{wf_i}} \right] = q'' \left( A_1 + A_2 + \cdots + A_i + \cdots + A_n \right)
\]

(5.11)

or,

\[
\Delta T_{wf} \left[ \sum_{i=1}^{n} \frac{A_i}{\Delta T_{wf_i}} \right] = A_1 + A_2 + \cdots + A_i + \cdots + A_n = A
\]

(5.12)

where \( A \) is the total heat transfer area.

Finally, the average temperature difference between the average wall temperature and fluid temperature can be expressed as:

\[
\frac{1}{\Delta T_{wf}} = \frac{1}{L} \sum_{i=1}^{n} \left( \frac{\Delta l_i}{\Delta T_{wf_i}} \right)
\]

(5.13)

Or, in another form,

\[
\Delta T_{wf} = \frac{L}{\sum_{i=1}^{n} \left( \frac{\Delta l_i}{\Delta T_{wf_i}} \right)}
\]

(5.14)

where \( \Delta l_i \) is the length of i-th section.

For the current test-section, the wall temperature is measured at 13 locations. Finite control volumes, therefore, can be divided into 13 sub-sections. A typical wall temperature and average fluid temperature distribution is shown in Figure 5.1. Because of the entrance effect, the wall temperature at the inlet portion should reflect a non-linear distribution. For a better representation, smaller sub-sections were taken. Therefore, the determination of sub-sections at the inlet portion was based on the wall temperature at the measured points. Figure 5.2 shows the finite control volume configuration of the current test-section. In this figure, the control volume in the inlet
and outlet portion were smaller than the middle portion because more temperatures were measured at the ends of the test-section. This method for obtaining the average temperature difference was thought to be a better representation than average of all-points method because it accounts for the thermal behavior of the entrance length. The sub-section length distribution is listed in Table 5.1.

Table 5.1: Sub-section length distribution

<table>
<thead>
<tr>
<th>sub-section, i</th>
<th>thermocouple point</th>
<th>distance from inlet, cm</th>
<th>Δl_i, cm</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>0</td>
<td>5</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>3</td>
<td>3</td>
<td>20</td>
<td>15</td>
</tr>
<tr>
<td>4</td>
<td>4</td>
<td>40</td>
<td>20</td>
</tr>
<tr>
<td>5</td>
<td>5</td>
<td>60</td>
<td>20</td>
</tr>
<tr>
<td>6</td>
<td>6</td>
<td>80</td>
<td>20</td>
</tr>
<tr>
<td>7</td>
<td>7</td>
<td>100</td>
<td>20</td>
</tr>
<tr>
<td>8</td>
<td>8</td>
<td>120</td>
<td>20</td>
</tr>
<tr>
<td>9</td>
<td>9</td>
<td>140</td>
<td>20</td>
</tr>
<tr>
<td>10</td>
<td>10</td>
<td>160</td>
<td>20</td>
</tr>
<tr>
<td>11</td>
<td>11</td>
<td>180</td>
<td>15</td>
</tr>
<tr>
<td>12</td>
<td>12</td>
<td>190</td>
<td>10</td>
</tr>
<tr>
<td>13</td>
<td>13</td>
<td>200</td>
<td>5</td>
</tr>
</tbody>
</table>

Heat transfer calculations

One of the thermodynamic properties, specific heat \( C_p \), is used to calculate a single phase heat transfer quantity, based upon an energy balance principle. This property, in general, is function of temperature and pressure. However, for a liquid at constant pressure, \( C_p \) is only a function of temperature. But, liquid \( C_p \) is not a strong function of temperature except above the reduced temperature \( T_R = 0.7 \) to 0.8. Detailed information on the description of \( C_p \) was given in the thermodynamic
uniform heat flux $q''$

Figure 5.1: A typical temperature distribution along a heated test-section
Figure 5.2: Finite control volume configuration of the test-section
and property textbooks [50, 59].

The net heat input rate for refrigerants of known \( C_p \) was calculated directly from the energy equation for flow of in the test-tube. That is, for refrigerants with known \( C_p \):

\[
\dot{Q}_{net} = \dot{m}C_p(T_o - T_i)
\]  

For refrigerants with unknown \( C_p \), the \( \dot{Q}_{net} \) is calculated from the energy balance of the test-tube. That is, for refrigerants with unknown \( C_p \), Equation 4.5 is applied. And, the \( \dot{q}_{net} \) can be calculated from Equation 4.3. In Equation 4.3, \( \dot{q}_{loss} \) is the heat loss of the test-section. A correlation for calculating \( \dot{q}_{loss} \) was described in the Equation 3.1 in Chapter 3. The heat transfer coefficients can be obtained by Equation 4.2. The parameters in this study, including \( \bar{N}u_D \), Pr, \( \bar{Re}_D \), and \( \Delta T^* \), were then calculated by their definitions presented earlier or defined in Chapter 4.

**Viscosity and thermal conductivity calculations**

The product of viscosity and density is directly measured by the viscometer. The viscosity is then obtained by dividing it by density which is measured by a densimeter or obtained from reliable data sources, such as the ASHRAE Handbook [93] or REFPROP [49], which are shown in Appendix A.

There are two approaches proposed in this study. The thermal conductivity can be obtained by either way. However, except for thermal conductivity being directly calculated from the Dittus-Boelter correlation in Approach 1, Equation 4.29, the other thermal conductivity correlations, Equations 4.30, 4.31, or Equation 4.34, are developed from Pr. Therefore, Pr must be obtained before the thermal conductivity can be calculated.
Correction of the pressure effect on thermal conductivity and viscosity

Pressure has some effect on the thermal conductivity and viscosity of a liquid. However, these effects are usually significant only at high pressures. In other words, thermal conductivity and viscosity are not only strong functions of temperature, but they are also strong functions of pressure when liquid is subjected to a high pressure. The following discussion provides corrections for the pressure effects on thermal conductivity and viscosity.

Pressure effect on liquid thermal conductivity

In general, at moderate pressures, up to 50 or 60 bar, the effect of pressure on the thermal conductivity of liquids is usually neglected, except near the critical point. Missenard has published a simple correlation for \( k \) which extends to higher pressures [50]. The thermal conductivity ratio is subjected to two different pressures but at the same temperature. They were found to correlate with reduced temperature and pressure, as shown in the following equation:

\[
\frac{k(P_r)}{k(\text{low pressure})} = 1 + Q \cdot P_r^{0.7}
\]  

where \( P_r \) is reduced pressure and \( Q \) is a functional parameter of \( P_r \) and \( T_r \). \( Q \) is given in Table 10-8 in R. C. Reid et al. [50]. A plot of thermal conductivity ratio against \( P_r \) is given by Figure 5.3.

Pressure effect on liquid viscosity

Liquid viscosity will be affected by pressure, especially at high pressures. Increasing the pressure over a liquid results in an increased viscosity. However, at
Figure 5.3: Pressure effect on liquid thermal conductivity
pressures less than the critical pressure, the pressure effect on viscosity is very weak. Lucas (1981) suggested that the change might be estimated from Equation 5.17 [50]:

\[
\frac{\mu}{\mu_{sat}} = \frac{1 + D (\Delta P_r/2.118)^A}{1 + C\omega\Delta P_r}
\]

(5.17)

where:

- \(\mu\) = viscosity of liquid at pressure
- \(\mu_{sat}\) = viscosity of liquid at saturated state
- \(\omega\) = acentric factor
- \(\Delta P_r = (P - P_{vp})/P_c\)
- \(A = 0.9991 - [4.674 \cdot 10^{-4}/(1.0523T_r - 0.03877 - 1.0513)]\)
- \(D = [0.3257/(1.0039 - T_r^{2.573})]^{0.2906} - 0.2086\)
- \(C = -0.07921 + 2.1616T_r - 13.4040T_r^2 + 44.1706T_r^3 - 84.8291T_r^4 + 96.1209T_r^5 - 59.812T_r^6 + 15.6719T_r^7\)

A plot of viscosity ratio versus \(\Delta P_r\) is shown in Figure 5.4 and 5.5.

Thermal conductivity and viscosity corrections for pressure effects

High pressure (higher than critical pressure) has a strong effect on viscosity and thermal conductivity, while low pressure has almost no effect or only a minor effect on them, as discussed in above sections. In the current study, refrigerants were circulated in the low pressure range. Therefore, pressure has a very small effect on viscosity and thermal conductivity. However, to obtain accurate properties, pressure effects on both viscosity and thermal conductivity were still taken into account. The manner for which pressure effects were considered is outlined in Equations 5.16 and 5.17 for thermal conductivity and viscosity, respectively. Because the operating state
pressure effect on liquid viscosity
Lucas correlation (1981)

Figure 5.4: Pressure effect on liquid viscosity ($\omega = 0.0$)
Figure 5.5: Pressure effect on liquid viscosity ($\omega = 0.2$)
must be controlled within a certain degree of subcooling at the outlet of test-section, the system pressure should also be greater than the saturation pressure to maintain a liquid phase throughout the test-section. The viscosity and thermal conductivity at that pressure were obtained from those at the saturation pressure by using Equations 5.16 and 5.17. On the other hand, viscosity and thermal conductivity at the saturation pressure were obtained by reversing the usage of these equations.

**Data presentation**

This section describes the format used to present the data in order to identify trends for the purpose of data comparison. In general, the thermal conductivity and viscosity are strong functions of temperature and weak functions of pressure in a low pressure region. Therefore, the thermal conductivity and viscosity were presented only as a function of temperature. All the plots shown in this study were plotted as saturated liquid thermal conductivity and viscosity versus temperature. Measured data compared with the data in ASHRAE Hankbook and REFPROP were also shown in the plots. The measured specific heat, $C_p$, and density, $\rho$, of interest versus temperature were also plotted and compared with ASHRAE Handbook or REFPROP values.
CHAPTER 6. REFRIGERANT CALIBRATIONS

In this chapter, calibration of thermal conductivity by using refrigerants of known properties will be shown and discussed for both approaches. Calibration Function (CF) in Approach 1 and correlation obtained by Approach 2 will be presented. In addition, the accuracy of the measured data by these approaches will also be presented.

Calibration of refrigerants and operating ranges

Before measuring the thermal conductivity of R-236ea, the calibration functions should be obtained. The calibration functions have been obtained from several fluids with known properties by the current method proposed in this study. These refrigerants of known properties were selected in order to obtain the calibration functions. They are R-22, R-12, R-113, and R-114. The properties used for these refrigerants are based on ASHRAE data and their curve fit correlations are presented in Appendix A. Table 6.1 shows the operating ranges used to obtain the calibration functions of these selected refrigerants. The fluids (R-236ea, blends A and B) of unknown thermal conductivity tests must be within the Pr number ranges of the basic refrigerant tests shown above. For R-236ea as an example, with operating temperatures between 0°C and 50°C, the Pr number varies from 5.60 to 8.12 while \( Re_D \) can be controlled by
Table 6.1: Prandtl number, Reynolds number, and $\Delta T^*$ ranges for selected fluids

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Reynolds number ($Re_D$)</th>
<th>Prandtl number (Pr)</th>
<th>$\Delta T^*$</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-22</td>
<td>38360 - 184852</td>
<td>2.28 - 2.56</td>
<td>0.81 - 1.42</td>
</tr>
<tr>
<td>R-12</td>
<td>31031-168007</td>
<td>2.82 - 3.08</td>
<td>0.97 - 1.30</td>
</tr>
<tr>
<td>R-113</td>
<td>7970 - 64354</td>
<td>7.00 - 10.58</td>
<td>0.64 - 0.80</td>
</tr>
<tr>
<td>R-114</td>
<td>16296 - 99517</td>
<td>4.46 - 6.34</td>
<td>0.79 - 0.99</td>
</tr>
<tr>
<td>Overall</td>
<td>7970 - 184852</td>
<td>2.28 - 10.58</td>
<td>0.64 - 1.42</td>
</tr>
</tbody>
</table>

regulating the mass flow rate as as that it falls within the ranges of basic refrigerants listed above. Under most conditions, the mass flow rate varied from 3 kg/min. to 15 kg/min.

**Calibration functions**

The calibration functions were found for the three Nusselt correlations mentioned earlier. A linear regression between $\ln(Nu/Pr^N)$ and $\ln(Re_D)$ was performed for the experimental curve fits. In order to verify the current method for thermal conductivity measurement, the calibration refrigerants can be selected such that the calibration functions were obtained only for R-22, R-12, and R-113, which covered the Pr from 2.28 to 10.58. These functions could then be applied to the calculation of unknown fluids such as R-114. Figures 6.1, 6.2, and 6.3 show the results for the three different Nusselt number correlations used for these calibration refrigerants.

The calibrated Nusselt number can be obtained from the curve fit equations such that:

\[
\tilde{Nu}_{D\text{experiment}} = CF \cdot \tilde{Nu}_{D\text{correlation}}
\]  

(6.1)

where the calibration function, CF can be obtained from Equation 4.24.
Figure 6.1: The Nusselt number relations from hD/k and Dittus-Boelter correlation for the calibration refrigerants
Figure 6.2: The Nusselt number relations from hD/k and the Petukhov correlation for the calibration refrigerants
Figure 6.3: The Nusselt number relations from hD/k and the Gnielinski correlation for the calibration refrigerants
Table 6.2: Curve fit coefficients for Nusselt numbers-3 calibration refrigerants

<table>
<thead>
<tr>
<th>coefficient</th>
<th>$hD/k$</th>
<th>Dittus-Boelter</th>
<th>Petukhov</th>
<th>Gnielinski</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C, C^e$</td>
<td>0.006758</td>
<td>0.023</td>
<td>0.017266</td>
<td>0.012093</td>
</tr>
<tr>
<td>$a, a^e$</td>
<td>0.903770</td>
<td>0.8</td>
<td>0.827298</td>
<td>0.862783</td>
</tr>
<tr>
<td>$b, b^e$</td>
<td>0.502199</td>
<td>0.4</td>
<td>0.455283</td>
<td>0.441244</td>
</tr>
</tbody>
</table>

The curve fit coefficients of the calibration functions are shown in Table 6.2 for $hD/k$, in which $h$ is experimentally measured, and for the three different Nusselt number correlations.

It should be noted that the calibration function, CF, is still expressed as a function of $Re_D$ and $Pr$. Therefore, like the $Nu_D$ expression in Equation 4.20, a linear relationship between $ln(CF/Pr^{b^e-b})$ and $ln(Re_D)$ holds true and can be described as follows:

$$ln \left( CF/Pr^{b^e-b} \right) = \ln \left( C^e/C \right) + \left( a^e - a \right) \ln \left( Re_D \right) \quad (6.2)$$

The plots of $ln(CF)$ versus $ln(Re_D)$ are shown in Figures 6.4, 6.5, and 6.6 for the CF obtained from the Dittus-Boelter, Petukhov, and Gnielinski correlations, respectively. As indicated in these figures, the relationships are all linear with various $Pr$.

If the four refrigerants: R-22, R-12, R-113, and R-114, were used as the calibration refrigerants, the calibration function, CF, would be slightly different but very close to that obtained for the three calibration refrigerants. The CF coefficients are listed in Table 6.3.
Figure 6.4: The calibration functions for the Dittus-Boelter correlation
Figure 6.5: The calibration functions for Petukhov correlation
Figure 6.6: The calibration functions for Gnielinski correlation
Table 6.3: Curve fit coefficients for Nusselt numbers-4 calibration refrigerants

<table>
<thead>
<tr>
<th>coefficient</th>
<th>$hD/k$</th>
<th>Dittus-Boelter</th>
<th>Petukhov</th>
<th>Gnielinski</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C, C^e$</td>
<td>0.006633</td>
<td>0.023</td>
<td>0.017372</td>
<td>0.012381</td>
</tr>
<tr>
<td>$a, a^e$</td>
<td>0.905027</td>
<td>0.8</td>
<td>0.826843</td>
<td>0.860922</td>
</tr>
<tr>
<td>$b, b^e$</td>
<td>0.507640</td>
<td>0.4</td>
<td>0.457499</td>
<td>0.442776</td>
</tr>
</tbody>
</table>

The Prandtl number regression

As mentioned earlier, Approach 2 for calculating the thermal conductivity is from the Prandtl number, $Pr$. The $Pr$ is derived as function of $Re_D$ and $\Delta T^*$ as pointed out previously. For the three basic calibration refrigerants: R-22, R-12, and R-113, and four basic calibration refrigerants: R-22, R-12, R-113, and R-114, the full term 3-degree polynomial function regressions for $ln(Pr)$ were fitted as shown in Table 6.4.

Table 6.4: Curve fit coefficients for $ln(Pr)$

<table>
<thead>
<tr>
<th>term</th>
<th>3 refrigerants</th>
<th>4 refrigerants</th>
</tr>
</thead>
<tbody>
<tr>
<td>interception</td>
<td>96.908792</td>
<td>132.495075</td>
</tr>
<tr>
<td>$ln(Re_D)$</td>
<td>-26.452258</td>
<td>-36.446721</td>
</tr>
<tr>
<td>$ln(\Delta T^*)$</td>
<td>83.304684</td>
<td>102.201609</td>
</tr>
<tr>
<td>$ln(Re_D^2)$</td>
<td>2.459228</td>
<td>3.392931</td>
</tr>
<tr>
<td>$ln(Re_D) \cdot ln(\Delta T^*)$</td>
<td>-16.020859</td>
<td>-19.449363</td>
</tr>
<tr>
<td>$ln(\Delta T^*^2)$</td>
<td>8.088965</td>
<td>14.165714</td>
</tr>
<tr>
<td>$ln(Re_D^3)$</td>
<td>-0.076756</td>
<td>-0.105753</td>
</tr>
<tr>
<td>$ln(Re_D^2) \cdot ln(\Delta T^*)$</td>
<td>0.750949</td>
<td>0.905506</td>
</tr>
<tr>
<td>$ln(Re_D) \cdot ln(\Delta T^*^2)$</td>
<td>-0.668378</td>
<td>-1.253316</td>
</tr>
<tr>
<td>$ln(\Delta T^*^3)$</td>
<td>2.213377</td>
<td>2.823725</td>
</tr>
</tbody>
</table>
Plots of Pr versus ΔT* are shown in Figures 6.7 and 6.8, for the three refrigerant and four refrigerant bases. As can be seen from these figures, the smaller the ΔT*, the larger the Pr for a fixed Re_D. On the other hand, at a fixed ΔT*, the smaller the Re_D and the larger the Pr. These plots furnish the trends of the relationship between Pr, Re_D, and ΔT* developed by Approach 2.

Comparison of measured thermal conductivity

It is of interest to compare the measured thermal conductivity with ASHARE handbook data for the calibration refrigerants (following calibration). To do this comparison, the thermal conductivity of the four calibration refrigerants were plotted for this purpose using Approach 1 and 2. Figure 6.9 shows the measured thermal conductivity using Approach 1 versus the ASHRAE handbook thermal conductivity plot, and Figure 6.10 shows the measured thermal conductivity by Approach 2 versus the ASHRAE handbook thermal conductivity. Lines of ±5% were used to present an indication of the deviation in the measured data. As shown in these figures, most of the measured points were located inside the ±5% band except for some points scattered outside this band but within ±10%. This could be due to experimental errors, which are not included in the experimental uncertainty, such as not actual steady-state or inaccurate property data. By comparing the distribution of the data points in these two figures, it is found that there seems to be more fluctuation in the results of Approach 1 over the results of Approach 2. However, these results show that the experimental uncertainties were within ±5% as predicted by theoretical uncertainties which were pointed out earlier. These results provide more confidence in measuring other refrigerants by this method.
Approach 2: Plot of Pr versus \((T_o-T_i)/(T_w-T_i)\)
(Three Refrigerants Base: R-22, R-12, R-113)

Legend

- \(\text{Re}=30000\)
- \(\text{Re}=50000\)
- \(\text{Re}=70000\)
- \(\text{Re}=90000\)
- \(\text{Re}=110000\)
- \(\text{Re}=130000\)
- \(\text{Re}=150000\)

Figure 6.7: Plots of Pr versus \(\Delta T^\ast\) (3-refrigerant base)
Approach 2: Plot of Pr versus (To-Ti)/(Tw-Tf)
(Four Refrigerants Base: R-22, R-12, R-113, R-114)

Figure 6.8: Plots of Pr versus ΔT*(4-refrigerant base)
Figure 6.9: Measured thermal conductivity compared with ASHRAE data by Approach 1 (4-refrigerant base)
Figure 6.10: Measured thermal conductivity compared with ASHRAE data by Approach 2 (4-refrigerant base)
CHAPTER 7. VERIFICATION OF METHODOLOGY

The purpose of this study is to determine the properties of refrigerant alternatives. Because R-236ea is the classified alternative for R-114, R-114 properties measured by the current approach are compared with the ASHRAE data in order to see whether this approach can be applied to fluids of unknown properties. The properties of R-114 being verified in this study are thermal conductivity, $k$, viscosity, $\mu$, and specific heat, $C_p$. However, it is first interesting to compare those properties between REFPROP and ASHRAE. Then, measured results are presented and compared with the ASHRAE data. The compared results are shown in the following sections.

R-114 properties of REFPROP and ASHRAE

REFPROP is a computer package for refrigerant properties recently developed by NIST, and REFPROP-4.0 is the most updated version. Because of the limited experimental data, theoretical predictions are thought to be the only method to obtain properties for this heat transfer study. In this section, interest is focused on a comparison of density, specific heat, viscosity, and thermal conductivity for R-114. Figures 7.1, 7.2, 7.3, and 7.4 show the plots of the ASHRAE data and REFPROP-4.0 data for these respective properties.
Figure 7.1: The comparison plot between ASHRAE and REFPROP for R-114 density.
R114 specific heat comparison between ASHRAE and REFPROP-4.0

Figure 7.2: The comparison plot between ASHRAE and REFPROP for R-114 specific heat
Figure 7.3: The comparison plot between ASHRAE and REFPROP for R-114 viscosity
Figure 7.4: The comparison plot between ASHRAE and REFPĐROP for R-114 thermal conductivity.
As shown in these comparison plots, some deviations exist. The deviations presented as percentages over a temperature range of $-50^\circ C$ to $120^\circ C$ for each of the four properties discussed here are listed in Table 7.1. As shown in these figures (Figures 7.1 through 7.4) and Table 7.1, these properties deviate somewhat from each other over the temperature ranges under study. Therefore, the results obtained might be different if other data sources are used.

Table 7.1: R-114 property comparison between ASHRAE and REF-PROP

<table>
<thead>
<tr>
<th>property</th>
<th>dev.(low temp.)</th>
<th>dev.(high temp.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>density, $\rho (kg/m^3)$</td>
<td>+5.2%</td>
<td>-0.02%</td>
</tr>
<tr>
<td>specific heat, $C_p (J/kg \cdot K)$</td>
<td>+6.8%</td>
<td>-2.2%</td>
</tr>
<tr>
<td>viscosity, $\mu (Pa \cdot s)$</td>
<td>+4.2%</td>
<td>-7.1%</td>
</tr>
<tr>
<td>thermal conductivity, $k (W/m \cdot K)$</td>
<td>-2.7%</td>
<td>-11.7%</td>
</tr>
</tbody>
</table>

$^a$dev. = $[(\text{REFPROP} - \text{ASHRAE}) / \text{ASHRAE}] \cdot 100\%$

Verification of R-114 properties

In this section, the measured properties of R-114 which include thermal conductivity, viscosity, specific heat, and density will be presented and compared with ASHRAE data.

R-114 thermal conductivity

The R-114 thermal conductivities were obtained from experimental data operating from near $0^\circ C$ to $50^\circ C$. Figure 7.5 shows the thermal conductivity of R-114 by Approach 1 and compared to ASHRAE data[93]. The deviations of thermal conductivity between measured and ASHRAE data are shown to be within $-3.8\%$ over the
measured temperature range. Figure 7.6 shows the thermal conductivity of R-114 by Approach 2 and also compares it to ASHRAE data. Again, the deviation is still within −3.8%. It is important to mention that the properties required in the thermal conductivity calculations such as viscosity and specific heat were all based on ASHRAE data. Deviation plots of thermal conductivity for measured and ASHRAE data are shown in Figures 7.7 and 7.8 for Approaches 1 and 2, respectively.

**R-114 viscosity**

The R-114 viscosity was measured directly by the inline viscometer. The accuracy of the viscometer had already been verified for several fluids. As mentioned previously, Figure 3.8 shows the plot of viscosity versus temperature for R-114. As can be seen from the figure, the measured viscosity and ASHRAE viscosity were in very close agreement with the deviation being ±2%. Figure 3.9 shows the deviation plot over the measured temperature range.

**R-114 specific heat**

The measured specific heat was calculated from an energy balance of the test-tube. With the heat loss calibration and total power input, the specific heat was calculated from Equation 4.5. Figure 7.9 shows the R-114 specific heat over a temperature range of 0°C to 50°C. As shown in this figure, the measured $C_p$ matches closely with the ASHRAE data. This suggests that the temperature and the net power measurements are quite good and provide much confidence in measuring unknown fluids such as R-236ea.

It should be noted that the R-114 specific heat, $C_p$, was calculated by using
R114 thermal conductivity vs. temperature

Figure 7.5: Plot of R-114 measured thermal conductivity by Approach 1
Figure 7.6: Plot of R-114 measured thermal conductivity by Approach 2
Figure 7.7: R-114 measured thermal conductivity (Approach 1) versus ASHRAE data
Figure 7.8: R-114 measured thermal conductivity (Approach 2) versus ASHRAE data
Figure 7.9: Plot of R-114 measured specific heat
the test-section heat loss estimation described above. The resulting plot is shown in Figure 3.7, and Equation 3.1 was used. The deviations of measured \( C_p \) compared to ASHRAE \( C_p \) are located within ±3%.

**R-114 density**

The density of R-114 was measured by a densimeter. The densimeter was verified with refrigerants, and results were compared with ASHRAE data as shown in Figure 3.8 in Chapter 3. The measured data are shown in Figure 7.10. The deviation percentage was calculated within ±1.8% compared with ASHRAE data.

**Summary**

In this chapter, the measured properties of R-114 have been presented and compared with ASHRAE data. The purpose of the R-114 property measurements is a verification of the methodology. The main properties of interest are thermal conductivity and viscosity. However, other properties such as specific heat and density were also measured and compared. Of which, specific heat is a required property for the thermal conductivity calculation in the methodology of this study. In general, the measured properties were matched closely with ASHRAE data. This shows the validity of the methodology developed in this study. A summary follows of this verification. Table 7.2 shows this summary of verified results for each property.

The deviations shown in Table 7.2 indicate larger deviations exist between REFPROP and ASHRAE data for transport properties (viscosity and thermal conductivity), while smaller deviations were detected for thermodynamic properties (density and specific heat). For measured properties, it was shown that there is less deviation
Figure 7.10: Plot of R-114 measured density
Table 7.2: Summary of deviation for R-114 properties

<table>
<thead>
<tr>
<th>properties</th>
<th>$Dev_1^a$ ((-50 \sim 150^\circ C))</th>
<th>$Dev_2^b$ ((0 \sim 50^\circ C))</th>
</tr>
</thead>
<tbody>
<tr>
<td>density, $\rho$</td>
<td>±5.2%</td>
<td>+1.9%</td>
</tr>
<tr>
<td>specific heat, $C_p$</td>
<td>±6.8%</td>
<td>±3%</td>
</tr>
<tr>
<td>viscosity, $\mu$</td>
<td>±7.1%</td>
<td>±2%</td>
</tr>
<tr>
<td>thermal conductivity$^c$, $k$</td>
<td>±11.7%</td>
<td>±5%</td>
</tr>
</tbody>
</table>

$^a Dev_1$: (REFPROP-ASHRAE)/ASHRAE
$^b Dev_2$: (Experiment-ASHRAE)/ASHRAE
$^c$ for both approaches

from ASHRAE data than REFPROP. It verifies that the methodology worked properly. However, to improve accuracy, more calibrations might be necessary. Other possible error sources, excluding experimental uncertainties such as operation errors (e.g. steady-state requirement), must be eliminated as much as possible during the experimental operation.
CHAPTER 8. PROPERTIES OF R-236ea

In the previous chapter, R-114 measured properties were verified with ASHRAE data. It illustrated confidence for applying the current approach to measure refrigerant properties. Based upon the thermal conductivity calibrations (both Approaches 1 and 2), same test-section conditions, and viscometer, the R-236ea properties were measured. The results were also compared with REFPROP-4.0[49] and are presented as follows.

R-236ea specific heat

Based on the same test-section heat loss estimation mentioned previously, the R-236ea $C_p$ was measured. Figure 8.1 shows the $C_p$ of R-236ea and compares it with the REFPROP-4.0 data. Because of the lack of the published experimental data, the REFPROP-4.0 data are used for comparison. The measured data show the deviation from REFPROP-4.0 is within $\pm 4.8\%$.

A linear curve fit equation is provided for the measured $C_p$, $kJ/kg \cdot ^{\circ}C$, versus temperature, $^\circ C$, as shown below:

$$C_p = 1.2048 + 0.001925 \cdot T$$

(8.1)
Figure 8.1: Plot of R-236ea measured and REFPROP specific heat
R-236ea viscosity

R-236ea viscosity was measured by a viscometer with an accuracy verified by a number of fluids as shown in Chapter 3 with an acceptable accuracy ±2%. Figure 8.2 shows the measured viscosity of R-236ea versus temperature over a temperature range from —5°C to nearing 60°C. A deviation from REFPROP is plotted in Figure 8.3. As indicated in this figure, the deviation percentage is +5%.

In order to conveniently use the results, a curve fit equation expressed in a 5 (or 2) degree polynomial is given below:

\[ \mu = \sum_{i=0}^{n} a_i \cdot T^i \]  (8.2)

where \( \mu \) is cp and \( T \) is °C. The curve fit coefficients are listed in Table 8.1.

<table>
<thead>
<tr>
<th>coefficient</th>
<th>value (5 degree)</th>
<th>value (2 degree)</th>
</tr>
</thead>
<tbody>
<tr>
<td>( a_0 )</td>
<td>5.65237164E-01</td>
<td>5.62097609E-01</td>
</tr>
<tr>
<td>( a_1 )</td>
<td>-1.06925368E-02</td>
<td>-8.41619726E-03</td>
</tr>
<tr>
<td>( a_2 )</td>
<td>2.31479004E-04</td>
<td>5.20047470E-05</td>
</tr>
<tr>
<td>( a_3 )</td>
<td>-4.57916803E-06</td>
<td>-</td>
</tr>
<tr>
<td>( a_4 )</td>
<td>3.80020317E-08</td>
<td>-</td>
</tr>
<tr>
<td>( a_5 )</td>
<td>-1.66399134E-11</td>
<td>-</td>
</tr>
</tbody>
</table>

R-236ea density

The density of R-236ea was also measured by densimeter. The accuracy of the densimeter has been verified with a number of refrigerants as discussed in Chapter 3. The resulting plot of density versus temperature is shown in Figure 8.4. The deviations were calculated within ±1% compared with REFPROP-4.0.
Figure 8.2: Plot of R-236ea measured and REFPROP viscosity
Figure 8.3: Plot of R-236ea measured and REFPROP viscosity deviation ranges
Figure 8.4: Plot of R-236ea measured and REFPROP density
The linear curve fit equation is given as the following equation:

\[ \rho = 1514.30969 - 3.13807 \cdot T \] (8.3)

where \( \rho \) is \( kg/m^3 \) and \( T \) is \( ^\circ C \).

**R-236ea thermal conductivity**

The thermal conductivity of R-236ea over a temperature range of \( 0^\circ C \) to \( 60^\circ C \) was measured by Approaches 1 and 2 in this study. The resulting plot is shown in Figure 8.5 using Approach 1 and in Figure 8.6 using Approach 2. As shown in these two figures, the measured results average 15% higher than REFPROP-4.0 results. Based on the verification data of R-114, it implies the REFPROP data might need to be increased over its current values for the measured temperature ranges. A linear curve fit equation is given for easy use below:

\[ k = 0.097013 - 0.000277 \cdot T \] Approach 1 (Dittus-Boelter CF) (8.4)

\[ k = 0.094944 - 0.000285 \cdot T \] Approach 1 (Petukhov-Popov CF) (8.5)

\[ k = 0.093838 - 0.000282 \cdot T \] Approach 1 (Gnielinski CF) (8.6)

\[ k = 0.0959604 - 0.0002684 \cdot T \] Approach 2 (8.7)

where \( k \) is \( W/m \cdot C \) and \( T \) is \( ^\circ C \).

**Other properties of extended calculations**

In the current study, there are four properites which can be measured or calculated. These include thermal conductivity, viscosity, specific heat, and density.
Figure 8.5: R-236ea thermal conductivity by Approach 1
Approach 2: R-236ea thermal conductivity versus temperature

Legend
- measured data
- REFP-4.0 data

Figure 8.6: R-236ea thermal conductivity by Approach 2
However, some properties extended from these four properties can be also obtained. These include thermal diffusivity and the Prandtl number. Thermal diffusivity is defined as $k/\rho C_p$, and Prandtl number is $\mu C_p/k$. Figures 8.7 and 8.8 show the thermal diffusivity and Pr of R-236ea versus temperature, respectively.

Summary

In this Chapter, R-236ea properties were measured by the current facility and methodology which were verified by the results of R-114 as discussed in the previous chapter. R-236ea thermodynamic properties (density and specific heat), showed good accuracy compared with REFPROP. However, transport properties (viscosity and thermal conductivity), showed larger deviations than the thermodynamic properties. A summary table of the deviations is shown in Table 8.2. As shown in this table, the deviations of density, specific heat, and viscosity are quite reasonable, while the thermal conductivity deviation is quite large. However, as shown in Figures 8.5 and 8.6, the trends of measured and REFPROP data were quite consistent. A modification between these two measures is necessary to achieve closer agreement.

<table>
<thead>
<tr>
<th>properties</th>
<th>deviation$^a$ (0 $\sim$ 50°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>density, $\rho$</td>
<td>$\pm 1%$</td>
</tr>
<tr>
<td>specific heat, $C_p$</td>
<td>$+4.8%$</td>
</tr>
<tr>
<td>viscosity, $\mu$</td>
<td>$-5%$</td>
</tr>
<tr>
<td>thermal conductivity$^b$, $k$</td>
<td>$+15%$</td>
</tr>
</tbody>
</table>

$^a$ deviation = (Experiment - REFPROP) / REFPROP

$^b$ for both approaches
Figure 8.7: R-236ea thermal diffusivity versus temperature
Figure 8.8: R-236ea Prandtl number versus temperature

R-236ea Pr versus temperature

Prandtl number, Pr

temperature, C
CHAPTER 9. PROPERTIES OF REFRIGERANT MIXTURES

In this chapter, the properties of the refrigerant mixtures, blend A and blend B, are presented. Some thermophysical properties of blend A are listed in Appendix C. Because these ternary mixtures are non-azeotropic mixtures, the compositions of constitution are very sensitive to the properties. Therefore, refrigerant mixtures must be charged into the system in the liquid form to prevent alteration of the composition. All properties were measured by the current approaches mentioned previously. The results were also compared with REFPROP and are presented in the following sections.

Properties of blend A: R-32 (0.23)/R-125 (0.25)/R-134a (0.52)

The properties measured for blend A include thermal conductivity, viscosity, specific heat, and density. They are shown in the following sections.

Liquid thermal conductivity of blend A

The thermal conductivity of blend A was measured by the same test-section as before and calculated by both approaches, Approach 1 and Approach 2. The test conditions covered a temperature range of -10°C to 40°C. Figure 9.1 shows the thermal conductivity of blend A versus temperature. Data points from REFPROP
were also plotted in this figure. As can be seen, the measured data show rather close agreement to the REFPROP data. The deviation from REFPROP is within ±8%. Moreover, the measured points calculated from Approaches 1 and 2 indicated pretty close agreement with each other. Because the temperature range is somewhat narrow, a linear regression equation would be adequate to fit the data. A representative curve fit equation is provided as follows:

$$k_{bA} = (1.02185875E - 01) + (-5.25052368E - 04) \cdot T$$

(9.1)

where $k$ is $W/m \cdot C$ and $T$ is °C. Although this equation is a linear regression, a somewhat extended extrapolation is still possible.

**Liquid viscosity of blend A**

Viscosity of blend A was measured and shown in Figure 9.2. As indicated in this figure, the viscosity as expected reduced with an increase in temperature. The measured viscosity was shown to be generally lower than REFPROP's prediction within 8%. A curve fit equation for measured viscosity is also provided in Equation 9.2:

$$\mu_{bA} = \sum_{i=0}^{5} a_i \cdot T^i$$

(9.2)

where the curve fit coefficients are shown in Table 9.1. The unit of viscosity is cp while temperature is °C.

It is noted that the above equation is only applicable for a temperature range of -15°C to 40°C.
Figure 9.1: Liquid thermal conductivity of blend A versus temperature
Figure 9.2: Liquid viscosity of blend A versus temperature
Table 9.1: Coefficients for blend A viscosity

<table>
<thead>
<tr>
<th>refrigerant</th>
<th>blend A</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>2.0345494E-01</td>
</tr>
<tr>
<td>$a_1$</td>
<td>-2.66797119E-03</td>
</tr>
<tr>
<td>$a_2$</td>
<td>1.61371918E-06</td>
</tr>
<tr>
<td>$a_3$</td>
<td>-1.29005286E-08</td>
</tr>
<tr>
<td>$a_4$</td>
<td>4.46874537E-08</td>
</tr>
<tr>
<td>$a_5$</td>
<td>-7.89911914E-10</td>
</tr>
<tr>
<td>determinant</td>
<td>9.96237278E-01</td>
</tr>
</tbody>
</table>

Liquid specific heat of blend A

A plot of specific heat of blend A versus temperature is shown in Figure 9.3. A REFPROP prediction line is also plotted. An average deviation of measured data from REFPROP was calculated within ±3% over a temperature range of $-10^\circ\text{C}$ to $40^\circ\text{C}$. Again, a linear equation is obtained to represent the measured line as follows:

$$C_{pbA} = (1.38011230E + 03) + (5.69799995E + 00) \cdot T \quad (9.3)$$

where $C_p$ is $J/kg \cdot ^\circ\text{C}$ and $T$ is $^\circ\text{C}$. Although this equation is also a linear regression, limited extrapolation is still possible.

Liquid density of blend A

A plot of the density of blend A versus temperature is shown in Figure 9.4. A REFPROP prediction line is also plotted. An average deviation of measured data from REFPROP was calculated within ±1.2% over a temperature range of $-10^\circ\text{C}$ to $40^\circ\text{C}$. A fifth degree polynomial equation is obtained to represent the measured
Figure 9.3: Liquid specific heat of blend A versus temperature
line as follows:

$$\rho_{bA} = \sum_{i=0}^{5} a_i \cdot T^i$$  \hspace{1cm} (9.4)$$

where the curve fit coefficients are shown in Table 9.2. The unit of density is $kg/m^3$ while temperature is °C.

<table>
<thead>
<tr>
<th>refrigerant</th>
<th>blend A</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>1.23781580E+03</td>
</tr>
<tr>
<td>$a_1$</td>
<td>-3.66802144E+00</td>
</tr>
<tr>
<td>$a_2$</td>
<td>-1.09664537E-03</td>
</tr>
<tr>
<td>$a_3$</td>
<td>-1.24647093E-04</td>
</tr>
<tr>
<td>$a_4$</td>
<td>-2.85761689E-05</td>
</tr>
<tr>
<td>$a_5$</td>
<td>7.22930338E-07</td>
</tr>
<tr>
<td>determinant</td>
<td>9.99969244E-01</td>
</tr>
</tbody>
</table>

Other properties such as thermal diffusivity which is defined as $k/\rho C_p$, and Prandtl number which is $\mu C_p/k$ can also be calculated from the measurements of the current study. Figure 9.5 shows the thermal diffusivity of blend A versus temperature and Figure 9.6 shows the Prandtl number versus temperature. As it can be seen from these figures, both properties are not significantly affected by the temperature of the current test range. However, they are somewhat affected in viscosity.

**Properties of blend B: R-125 (0.44)/R134a (0.04)/R-143a (0.52)**

The properties measured for blend B include thermal conductivity, viscosity, specific heat, and density and results are outlined in the following sections.
Figure 9.4: Liquid density of blend A versus temperature
Figure 9.5: Liquid thermal diffusivity of blend A versus temperature
blend A Prandtl number versus temperature

Figure 9.6: Liquid Prandtl number of blend A versus temperature
Liquid thermal conductivity of blend B

Again, the thermal conductivity of blend B was measured by the same test-section and calculated by both approaches, Approach 1 and Approach 2. The test results covered a temperature range of 0°C to 40°C. Figure 9.7 shows the thermal conductivity of blend B versus temperature. Data points from REFPROP were also plotted in this figure. As can be seen, the measured data show rather close agreement to the REFPROP data. The deviation from REFPROP is within ±12%. Moreover, the measured points calculated from Approaches 1 and 2 are also fairly close to each other. Because the temperature range is somewhat narrow, a linear regression equation would be adequate to characterize the data. A representative curve fit equation is provided as follows:

\[ k_{bB} = (7.64370412E - 02) + (-2.39820831E - 04) \times T \]  \hspace{1cm} (9.5)

where \( k \) is \( W/m \cdot °C \) and \( T \) is °C. Although this equation is a linear regression, a somewhat linear extrapolation is still valid.

Liquid viscosity of blend B

Viscosity of blend B was measured and shown in Figure 9.8. As indicated in this figure, the viscosity as expected reduced with an increase in temperature. The measured viscosity was shown generally lower than REFPROP's prediction within 8%. A curve fit equation for the measured viscosity is also provided in Equation 9.6.

\[ \mu_{bB} = \sum_{i=0}^{5} a_i \cdot T^i \]  \hspace{1cm} (9.6)

where the curve fit coefficients are shown in Table 9.3. The unit of viscosity is cp, while temperature is °C.
Figure 9.7: Liquid thermal conductivity of blend B versus temperature
Table 9.3: Coefficients for blend B

<table>
<thead>
<tr>
<th>refrigerant</th>
<th>blend B</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>1.65990159E-01</td>
</tr>
<tr>
<td>$a_1$</td>
<td>-1.80001778E-03</td>
</tr>
<tr>
<td>$a_2$</td>
<td>-6.4622907E-05</td>
</tr>
<tr>
<td>$a_3$</td>
<td>5.65852315E-06</td>
</tr>
<tr>
<td>$a_4$</td>
<td>-1.64196109E-07</td>
</tr>
<tr>
<td>$a_5$</td>
<td>1.76490345E-09</td>
</tr>
<tr>
<td>determinant</td>
<td>9.93308663E-01</td>
</tr>
</tbody>
</table>

It is noted that the above equation is only applicable for the temperature range of $0^\circ C$ to $40^\circ C$.

Liquid specific heat of blend B

A plot of specific heat of blend B versus temperature is shown in Figure 9.9. A REFPROP prediction line is also plotted. An average deviation of measured data from REFPROP was calculated within $\pm 1.7\%$ over the temperature range of $0^\circ C$ to $40^\circ C$. Again, a linear equation is obtained to represent the measured line as follows:

$$C_{p_bB} = (1.35658789E + 03) + (6.55297089E + 00) \cdot T \quad (9.7)$$

where $C_p$ is $J/kg \cdot C$ and $T$ is $^\circ C$. Although this equation is a linear regression, limited extended extrapolation is still considered accurate.

Liquid density of blend B

A plot of density of blend B versus temperature is shown in Figure 9.10. A REFPROP prediction line is also plotted. An average deviation of measured data
Figure 9.8: Liquid viscosity of blend B versus temperature
Figure 9.9: Liquid specific heat of blend B versus temperature
from REFPROP was calculated within ±0.5% over the temperature range of 0°C to 40°C. A fifth degree polynomial equation is obtained to represent the measured line as follows:

$$\rho_{bB} = \sum_{i=0}^{5} a_i \cdot T^i$$ (9.8)

where the curve fit coefficients are shown in Table 9.4. The unit of density is $kg/m^3$ while temperature is °C.

**Table 9.4: Coefficients for blend B**

<table>
<thead>
<tr>
<th>density</th>
<th>blend B</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>1.15579333E+03</td>
</tr>
<tr>
<td>$a_1$</td>
<td>-3.88016772E+00</td>
</tr>
<tr>
<td>$a_2$</td>
<td>-9.93751734E-02</td>
</tr>
<tr>
<td>$a_3$</td>
<td>1.07804816E-02</td>
</tr>
<tr>
<td>$a_4$</td>
<td>-4.50898428E-04</td>
</tr>
<tr>
<td>$a_5$</td>
<td>6.09034350E-06</td>
</tr>
<tr>
<td>determinant</td>
<td>9.99526978E-01</td>
</tr>
</tbody>
</table>

Other properties such thermal diffusivity and Prandtl number, extended from these measured data were also calculated and are shown in Figures 9.11 and 9.12.

**Summary**

Blend A and blend B are two possible refrigerant alternatives for R-22 and R-502. Property information is required for evaluating the performance of new refrigerant mixtures. Liquid properties such as thermal conductivity, viscosity, specific heat, and density were measured over a temperature range of −10°C to 40°C (blend A) and of 0°C to 40°C (blend B). A comparison of measured data with REFPROP data
Figure 9.10: Liquid density of blend B versus temperature
Figure 9.11: Liquid thermal diffusivity of blend B versus temperature
Figure 9.12: Liquid Prandtl number of blend B versus temperature
was also made. Equations for these properties, as a function of temperature, were also provided as a possible improvement. In general, the specific heat and density matched REFPROP data pretty well. However, viscosity and thermal conductivity showed a somewhat deviation from REFPROP. A summary of the deviations for all these properties is shown in Table 9.5:

Table 9.5: Summary of deviation for measured properties and REFPROP

<table>
<thead>
<tr>
<th>properties</th>
<th>blend A</th>
<th>blend B</th>
</tr>
</thead>
<tbody>
<tr>
<td>thermal conductivity</td>
<td>±8%</td>
<td>±12%</td>
</tr>
<tr>
<td>viscosity</td>
<td>−8%</td>
<td>−8%</td>
</tr>
<tr>
<td>specific heat</td>
<td>±3%</td>
<td>±1.7%</td>
</tr>
<tr>
<td>density</td>
<td>±1.2%</td>
<td>±0.5%</td>
</tr>
</tbody>
</table>
CHAPTER 10. PROPERTIES OF BLEND A AND LUBRICANT MIXTURE

Property characteristics of a refrigerant mixture mixed with a polyol ester (POEs) lubricant (ICI Emkarate RL-32S) were investigated. Properties studied included thermal conductivity and viscosity. However, specific heat, density, thermal diffusivity, and Prandtl number were also provided. A procedure of lubricant injection and mass fraction sampling were described in Chapter 3. Five lubricant concentrations were studied in this work. Each concentration sample was taken after ten hours of refrigerant loop circulation following the injection of the lubricant. A summary table for lubricant concentrations is shown in Table 10.1.

Table 10.1: List of blend A-lubricant concentration samples

<table>
<thead>
<tr>
<th>number of run</th>
<th>empty cell weight (g)</th>
<th>total cell weight (g)</th>
<th>net ref.+ oil weight (g)</th>
<th>net oil+ cell weight (g)</th>
<th>net oil weight (g)</th>
<th>net ref. weight (g)</th>
<th>oil mass fraction (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>521.67</td>
<td>607.21</td>
<td>85.5325</td>
<td>523.47</td>
<td>1.7925</td>
<td>83.74</td>
<td>2.09</td>
</tr>
<tr>
<td>2</td>
<td>521.67</td>
<td>606.72</td>
<td>85.0475</td>
<td>525.21</td>
<td>3.5975</td>
<td>81.45</td>
<td>4.23</td>
</tr>
<tr>
<td>3</td>
<td>521.67</td>
<td>607.28</td>
<td>85.6075</td>
<td>526.53</td>
<td>4.8775</td>
<td>80.73</td>
<td>5.70</td>
</tr>
<tr>
<td>4</td>
<td>521.67</td>
<td>608.31</td>
<td>86.6325</td>
<td>527.58</td>
<td>5.9875</td>
<td>80.65</td>
<td>6.91</td>
</tr>
<tr>
<td>5</td>
<td>521.67</td>
<td>607.00</td>
<td>85.3275</td>
<td>531.58</td>
<td>9.9075</td>
<td>75.42</td>
<td>11.61</td>
</tr>
</tbody>
</table>
Effects of lubricant concentration on thermal conductivity of mixture

Because of limited lubricant data sources, pure lubricant thermal conductivity could not be obtained. However, in general, the lubricant thermal conductivity is usually 2-3 times that of most of refrigerants. So far, there have been very limited papers published in this area. A theoretical predictive mixing rule was mentioned in previous studies [94, 96]. This rule is described by the Equation 2.6. Although this equation has been widely used in refrigerant mixtures, its accuracy with lubricant-refrigerant mixtures of multiple compositions remains questionable and in need of further verification. In the current study, the thermal conductivity of blend A with a selected lubricant (ICI Emkarate RL-32S) mixture was measured. For this study, the thermal conductivity calculation was based on Approach 2 (Chapter 5) because it showed less uncertainty and was easier to apply from previous results.

Figure 10.1 shows the lubricant-blend A mixture thermal conductivity versus temperature for various lubricant concentrations. As shown in this figure, the thermal conductivity increases with lubricant concentration. At lower temperatures, the increase is more significant than it is at higher temperatures. It also shows the lubricant concentration effect on the thermal conductivity. At higher temperatures, a minor effect of lubricant concentration on mixture thermal conductivity is indicated. A smooth plot based on curve fit equations was also plotted in Figure 10.2. The curve fit equations are provided at later point in this discussion. Figure 10.3 shows the thermal conductivity of the mixture increasing with lubricant concentration. A nearly 30% thermal conductivity increase was found at a lower temperature (−10°C) and at a higher lubricant concentration (11.6%). As indicated, at a high temperature (40°C), the effect is not significant at a concentration less than 10%.
At higher temperatures, over 40°C, an even smaller effect would be expected. This tendency would match statements by Baustian et. al. (1986) [96] and Jensen and Jackman (1984) [94] who tested R-113 with napthenic lubricant at a high temperature (47.7°C) and experienced a thermal conductivity increase under 3%. However, different lubricant-refrigerant mixtures might have different mixing characteristics.

An alternative expression of lubricant mixture thermal conductivity plotted versus lubricant concentration is shown in Figure 10.4. In this figure, a number of temperature lines were plotted versus lubricant concentration. As shown, thermal conductivity increases with higher lubricant concentration and decreases with higher temperatures. This is true and is expected because the thermal conductivity of the lubricant is about 2-3 times higher than refrigerants and thermal conductivity generally decreased with increasing temperature.

A two variable correlation of temperature (T) and lubricant concentration ($C_{oil}$) for thermal conductivity of lubricant-blend A mixture was developed by a curve fit scheme as shown below:

$$k_m = a_0 + a_1T + a_2C_{oil} + a_3T^2 + a_4TC_{oil} + a_5C_{oil}^2 + a_6T^3 + a_7T^2C_{oil} + a_8TC_{oil}^2 + a_9C_{oil}^3$$

(10.1)

where $k$ is $W/m \cdot K$, $T$ is °C, and $C_{oil}$ is the mass fraction. The $a_i$ are listed in Table 10.2.

It should be noted that some terms in Equation 10.2 seem unimportant to the predicted value of the mixture thermal conductivity from viewing the coefficients, which are relatively small and are listed in Table 10.2. Three terms, including $T^2$, $T^3$, and $T^2 \cdot C_{oil}$, can be omitted and will not affect the accuracy of the prediction.
Thermal conductivity of lubricant-blend A mixtures
(Lubricant: ICI EMKARATE RL 32S)

Figure 10.1: Liquid thermal conductivity of lubricant-blend A mixture versus temperature
Figure 10.2: Smooth plot of liquid thermal conductivity of lubricant-blend A mixture versus temperature
**Figure 10.3:** Thermal conductivity increasing percentage of lubricant-blend A mixture versus temperature.
Figure 10.4: Thermal conductivity of lubricant-blend A mixture versus lubricant mass fraction
### Table 10.2: Coefficients of thermal conductivity for lubricant-blend A mixtures

<table>
<thead>
<tr>
<th>polynomial terms</th>
<th>coefficients</th>
<th>values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intercept</td>
<td>$a_0$</td>
<td>0.104180</td>
</tr>
<tr>
<td>$T$</td>
<td>$a_1$</td>
<td>-0.000565</td>
</tr>
<tr>
<td>$C_{oil}$</td>
<td>$a_2$</td>
<td>0.200480</td>
</tr>
<tr>
<td>$T^2$</td>
<td>$a_3$</td>
<td>3.26E-10</td>
</tr>
<tr>
<td>$T \cdot C_{oil}$</td>
<td>$a_4$</td>
<td>-0.006018</td>
</tr>
<tr>
<td>$C_{oil}^2$</td>
<td>$a_5$</td>
<td>3.049586</td>
</tr>
<tr>
<td>$T^3$</td>
<td>$a_6$</td>
<td>-3.62E-12</td>
</tr>
<tr>
<td>$T^2 \cdot C_{oil}$</td>
<td>$a_7$</td>
<td>-1.37E-09</td>
</tr>
<tr>
<td>$T \cdot C_{oil}^2$</td>
<td>$a_8$</td>
<td>0.019484</td>
</tr>
<tr>
<td>$C_{oil}^3$</td>
<td>$a_9$</td>
<td>-24.221797</td>
</tr>
</tbody>
</table>

A comparison of the deviation between experimental data and predicted values by the currently developed equation are shown in Figure 10.5. As shown in this figure, the predicted values agree with the experimental data within ±2%.

An individual equation for the thermal conductivity of lubricant-blend A mixture is also provided for various lubricant concentrations and is shown below:

- **0.0% lubricant**: $k_m = (1.03812814E - 01) + (-5.56966057E - 04)T$ (10.2)
- **2.1% lubricant**: $k_m = (1.10714816E - 01) + (-7.05215149E - 04)T$ (10.3)
- **4.2% lubricant**: $k_m = (1.14966728E - 01) + (-7.68150843E - 04)T$ (10.4)
- **5.7% lubricant**: $k_m = (1.20852523E - 01) + (-8.34880862E - 04)T$ (10.5)
- **6.9% lubricant**: $k_m = (1.25194848E - 01) + (-8.98679136E - 04)T$ (10.6)
- **11.6% lubricant**: $k_m = (1.30798012E - 01) + (-9.94335976E - 04)T$ (10.7)
Figure 10.5: Predicted thermal conductivity versus experimental data of lubricant-blend A mixtures
Effects of lubricant concentration on the viscosity of mixture

Lubricant has a very high viscosity compared to refrigerant. In general, it is several thousand times greater than most refrigerants at low temperatures. The viscosity of lubricant (ICI Emkarate RL 32S) in this study was measured by a capillary viscometer with the temperature controlled by a constant temperature bath. The plot of lubricant viscosity versus temperature is shown in Figure 10.6. As indicated in the figure, the viscosity changes rapidly with temperature, especially in the low temperature range. However, the viscosity in the high temperature range, although not significantly changing with temperature, is still very high and could be several hundred times the refrigerant viscosity. When lubricant is mixed with refrigerant, the viscosity of the mixture rapidly increases due to the high viscosity of the lubricant.

In the present study, the viscosity of lubricant-blend A mixture was measured. Figure 10.7 shows the mixture viscosity versus temperature for various lubricant concentrations. As shown in this figure, the viscosity increases with a lubricant concentration increase, and with a temperature decrease. An interesting result found that the mixture viscosity rapidly increased with a high lubricant concentration of 11.6%. In low temperatures (below 0°C) and lubricant concentrations under 7%, it was found that its effect on the mixture viscosity was not as significant, but still has some degree of influence. At high temperature situations, the lubricant concentration was shown not so obviously to affect the mixture viscosity as at low temperatures. Figure 10.8 shows the smooth curve fit plot for viscosity versus temperature, based upon the experimental results. An alternative plot for viscosity versus concentration for various temperatures is shown in Figure 10.9. Again, the phenomena still holds true.
Figure 10.6: Pure lubricant viscosity versus temperature
Figure 10.7: Viscosity versus temperature for lubricant-blend A mixture under various lubricant concentrations
Figure 10.8: Smooth plot of viscosity versus temperature for lubricant-blend A mixture
Figure 10.9: Viscosity versus lubricant mass fraction for lubricant-blend A mixture under various temperatures
Equations developed for applying these results are useful and they are provided in the following discussion. An individual lubricant concentration equation was fit and listed below:

\[
\ln(\mu) = -561.097336 + 58176\frac{1}{T} + 1.80002T - 0.0019377T^2 \quad (10.8)
\]

\[
\ln(\mu) = -47.465448 + 5116.295806\frac{1}{T} + 0.147864T - 0.000173T^2 \quad (10.9)
\]

\[
\ln(\mu) = 91.768918 - 7969.560719\frac{1}{T} - 0.346109T + 0.00041T^2 \quad (10.10)
\]

\[
\ln(\mu) = 152.920278 - 13973\frac{1}{T} - 0.553686T + 0.000644T^2 \quad (10.11)
\]

\[
\ln(\mu) = 90.882423 - 8000.603788\frac{1}{T} - 0.340262T + 0.00047T^2 \quad (10.12)
\]

\[
\ln(\mu) = 120.641368 - 10814\frac{1}{T} - 0.446502T + 0.000527T^2 \quad (10.13)
\]

where \( T \) is K and \( \mu \) is cp.

A two variable correlation, temperature (T) and lubricant concentration (\( C_{oil} \)), for the viscosity of lubricant-blend A mixture was also developed by the curve fit scheme as shown below:

\[
\ln(\mu_m) = a_0 + a_1T + a_2C_{oil} + a_3T^2 + a_4TC_{oil} + a_5C_{oil}^2 \\
+ a_6T^3 + a_7T^2C_{oil} + a_8TC_{oil}^2 + a_9C_{oil}^3 \quad (10.14)
\]
where $\mu$ is cp, $T$ is °C, and $C_{oil}$ is the mass fraction. The $a_i$ are listed in Table 10.3. It should be noted that some terms such as $T^2$ and $T^3$ were shown to be insignificant on the calculated values. An accuracy test for using this equation was verified within ±5% with the experimental data covering a viscosity range of 0 to 0.6 cp. Figure 10.10 shows the predicted viscosity by using Equation 10.14 versus experimental data. It shows that at the lower viscosity range (0 to 0.6 cp), the prediction equation is more suitable for use with the experimental data.

Other properties of lubricant-blend A mixture

In this section, specific heat, density, thermal diffusivity, and Prandtl (Pr) are calculated and presented with the lubricant effects. Figure 10.11 shows the specific heat of lubricant-blend A mixtures. As shown, the specific heat is not so significantly affected by lubricant concentrations except at high concentrations of lubricant.
Figure 10.10: Predicted viscosity versus experimental data of lubricant-blend A mixture
in the lower temperature range. Although pure lubricant specific heat is difficult to obtain, it can be assumed that the specific heat for both materials should be fairly close. However, further verification will be required when specific heat data of pure lubricants become available.

The density of the mixture is shown in Figure 10.12. As expected, the mixture density does not significantly change, except in the lower temperature range, because the lubricant density is fairly close to blend A (REFPROP data) in the high temperature range but gradually decreases in the lower temperature range. Therefore, the density of the mixture becomes smaller when the temperature decreases.

Thermal diffusivity, $\alpha$, is another property for heat transfer study. It is defined as $k/\rho C_p$. Therefore, $\alpha$ can also be calculated in this study. The plot of $\alpha$ versus temperature for lubricant-blend A mixture is shown in Figure 10.13. The trend is fairly similar to thermal conductivity because $\rho$ and $C_p$ were not obviously affected by lubricant concentration.

Finally, the Pr was also calculated by its definition, $\mu C_p/k$. A plot of thermal diffusivity versus temperature is shown in Figure 10.14. In this figure, a similar tendency is shown with the viscosity because, unlike viscosity, thermal conductivity and specific heat are not significantly changed. Therefore, the viscosity dominates Pr, as expected.

**Summary**

In this chapter, the transport properties of lubricant-blend A mixture are presented. Lubricant concentration affects on both thermal conductivity and viscosity are discussed. However, viscosity is more significantly affected by lubricant concen-
Figure 10.11: Specific heat versus temperature of lubricant-blend A mixture
Figure 10.12: Density versus temperature of lubricant-blend A mixture
Figure 10.13: Thermal diffusivity versus temperature of lubricant-blend A mixture
Figure 10.14: Prandtl (Pr) versus temperature of lubricant-blend A mixture
tration, especially at a lower temperature range. High lubricant concentration showed a very significant effect at low temperatures. Thermal conductivity is affected by lubricant concentrations from 8 to 28% at \(-10^\circ C\) and only from 1 to 10% at \(40^\circ C\) for a lubricant concentration varying from 2 to 11%.

Other properties such as density, specific heat, thermal diffusivity, and Pr were also calculated and discussed. They all showed a reasonable pattern. Of which, Pr is significantly affected by lubricant concentration due to viscosity changes.
In this chapter, conclusions of this current work will be summarized. Finally, the potential improvement of current work and possible extended research by using the current test facility will be outlined and suggested.

Conclusions

This study proposes a new approach for simultaneously measuring several thermophysical properties, such as thermal conductivity, viscosity, specific heat, and density. This approach uses single-phase in-tube heat transfer knowledge to obtain thermal conductivity. Viscosity is measured by a viscometer placed in-line with the heat transfer test section. There are two approaches; Approach 1: the Nusselt number method and Approach 2: the Prandtl number method. The uncertainty analysis was presented in this study. Approach 2 seems to have less uncertainty than Approach 1 and $T_i$ and $\Delta T$ do not significantly affect the uncertainties. However, $\Delta T_{wf}$ is a significant parameter that affects the uncertainties.

In Approach 1, the determination of a calibration function (CF) by experiments using fluids with known properties, is shown to be important for accurate thermal conductivity measurements. Three different Nusselt number correlations were used for calculating thermal conductivity in this study, and they were examined and dis-
discussed. Four refrigerants: R-22, R-12, R-113, and R-114 were used for calibration and verification purposes which cover the Pr from 2.28 to 10.58 and Re_D from 8000 to 18000. Based on the calibration results, the CF functions were found for three different correlations which were examined in this study. In Approach 2, bypassing the Nusselt number, the thermal conductivity was found from Pr which is directly related to Re_D and non-dimensional temperature, AT*. This approach was shown to be more accurate and convenient to use because less variables were involved. A theoretical uncertainty analysis also showed this approach to have less uncertainty than Approach 1. The measured results were also compared and discussed for both approaches and consistency was shown between them.

Viscosity was measured by a torsional oscillation inline viscometer. The accuracy of the viscosity measurement was verified with R-113, R-12, and pure water, and shown within ±2% when compared with the ASHRAE data. The measured properties included specific heat, density, viscosity, and thermal conductivity. They were also examined for R-114, compared with ASHRAE data, and shown to be matched closely within ±5% for thermal conductivity, ±3% for specific heat, and within ±1% for density. For R-236ea property measurements, REFPROP-4.0 data was used as a comparison with the measured data. The deviations of measured properties from REFPROP-4.0 are +4.8% for specific heat, −5.0% for viscosity, ±1% for density, and +15% for thermal conductivity.

Two ternary blends: blend A (R-32 (0.23)/R-125 (0.25)/R-134a (0.52)) and blend B (R-125 (0.44)/R134a (0.04)/R-143a (0.52)) were also tested by the current approach for thermal conductivity and viscosity measurements. The thermal conductivity, viscosity, density, and specific heat for both blends were measured and
compared with the REFPROP data source. In general, thermodynamic properties agree with the REFPROP data quite well, while transport properties deviate somewhat (±12% in R-114 thermal conductivity) from each other. Equations based on measured data are provided for convenient use.

A lubricant (Emkarate RL 32S) was selected to mix with blend A. Properties were measured for five lubricant concentrations over a temperature range of -10 to 40°C. Thermal conductivity effects due to the lubricant seemed to be more significant at low temperatures than high temperatures. The thermal conductivity was found to increase from 8% to 28% compared with the pure mixture at a low temperature (−10°C). This increase of thermal conductivity was less than 10% at a high temperature (40°C). Viscosity was obviously affected by lubricant concentration, especially at a low temperature and high lubricant concentration. It increased over 300% from the pure mixture at the low temperature for a lubricant concentration of 11.6% while it increased less than 100% at the high temperature. Curve fit equations for both one variable (temperature) and two variables (temperature and lubricant concentration) were provided for convenient use. Other properties such as density, specific heat, thermal diffusivity, and Pr were also calculated.

Transport properties are important for the evaluation of refrigerant alternatives. The method developed in this study shows that several properties can be quickly and simultaneously measured. Today, refrigerant alternatives are being screened and tested for application. The methods developed in this study can provide a quick way for obtaining these properties.
Recommendations

The following points are the suggestions for further improvement of the current study and possible extended future research using the current test facility:

1. The Prandtl number (Pr) and Reynolds number (Re) ranges of calibration can be further extended by testing more fluids of known properties to extend the ranges of application for the proposed methodology in this study.

2. The limitations of the current test facility can be extended by installing higher performance refrigerant pumps or changing the heating and cooling manner so that a wider range of mass flow rates and operating temperatures can be obtained.

3. The accuracy of the thermal conductivity and viscosity measurements can be further improved by using more accurate sensors or re-calibrating the current instruments to higher accuracy.

4. Property and heat transfer correlations can be developed if more data are taken and accumulated.

5. Although properties are the primary interest in this study, the single-phase heat transfer and flow characteristics of lubricant-refrigerant mixtures can be further investigated by the current test facility.

6. The miscibility and solubility are related to heat transfer and flow pattern. A further study of the relationships between them would be another interesting topic.
7. The property studies at various compositions of binary and ternary blends can be further investigated by minor modifications of current test facility.

8. An obvious extension of this study would be to use different lubricants and refrigerants which include binary or ternary refrigerant mixtures subjected to a wide range of temperatures. A study of partly miscible lubricant-refrigerant mixtures related to the temperature change would be another interesting study.

9. A real time determination of concentration of lubricant (or oil) dissolved in the refrigerant flow stream can be further studied by properly extending the usage of the current test facility.

10. By installing different types of micro-fin test-tubes in the test-section, the in-tube single-phase heat transfer characteristics and entrance effects can be further investigated for lubricant-refrigerant mixtures, especially for alternative refrigerants for the purpose of real applications.
BIBLIOGRAPHY


[35] Diller, Dwain E., Aragon, Anthong S., and Laesecke, Arno "Measurements of The Viscosities of Saturated and Compressed Liquid 1,1,1,2 - Tetrafluoro-ethane (R-134a), 2,2 - dichloro - 1, 1 - Trifluoroethane (R123), and 1,1 - Dichluo - 1 - Fluoroethane (R142b)." Fluid Phase Equilibria, Vol. 88, pp. 251-262, 1993.


This appendix discusses the use of thermophysical properties such as saturated pressure \( P_{sat} \), density \( \rho \), specific heat \( C_P \), viscosity \( \mu \), and thermal conductivity \( k \). The data source of these properties can be found in a number of references, such as ASHRAE, NIST (REFPROP), Chemistry Physics Handbook, etc. However, for purposes of refrigerant study, ASHRAE data and REFPROP data are usually the preferred sources. It should be noted that REFPROP is a recently developed package which uses theoretical prediction methods. Because of limited experimental data for new refrigerants, such as R-236ea, blend A, and blend B, REFP4.0 (REFPROP version 4.0) was employed as a reference data source for comparison. Although REFPROP data includes almost all of the refrigerants and their mixtures in any combination, it was discovered that the transport properties between these two data sources were sometimes quite different from each other. Therefore, proper care should be taken before using them. In this study, those properties used for thermal conductivity measurements are all based on the ASHRAE data source [93] while comparisons between measured properties and REFPROP [49] were made for the fluids of unknown properties.

In order to use property data conveniently, based on ASHRAE or REFP40 data source, curve fit equations were developed by using polynominal functions de-
scribed by:

\[
property = \sum_{i=0}^{5} a_i \cdot T^i
\]  

(A.1)

where temperature, \( T \) is in °C here and in the following discussions unless otherwise stated. The applied temperature range of regression for all properties is at the saturation temperature, and the pressure is also at the saturated condition. The curve fit temperature range is listed in Table A.1.

### Table A.1: Curve fit temperature range for test refrigerants

<table>
<thead>
<tr>
<th>refrigerant</th>
<th>( P_{\text{sat}} )</th>
<th>density</th>
<th>specific heat</th>
<th>viscosity</th>
<th>conductivity</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-12</td>
<td>-50 ~ 100</td>
<td>-50 ~ 100</td>
<td>-40 ~ 80</td>
<td>-70 ~ 68</td>
<td>-60 ~ 68</td>
</tr>
<tr>
<td>R-22</td>
<td>-130 ~ 96.14</td>
<td>-130 ~ 96.14</td>
<td>-90 ~ 70</td>
<td>-20 ~ 40</td>
<td>-70 ~ 48</td>
</tr>
<tr>
<td>R-113</td>
<td>-30 ~ 214</td>
<td>-30 ~ 214</td>
<td>0 ~ 180</td>
<td>0 ~ 175</td>
<td>0 ~ 170</td>
</tr>
<tr>
<td>R-114</td>
<td>-40 ~ 115</td>
<td>-40 ~ 115</td>
<td>-40 ~ 115</td>
<td>-40 ~ 115</td>
<td>-40 ~ 115</td>
</tr>
<tr>
<td>R-236ea</td>
<td>-40 ~ 100</td>
<td>-40 ~ 100</td>
<td>-40 ~ 100</td>
<td>-40 ~ 100</td>
<td>-40 ~ 100</td>
</tr>
<tr>
<td>blend A</td>
<td>-40 ~ 85</td>
<td>-40 ~ 85</td>
<td>-40 ~ 85</td>
<td>-40 ~ 85</td>
<td>-40 ~ 85</td>
</tr>
<tr>
<td>blend B</td>
<td>-40 ~ 70</td>
<td>-40 ~ 70</td>
<td>-40 ~ 70</td>
<td>-40 ~ 70</td>
<td>-40 ~ 70</td>
</tr>
</tbody>
</table>

**Saturation pressure**

The \( P_{\text{sat}} \) versus temperature curve fit equations are provided in the following sections for the tested refrigerants. The unit for saturation pressure, \( P_{\text{sat}} \) is MPa for ASHRAE and psia for REFPROP data, while the temperature unit is °C.

\( P_{\text{sat}} \) for calibration refrigerants - ASHRAE

Table A.2 lists the saturation pressures of refrigerants based on the ASHRAE data. These data were used to verify the subcooled conditions and to verify all operating conditions were in the liquid phase.
Table A.2: $P_{sat}$ curve fit coefficients for R-22, R-12, R-113, and R-114

<table>
<thead>
<tr>
<th>refrigerant</th>
<th>R-22</th>
<th>R-12</th>
<th>R-113</th>
<th>R-114</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>5.00005E-01</td>
<td>2.91020E-01</td>
<td>1.58554E-02</td>
<td>8.76786E-02</td>
</tr>
<tr>
<td>$a_1$</td>
<td>1.61909E-02</td>
<td>1.00845E-02</td>
<td>6.77690E-04</td>
<td>3.47903E-03</td>
</tr>
<tr>
<td>$a_2$</td>
<td>1.95017E-04</td>
<td>1.49920E-04</td>
<td>1.25848E-05</td>
<td>5.45875E-05</td>
</tr>
<tr>
<td>$a_3$</td>
<td>1.03242E-06</td>
<td>5.64051E-07</td>
<td>2.28213E-07</td>
<td>3.74883E-07</td>
</tr>
<tr>
<td>$a_4$</td>
<td>2.23206E-09</td>
<td>-5.91877E-09</td>
<td>-1.47081E-10</td>
<td>4.43447E-10</td>
</tr>
<tr>
<td>$a_5$</td>
<td>1.71047E-12</td>
<td>5.79095E-11</td>
<td>1.60668E-12</td>
<td>1.88150E-12</td>
</tr>
</tbody>
</table>

$P_{sat}$ for R-236ea - REFPROP

Table A.3 lists the saturation pressure (psia) of refrigerants based on the REFPROP data.

Table A.3: $P_{sat}$ curve fit coefficients for R-236ea

<table>
<thead>
<tr>
<th>polynominal terms</th>
<th>R-236ea</th>
</tr>
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<tbody>
<tr>
<td>$a_0$</td>
<td>1.13349352E+01</td>
</tr>
<tr>
<td>$a_1$</td>
<td>4.84870791E-01</td>
</tr>
<tr>
<td>$a_2$</td>
<td>8.30436591E-03</td>
</tr>
<tr>
<td>$a_3$</td>
<td>6.78510114E-05</td>
</tr>
<tr>
<td>$a_4$</td>
<td>2.04624143E-07</td>
</tr>
<tr>
<td>$a_5$</td>
<td>-2.19035401E-10</td>
</tr>
<tr>
<td>determinant</td>
<td>1.00000000E+00</td>
</tr>
</tbody>
</table>

$P_{sat}$ for blends A and B - REFPROP

For refrigerant mixtures, blends A and B, used in this study, the curve fit equations are also shown in Table A.4.
Table A.4: $P_{\text{sat}}$ curve fit coefficients for blends A and B

<table>
<thead>
<tr>
<th>refrigerant</th>
<th>blend A$^a$</th>
<th>blend B$^b$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>8.24902725E+01</td>
<td>8.87777023E+01</td>
</tr>
<tr>
<td>$a_1$</td>
<td>2.67322135E+00</td>
<td>2.79479003E+00</td>
</tr>
<tr>
<td>$a_2$</td>
<td>3.26958708E-02</td>
<td>3.31294164E-02</td>
</tr>
<tr>
<td>$a_3$</td>
<td>1.71903855E-04</td>
<td>1.66634287E-04</td>
</tr>
<tr>
<td>$a_4$</td>
<td>2.21557997E-07</td>
<td>1.94832268E-07</td>
</tr>
<tr>
<td>$a_5$</td>
<td>-1.68329772E-09</td>
<td>-5.78850690E-10</td>
</tr>
<tr>
<td>determinant</td>
<td>1.00000000E+00</td>
<td>1.00000000E+00</td>
</tr>
</tbody>
</table>

$^a$ blend A: R-32 (0.23)/R-125 (0.25)/R134a (0.52)

$^b$ blend B: R-125 (0.44)/R-134a (0.04)/R-143a (0.52)

**Liquid density**

The liquid density versus temperature curve fit equations are provided in the following sections for the tested refrigerants. The unit for density, $\rho$, is $(kg/m^3)$ while the unit for temperature is °C.

**Density for calibration refrigerants - ASHRAE**

Because the type of viscometer used in this study measures the product of density ($g/cm^3$) and kinematic viscosity (cp), therefore, density is a property needed to be measured in the present study in order to obtain the viscosity. For the calibration refrigerants of known properties, the ASHRAE data for density was used to verify both measured density and viscosity. Table A.5 lists the density curve fit equation coefficients at the saturated liquid state based on ASHRAE data.

**Liquid density for R-236ea - REFPROP**

The curve fitting coefficients of liquid density based on REFPROP are listed in Table A.6.
Table A.5: Liquid density curve fit coefficients for R-22, R-12, R-113, R-114

<table>
<thead>
<tr>
<th>refrigerant</th>
<th>R-22</th>
<th>R-12</th>
<th>R-113</th>
<th>R-114</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>1.27397E+03</td>
<td>1.39594E+03</td>
<td>1.61316E+03</td>
<td>1.50572E+03</td>
</tr>
<tr>
<td>$a_1$</td>
<td>-3.39602E+00</td>
<td>-3.22976E+00</td>
<td>-2.00082E+00</td>
<td>-1.76217E+00</td>
</tr>
<tr>
<td>$a_2$</td>
<td>5.30874E-03</td>
<td>-7.53629E-03</td>
<td>1.26301E-02</td>
<td>-1.78123E-02</td>
</tr>
<tr>
<td>$a_3$</td>
<td>1.21421E-05</td>
<td>-7.65376E-06</td>
<td>-4.93012E-04</td>
<td>-5.05889E-05</td>
</tr>
<tr>
<td>$a_4$</td>
<td>-2.76909E-06</td>
<td>3.27685E-07</td>
<td>4.29266E-06</td>
<td>1.49907E-06</td>
</tr>
<tr>
<td>$a_5$</td>
<td>-1.82476E-08</td>
<td>-1.14387E-08</td>
<td>9.94815E-01</td>
<td>-8.35448E-09</td>
</tr>
</tbody>
</table>

Table A.6: Liquid density curve fit coefficients for R-236ea

<table>
<thead>
<tr>
<th>polynomial terms</th>
<th>R-236ea</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>1.50280E+03</td>
</tr>
<tr>
<td>$a_1$</td>
<td>-2.84734E+00</td>
</tr>
<tr>
<td>$a_2$</td>
<td>-5.44754E-03</td>
</tr>
<tr>
<td>$a_3$</td>
<td>-2.65424E-05</td>
</tr>
<tr>
<td>$a_4$</td>
<td>1.76196E-08</td>
</tr>
<tr>
<td>$a_5$</td>
<td>-1.89195E-09</td>
</tr>
<tr>
<td>determinant</td>
<td>9.99996E-01</td>
</tr>
</tbody>
</table>

Liquid density for blends A and B - REFPROP

The liquid density curve fit equation for blends A and B, based on REFPROP, was obtained as follows. Table A.7 shows the curve fit coefficients for blends A and B.

Liquid specific heat

The liquid specific heat versus temperature curve fit equations are provided in the following sections for the tested refrigerants. The units for specific heat, $C_p$, are $kJ/m^3$, while the unit for temperature is °C.

Specific heat for calibration refrigerants - ASHRAE
Table A.7: Liquid density curve fit coefficients for blends A and B

<table>
<thead>
<tr>
<th>refrigerant</th>
<th>blend A</th>
<th>blend B</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>1.25249316E+03</td>
<td>1.15292554E+03</td>
</tr>
<tr>
<td>$a_1$</td>
<td>-3.64262772E+00</td>
<td>-3.90139627E+00</td>
</tr>
<tr>
<td>$a_2$</td>
<td>-1.10913366E-02</td>
<td>-1.56064043E-02</td>
</tr>
<tr>
<td>$a_3$</td>
<td>-5.23424715E-05</td>
<td>-3.50880182E-05</td>
</tr>
<tr>
<td>$a_4$</td>
<td>7.59501972E-08</td>
<td>1.76500635E-07</td>
</tr>
<tr>
<td>$a_5$</td>
<td>-1.06245919E-08</td>
<td>-3.39652217E-08</td>
</tr>
<tr>
<td>determinant</td>
<td>9.99996483E-01</td>
<td>9.99996543E-01</td>
</tr>
</tbody>
</table>

Table A.8: Liquid specific heat curve fit coefficients for R-22, R-12, R-113, R-114

<table>
<thead>
<tr>
<th>refrigerant</th>
<th>R-22</th>
<th>R-12</th>
<th>R-113</th>
<th>R-114</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>1.17392E+00</td>
<td>9.27994E-01</td>
<td>9.21305E-01</td>
<td>9.30861E-01</td>
</tr>
<tr>
<td>$a_1$</td>
<td>2.72938E-03</td>
<td>1.87657E-03</td>
<td>1.70157E-03</td>
<td>1.13601E-03</td>
</tr>
<tr>
<td>$a_2$</td>
<td>1.46720E-05</td>
<td>1.52956E-05</td>
<td>-1.40780E-05</td>
<td>-3.84778E-06</td>
</tr>
<tr>
<td>$a_3$</td>
<td>1.18154E-07</td>
<td>-1.12761E-08</td>
<td>9.15992E-08</td>
<td>1.96439E-07</td>
</tr>
<tr>
<td>$a_4$</td>
<td>6.23786E-09</td>
<td>-5.42684E-10</td>
<td>6.44030E-11</td>
<td>8.34917E-11</td>
</tr>
<tr>
<td>$a_5$</td>
<td>5.16438E-11</td>
<td>4.66421E-11</td>
<td>-1.80450E-13</td>
<td>-4.94558E-13</td>
</tr>
</tbody>
</table>

Table A.8 lists the specific heat curve fit equation coefficients for the saturated liquid state of R-22, R-12, R-113, and R-114 based on the ASHRAE data.

Specific heat for R-236ea - REFPROM

The coefficients of specific heat curve fit based on REFPROM are listed in Table A.9.

Specific heat for blend A and B - REFPROM

The liquid density curve fit equation for blends A and B, based on REFPROM,
was obtained as follows. Table A.10 shows the curve fit coefficients for blends A and B.

**Liquid viscosity**

The liquid viscosity versus temperature curve fit equations are provided in the following sections for the tested refrigerants. The units for viscosity, $\mu$, are $Pa \cdot S$ while the unit for temperature is $^\circ C$.

**Viscosity for calibration refrigerants - ASHRAE**

Table A.11 lists the kinematic viscosity (in $Pa \cdot S$) curve fit equation coefficients at the saturated liquid state based on ASHRAE data.

**Liquid viscosity for R-236ea - REFPROP**

The coefficients of viscosity curve fit equation based on REFPROP are listed in Table A.12.

**Liquid viscosity for blends A and B - REFPROP**

The liquid density curve fit equation for blends A and B, based on REFPROP, was obtained from the following. Table A.13 shows the curve fit coefficients for blends

<table>
<thead>
<tr>
<th>polynominal terms</th>
<th>coefficients</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>1.15099E+03</td>
</tr>
<tr>
<td>$a_1$</td>
<td>2.45936E+00</td>
</tr>
<tr>
<td>$a_2$</td>
<td>4.03757E-03</td>
</tr>
<tr>
<td>$a_3$</td>
<td>1.64306E-05</td>
</tr>
<tr>
<td>$a_4$</td>
<td>-7.58220E-07</td>
</tr>
<tr>
<td>$a_5$</td>
<td>1.53071E-08</td>
</tr>
<tr>
<td>determinant</td>
<td>9.99988E-01</td>
</tr>
</tbody>
</table>
Table A.10: Liquid specific heat curve fit coefficients for blends A and B

<table>
<thead>
<tr>
<th>polynomial terms</th>
<th>blend A</th>
<th>blend B</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a_0)</td>
<td>1.36183484E+03</td>
<td>1.34657886E+03</td>
</tr>
<tr>
<td>(a_1)</td>
<td>4.04480314E+00</td>
<td>6.05644703E+00</td>
</tr>
<tr>
<td>(a_2)</td>
<td>7.27501735E-02</td>
<td>9.71828103E-02</td>
</tr>
<tr>
<td>(a_3)</td>
<td>-7.23460515E-04</td>
<td>-2.65167817E-03</td>
</tr>
<tr>
<td>(a_4)</td>
<td>-2.53525941E-05</td>
<td>-3.25687506E-05</td>
</tr>
<tr>
<td>(a_5)</td>
<td>5.50435402E-07</td>
<td>1.48304116E-06</td>
</tr>
<tr>
<td>determinant</td>
<td>9.99096692E-01</td>
<td>9.7995794E-01</td>
</tr>
</tbody>
</table>

Table A.11: Liquid viscosity curve fit coefficients for R-22, R-12, R-113, R-114

<table>
<thead>
<tr>
<th>terms</th>
<th>R-22</th>
<th>R-12</th>
<th>R-113</th>
<th>R-114</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a_0)</td>
<td>2.10126E-04</td>
<td>2.57691E-04</td>
<td>9.54513E-04</td>
<td>4.91168E-04</td>
</tr>
<tr>
<td>(a_1)</td>
<td>-2.27812E-06</td>
<td>-2.77653E-06</td>
<td>-1.31618E-05</td>
<td>-6.55940E-06</td>
</tr>
<tr>
<td>(a_2)</td>
<td>1.14319E-08</td>
<td>1.89625E-08</td>
<td>1.11619E-07</td>
<td>4.48919E-08</td>
</tr>
<tr>
<td>(a_3)</td>
<td>3.80621E-12</td>
<td>-8.65530E-11</td>
<td>-6.34269E-10</td>
<td>-1.26949E-10</td>
</tr>
<tr>
<td>(a_4)</td>
<td>-7.75142E-13</td>
<td>-2.19818E-14</td>
<td>2.15999E-12</td>
<td>-4.40069E-13</td>
</tr>
<tr>
<td>(a_5)</td>
<td>7.72755E-15</td>
<td>2.96471E-15</td>
<td>-3.23601E-15</td>
<td>3.18456E-15</td>
</tr>
</tbody>
</table>

A and B.

**Liquid thermal conductivity**

The liquid thermal conductivity versus temperature curve fit equations are provided in the following sections for the tested refrigerants. The units for thermal conductivity, \(k\), are \(W/m \cdot K\) while the unit for temperature is \(°C\).

**Thermal conductivity for calibration refrigerants - ASHRAE**

Thermal conductivity quoted for the calibration refrigerants was based on ASHRAE
Table A.12: Liquid viscosity curve fit coefficients for R-236ea

<table>
<thead>
<tr>
<th>polynomial terms</th>
<th>R-236ea</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>5.83657E-04</td>
</tr>
<tr>
<td>$a_1$</td>
<td>-9.32850E-06</td>
</tr>
<tr>
<td>$a_2$</td>
<td>1.03792E-07</td>
</tr>
<tr>
<td>$a_3$</td>
<td>-1.08120E-09</td>
</tr>
<tr>
<td>$a_4$</td>
<td>8.20849E-12</td>
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<tr>
<td>$a_5$</td>
<td>-2.86313E-14</td>
</tr>
<tr>
<td>determinant</td>
<td>9.99997E-01</td>
</tr>
</tbody>
</table>

Table A.13: Liquid viscosity curve fit coefficients for blends A and B

<table>
<thead>
<tr>
<th>refrigerant</th>
<th>blend A</th>
<th>blend B</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>2.24262447E-04</td>
<td>1.79984272E-04</td>
</tr>
<tr>
<td>$a_1$</td>
<td>-2.76354513E-06</td>
<td>-2.26994439E-06</td>
</tr>
<tr>
<td>$a_2$</td>
<td>1.73987864E-08</td>
<td>3.40445361E-09</td>
</tr>
<tr>
<td>$a_3$</td>
<td>-1.34042943E-10</td>
<td>2.21182239E-10</td>
</tr>
<tr>
<td>$a_4$</td>
<td>1.04177799E-12</td>
<td>5.73477160E-12</td>
</tr>
<tr>
<td>$a_5$</td>
<td>-4.04459588E-15</td>
<td>-1.33448184E-13</td>
</tr>
<tr>
<td>determinant</td>
<td>9.99943137E-01</td>
<td>9.98672426E-01</td>
</tr>
</tbody>
</table>

data. Table A.14 lists the thermal conductivity (in W/m · K) curve fit equation coefficients at the saturated liquid state.

**Thermal conductivity for R-236ea - REFPROP**

The coefficients of thermal conductivity curve fit equation based on REFPROP are created in Table A.15.

**Liquid thermal conductivity for blends A and B - REFPROP**

The liquid density curve fit equation for blends A and B, based on REFPROP,
Table A.14: Liquid thermal conductivity curve fit coefficients for R-22, R-12, R-113, and R-114

<table>
<thead>
<tr>
<th>refrigerant</th>
<th>R-22</th>
<th>R-12</th>
<th>R-113</th>
<th>R-114</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a_0)</td>
<td>9.61623E-02</td>
<td>7.71084E-02</td>
<td>8.19249E-02</td>
<td>7.04566E-02</td>
</tr>
<tr>
<td>(a_1)</td>
<td>-4.24513E-04</td>
<td>-3.66836E-04</td>
<td>-1.74554E-04</td>
<td>-2.41089E-04</td>
</tr>
<tr>
<td>(a_2)</td>
<td>4.25538E-07</td>
<td>1.77984E-07</td>
<td>-6.07317E-08</td>
<td>2.54213E-07</td>
</tr>
<tr>
<td>(a_3)</td>
<td>4.88343E-10</td>
<td>-4.95688E-10</td>
<td>1.01692E-09</td>
<td>-9.79673E-10</td>
</tr>
<tr>
<td>(a_4)</td>
<td>-2.09592E-11</td>
<td>6.91404E-13</td>
<td>-6.47263E-12</td>
<td>-2.41232E-11</td>
</tr>
<tr>
<td>(a_5)</td>
<td>-4.18749E-13</td>
<td>1.10221E-13</td>
<td>1.55361E-14</td>
<td>1.19913E-13</td>
</tr>
</tbody>
</table>

Table A.15: Liquid thermal conductivity curve fit coefficients for R-236ea

<table>
<thead>
<tr>
<th>polynomial terms</th>
<th>R-236ea</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a_0)</td>
<td>7.98508E-02</td>
</tr>
<tr>
<td>(a_1)</td>
<td>-3.22129E-04</td>
</tr>
<tr>
<td>(a_2)</td>
<td>-5.87303E-07</td>
</tr>
<tr>
<td>(a_3)</td>
<td>8.10775E-09</td>
</tr>
<tr>
<td>(a_4)</td>
<td>2.08164E-11</td>
</tr>
<tr>
<td>(a_5)</td>
<td>-4.47036E-13</td>
</tr>
<tr>
<td>determinant</td>
<td>9.99743E-01</td>
</tr>
</tbody>
</table>

was obtained as follows. Table A.16 shows the curve fit coefficients for blends A and B.
Table A.16: Liquid thermal conductivity curve fit coefficients for blends A and B

<table>
<thead>
<tr>
<th>refrigerant</th>
<th>blend A</th>
<th>blend B</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>1.03215024E-01</td>
<td>7.81119093E-02</td>
</tr>
<tr>
<td>$a_1$</td>
<td>-6.91256893E-04</td>
<td>-6.11332827E-04</td>
</tr>
<tr>
<td>$a_2$</td>
<td>3.69709596E-10</td>
<td>-2.91751167E-06</td>
</tr>
<tr>
<td>$a_3$</td>
<td>1.01864988E-08</td>
<td>1.21877434E-07</td>
</tr>
<tr>
<td>$a_4$</td>
<td>7.46210316E-11</td>
<td>2.00395212E-09</td>
</tr>
<tr>
<td>$a_5$</td>
<td>-2.79397837E-12</td>
<td>-5.31645526E-11</td>
</tr>
<tr>
<td>determinant</td>
<td>9.99992311E-01</td>
<td>9.96164799E-01</td>
</tr>
</tbody>
</table>
APPENDIX B. UNCERTAINTY ANALYSIS FOR THERMAL CONDUCTIVITY MEASUREMENT

This appendix analyzes the uncertainty of the thermal conductivity measurement for the methods of Approach 1: the Nusselt number method, and Approach 2: the Prandtl number method.

Uncertainty analysis of Approach 1

Three different single-phase correlations employed in the thermal conductivity calculations will be analyzed for their uncertainty. The uncertainties of measured thermal conductivity will arise due to both sensor (or instrument) uncertainty as well as the uncertainty in the Nusselt number correlations. However, in this analysis, only the sensor uncertainties are used for the uncertainty calculation, while the Nusselt number uncertainty induced is not discussed in this current analysis.

Uncertainty analysis of \( k \) by the Dittus-Boelter correlation

The thermal conductivity obtained from the Dittus-Boelter correlation is:

\[
k_d = \left[ \frac{\bar{h}D}{0.023 Re^{-0.8} \mu^{-0.4} C_p^{-0.4}} \right]^{\frac{5}{3}}
\]  

\( (B.1) \)

\[
k_d = f \left( \bar{h}, D, Re_D, \mu, C_p \right)
\]  

\( (B.2) \)
The thermal conductivity presented here is a function of $\bar{h}$, $D$, $Re_D$, $\mu$, and $C_p$. Therefore, the uncertainty of the sum of $U_{kd}/k_d$ can be expressed as follows:

$$
\left( \frac{U_{kd}}{k_d} \right)^2 = \left( \frac{5 U_h}{3 \bar{h}} \right)^2 + \left( \frac{5 U_D}{3 D} \right)^2 + \left( -\frac{4 U_{Re_D}}{3 Re_D} \right)^2 + \\
\left( \frac{-2 U_{\mu}}{3 \mu} \right)^2 + \left( \frac{-2 U_{C_p}}{3 C_p} \right)^2
$$

(B.3)

$$
= SS_{k_d}
$$

(B.4)

where $U_h/h$, and $U_{Re_D}/Re_D$, are discussed in the following sections. Then, $U_{kd}/k_d$ is obtained by taking the square roots of the sums of squares of each uncertainty source, $SS_{k_d}$. That is:

$$
\frac{U_{kd}}{k_d} = (SS_{k_d})^{\frac{1}{2}}
$$

(B.5)

Uncertainty analysis of thermal conductivity by the Petukhov-Popov correlation

The thermal conductivity from the Petukhov correlation is found to be:

$$
k_p = \mu C_p \left[ \frac{(8f)^{\frac{1}{2}} \mu C_p Re_D}{101.6 h D} - \frac{1.07}{12.7} \left( \frac{f}{8} \right)^{-\frac{1}{2}} + 1 \right]^{-\frac{3}{2}}
$$

(B.6)

$$
= f(\mu, C_p, f, Re_D, \bar{h}, D)
$$

(B.7)

In this equation, $k_p$ is a function of $\mu$, $C_p$, $f$, $Re_D$, $\bar{h}$, and $D$. Therefore, the uncertainty of $k_p$ is the sum of squares of $U_{kp}/k_p$ and can be expressed as follows:

$$
\left( \frac{U_{kp}}{k_p} \right)^2 = \left( \frac{\partial k_p U_{\mu}}{\partial \mu} \right)^2 + \left( \frac{\partial k_p U_{C_p}}{\partial C_p} \right)^2 + \left( \frac{\partial k_p U_f}{\partial f} \right)^2 + 
$$
\[
\left( \frac{\partial k_p}{\partial \text{Re}_D} \frac{U \text{Re}_D}{k_p} \right)^2 + \left( \frac{\partial k_p}{\partial h} \frac{U_h}{k_p} \right)^2 + \left( \frac{\partial k_p}{\partial D} \frac{U_D}{k_p} \right)^2 = SS_{k_p} \quad (B.8)
\]

where each item in the above equation is derived as follows:

\[
\frac{\partial k_p}{\partial \mu} \frac{U_{\mu}}{k_p} = \frac{U_{\mu}}{\mu} - \frac{3}{2} \left[ \frac{(8f)^{1/2}}{101.6hD} - \frac{1.07 \left( \frac{f}{8} \right)^{-1/2} + 1}{12.7} \right]^{1/2} \left( \frac{f \mu C_p \text{Re}_D}{101.6hD (f/8)^{1/2}} \right) \quad (B.10)
\]

\[
\frac{\partial k_p}{\partial C_p} \frac{U_{C_p}}{k_p} = \frac{U_{C_p}}{C_p} - \frac{3}{2} \left[ \frac{(8f)^{1/2}}{101.6hD} - \frac{1.07 \left( \frac{f}{8} \right)^{-1/2} + 1}{12.7} \right]^{1/2} \left( \frac{f \mu C_p \text{Re}_D}{101.6hD (f/8)^{1/2}} \right) \quad (B.11)
\]

\[
\frac{\partial k_p}{\partial f} \frac{U_f}{k_p} = -\frac{3}{2} \left[ \frac{(8f)^{1/2}}{101.6hD} - \frac{1.07 \left( \frac{f}{8} \right)^{-1/2} + 1}{12.7} \right]^{1/2} \left( \frac{g^{1/2} f^{3/2} \mu C_p \text{Re}_D}{203.2hD} + \frac{g^{1/2} (1.07)^{1/2}}{25.4} \right) \left( \frac{f \mu C_p \text{Re}_D}{101.6hD (f/8)^{1/2}} \right) \quad (B.12)
\]
\[ \frac{\partial k_p}{\partial Re_D} \frac{U_{Re_D}}{k_p} = -\frac{3}{2} \left[ \frac{(8f)^{1/2}}{101.6hD^2} - \frac{1.07}{12.7} \left( \frac{f}{8} \right)^{-1/2} + 1 \right]^{-1} \]

\[ \left[ \frac{8^{1/2}f^{1/2}MCpRe_D}{101.6hD} \right] \frac{U_{Re_D}}{Re_D} \tag{B.13} \]

\[ \frac{\partial k_p}{\partial h} \frac{U_h}{k_p} = -\frac{3}{2} \left[ \frac{(8f)^{1/2}}{101.6hD^2} - \frac{1.07}{12.7} \left( \frac{f}{8} \right)^{-1/2} + 1 \right]^{-1} \]

\[ \left[ \frac{8^{1/2}f^{1/2}MCpRe_D}{101.6hD} \right] \frac{U_h}{h} \tag{B.14} \]

\[ \frac{\partial k_p}{\partial D} \frac{U_D}{k_p} = -\frac{3}{2} \left[ \frac{(8f)^{1/2}}{101.6hD^2} - \frac{1.07}{12.7} \left( \frac{f}{8} \right)^{-1/2} + 1 \right]^{-1} \]

\[ \left[ \frac{8^{1/2}f^{1/2}MCpRe_D}{101.6hD} \right] \frac{U_D}{D} \tag{B.15} \]

Then, \( U_{kp}/k_p \) is obtained by taking the square root of the sums of squares of each uncertainty source, \( SS_{kp} \). That is,

\[ \frac{U_{kp}}{k_p} = \left( SS_{kp} \right)^{1/2} \tag{B.16} \]
Uncertainty analysis of $k$ by the Gnielinski correlation

The thermal conductivity from the Gnielinski correlation is found to be:

$$k_g = \mu C_p \left[ \frac{\left(8f\right)^{\frac{1}{2}} \mu C_p \left(Re_D - 1000\right)}{101.6hD} - \frac{1}{12.7} \left(\frac{f}{8}\right)^{-\frac{1}{2}} + 1 \right]^{-\frac{3}{2}}$$  \hspace{1cm} (B.17)

$$= f(\mu, C_p, f, Re_D, \bar{h}, D)$$  \hspace{1cm} (B.18)

In this equation, again, $k_g$ is function of $\mu$, $C_p$, $f$, $Re_D$, $\bar{h}$, and $D$. Therefore, the uncertainty of the sum of the squares of $U_{k_g}/k_g$ can be expressed as follows:

$$\left(\frac{U_{k_g}}{k_g}\right)^2 = \left(\frac{\partial k_g}{\partial \mu} \frac{U_\mu}{k_g}\right)^2 + \left(\frac{\partial k_g}{\partial C_p} \frac{U_{C_p}}{k_g}\right)^2 + \left(\frac{\partial k_g}{\partial f} \frac{U_f}{k_g}\right)^2 +$$

$$\left(\frac{\partial k_g}{\partial Re_D} \frac{U_{Re_D}}{k_g}\right)^2 + \left(\frac{\partial k_g}{\partial \bar{h}} \frac{U_{\bar{h}}}{k_g}\right)^2 + \left(\frac{\partial k_g}{\partial D} \frac{U_D}{k_g}\right)^2$$  \hspace{1cm} (B.19)

$$= SS_{k_g}$$  \hspace{1cm} (B.20)

where each term in the above equation is derived as follows:

$$\frac{\partial k_g}{\partial \mu} \frac{U_\mu}{k_g} = \frac{U_\mu}{\mu} - 3 \frac{\left(8f\right)^{\frac{1}{2}} \mu C_p \left(Re_D - 1000\right)}{2 \cdot 101.6hD} - \frac{1}{12.7} \left(\frac{f}{8}\right)^{-\frac{1}{2}} + 1 \right]^{-1} \cdot \frac{f \mu C_p \left(Re_D - 1000\right)}{101.6hD(f/8)^{1/2}} \frac{U_\mu}{\mu}$$  \hspace{1cm} (B.21)

$$\frac{\partial k_g}{\partial C_p} \frac{U_{C_p}}{k_g} = \frac{U_{C_p}}{C_p} - 3 \frac{\left(8f\right)^{\frac{1}{2}} \mu C_p \left(Re_D - 1000\right)}{2 \cdot 101.6hD} - \frac{1}{12.7} \left(\frac{f}{8}\right)^{-\frac{1}{2}} + 1 \right]^{-1} \cdot \frac{f \mu C_p \left(Re_D - 1000\right)}{101.6hD(f/8)^{1/2}} \frac{U_{C_p}}{C_p}.$$
\[
\frac{\partial k_g U_f}{\partial f \ k_g} = -\frac{3}{2} \left[ \frac{(8f)^{1/2} \mu C_p \left( Re_D - 1000 \right)}{101.6 h D} - \frac{1}{12.7} \left( \frac{f}{8} \right)^{-1/2} + 1 \right]^{-1} \left( \frac{8^{1/2} f^{3/2} \mu C_p \left( Re_D - 1000 \right)}{203.2 h D} + \frac{8^{1/2} f^{-1/2}}{25.4} \right) \frac{U_f}{f} 
\]  
\[\text{(B.23)}\]

\[
\frac{\partial k_g U_{Re_D}}{\partial Re_D \ k_g} = -\frac{3}{2} \left[ \frac{(8f)^{1/2} \mu C_p \left( Re_D - 1000 \right)}{101.6 h D} - \frac{1}{12.7} \left( \frac{f}{8} \right)^{-1/2} + 1 \right]^{-1} \left[ \frac{8^{1/2} f^{1/2} \mu C_p Re_D}{101.6 h D} \right] \frac{U_{Re_D}}{Re_D} 
\]  
\[\text{(B.24)}\]

\[
\frac{\partial k_g U_h}{\partial h \ k_g} = -\frac{3}{2} \left[ \frac{(8f)^{1/2} \mu C_p \left( Re_D - 1000 \right)}{101.6 h D} - \frac{1}{12.7} \left( \frac{f}{8} \right)^{-1/2} + 1 \right]^{-1} \left[ -\frac{8^{1/2} f^{1/2} \mu C_p \left( Re_D - 1000 \right)}{101.6 h D} \right] \frac{U_h}{h} 
\]  
\[\text{(B.25)}\]

\[
\frac{\partial k_g U_D}{\partial D \ k_g} = -\frac{3}{2} \left[ \frac{(8f)^{1/2} \mu C_p \left( Re_D - 1000 \right)}{101.6 h D} - \frac{1}{12.7} \left( \frac{f}{8} \right)^{-1/2} + 1 \right]^{-1} \left[ -\right] 
\]
Then, \( U_{kg}/k_g \) is obtained by taking the square root of the sums of squares of each uncertainty source, \( SS_{kg} \). That is:

\[
\frac{U_{kg}}{k_g} = \left( SS_{kg} \right)^{\frac{1}{2}} \tag{B.27}
\]

**Uncertainty analysis for heat and flow measurements**

The uncertainty calculation in heat and flow measurements would also contribute to those uncertainty quantities in the thermal conductivity uncertainty mentioned in the previous three sections. In this section, these heat and flow uncertainties will be discussed. The uncertainties in heat transfer measurements include those from the heat transfer rate, \( \dot{Q} \), and heat transfer coefficient, \( h \), while the uncertainties in the fluid flow measurements include those from the calculations of the Reynolds number, \( Re_D \), and friction factor, \( f \).

**The uncertainty of \( \dot{Q} \):**

The net heat transfer rate is calculated from the following equation:

\[
\dot{Q} = hA(T_w - T_f) \tag{B.28}
\]
\[
= \dot{m}C_p(T_o - T_i) \tag{B.29}
\]
\[
= f(\dot{m}, C_p, T_o, T_i) \tag{B.30}
\]
Therefore, the uncertainty of the sum of the squares of \( U_\dot{Q}/\dot{Q} \) is given as:

\[
\left( \frac{U_\dot{Q}}{\dot{Q}} \right)^2 = \left( \frac{U_m}{m} \right)^2 + \left( \frac{U_{Cp}}{C_p} \right)^2 + \left( \frac{U_{T_0}}{T_0} \right)^2 + \left( \frac{U_{T_1}}{T_1} \right)^2
\]

(B.31)

\[
= SS_{\dot{Q}}
\]

(B.32)

The resulting \( U_\dot{Q}/\dot{Q} \) is given by:

\[
\frac{U_\dot{Q}}{\dot{Q}} = \left( SS_{\dot{Q}} \right)^{\frac{1}{2}}
\]

(B.33)

The uncertainty of \( h \):

The heat transfer coefficient is obtained from the following equation:

\[
\bar{h} = \frac{\dot{Q}}{A(T_w - T_f)}
\]

(B.34)

\[
= \frac{\dot{Q}}{\pi DL(T_w - T_f)}
\]

(B.35)

\[
= f(\dot{Q}, D, L, T_w, T_f)
\]

(B.36)

The uncertainty of the sum of the squares of \( U_{\bar{h}}/\bar{h} \) is given by:

\[
\left( \frac{U_{\bar{h}}}{\bar{h}} \right)^2 = \left( \frac{U_{\dot{Q}}}{\dot{Q}} \right)^2 + \left( -\frac{U_D}{D} \right)^2 + \left( -\frac{U_L}{L} \right)^2 + \left( -\frac{U_{T_w}}{T_w - T_f} \right)^2 + \left( -\frac{U_{T_f}}{T_w - T_f} \right)^2
\]

(B.37)

\[
= SS_{\bar{h}}
\]

(B.38)
Then, the uncertainty of $\frac{U_h}{h}$ is given by:

$$\frac{U_h}{h} = (SS_h)^{\frac{1}{2}}$$  \hspace{1cm} (B.39)

The uncertainty of fluid flow includes the calculation of the Reynolds number, $Re_D$, and the friction factor, $f$. They are discussed as follows:

**The uncertainty of $Re_D$:**

The Reynolds number is calculated from the following definition:

$$Re_D = \frac{4\bar{m}}{\pi D \mu}$$  \hspace{1cm} (B.40)

$$= f(\bar{m}, D, \mu)$$  \hspace{1cm} (B.41)

The uncertainty of the sum of the squares of $U_{Re_D}/Re_D$ is given by:

$$\left( \frac{U_{Re_D}}{Re_D} \right)^2 = \left( \frac{U_{\bar{m}}}{\bar{m}} \right)^2 + \left( \frac{U_D}{D} \right)^2 + \left( \frac{U_\mu}{\mu} \right)^2$$  \hspace{1cm} (B.42)

$$= SS_{Re_D}$$  \hspace{1cm} (B.43)

Therefore, the uncertainty of $U_{Re_D}/Re_D$ is given by:

$$\frac{U_{Re_D}}{Re_D} = \left( SS_{Re_D} \right)^{\frac{1}{2}}$$  \hspace{1cm} (B.44)

**The uncertainty of friction factor, $f$:**

The friction factor is calculated from the following curve fit equation:

$$f = \left( 0.79 \ln \left( Re_D \right) - 1.64 \right)^{-2}$$  \hspace{1cm} (B.45)

$$= f(Re_D)$$  \hspace{1cm} (B.46)

$$= f(Re_D)$$  \hspace{1cm} (B.47)
Therefore, the uncertainty of the sum of the squares of the friction factor, \( U_f/f \), is given by:

\[
\left( \frac{U_f}{f} \right)^2 = \left( \frac{\partial f}{\partial Re_D} \right)^2 \frac{U_{Re_D}}{f} \quad \text{(B.48)}
\]

\[
= SS_f \quad \text{(B.49)}
\]

Or, the resulting uncertainty of \( U_f/f \) is derived as:

\[
\frac{U_f}{f} = \frac{-1.58}{0.79 \ln(Re_D) - 1.64} \frac{U_{Re_D}}{Re_D} \quad \text{(B.50)}
\]

### Uncertainty analysis of Approach 2

Approach 2 for thermal conductivity is based in the Prandtl number, \( Pr \), measurement. The \( Pr \) is expressed in Equation 4.34, which means:

\[
Pr = f(D, L, \bar{Re}_D, \Delta T^*) \quad \text{(B.51)}
\]

Because \( Pr = \mu C_p/k \), \( k \) is a function of \( \mu, C_p \), and \( Pr \). In other words:

\[
k = \frac{\mu C_p}{Pr} = f(\mu, C_p, Pr) \quad \text{(B.52)}
\]

The uncertainty of \( k \) is expressed as follows:

\[
\frac{U_k}{k} = \left[ \left( \frac{\partial k}{\partial \mu} \frac{U_\mu}{k} \right)^2 + \left( \frac{\partial k}{\partial C_p} \frac{U_{C_p}}{k} \right)^2 + \left( \frac{\partial k}{\partial Pr} \frac{U_{Pr}}{k} \right)^2 \right]^{\frac{1}{2}} \quad \text{(B.53)}
\]

After taking the partial derivative of each term and rearranging, this equation can be written as:

\[
\frac{U_k}{k} = \left[ \left( \frac{U_\mu}{\mu} \right)^2 + \left( \frac{U_{C_p}}{C_p} \right)^2 + \left( -\frac{U_{Pr}}{Pr} \right)^2 \right]^{\frac{1}{2}} \quad \text{(B.54)}
\]
Uncertainty of Pr

The measured Pr is a function of D, L, \( \overline{Re}_D \), and \( \Delta T^* \). Therefore, the uncertainty of Pr is expressed as:

\[
\frac{U_{Pr}}{Pr} = \left[ \left( \frac{\partial Pr}{\partial D} \cdot \frac{U_D}{D} \right)^2 + \left( \frac{\partial Pr}{\partial L} \cdot \frac{U_L}{L} \right)^2 + \left( \frac{\partial Pr}{\partial Re_D} \cdot \frac{U_{Re_D}}{Re_D} \right)^2 + \left( \frac{\partial Pr}{\partial \Delta T^*} \cdot \frac{U_{\Delta T^*}}{\Delta T^*} \right) \right]^{\frac{1}{2}}
\]

(B.55)

where \( U_{Re_D}/Re_D \) is found from Equation B.42 and \( U_{\Delta T^*}/\Delta T^* \) is derived in the following section.

Uncertainty of \( \Delta T^* \)

The \( \Delta T^* \) is defined as \( (T_o - T_i)/(T_w - T_f) \). The uncertainty of \( \Delta T^* \) is expressed in the following equation:

\[
\frac{U_{\Delta T^*}}{\Delta T^*} = \left[ \left( \frac{U_{T_o}}{T_o - T_i} \right)^2 + \left( \frac{U_{T_i}}{T_o - T_i} \right)^2 + \left( \frac{U_{T_w}}{T_w - T_f} \right)^2 + \left( \frac{U_{T_f}}{T_w - T_f} \right) \right]^{\frac{1}{2}}
\]

(B.56)
APPENDIX C. SOME PHYSICAL PROPERTIES OF TEST REFRIGERANTS

Table C.1: Some physical properties of test refrigerants

<table>
<thead>
<tr>
<th>test refrigerants</th>
<th>molecular weight</th>
<th>acentric factor, ( \omega )</th>
<th>normal B.P. temp. °C</th>
<th>critical temp. °C</th>
<th>critical pressure, psia</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-12</td>
<td>120.91</td>
<td>0.1814</td>
<td>-29.76</td>
<td>111.80</td>
<td>606.26</td>
</tr>
<tr>
<td>R-22</td>
<td>86.47</td>
<td>0.2211</td>
<td>-40.86</td>
<td>96.15</td>
<td>733.02</td>
</tr>
<tr>
<td>R-32</td>
<td>52.02</td>
<td>0.2671</td>
<td>-51.75</td>
<td>78.21</td>
<td>839.92</td>
</tr>
<tr>
<td>R-113</td>
<td>187.38</td>
<td>0.2563</td>
<td>47.65</td>
<td>214.35</td>
<td>501.25</td>
</tr>
<tr>
<td>R-114</td>
<td>170.92</td>
<td>0.2511</td>
<td>3.65</td>
<td>145.65</td>
<td>471.08</td>
</tr>
<tr>
<td>R-125</td>
<td>120.03</td>
<td>0.2953</td>
<td>-48.57</td>
<td>66.18</td>
<td>526.34</td>
</tr>
<tr>
<td>R-134a</td>
<td>102.03</td>
<td>0.3235</td>
<td>-26.15</td>
<td>101.15</td>
<td>589.87</td>
</tr>
<tr>
<td>R-143a</td>
<td>84.04</td>
<td>0.2566</td>
<td>-47.35</td>
<td>73.10</td>
<td>552.74</td>
</tr>
<tr>
<td>R-152a</td>
<td>66.05</td>
<td>0.2573</td>
<td>-24.65</td>
<td>113.55</td>
<td>651.51</td>
</tr>
<tr>
<td>R-236ea</td>
<td>152.05</td>
<td>0.3860</td>
<td>6.50</td>
<td>139.29</td>
<td>512.42</td>
</tr>
<tr>
<td>blend A</td>
<td>86.19</td>
<td>0.2169</td>
<td>-45.00</td>
<td>86.80</td>
<td>664.46</td>
</tr>
<tr>
<td>blend B</td>
<td>97.60</td>
<td>0.2663</td>
<td>-46.57</td>
<td>72.71</td>
<td>541.28</td>
</tr>
</tbody>
</table>

In Table C.1, blend A (AC-9000) and blend B (HP-62) are two 3-component refrigerant mixtures. Their components and compositions (by mass) are listed in the following table, Table C.2. In Table C.1, the acentric factor is a component constant, which is defined as [50]:

\[ \omega = -\log P_{vpr} \text{(at } T_r = 0.7) - 1.000 \]  \hfill (C.1)
Table C.2: Components and compositions of test refrigerant mixtures

<table>
<thead>
<tr>
<th>mixture</th>
<th>R-32</th>
<th>R-125</th>
<th>R-134a</th>
<th>R-143a</th>
</tr>
</thead>
<tbody>
<tr>
<td>blend A</td>
<td>0.23</td>
<td>0.25</td>
<td>0.52</td>
<td>-</td>
</tr>
<tr>
<td>blend B</td>
<td>-</td>
<td>0.44</td>
<td>0.04</td>
<td>0.52</td>
</tr>
</tbody>
</table>

To obtain values of \( \omega \), the reduced vapor pressure \( (P_r = P/P_c) \) at \( T_r = T/T_c = 0.7 \) is required. One of the calculated methods was developed from Lee-Kesler vapor pressure relations, which was:

\[
\omega = \frac{\alpha}{\beta} \quad (C.2)
\]

\[
\alpha = -\ln(P_c) - 5.92714 + 6.09648\theta^{-1} + 1.28862\ln(\theta) - 0.169347\theta^6 \quad (C.3)
\]

\[
\beta = 15.2518 - 15.6875\theta^{-1} - 13.4721\ln(\theta) + 0.43577\theta^6 \quad (C.4)
\]

\[
\theta = \frac{T_b}{T_c} \quad (C.5)
\]

In \( \alpha \) equation, \( P_c \) must use the unit of atmospheric pressure (atm).
APPENDIX D. EQUIPMENT AND INSTRUMENTATION
SPECIFICATIONS

The manufacturer and model number of each component in the test apparatus is shown in Table D.1. Electronic instrumentations are listed in Table D.2.

Table D.1: Components of the test rig

<table>
<thead>
<tr>
<th>component</th>
<th>manufacturer</th>
<th>description of type</th>
<th>model</th>
</tr>
</thead>
<tbody>
<tr>
<td>refrigerant</td>
<td>Wanner Eng.</td>
<td>positive displ.</td>
<td>D-10</td>
</tr>
<tr>
<td>pump</td>
<td>GE</td>
<td>diaphragm</td>
<td>58PB56SAA42A</td>
</tr>
<tr>
<td>motor</td>
<td>GE</td>
<td>1hp, dc</td>
<td>58PB56SAA42A</td>
</tr>
<tr>
<td>motor controller</td>
<td>Dart Inc.</td>
<td>variable current</td>
<td>250G</td>
</tr>
<tr>
<td>filter drier</td>
<td>Sporlan</td>
<td>desiccant, 16in^3</td>
<td>C-164-S</td>
</tr>
<tr>
<td>auxiliary</td>
<td>Omega</td>
<td>heat tape, 627 Watts</td>
<td>STH051-080</td>
</tr>
<tr>
<td>heater</td>
<td>Omega</td>
<td>120V, 1/2 inch W x 8 ft. L</td>
<td>STH051-080</td>
</tr>
<tr>
<td>teat-section heater</td>
<td>Omega</td>
<td>heat tape, 2@627 Watts</td>
<td></td>
</tr>
<tr>
<td>mixer</td>
<td>Omega</td>
<td>120V, 1/2 inch W x 8 ft. L</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Omega</td>
<td>rugged static type</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>3/8 inch I.D., 3 inch long</td>
<td>FMX 7301</td>
</tr>
<tr>
<td>sight glasses</td>
<td>Sporlan</td>
<td>1/2 inch, solder fit</td>
<td></td>
</tr>
<tr>
<td>condenser</td>
<td>Doucete</td>
<td>coaxial coil</td>
<td>CX-H 050</td>
</tr>
<tr>
<td>accumulator</td>
<td>Oil-Air Ind.</td>
<td>Bladder (butyl, Buna-N)</td>
<td>1-100-1</td>
</tr>
<tr>
<td>charging valve</td>
<td>Henry Co.</td>
<td>packed angle</td>
<td>9271</td>
</tr>
<tr>
<td>pres. relief</td>
<td>Nupro Co.</td>
<td>spring set type</td>
<td>SS-4R3A-A</td>
</tr>
<tr>
<td>chiller</td>
<td>Heat-X Inc.</td>
<td>R-12, 3 HP</td>
<td>PC-300</td>
</tr>
<tr>
<td>line conditioner</td>
<td>Tripp Lite Co.</td>
<td>spike, surge suppressor</td>
<td>LC1800</td>
</tr>
<tr>
<td>oil-injection cyl.</td>
<td>Clippard Inc.</td>
<td>pneumatic cylinder type</td>
<td>UDR-SS-32-6</td>
</tr>
<tr>
<td>sample cylinder</td>
<td>swagelok Co.</td>
<td>closed-ends cylinder</td>
<td>304L-HDF4-75</td>
</tr>
</tbody>
</table>
Table D.2: Data acquisition and instrumentation components

<table>
<thead>
<tr>
<th>component</th>
<th>manufacturer</th>
<th>description of type</th>
<th>model</th>
</tr>
</thead>
<tbody>
<tr>
<td>computer</td>
<td>Zenith</td>
<td>386 PC</td>
<td>Z-386-20</td>
</tr>
<tr>
<td>multimeter</td>
<td>HP Inc.</td>
<td>digital multimeter</td>
<td>3457A</td>
</tr>
<tr>
<td>scanner</td>
<td>HP Inc.</td>
<td>digital control unit</td>
<td>3488A</td>
</tr>
<tr>
<td>meter board</td>
<td>HP Inc.</td>
<td>armature relay</td>
<td>44491A</td>
</tr>
<tr>
<td>scanner board</td>
<td>Hewlett-Packard</td>
<td>multiplexer, 10 channel</td>
<td>44470A</td>
</tr>
<tr>
<td>interface card</td>
<td>National Instr.</td>
<td>AT-GPIB (FORTRAN)</td>
<td>776207-01</td>
</tr>
<tr>
<td>thermocouple</td>
<td>Omega</td>
<td>bare bead, teflon shield</td>
<td>T-type</td>
</tr>
<tr>
<td>cold junction</td>
<td>Omega</td>
<td>electric ice point</td>
<td>CJ-T</td>
</tr>
<tr>
<td>RTD probe</td>
<td>Hy-Cal Eng.</td>
<td>100 Ohm platinum</td>
<td>RTS-36-T-100-A</td>
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<td></td>
<td></td>
<td>0.00385/Ω/°C</td>
<td>-5-2-36-X55</td>
</tr>
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<td>RTD transmitter</td>
<td>Hy-Cal Eng.</td>
<td>2-wire, 4-20 mA, 100 Ohm</td>
<td>CT-801-A-S-X5</td>
</tr>
<tr>
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<td>Jemtec Co.</td>
<td>2-element, 3-wire,</td>
<td>XL31K5A2</td>
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<td></td>
<td></td>
<td>3-phase, accuracy:±0.2%</td>
<td></td>
</tr>
<tr>
<td>mass flow meter</td>
<td>Micro Motion</td>
<td>0-1800 kg/hr, accuracy:±0.15%</td>
<td>RFT9739</td>
</tr>
<tr>
<td>densitometer</td>
<td>Micro Motion</td>
<td>0-1300 kg/m³, accuracy:±0.15%</td>
<td>RFT9739</td>
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<tr>
<td>viscometer</td>
<td>Nametre Co.</td>
<td>0.1-500 cp × g/cm³</td>
<td>1810-LV</td>
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<tr>
<td></td>
<td></td>
<td>4-20 mA, accuracy:±2%</td>
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<tr>
<td>abs. pressure</td>
<td>Setra Co.</td>
<td>0-250 psia, ±0.11%FS</td>
<td>C280E</td>
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<tr>
<td>diff. pressure</td>
<td>Setra Co.</td>
<td>0-1 psid, ±0.15%FS</td>
<td>C228-1</td>
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<td>voltage regulator</td>
<td>Staco Energy Co.</td>
<td>120 V, 1.4 KVA</td>
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APPENDIX E. SAS REGRESSION OUTPUTS

This appendix lists the output of SAS for three correlation regressions in the Nusselt number method (Approach 1) and two variables ($Re_D$ and $\Delta T^*$) regression in the Prandtl number method (Approach 2) for three calibration refrigerants (R-22, R-12, and R-113) and four calibration refrigerants (R-22, R-12, R-113, and R-114). The SAS program (version 6.07) was accessed from the campus vincent computer work station network. The first section in this appendix is the SAS output for the three correlations discussed in Approach 1, whereas the second section is the SAS output for the $ln(Pr)$ regression with variables of $ln(Re_D)$ and $\Delta T^*$ discussed in Approach 2. Both sections contain three and four calibration refrigerant cases.
Nusselt number regression in Approach 1

Three calibration refrigerants

The SAS System
22:35 Saturday, September 10, 1994

Model: MODEL1
Dependent Variable: NUE

Analysis of Variance

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<td>C Total</td>
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<td>31.29279</td>
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Root MSE 0.04914  R-square 0.9907
Dep Mean 5.53266  Adj R-sq 0.9906
C.V. 0.88821

Parameter Estimates

| Variable | DF | Parameter Estimate | Standard Error | T for H0: Parameter=0 | Prob > |T| |
|----------|----|--------------------|----------------|-----------------------|--------|---|
| INTERCEP | 1  | -4.997045          | 0.10783630     | -46.339               | 0.0001 |
| LNRE     | 1  | 0.903770           | 0.00876542     | 103.106               | 0.0001 |
| LNPR     | 1  | 0.502199           | 0.01131934     | 44.366                | 0.0001 |
The SAS System
22:35 Saturday, September 10, 1994

Model: MODEL2
Dependent Variable: NUD

Analysis of Variance

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<td>25.54522</td>
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Root MSE 0.00000  R-square 1.0000  Dep Mean 5.48384  Adj R-sq 1.0000  C.V. 0.00000

Parameter Estimates

| Variable  | DF | Parameter Estimate | Standard Error | T for H0: Parameter=0 | Prob > |T| |
|-----------|----|--------------------|----------------|-----------------------|--------|
| INTERCEP  | 1  | -3.772261          | 0.0000000000   | .                     | .      |
| LNRE      | 1  | 0.800000000       | 0.0000000000   | .                     | .      |
| LNPR      | 1  | 0.400000000       | 0.0000000000   | .                     | .      |
Model: MODEL3
Dependent Variable: NUP

Analysis of Variance

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Root MSE 0.00829  R-square 0.9997
Dep Mean 5.57329  Adj R-sq 0.9997
C.V. 0.14873

Parameter Estimates

| Variable  | DF | Parameter Estimate | Standard Error | T for H0: Parameter=0 | Prob > |T| |
|-----------|----|--------------------|----------------|------------------------|--------|---|
| INTERCEP  | 1  | -4.059037          | 0.01819021     | -223.144               | 0.0001 |
| LNRE      | 1  | 0.827298           | 0.00147858     | 559.521                | 0.0001 |
| LNPR      | 1  | 0.455283           | 0.00190939     | 238.445                | 0.0001 |
Model: MODEL4
Dependent Variable: NUG

Analysis of Variance

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Root MSE: 0.01685
Dep Mean: 5.58170
C.V.: 0.30185
R-square: 0.9988
Adj R-sq: 0.9988

Parameter Estimates

| Variable | DF | Parameter Estimate | Standard Error | T for H0: Parameter=0 | Prob > |T| |
|----------|----|--------------------|----------------|-----------------------|--------|
| INTERCEP | 1  | -4.415096          | 0.03697204     | -119.417              | 0.0001 |
| LNRE     | 1  | 0.862783           | 0.00300526     | 287.091               | 0.0001 |
| LNPR     | 1  | 0.441244           | 0.00388087     | 113.697               | 0.0001 |
Four calibration refrigerants

The SAS System 1
23:37 Saturday, September 10, 1994

Model: MODEL1
Dependent Variable: NUE

Analysis of Variance

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Root MSE  0.04604  R-square  0.9919
Dep Mean  5.48795  Adj R-sq  0.9918
C.V.      0.83884

Parameter Estimates

| Variable | DF | Parameter Estimate | Standard Error | T for H0: Parameter=0 | Prob > |T| |
|----------|----|--------------------|----------------|-----------------------|--------|---|
| INTERCEP | 1  | -5.015673          | 0.08491757     | -59.065               | 0.0001 |
| LNRE     | 1  | 0.905027           | 0.00687972     | 131.550               | 0.0001 |
| LNPR     | 1  | 0.507640           | 0.00970171     | 52.325                | 0.0001 |
Model: MODEL2  
Dependent Variable: NUD

Analysis of Variance

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Root MSE 0.00000  
Dep Mean 5.43887  
C.V. 0.00000

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Model: MODEL3  
Dependent Variable: NUP

### Analysis of Variance

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Root MSE: 0.01037  
Dep Mean: 5.53387  
C.V.: 0.18737

### Parameter Estimates

| Variable | DF | Parameter Estimate | Standard Error | T for H0: Parameter=0 | Prob > |T| |
|----------|----|--------------------|----------------|------------------------|--------|
| INTERCEP | 1  | -4.052869          | 0.01912666     | -211.896               | 0.0001 |
| LNRE     | 1  | 0.826843           | 0.00154957     | 533.594                | 0.0001 |
| LNPR     | 1  | 0.457499           | 0.00218519     | 209.363                | 0.0001 |
Analysis of Variance

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Root MSE 0.01720  R-square 0.9988
Dep Mean 5.53955  Adj R-sq 0.9988
C.V. 0.31045

Parameter Estimates

| Variable | DF | Parameter | Standard Error | T for H0: Parameter=0 | Prob > |T| |
|----------|----|-----------|----------------|-----------------------|--------|---|
| INTERCEP | 1  | -4.391624 | 0.03172280     | -138.437              | 0.0001 |
| LNRE     | 1  | 0.860922  | 0.00257007     | 334.980               | 0.0001 |
| LNPR     | 1  | 0.442776  | 0.00362428     | 122.169               | 0.0001 |
Prandtl number regression in Approach 2

Three calibration refrigerants

The SAS System 12:56 Sunday, August 21,

Model: MODEL1
Dependent Variable: PR

Analysis of Variance

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<td>41.37167</td>
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Root MSE 0.06843 R-square 0.9872
Dep Mean 1.45033 Adj R-sq 0.9862
C.V. 4.71827

Parameter Estimates

| Variable | DF | Parameter Estimate | Standard Error | T for HO: Parameter=0 | Prob>|T| |
|----------|----|--------------------|----------------|-----------------------|-------|
| INTERCEP | 1  | 96.908792          | 38.77678932    | 2.499                 | 0.0139 |
| X        | 1  | -26.452258         | 10.79544008    | -2.450                | 0.0158 |
| Y        | 1  | 83.304684          | 26.64656684    | 3.126                 | 0.0023 |
| X2       | 1  | 2.459228           | 0.99959736     | 2.460                 | 0.0154 |
| XY       | 1  | -16.020859         | 4.93940149     | -3.243                | 0.0016 |
| Y2       | 1  | 8.088965           | 7.10238635     | 1.139                 | 0.2572 |
| X3       | 1  | -0.076756          | 0.03078419     | -2.493                | 0.0141 |
| X2Y      | 1  | 0.750949           | 0.22852190     | 3.286                 | 0.0014 |
| XY2      | 1  | -0.668378          | 0.65967019     | -1.013                | 0.3131 |
| Y3       | 1  | 2.213377           | 1.01672374     | 2.177                 | 0.0316 |
Four calibration refrigerants

The SAS System

23:58 Saturday, September 10, 1994

Model: MODEL1
Dependent Variable: PR

Analysis of Variance

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Root MSE: 0.06845  R-square: 0.9829
Dep Mean: 1.50972  Adj R-sq: 0.9819
C.V.: 4.53416

Parameter Estimates

| Variable | DF | Parameter Estimate | Standard Error | T for HO: Parameter=0 | Prob > |T| |
|----------|----|--------------------|----------------|-----------------------|--------|---|
| INTERCEP | 1  | 132.495075         | 34.02680322    | 3.894                 | 0.0001 |
| X        | 1  | -36.446721         | 9.44835075     | -3.857                | 0.0002 |
| Y        | 1  | 102.201609         | 23.81121972    | 4.292                 | 0.0001 |
| X2       | 1  | 3.392931           | 0.87286660     | 3.887                 | 0.0001 |
| XY       | 1  | -19.449363         | 4.40931003     | -4.411                | 0.0001 |
| Y2       | 1  | 14.165714          | 6.12753638     | 2.312                 | 0.0221 |
| X3       | 1  | -0.105753          | 0.02682882     | -3.942                | 0.0001 |
| X2Y      | 1  | 0.905506           | 0.20379577     | 4.443                 | 0.0001 |
| XY2      | 1  | -1.253316          | 0.57134480     | -2.194                | 0.0297 |
| Y3       | 1  | 2.823725           | 0.81305846     | 3.473                 | 0.0007 |
APPENDIX F. DATA ANALYSIS PROGRAM

This appendix contains a copy of the FORTRAN program used to reduce the raw data from transport property tests. This program opened the raw data files which were collected from the data acquisition system used to calculate all the property and heat transfer data. Two approaches, the Nusselt Number Method and Prandtl Number Method, were also used to analyze the thermal conductivity in this program. All the equations referred to in this program were mentioned and discussed in the previous chapters.

Properties needed are based on the ASHRAE Handbook or REFPROP (version 4.0), which were mentioned in the Appendix A, and written in subroutine form. Calculated or measured properties in this program include thermal conductivity, viscosity, specific heat, and density.
program analysis
implicit real*8(a-h,o-z)
common/acp/acp0,acp1,acp2,acp3,acp4,acp5
common/av/av0,av1,av2,av3,av4,av5
common/ak/ak0,ak1,ak2,ak3,ak4,ak5
common/gpda/ga,pa,da
common/gpdb/gb,pb,db
common/gpdc/gc,pc,dc
common/cc/ccs,ccg,ccp,ccd
dimension tw(15),ttw(13),two(13),twi(13)
real*8 kt,ktt,nu
data di,d1,pai/0.009525,2.0,3.141592654/

c   toamp=(ti+81.4854431)/1.00110283e+04 !to amp(current)
c   toamp=(ti+81.4854431)/1.00110283e+04

c......open output files
   open(unit=9,file='pr.dat',status='unknown')
   open(unit=10,file='lognu.dat',status='unknown')
   open(unit=11,file='ynu.dat',status='unknown')
   open(unit=12,file='cfh.h',status='unknown')
   open(unit=13,file='yy1u.dat',status='unknown')
   open(unit=14,file='raw1.dat',status='unknown')
   open(unit=15,file='raw2.dat',status='unknown')
   open(unit=16,file='refined.dat',status='unknown')
   open(unit=17,file='sf.dat',status='unknown')
   open(unit=18,file='loss.dat',status='unknown')
   open(unit=19,file='cp.dat',status='unknown')
   open(unit=21,file='ycf.dat',status='unknown')
   open(unit=8,file='ypgnu.dat',status='unknown')
   open(unit=7,file='modified.k.dat',status='unknown')
   open(unit=23,file='new.k.dat',status='unknown')
   open(unit=22,file='hk.dat',status='unknown')
   open(unit=24,file='viscp.dat',status='unknown')

C------------------------------------------------------------------------------
C......Cp, viscosity, K curve fitting coefficient from ASHRAE or REFPROP
C......nr=1 for r22; nr=2 for r12; nr=3 for r113; nr=4 for r114;
C......nr=5 for r236ea(REFPROP);
C------------------------------------------------------------------------------
write(*,*),'input refrigerant number, nr=?',
&'nr=1(r22); nr=2(r12); nr=3(r113); nr=4(r114);',
&'nr=5(r236ea); nr=6(90% E.G.); nr=7(80% E.G.);',
&'nr=8(70% E.G.); nr=9(60% E.G.); nr=10(50% E.G.)'
read(*,*),nr0
write(*,*)'input ending refrigerant number, nend=?'
read(*,*)nend
nr=nr0
write(*,*)'input number of data set, NTDATA=?'
read(*,*)ntdata
ndata=1
90 if(nr.eq.1) then
         r22 transport properties data
         r22 Cp, viscosity, K curve fitting coefficient from ASHRAE
         1  tc=96.15+273.33
            tb=-40.86+273.33
            pc=733.02
            wm=86.47
            w=0.2211
            call prop(nr) !call prop(nr) to properties coefficients
            open(unit=5,file='r22.dat',status='old')
            go to 50
   c
         r12 transport properties data
         r12 Cp, viscosity, K curve fitting coefficient from ASHRAE
         else if(nr.eq.2) then
         2  tc=111.8+273.33
            tb=-29.76+273.33
            pc=606.26
            wm=120.91
            w=0.1814
            call prop(nr) !call prop(nr) to properties coefficients
            open(unit=5,file='r12.dat',status='old')
            go to 50
   c
         r113 transport properties data
         r113 Cp, viscosity, K curve fitting coefficient from ASHRAE
         else if(nr.eq.3) then
         3  tc=214.35+273.33
            tb=47.65+273.33
            pc=187.38
            wm=0.2563
            w=0.2563
            call prop(nr) !call prop(nr) to properties coefficients
            open(unit=5,file='r113.dat',status='old')
            go to 50
246

c......r114 transport properties data
c......r114 Cp, viscosity, K curve fitting coefficient from ASHRAE
c
   else if(nr.eq.4)then
4  tc=145.65+273.33
   tb=3.65+273.33
   pc=471.08
   wm=170.92
   w=0.2611
   call prop(nr) !call prop(nr) to properties coefficients
   open(unit=5,file='r114.dat',status='old')
go to 50

c......r236ea transport properties data
c......r236ea Cp, viscosity, K curve fitting coefficient from REFPROP-4.0
c
   else if(nr.eq.5)then
5  tc=139.29+273.33
   tb=6.5+273.33
   pc=512.42
   wm=152.05
   w=0.3860
   call prop(nr) !call prop(nr) to properties coefficients
   open(unit=5,file='r236ea.dat',status='old')
   open(unit=5,file='/home/yuan/viscometer/r236eat1.dat',
   &status='old')
go to 50
end if

c-----------------------------

c......output head lines
50 write(*,*)'Tave(C) mass(kg/s) heat(W) h(W/m^2.C) Nu'
write(*,*)' T(C) Kg(W/m.C) Kp(W/m.C) Kd(W/m.C) Kt(W/m.C) E(\% &)-G E(\%)-P E(\%)-D'
write(*,*)'--------------------------------------'
write(*,*)
area=pi*di*dl
nexp=20
do 1000 i=ndata,ntdata

c......reading data over nexp(=20) times
do 200 k=1,nexp
read(5,*,end=1001)(tw(j),j=1,15)
read(5,*,end=1001)power,ti,to,td,pi,den,tsen,rmass

rtd anti-calibration for checking accuracy purpose

tiamp=(ti+81.2238617)/1.00184971e+04

ti=-80.0+(1.0e+04)*tiamp

toamp=(ti+81.4854431)/1.00110283e+04

to=-80.0+(1.0e+04)*toamp
tf=0.5*(ti+to)

...fluid transport properties at Tsat: Cp, vis, and K
...(ASHRAE data from curve fitting)
...coefficients obtained from calling prop(nr) previously

c
fcp=acp0+acpl*tf+acp2*(tf**2)+acp3*(tf**3)+acp4*(tf**4)+
&acp5*(tf**5) !Cp
fvis=av0+avl*tf+av2*(tf**2)+av3*(tf**3)+av4*(tf**4)+
&av5*(tf**5) !viscosity
fkt=ak0+akl*tf+ak2*(tf**2)+ak3*(tf**3)+ak4*(tf**4)+
&ak5*(tf**5) !thermal conductivity

...net heat input to the test-section
heat=rmass*fcp*(to-ti) !real heat input to the test-section
hflux=heat/area !heat flux of the test-section

...inner wall temperature correction by heat conduction through wall

r1=(3.0/16.0)*0.0254 !tube inside diameter
r2=(4.0/16.0)*0.0254 !tube outside diameter
twc=hflux*r1*log(r2/r1)/401.0 !wall temp. correction(wall conduction)
do 300 n=1,13
twi(n)=tw(n)-twc !inner wall temp.
300 continue

twi(10)=0.5*(twi(9)+twi(11)) !bad twi(10), so use 9,11 for average

call dtwf(twi,ti,to,dt) !call dtwf to get dt
h=heat/(area*dt) !heat transfer coefficient

...total quantities for wall temp., fluid, power, insulation surface,
...and air temperature

ht=ht+h
tti=tti+ti
ttw(n) = ttw(n) + tw(n)   !each outer wall temp. total quantities
180 continue

ttwa = ttwa + twa   !twa=average wall temp.
tfa = tfa + tf   !tfa=average fluid temp.
ttw(14) = ttw(14) + tw(14)   !tw(14)=insulation surface temp.
ttw(15) = ttw(15) + tw(15)   !tw(15)=ambient temp.
t=tt+t   !t=viscometer temperature (C)
pi = tpi + pi   !pi=inlet pressure (psia)
tpower = tpower + power   !power=power input (I*V)
tvismm = tvismm + vismm   !vismm=viscosity measured by viscometer (cp)
trmass = trmass + rmass   !rmass=mass flow rate
tden = tden + den   !den=density
cpt = cpt + cp   !cp=specific heat capacity
vist = vist + vis   !vis=viscosity
ktt = ktt + kt   !kt=thermal conductivity

200 continue

.....average quantities for nexp (=20) times

do 190 n=1,13
two(n) = ttw(n) / nexp   !each average outer wall temp. of nexp times
	wi(n) = twi(n) - twc
190 continue

tti = tti / nexp   !average inlet temp.
tto = to / nexp   !average outlet temp.
twa = twa / nexp   !average wall temp.
tfa = tfa / nexp   !average fluid temp.
t = tt / nexp   !average sample cell temperature (viscometer)
pi = tpi / nexp   !average inlet pressure
tsave = tts / nexp   !average insulation surface temp.
tave = tta / nexp   !average ambient temp.
tpower = tpower / nexp   !average power input
tvismm = tvismm / nexp   !average viscosity (cp) measured
trmass = trmass / nexp   !average mass flow rate
tden = tden / nexp   !average density
have = ht / nexp   !average heat transfer coefficient
cpave = cpt / nexp   !average cp
cvisave = vist / nexp   !average viscosity
cktave = ktt / nexp   !average thermal conductivity
c...Psat
  if(nr.eq.1)then
    w=0.2211
    pc=733.02
    psat=((5.00004649E-01)+(1.61909014E-02)*tfa+
    & (1.95017230E-04)*tfa**2+
    & (1.03241905E-06)*tfa**3+(2.23205832E-09)*tfa**4+
    & (1.71046532E-12)*tfa**5)*1.0E+06*1.45038E-04
  else if(nr.eq.2)then
    w=0.1814
    pc=606.26
    psat=((2.91020423E-01)+(1.00844698E-02)*tfa+
    & (1.49919608E-04)*tfa**2+
    & (5.64050538E-07)*tfa**3+(-5.91876770E-09)*tfa**4+
    & (5.7994885E-11)*tfa**5)*1.0E+06*1.45038E-04
  else if(nr.eq.3)then
    w=0.2563
    pc=501.25
    psat=((1.58554465E-02)+(6.77690492E-04)*tfa+
    & (1.25848301E-05)*tfa**2+
    & (2.28212969E-07)*tfa**3+(-1.47080736E-10)*tfa**4+
    & (1.6066746E-12)*tfa**5)*1.0E+06*1.45038E-04
  else if(nr.eq.4)then
    w=0.2511
    pc=471.08
    psat=((8.76755517E-02)+(3.47902835E-03)*tfa+
    & (5.4587973E-05)*tfa**2+
    & (3.74883115E-07)*tfa**3+(4.43446752E-10)*tfa**4+
    & (1.88150016E-12)*tfa**5)*1.0E+06*1.45038E-04
  else if(nr.eq.5)then
    w=0.3860
    pc=512.42
    psat=(1.13349352E+01)+(4.84870791E-01)*tfa+
    & (8.30436591E-03)*tfa**2+
    & (6.78510114E-05)*tfa**3+(2.04624143E-07)*tfa**4+
    & (-2.19035401E-10)*tfa**5
  end if

c...fluid saturated transport properties based on temperature at tfa:Cp, vis, and K

c...(ASHRAE data from curve fitting)
c...coefficients obtained from calling prop(nr) previously

c
  cp=acp0+acp1*tfa+acp2*(tfa**2)+acp3*(tfa**3)+acp4*(tfa**4)+
&acp5*(tfa**5) !Cp
vis=av0+av1*tfa+av2*(tfa**2)+av3*(tfa**3)+av4*(tfa**4)+
&av5*(tfa**5) !viscosity
kt=ak0+ak1*tfa+ak2*(tfa**2)+ak3*(tfa**3)+ak4*(tfa**4)+
&ak5*(tfa**5) !thermal conductivity

c...non-dimensionized temperature and pressure
c
tr=(tfa+273.33)/tc !tc is critical Temp.
ppr=pi/pc !pc is critical pressure
dpr=(pi-psat)/pc !psat is saturated pressure
call kfactor(tr,ppr,ratiok)!k midified factor based on current pressure pi
call vfactor(w,tr,dpr,ratiov)!viscosity modified factor based on pi
write(*,*)'ratiok='ratiok,'ratiov='ratiov
vis=vis*ratiov !modifying viscosity from saturated state to pi state
viscp=vis*cp
kt=kt*ratiok !modifying k from saturated state to pi state
call dtwf(twi,tiave,toave,dt)!call dtwf to get dt
dtn=(toave-tiave)/dt !non-dimensionized dt
twave=tfa+dt
h=rmass*cp*(toave-tiave)/(area*dt)!heat transfer coefficient
c......write aveaged data
write(14,3000)i,tfa,rmass,tiave,toave,twave,dt,taave,tsave,powera
3000 format(lx,i3,lx,f6.3,lx,f8.5,lx,6(f7.3,lx),fl0.4)
c
c......heat loss calculations and calibrations
c
tak=taave+273.33
call airprop(tak,dena,cpa,visa,tka)!call airprop to get prop.
c
c......heat loss calculations
c
pra=visa*cpa/tka !air Prantdl number
beta=1.0/taave !air expansion coefficent
g=9.8 !gravitational accleration
dvisa=visa/dena !air kinematic viscosity
alpha=tka/(dena*cpa) !air thermal diffusivity
dia=2.5*2.54/100.0 !outer insulation diameter
aco=pai*dia*dl !outer insulation surface area
dtsa=tsave-taave !temp. difference between surf/air
dtwave=wave-taave !temp. difference between wall/air
rdtswa=dtsa/dtwave !temp. diff. ratio between surf-wall-air
rada = (g * beta * dtsa * dia**3.0) / (dvisa * alpha)  ! Ra number

c

c..... churchill free convection's Nusselt number calculations

c

heat = rmass * cp * (toave - tiave)  ! real heat input to the test-section
hflux = heat / area  ! heat flux of the test-section
prn = (1.0 + ((0.559 / pra)**(9.0 / 16.0)))**27.0
anuic = (0.6 + (0.387 * (rada**(1.0 / 6.0)) / prn))**2.0  ! air Nu (unmodify)
yanui = log(anuic)  ! log(anuic)
anu = 1.2 * anuic  ! modified Nu number, update 1.2
ha = anu * tka / dia  ! air heat loss coefficient, ha
qloss = ha * ao * (tsave - taave)  ! heat loss calculated from C.C. eq.
qlossx = powera - heat  ! true heat loss
hac = qlossx / ao / (tsave - taave)  ! true ha
anuc = hac * dia / tka  ! true Nuair
yanuc = log(anuc)  ! log(anuc)
cf = qlossx / qloss  ! heat loss ratio
ylncf = log(cf)  ! log of heat loss ratio
xlnra = log(rada)  ! log of Ra
write(18, 2250) log(dtsa), log(qlossx)  ! write to loss.dat

2250 format (lx, 3(fl5.6, 2x))

2260 format (lx, fl0.5, 2x, 2fl2.6, 2x, f7.3)

net heat input

c

qloss = (exp(0.612001419)) * (dtsa)**1.33022535  ! heat loss prediction
qnet = powera - qloss  ! net heat to the test-section
qlratio = (qloss / powera) * 100.0  ! heat loss percentage ratio
 CPC = qnet / (rmass * (toave - tiave))  ! calculated Cp
rcp = ((cp / cp)) / cp) * 100.0  ! Cp deviation percentage
write(19, 2260) tfa, cp, CPC, RCP  ! write to cp.dat

2260 format (lx, f10.5, 2x, f12.6, 2x, f7.3)

Re and Pr number

c

re = (4.0 * rmass) / (pai * di * vis)  ! Reynolds numbers
xre = log(re)  ! logarithm Re transformation
pr = vis * cp / kt  ! fluid Prandtl number
ec = rmass**2 / (den**2 * di**2 * cp * dt)
write (24, 1013) tfa, kt, rmass, vis, cp
write (23, *) log(pr), log(re), log(dt)
c write (23, 1013) kt, rmass, vis*1.0E+03, cp/1000.0, dt

c write (*, *) 'vis=', vis, 'cp=', cp, 'rmass=', rmass
call newk(pr, re, dt)
c
if (nr.eq.5) then
hk=vis*cpc/pr
else
hk=vis*cp/pr
end if
hk=hk/ratiok
rhk=((hk-kt)/kt)*100.0
write(22,1014)tfa,hk,kt,rhk
1014 format(1x,f9.6,2(f12.6,lx),f10.6)
c
......parameter logarithm transformation for corrected Nusselt number
c
pr=vis*cp/kt !fluid Prantdl number
xpr=log(pr) !log Prandtl
nu=h*di/kt !corrected Nu number
ynu=log(nu) !log Nu
yynu=log(nu/(pr**0.507056))
write(15,3001)i,tfa,tr,pr,cp
3001 format(1x,i3,1x,2(f6.3,lx),lx,f10.2,1x,f8.4,1x,f10.5)
c
......correlation Nusselt number calibration functions
.....call subroutine cnucf(re,gcf,pcf,cf)
c
call cfnu(pr,re,enu, cfg, cf, cfh)
c
c......modified h values from enu(curve fitting equation) values
c
hstar=(enu*kt)/di
cfh=hstar/h
yhhstar=log(hstar)
yh=log(h)
h1=h*cfh
yyh=log(h/pr**(-0.477820))
yyhhstar=log(hstar/pr**(-0.477820))
c
c......Dittus Boelter correlation
c
c h=have
a1=0.905027
b1=0.507640
c1=exp(-5.015673)
a2=0.800013840
b2 = 0.4

c2 = \exp(-3.772261)

em1 = a1 - a2 + 0.8

em2 = b1 - b2 + 0.4

en = 1.0 / (0.6 - b1 + b2)

dks = ((c2*h*di) / (0.023*re**em1*(vis*cp)**em2))**en

dks = dks / ratiok

dnustar = h*di / kt

dnu = 0.023*re**0.8*pr**0.4  ! Dittus-Boelter eq. Nu

ydnu = log(dnu)  ! \ln(Nud)

ycdf = ynu - ydnu

dcf = nnu / dnu

dk = ((h*di) / (0.023*re**0.8*(vis*cp)**0.4))**(5.0/3.0)

dk = dk / ratiok

1980 format (ix, f9.5, 1x, f9.5, 1x, f12.6, 1x)

prd = vis*cp / dk

c prstar = 7.413721086 - 2 + 9.85503972e-1 * prd

c dk = vis*cp / prstar

rdk = ((dks - kt) / kt) * 100.0  ! deviation ratio

if (abs(rdk) .le. 5.0) then
     write (16, 1800) powera, rmass, den,
     & tfa, tave, toave, tave, dt, taave, tsave
else if (abs(rdk) .gt. 5.0) then
     go to 800
end if

800 write (*, *) re, pr, xre, xpr, nu, dnu

write (10, 2010) re, pr, xre, xpr, nu, dnu

2010 format (ix, f15.6, 1x, f8.4, 2x, 2(f8.3, 2x), f8.4, 2x, 2(f8.3, 2x))

write (*, 900) tr, tfa, pr, kt, dk, rdk

write (9, 900) tr, tfa, pr, kt, dk, rdk, dens, den  ! write to pr.dat

900 format (ix, f5.2, 2x, f5.2, 2x, f8.4, 2x, 2(f8.3, 2x))

c........Petukhov Popov correlation

f = (1.82*log10(re) - 1.64)**-2

pu1 = (f/8) * re * pr

pu2 = 1.07 + 12.7*((f/8)**(0.5))*(pr**2.3 - 1.0)

pu = pu1 / pu2

ypnu = log(pnu)

ypcf = ynu - ypnu

pcf = nu / pnu

cstar = cl

astar = a1
cstar=b1
cpe=exp(-4.052869)
apc=0.826843
bpe=0.457499
proold=pr

850 fpi=(cstar*(f/8.)**0.5*vis*cp*re**(astar-ape+1))/
(12.7*cpe*h*di)
fp2=(1.07/(12.7*(f/8.)**0.5))
fp=prpold**(2./3.)-fp1*prpold**(bstar-bpe)+fp2-1
fpp=(2./3.)*prpold**(-1./3.)-(bstar-bpe)*fp1*
&prpold**(bstar-bpe-1)
prpnew=prpold-(fp/fpp)
epsp=prpnew-prpold
if(abs(epsp).le.1.0e-12)go to 920
prpold=prpnew
go to 850

920 pk=vis*cp/prpnew
pk=pk/ratiok
rpk=((pk-kt)/kt)*100.0

......Gnielinski correlation
fg=(0.79*log(re)-1.64)**-2
gnu=(fg/8.)*(re-1000.0)*pr
gnu2=1.0+12.7*((fg/8)**0.5)*pr**(2./3.-1.0)
gnu=gnu1/gnu2 !Gnielinski Nu
gkc=vis*cp*((gnu1/(12.7*(fg/8)**(0.5)*h*di))- &((1/(12.7*(fg/8)**0.5))+1)**(-1.5)
ygnu=log(gnu) !ln(Nug)
ygcf=ynu-ygnu
gcf=gnu/ygnu
cge=exp(-4.391624)
age=0.860922
bge=0.442776
prgold=pr

860 fg1=(cstar*(f/8.)**0.5*vis*cp*(re-1000.0)**(astar-age+1))/
(12.7*cge*h*di)
fg2=(1.0/(12.7*(f/8.)**0.5))
ffg=prgold**(2./3.-fg1*prgold**bstar-bge)+fg2-1
ffgp=(2./3.)*prgold**(-1./3.-bstar-bge)*fg1*prgold**
&bstar-bge-1)
prgnew=prgold-(ffg/ffgp)
epsg=prgnew-prgold
if(abs(epsg).le.1.0e-12)go to 930
Nusselt number curve fittings

n for Pr'n in 3 and 4 calibration fluids (R22, R12, R113, add R114) are:

- \( \text{expriment:} 0.498730; 0.504359 \)
- \( \text{Dittus:} 0.4; 0.4 \)
- \( \text{Petukhov:} 0.455420; 0.457575 \)
- \( \text{Gnielinski:} 0.441340; 0.442798 \)

\[
yenu = \log\left(\frac{\nu}{pr^{0.523086}}\right)
\]
\[
ydnu = \log\left(\frac{dnu}{pr^{0.4}}\right)
\]
\[
ypnu = \log\left(\frac{pnu}{pr^{0.457575}}\right)
\]
\[
ygnu = \log\left(\frac{gnu}{pr^{0.442798}}\right)
\]

!write to ynu.dat

\[ 2600 \text{ format}(2x,5(f12.8,2x)) \]

!total quantity of each wall temp.

700 do 701 n=1,13

ttw(n)=0.0

701 continue

ht=0.0  !total quantity of heat transfer coef.
tti=0.0  !total quantity of inlet temp.
tto=0.0  !total quantity of outlet temp.
ttwa=0.0  !total quantity of average wall temp.
ttfa=0.0  !total quantity of average fluid temp.
tts=0.0  !total quantity of insulation surface temp.
tta=0.0  !total quantity of ambient air temp.

!total quantity of sample cell temperature

tt=0.0  !total quantity of sample cell temperature

tpi=0.0  !total quantity of inlet pressure
```fortran
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!total quantity of viscosity
tpower=0.0     !total quantity of power input
tmass=0.0      !total quantity of mass flow rate
tden=0.0       !total quantity of density
ndata=i+1      !number of data set(each set has nexp times)
1000 continue

1001 if(nr.eq.nend)go to 2000      !end of data file detected come here and check nr
   close(unit=5)                  !temporary close 5 and ready open 5 for another fluid
   ndata=i                       !set data number continuously
   nr=nr+1                       !number of refrigerant increased by 1
   go to 90
1010 format(1x,i2,2x,f15.8))
1011 format(1x,f6.3,7(f10.5,1x))
1012 format(1x,i2,2x,6(f15.8,lx))
1013 format(1x,5(f15.8,lx))
2000 close(unit=5)
stop
end

C This subroutine calculates the refrigerant properties, such as specific *
C heat capacity, viscosity, and thermal conductivity, by curve fittings. *
C The source data are based upon ASHRAE except R236ea based on REFPROP4.0 *
C***********************************************************************

subroutine prop(nr) !input number of refrigerant
   implicit real*8(a-h,o-z)
   common/acp/acp0,acp1,acp2,acp3,acp4,acp5  !Cp coefficients
   common/av/av0,av1,av2,av3,av4,av5        !Viscosity coef.
   common/ak/ak0,ak1,ak2,ak3,ak4,ak5       !K coef.
   if(nr.eq.1)then
      open(unit=20,file='prop.22',status='old') !coef. file for R-22
      c......coefficients for R-22 Cp
      read(20,*)acp0,acp1,acp2,acp3,acp4,acp5
   c......coefficients for R-22 viscosity
      read(20,*)av0,av1,av2,av3,av4,av5
   c......coefficients for R-22 K
      read(20,*)ak0,ak1,ak2,ak3,ak4,ak5
      close(unit=20)
      return
   else if(nr.eq.2)then
      open(unit=20,file='prop.12',status='old') !coef. file for R-12
      c......coefficients for R-12 Cp
```

---

The text is a Fortran subroutine that calculates refrigerant properties such as specific heat capacity, viscosity, and thermal conductivity. It reads data from files for different refrigerants (R-22 and R-12) and uses curve fittings for these calculations. The subroutine is part of a larger program that deals with the analysis of working fluids in thermodynamic systems.
read(20,*)acp0,acp1,acp2,acp3,acp4,acp5
  c.....coefficients for R-12 viscosity
  read(20,*)av0,av1,av2,av3,av4,av5
  c.....coefficients for R-12 K
  read(20,*)ak0,ak1,ak2,ak3,ak4,ak5
  close(unit=20)
  return
else if(nr.eq.3)then
  open(unit=20,file='prop.ll3',status='old') ! coef. file for R-113
  c coefficients for R-113 Cp
  read(20,*)acp0,acp1,acp2,acp3,acp4,acp5
  c.....coefficients for R-113 viscosity
  read(20,*)av0,av1,av2,av3,av4,av5
  c.....coefficients for R-113 K
  read(20,*)ak0,ak1,ak2,ak3,ak4,ak5
  close(unit=20)
  return
else if(nr.eq.4)then
  open(unit=20,file='prop.ll4',status='old') ! coef. file for R-114
  c.....coefficients for R-114 Cp
  read(20,*)acp0,acp1,acp2,acp3,acp4,acp5
  c.....coefficients for R-114 viscosity
  read(20,*)av0,av1,av2,av3,av4,av5
  c.....coefficients for R-114 K
  read(20,*)ak0,ak1,ak2,ak3,ak4,ak5
  close(unit=20)
  return
else if(nr.eq.5)then
  open(unit=20,file='prop.236',status='old') ! coef. file for R-236ea
  c.....coefficients for R-236ea Cp
  read(20,*)acp0,acp1,acp2,acp3,acp4,acp5
  acp0= 1.19340710E+03
  acp1= 6.33325863E+00
  acp2=-4.13577229E-01
  acp3= 1.47026302E-02
  acp4=-2.22029586E-04
  acp5= 1.21074845E-06
  c acp0=1.20477688E+03
  c acp1=1.92495738E-00
  c acp2=0.0
  c acp3=0.0
  c acp4=0.0
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c        acp5=0.0

c.....coefficients for R-236ea viscosity
read(20,*)av0,av1,av2,av3,av4,av5
av0=( 5.63759208E-01)*1.0e-3
av1=( -1.03053898E-02)*1.0e-3
av2=( 2.14341562E-04)*1.2e-3
av3=( -5.06344441E-06)*1.0e-3
av4=( 6.69040219E-08)*1.0e-3
av5=( -3.19661297E-10)*1.0e-3

c.....coefficients for R-236ea K
read(20,*)ak0,ak1,ak2,ak3,ak4,ak5
close(unit=20)
return
end if
end

C This subroutine calculates the air properties, such as density, Cp, viscosity, and thermal conductivity, by curve fittings. The source data are based on back tables in Appendix of the book by Incoperea and DeWitt.*

C****************************************************************************************************************************
C subroutine airprop(tak,dena,cpa,visa,tka)
implicit real*8(a-h,o-z)
data d0,d1,d2,d3,d4,d5/8.40219021e+00,-7.31001496e-02,
&3.01471766e-04,-6.34908758e-07,6.5867756e-10,-2.66433143e-13/
data c0,c1,c2,c3,c4,c5/1.11507312e+03,-1.31544113e+00,
&5.91507275e-03,-1.29040736e-05,1.44722101e-08,-6.3939549e-08,
&9.515507275e-01,1.33037244e-13,-1.27004888e-16,5.52250076e-20/
data v0,v1,v2,v3,v4,v5/-3.79514205e-07,8.36939549e-08,
&9.95046476e-11,1.33037244e-13,-1.27004888e-16,5.52250076e-20/
data t0,t1,t2,t3,t4,t5/-4.19377378e-04,1.03869352e-04,
&7.47035287e-08,1.22800797e-10,-1.5558361e-13,7.10215355e-17/
c.....getting density, specific heat, viscosity, and thermal conductivity of air at temperature tak(in K)
dena=d0+d1*tak+d2*tak**2+d3*tak**3+d4*tak**4+d5*tak**5
cpa=c0+c1*tak+c2*tak**2+c3*tak**3+c4*tak**4+c5*tak**5
visa=v0+v1*tak+v2*tak**2+v3*tak**3+v4*tak**4+v5*tak**5
tka=t0+t1*tak+t2*tak**2+t3*tak**3+t4*tak**4+t5*tak**5
return
end

C****************************************************************************************************************************
C This subroutine calculates the average wall and fluid temperature difference by using finite control volume method

C****************************************************************************************************************************
subroutine dtwf(twi,tiave,teave,dt)
implicit real*8(a-h,o-z)
dimension twi(13),tfi(13),dx(13),x(13)

c
....finite control volume grid increment and coordinates
c
dx(1)=0.05
x(1)=0.0
dx(2)=0.1
x(2)=0.1
dx(3)=0.15
x(3)=0.2
do 10 i=4,10
dx(i)=0.2
x(i)=x(i-1)+0.2
10 continue
edx(11)=0.15
x(11)=0.18
dx(12)=0.1
x(12)=0.19
dx(13)=0.05
x(13)=2.0
dl=2.0
c
c....fluid temperature distribution(assume linear)
c
tfi(1)=tiave
tfi(13)=teave
do 20 i=2,12
tfi(i)=tfi(i-1)+(x(i)-x(i-1))*((tfi(13)-tfi(i))/dl)
20 continue
c
c.....dt calculations
c
dtt=0.0
do 30 i=1,13
dtt=dtt+(dx(i)/(twi(i)-tfi(i)))
30 continue
dt=dl/dtt
return
end
This subroutine calculates the calibration functions for the Nusselt number's equations, such as Gnielinski, Petukhov, and Dittus-Boelter equations. Also, the experimental data forms the Nusselt number correlation, Nue. Those are obtained by curve fittings from SAS.

SUBROUTINE CFNU(PR,RE,ENU,CFG,CFP,CFD,CFH)
IMPLICIT REAL*8(A-H,O-Z)
COMMON/GPDA/GA,PA,DA
COMMON/GPDB/GB,PB,DB
COMMON/GPDC/GC,PC,DC
COMMON/CC/CCS,CCG,CCP,CCD
DATA AE,BE,CE/0.850544751,0.507056,-4.39778757/
DATA AG,BG,CG/0.835702658,0.495804,-4.16347694/
DATA AP,BP,CP/0.819511116,0.516636,-4.03165674/
DATA AD,BD,CD/0.799977124,0.400000,-3.77200365/
DATA HA,HB,HC/0.746661,-0.477820,-0.159677/
DATA HAL,HBL,HCL/0.746638,-0.477820,-0.159415/
GA=AE-AG
PA=AE-AP
DA=AE-AD
GB=BE-BG
PB=BE-BP
DB=BE-BD
CCS=EXP(CE)
CCG=EXP(CG)
CCP=EXP(CP)
CCD=EXP(CD)
GC=CE-CG
PC=CE-CP
DC=CE-CD
ENU=PR**AE*RE**BE*EXP(CE)
CFG=(PR**BE-BG)*(RE**AE-AG)*(EXP(CE-CG))
CFP=(PR**BE-BP)*(RE**AE-AP)*(EXP(CE-CP))
CFD=(PR**BE-BD)*(RE**AE-AD)*(EXP(CE-CD))
CFH=(PR**HB-HBI)*(RE**HA-HAI)*(EXP(HC-HC1))
RETURN
END

The following subroutine calculates the thermal conductivity ratio at different pressure but at the same temperature. This is so called pressure effects on thermal conductivity. The equation used here is Missenard correlation(1970)
c******************************************************************************
subroutine kfactor(tr,pr,ratiok)
implicit real*8(a-h,o-z)
dimension a(21)
data a(1),a(2),a(3),a(4),a(5),a(6),a(7),a(8),
& a(9),a(10),a(11),a(12),a(13),a(14),a(15),a(16),
& a(17),a(18),a(19),a(20),a(21)
& /-0.376480,1.998078,0.007849,-3.419245,-0.053279,
& 0.000131,1.956588,0.115588,-0.000281,-0.000001554,
& 0.0,-0.075644,-0.0000291,0.000005688,-5.59027E-10,
& 0.0,0.0,0.000517,-0.000004349,0.46296E-11,2.46059E-13/
q=a(1)+a(2)*tr+a(3)*pr+a(4)*tr**2+a(5)*tr*pr+a(6)*pr**2+
& a(7)*tr**3+a(8)*tr**2*pr+a(9)*tr*pr**2+a(10)*pr**3+
& a(11)*tr**4+a(12)*tr**3*pr+a(13)*tr**2*pr**2+a(14)*tr*pr**3+
& a(15)*pr**4+a(16)*tr**5+a(17)*tr**4*pr+a(18)*tr**3*pr**2+
& a(19)*tr**2*pr**3+a(20)*tr*pr**4+a(21)*pr**5
ratiok=l.0+q*pr**0.7
return
end

C******************************************************************************
c The following subroutine calculates the liquid viscosity *
c ratio at different pressure but at the same temperature. This is*
c so called pressure effects on liquid viscosity. The equation *
c used here is Lucas correlation(1981) *
c******************************************************************************
subroutine vfactor(w,tr,dpr,ratiov)
implicit real*8(a-h,o-z)
a=0.9991-(4.674E-04/(1.0523*tr**(-0.03877)-1.0513))
d=(0.3257/(1.0039-tr**2.573)**0.2960)-0.2086
c=-0.07921+2.1616*tr-13.4040*tr**2+44.1706*tr**3-
& 84.8291*tr**4+96.1209*tr**5-59.8127*tr**6+15.6719*tr**7
ratiov=(1.0+d*(dpr/2.118)**a)/(1.0+c*w*dpr)
return
end

C******************************************************************************
c The following program is used to calculate k by directly curve *
c fitting by function log(h), log(vis*cp), and log(re) *
c******************************************************************************
subroutine newk(pr,re,dtn)
implicit real*8(a-h,o-z)
data a0,a1,a2,a3,a4,a5,a6,a7,a8,a9
& /96.908792,-26.452258,83.304684,2.459228,
\& \begin{array}{l}
-16.020859, 8.088965, -0.076756, 0.750949, \\
-0.668378, 2.213377/
\end{array}
\]

\begin{verbatim}
x = \log(re)
y = \log(dtn)
pr = a0 + a1*x + a2*y + a3*x**2 + a4*x*y + a5*y**2 + \\
a6*x**3 + a7*x**2*y + a8*x*y**2 + a9*y**3
pr = \exp(pr)
return
end
\end{verbatim}