Experimental Determination of Thermal Accommodation Coefficients of Rarefied Argon, Helium, Nitrogen, and Water Vapor on Stainless Steel Concentric Cylinders

A thesis submitted in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering

by

Mitchell G. Lane

Dr. Mustafa Hadj-Nacer/Thesis Advisor

December 2020
THE GRADUATE SCHOOL

We recommend that the thesis prepared under our supervision by

Mitchell G. Lane

Entitled

Experimental Determination of Thermal Accommodation Coefficients of Rarefied Argon, Helium, Nitrogen, and Water Vapor on Stainless Steel Concentric Cylinders

be accepted in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

Mustafa Hadj-Nacer, Ph.D., Advisor

Miles Greiner, Ph.D., Co-advisor

Dev Chidambaram, Ph.D., Graduate School Representative

David W. Zeh, Ph.D., Dean, Graduate School

December 2020
Abstract

During vacuum drying of used nuclear fuel canisters, the temperature of the assemblies may considerably increase due to the effect of gas rarefaction at low pressure, and the lack of buoyancy induced gas motion. The effect of rarefaction can manifest as a temperature jump thermal resistance at the gas-solid interfaces that causes the fuel temperature to increase compared to non-rarefied conditions (no-temperature jump). It is important to maintain the temperature of the assemblies below certain limits to ensure the integrity and retrievability of the used nuclear fuel and internal components of the canister. Therefore, it is crucial to accurately determine the effect of gas rarefaction on the fuel temperature during vacuum drying process.

The objective of this thesis is to experimentally and numerically study the effect of gas rarefication on heat transfer across a 2 mm annular gap filled with different gases (argon, helium, nitrogen, and water vapor) at different pressures. The results from the experiments are used to determine the temperature jump and thermal accommodation coefficients for these gases on a stainless steel surface. The thermal accommodation coefficients were determined to be 0.35-0.38 for helium, 0.95-0.97 for argon, 0.88 for nitrogen, and 1.01 for water vapor. These determinations compare well with literature sources and provide a first determination of temperature jump and thermal accommodation coefficient of water vapor on a stainless steel surface. These results will be then used to predict the temperature of the fuel assemblies during vacuum drying and will be used to make rough determinations of temperature jump and thermal accommodation coefficients of gas mixtures.
Acknowledgements

First of all, I want to thank my advisor Dr. Mustafa Hadj-Nacer for his numerous hours of guidance and assistance which has helped me grow to become a better engineer and person. I would also like to thank my co-advisor Dr. Miles Greiner, who has allowed me to further myself as a professional and help me pay close attention to details. And I would like to thank Dr. Dev Chidambaram for being a member of my thesis committee.

I would like to thank my parents, Richard Lane and Tracey Greenwood for the love and encouragement throughout my education. Without the support of my family, I would not have not been able to complete this challenging journey.

Many thanks to my colleagues throughout my degree: Cody Zampella, Megan Higley, Miffat Hassan, Kegan Rahe, and Harrison Griffin. Their kindness, knowledge, and patience has been greatly appreciated. To my friends; Bruno Reyes, Joel Kaderka, Jose Carillo, Justin Siao, Skyler Russ and William Crain, our friendships have been invaluable and I will continue to hold them close to me.

This research has been funded by the US Department of Energy [DENE0008713] and the Nuclear Regulatory Commission [31310018M0032].
Table of Contents

Abstract ................................................................................................................................. i

Acknowledgments .............................................................................................................. ii

Table of Contents ................................................................................................................. iii

List of Tables .......................................................................................................................... vi

List of Figures ....................................................................................................................... viii

Nomenclature ........................................................................................................................ xii

Chapter 1: Introduction ........................................................................................................ 1

1.1 Gas Rarefaction ............................................................................................................ 1

1.2 Temperature Jump Effect ............................................................................................ 4

1.3 Thermal Accommodation Coefficient .......................................................................... 5

1.4 Applications of Rarefied Gas ....................................................................................... 6

1.5 Objectives of the Thesis ............................................................................................... 8

Chapter 2: Literature Review .............................................................................................. 10

2.1. Experimental Approaches for Thermal Accommodation Coefficient Determination ......10

2.1.1. Parallel Plate Geometry ....................................................................................... 10

2.1.2. Concentric Cylinder Geometry .......................................................................... 12

2.1.3. Cooling Sphere .................................................................................................. 12

2.2. Mathematical and Computational Models for Thermal Accommodation Coefficient Determination Based on Temperature Jump Coefficient .................................................. 14
Chapter 3: Experimental Setup ................................................................................................. 20

3.1. Geometry and Dimensions ............................................................................................... 20

3.2. Instrumentation .................................................................................................................. 26

3.3. Water Vapor Injection System ............................................................................................ 28

3.4. Experiment Modifications ................................................................................................. 29

Chapter 4: Analytical Solution of Heat Transfer ....................................................................... 30

4.1. Concentric Cylinder Solution of Heat Transfer ................................................................. 30

4.2. Direct Determination of Temperature Jump and Thermal Accommodation Coefficients . 33

4.3. Linear Regression Determination of Temperature Jump and Thermal Accommodation Coefficients .......................................................................................................................... 34

4.4. Uncertainty Analysis on Temperature Jump and Thermal Accommodation Coefficients 35

Chapter 5: Results and Discussion ........................................................................................... 37

5.1. Experimental Results ......................................................................................................... 37

5.1.1. Inner and Outer Cylinder Axial Temperature Profiles ..................................................... 38

5.1.2. Effect of Gas Rarefaction .................................................................................................. 43

5.2. Determination of Temperature Jump and Thermal Accommodation Coefficients from Experimental Data .................................................................................................................. 50

5.3. Comparison of Computational Fluid Dynamics Simulations with Experimental Results 60

Chapter 6: Conclusions and Future Work .................................................................................. 65

6.1. Conclusions ......................................................................................................................... 65
6.2. Future Work ...........................................................................................................67

Appendix A: Material Properties .............................................................................68

References ....................................................................................................................71
List of Tables

Table 2.1: Summary of thermal accommodation coefficient obtained from the literature for stainless steel ...........................................................................................................................................................................18

Table 3.1: Inner cylinder component dimensions of the cylindrical section ..................................................22

Table 3.2: Outer cylinder component dimensions ...............................................................................................23

Table 3.3: Vacuum tree valve component and gas specifications .................................................................25

Table 3.4: Specifications of thermocouples and capacitance manometers .........................................................27

Table 3.5: Data acquisition, power supply, and control systems .................................................................27

Table 3.6: Water vapor injection part list ........................................................................................................29

Table 5.1: The thermal accommodation ($\alpha$) and temperature-jump ($\zeta_T$) coefficients with their uncertainties for all heat generation rates of helium using regression approach ......................................................56

Table 5.2: The thermal accommodation ($\alpha$) and temperature-jump ($\zeta_T$) coefficients with their uncertainties for all heat generation rates of argon using regression approach ..........................................................56

Table 5.3: The thermal accommodation ($\alpha$) and temperature-jump ($\zeta_T$) coefficients with their uncertainties for all heat generation rates of nitrogen using regression approach ..........................................................56

Table 5.4: The thermal accommodation ($\alpha$) and temperature-jump ($\zeta_T$) coefficients with their uncertainties for all heat generation rates of water vapor using regression approach ..........................................................56

Table A.1: Material properties of Magnesium Oxide ..........................................................................................68

Table A.2: Material properties of Stainless Steel .............................................................................................68

Table A.3: Material properties of Aluminum ..................................................................................................68

Table A.4: Material properties of Helium ........................................................................................................69
Table A.5: Material properties of Argon .................................................................69

Table A.6: Material properties of Nitrogen ...............................................................70

Table A.7: Material properties of Water Vapor ........................................................70
List of Figures

Figure 1.1: Gas rarefaction regimes based on the Knudsen Number ..................................................2

Figure 1.2: Temperature jump profile for rarefied and non-rarefied cases.................................4

Figure 3.1: Bottom view of the inner cylinder geometry comprised of a cartridge heater, a thick aluminum cylinder, and a thin stainless steel sheath .................................................................21

Figure 3.2: Bottom view of the inner cylinder with the support piece attached .....................21

Figure 3.3: Inner cylinder attached to the outer cylinder system, with thermocouple and cartridge heater power wires .........................................................................................................................22

Figure 3.4: (a) Full cross-section and (b) zoomed views of the cylindrical section of the experimental apparatus schematic ..............................................................24

Figure 3.5: Vacuum tree section with capacitance manometers, gas inlet, vacuum pump, and water evaporation section ..........................................................................................................................25

Figure 3.6: Picture of the water vapor injection system .................................................................28

Figure 5.1: Temperature and pressure response over time for Nitrogen at $Q = 100$ W and $P = 1837$ Pa..................................................................................................................................................38

Figure 5.2: Temperature versus axial location for helium at the lowest and highest heat generations and pressures. The x symbols are for $Q=500$W and open square symbols are for $Q=100$W. The dashed lines represent the average outer wall temperature, and the solid lines represent the temperature profiles from the CFD simulations.........................................................39

Figure 5.3: Axial temperature profiles for helium and argon at $Q = 100$ W at the lowest and highest pressures. The x symbols are for argon and open square symbols are for helium. The
Figure 5.4: Axial temperature profiles for helium and nitrogen at $Q = 100$ W at the lowest and highest pressures. The x symbols are for nitrogen and open square symbols are for helium. The dashed lines represent the average outer wall temperature, and the solid lines represent the temperature profiles from the CFD simulations.

Figure 5.5: Axial temperature profiles for helium and water vapor at $Q = 100$ W at the lowest and highest pressures. The x symbols are for water vapor and open square symbols are for helium. The dashed lines represent the average outer wall temperature, and the solid lines represent the temperature profiles from the CFD simulations.

Figure 5.6: Experimental steady state average temperature differences as a function of pressure for helium at five heat generation rates, $Q = 100, 200, 300, 400, \text{ and } 500W$. The data points with squares, diamonds, triangles, circles, and x symbols are experimental data, and the solid and dashed lines connect simulated data points.

Figure 5.7: Experimental steady state average temperature differences as a function of pressure for argon at two heat generation rates, $Q = 50$ and 100W. The data points with squares, and triangles are from the experimental results, and the solid and dashed lines connect simulated data points.

Figure 5.8: Experimental steady state average temperature differences as a function of pressure for nitrogen at the heat generation rate of $Q = 100W$. The square symbols represent the experimental data, and the solid line connect simulated data points.
Figure 5.9: Experimental steady state average temperature differences as a function of pressure for water vapor at the heat generation rate of $Q = 100\text{W}$. The square symbols represent the experimental data. 48

Figure 5.10: Average temperature difference minus continuum temperature difference divided by continuum temperature difference as a function of rarefaction parameter $\delta$ for helium, argon, and nitrogen at $Q = 100\text{W}$. 50

Figure 5.11: Helium temperature difference as a function of the inverse of pressure fitted with linear regressions and equations. 51

Figure 5.12: Argon temperature difference as a function of the inverse of pressure fitted with linear regressions and equations. 52

Figure 5.13: Nitrogen temperature difference as a function of the inverse of pressure fitted with linear regressions and equations. 53

Figure 5.14: Water vapor temperature difference as a function of the inverse of pressure fitted with linear regressions and equations. 54

Figure 5.15: Helium thermal accommodation coefficients versus temperature. The dashed and solid lines represent the thermal accommodation coefficient value obtained using the regression method. The open square, diamond, triangle, circle, and x symbols are the thermal accommodation coefficient values for the five lowest pressures in a heat generation and the closed symbols are data collected from literature sources: Trott [14], Byung [17], Loyalka [18], and Song & Yovanovich [31]. 58

Figure 5.16: Argon thermal accommodation coefficients vs. temperature. The dashed and solid lines represent the thermal accommodation coefficient value using the regression method. The
open box and diamond symbols are the thermal accommodation coefficient values for the five lowest pressures in a heat generation and the closed symbols are data collected from literature sources: Trott [14], Byung [17], Loyalka [18], and Song & Yovanovich [31] ..................................59

Figure 5.17: Nitrogen thermal accommodation coefficients vs. temperature. The dashed and solid lines represent the thermal accommodation coefficient value using the regression method. The open box and diamond symbols are the thermal accommodation coefficient values for the five lowest pressures in a heat generation and the closed symbols are data collected from literature source: Trott [14] ...........................................................................................................................................60

Figure 5.18: Isometric view of one-third 3D CFD model .................................................................61

Figure 5.19: (a) Cross-sectional temperature contour at the axial center, z = 0 cm, and (b) isometric temperature contour plot obtained from the ANSYS Fluent CFD simulations for Argon at Q= 100W and P=37 Pa .............................................................................................................................................63
## Nomenclature

### Roman Letters

- \( a \) Slope Term of Linear Regression
- \( b \) Intercept Term of Linear Regression
- \( c_p \) Specific Heat at Constant Pressure
- \( c_v \) Specific Heat at Constant Volume
- \( k_B \) Boltzmann Constant
- \( Kn \) Knudsen Number
- \( L \) Cylinder Length
- \( L_c \) Characteristic Length
- \( m \) Molecular Mass
- \( P \) Pressure
- \( Pr \) Prandtl Number
- \( R \) Thermal Resistance
- \( T \) Temperature
- \( Q \) Heat Generated

### Greek Letters

- \( \alpha \) Thermal Accommodation Coefficient
- \( \gamma \) Ratio of Specific Heats
- \( \Delta T \) Temperature Difference Between the Inner and Outer Cylinder
- \( \delta \) Gas Rarefaction Parameter
- \( \delta f \) Uncertainty a Variable f
- \( \varepsilon \) Surface Emissivity
$k$  
Thermal Conductivity

$\lambda$  
Mean Free Path

$\mu$  
Dynamic Viscosity

$\mu_0$  
Reference Dynamic Viscosity

$\omega$  
Viscosity Index

$\zeta_T$  
Temperature Jump Coefficient

Subscripts

0  
Reference Value

$C$  
Heat Transferred by Conduction

$g$  
Gas

$i$  
Inner or Incident

$j$  
Jump

$o$  
Outer

$R$  
Heat Transferred by Radiation

$w$  
Wall

Acronyms

BGK  
Bhatnagar-Gross-Krook

CFD  
Computational Fluid Dynamics

HS  
Hard Sphere

MEMS  
Microelectromechanical Systems

TAC  
Thermal Accommodation Coefficient

VHS  
Variable Hard Sphere
Chapter 1: Introduction

In some heat transfer problems through gases that involve low-pressure conditions and/or small gaps between solid surfaces, some unique and interesting phenomena may occur at the walls. When the gas pressure is low, the average distance that gas molecules will travel between two successive collisions becomes comparable to the size of the small gaps. When this occurs, the gas is said to be rarefied and gas molecules will not fully accommodate the temperature of solid walls, which leads to a temperature discontinuity at the walls. This chapter provides a formal definition of gas rarefaction, describes how thermal energy is transferred under rarefied conditions, and offers examples of gas rarefaction in real systems.

1.1. Gas Rarefaction

There are two approaches when modeling gas heat transfer; macroscopic and microscopic. The macroscopic approach assumes that the gas will behave in a continuous manner. The governing equations that model continuum fluid flow and heat transfer are the Navier-Stokes and Convective Heat Transfer Equations, respectively. In contrast, in the microscopic approach, gas molecules must be modeled as a vast collection of individual molecules. The governing equation used to model discrete particle fluid flow and heat transfer is the Boltzmann Transport Equation [1]. Both approaches, macroscopic and microscopic, vary wildly in complexity but are both used in industry as well as academia.

For a continuous flow, the average distance that gas molecules travel between two successive collisions, known as the mean free path $\lambda$, is very small compared to the characteristic length of the system. A gas molecule will collide many times with other gas molecules before a collision with a solid wall. Contrary to the continuum approach, the mean free path in a so-called rarefied gas is comparable or larger than the characteristic length of the system. In this case, the
gas molecules will collide more often with the walls than with other gas molecules. The parameter that characterizes gas rarefaction is called the Knudsen number, and is the ratio of the mean free path, $\lambda$, to the characteristic length of the system, $L_c$ [1],

$$Kn = \frac{\lambda}{L_c}$$ (1.1)

The characteristic length, $L_c$, is generally the smallest dimension of a system. The mean free path takes on many interpretations based on application, but for rarefied gas dynamics problems, the mean free path is defined as [2]:

$$\lambda = \frac{\mu}{P} \sqrt{\frac{2k_B}{m}T}$$ (1.2)

where $\mu$ is the dynamic viscosity of the gas, $P$ is the gas pressure, $k_B$, is the Boltzmann Constant ($k_B = 1.3806 \times 10^{-23} m^2 kgs^{-2} K^{-1}$), $m$ is the molecular mass of the gas, and $T$ is the gas temperature. As pressure decreases, the mean free path increases and causes the Knudsen number to increase as well. A higher value of the Knudsen number means that gas is more rarefied.

Figure 1.1: Gas rarefaction regimes based on the Knudsen Number.
Figure 1.1 shows the different regimes of gas rarefaction based on the Knudsen number, as it was defined by Schaff and Chambre [3]. There are four regimes of rarefaction that have distinct physics and different numerical models can be used to solve them. These regimes are:

- The Continuum regime \((Kn \leq 0.001)\), where the gas can be treated as continuum, and properties such as temperature, velocity, and pressure are continuous in localized regions. In this regime, the Navier-Stokes and Convective Heat Transfer Equations can be used to model the flow and heat transfer.

- The Slip regime \((0.001 < Kn \leq 0.1)\), where the number of interactions between gas molecules and solid surfaces becomes small. This leads to discontinuities of macroscopic parameters such as temperature, velocity, and pressure at gas/solid interfaces. In this regime, the Navier-Stokes and Convective Heat Transfer Equations can be used but with modified boundary conditions to account for these discontinuities at the interfaces.

- Transitional regime \((0.1 < Kn \leq 10)\), where the mean free path becomes comparable to the characteristic length of the system, and gas molecule interactions with solid surfaces occur at approximately the same frequency as interactions between molecules. In this regime, large discontinuities in the macroscopic properties exist at the interfaces. In this regime, the Navier-Stokes and Convective Heat Transfer Equations are no longer valid, but the collisional Boltzmann Equation must be solved to model the flow and heat transfer.

- Free Molecular Regime \((Kn > 10)\), where gas flow properties are mainly driven by the collision of molecules with solid surfaces. Collisions of gas molecules with each other can be neglected in this regime. In this regime, the collisionless Boltzmann Equation can be used and can be solved analytically.
It must be noted that the Boltzmann Equation can be used to solve fluid flow and heat transfer in all regimes of rarefaction but is too computationally expensive in the continuum and slip regimes, even with modern computers. The experimental results shown in this thesis are conducted in the continuum and slip regimes.

1.2. Temperature Jump Effect

As explained above, the rarefaction effect may cause a discontinuity of temperature at the gas/solid interfaces. This discontinuity is known as “temperature jump.” It can be seen as a thermal resistance to heat transfer at the interfaces and it can significantly affect heat transfer in the slip, transitional, and free molecular regimes. The driving mechanism behind temperature jump at gas-solid interfaces is due to a loss in thermal energy by inelastic collisions between the gas molecules and the solid surface.

![Figure 1.2: Temperature jump profile for rarefied and non-rarefied cases.](image)

Figure 1.2 shows a schematic of the temperature jump model at the gas/solid interface in the slip regime (rarefied case) compared to a non-rarefied case (continuum regime). The model shows a solid and a gas region. Heat is applied to the solid region from the left and the temperature at the right of the gas region is held constant at $T_c$. Distance from the heat source, $Q$, is plotted on
the horizontal axis, and the temperature, $T$, through the media is on the vertical axis. The non-rarefied case is shown with a dashed line and it exhibits a temperature gradient through the solid and another temperature gradient through the gas. At the gas/solid interface, the gas and wall temperatures are equal (no temperature jump). For the rarefied case, the temperature gradient through the solid and gas regions is the same as the non-rarefied case. However, a temperature discontinuity or jump exists at the gas-solid interface indicated by $\Delta T_j$. The temperature jump is due to rarefaction effects and can be expressed as [4].

$$\Delta T_j = T_w - T_g = -\zeta_T \frac{\partial T}{\partial x}|_{x=w}$$  \hspace{1cm} (1.3)

where $T_w$ is the wall temperature, $T_g$ is the gas temperature near the wall, and $\zeta_T$ is the temperature jump coefficient. The temperature jump coefficient multiplied by the mean free path is often referred to the Knudsen layer thickness. As a result of temperature jump, the temperature in the rarefied case is higher than in the non-rarefied case.

**1.3. Thermal Accommodation Coefficient**

The parameter that describes how much thermal energy gas molecules will transfer when colliding with solid interfaces is called the thermal accommodation coefficient (TAC). The temperature jump coefficient $\zeta_T$ in Eq. 1.3 is a function of the TAC. A commonly used relation for thermal accommodation coefficient and temperature jump coefficient will be discussed in Sec. 4.2. The thermal accommodation coefficient can be defined as [3],

$$\alpha = \frac{T_r - T_i}{T_w - T_i}$$  \hspace{1cm} (1.4)

where $T_i$ is the incident gas molecule temperature and $T_r$ is the temperature of the reflected gas molecule. This ratio states how well gas molecules will exchange thermal energy upon collision with a solid interface. A specular reflection of a gas molecule has a thermal accommodation
coefficient value of zero, and the gas molecule will transfer no thermal energy upon collision with the solid interface. A diffuse reflection of a gas molecule has a thermal accommodation coefficient of one, and the gas molecule accommodates its thermal energy with the wall. The thermal accommodation coefficient of a given gas (collection of molecules) is between zero and one. This means that some of the molecules will be reflected specularly and the rest of the molecules will be reflected diffusely from the wall. A thermal accommodation coefficient that is close to zero will yield large temperature jumps at gas-solid interfaces, while a value close to one will yield small temperature jumps due to the Knudsen layer thickness.

The thermal accommodation coefficient at a gas-solid interface can be a function of material, surface characteristics, and temperature. By changing the solid interface material, as well as the gas type [5], the thermal accommodation coefficient can be modulated. A cleaned surface will have a lower thermal accommodation coefficient than that of a dirty (engineering) surface.

1.4. Applications of Rarefied Gas

Rarefied gas flows are uncommon in daily life, but they can exist in engineering or industrial applications. In some cases, gas rarefaction is not desirable and may even lead to a malfunction. In contrast, some products have been designed with gas rarefaction in mind to accomplish a particular design parameter. While there are many examples of gas rarefaction, only examples pertinent to this research will be discussed in this section.

The first example of gas rarefaction stems from the storage of used nuclear fuel (UNF) assemblies. UNF assemblies are stored underwater in large cooling pools for many years. They are then transferred to dry storage canisters. The transfer operation takes place underwater. The canisters are then sealed, lifted out of the water, and drained. The remaining water in the canister must be completely removed to avoid corrosion of the canister’s internal components or the
formation of a flammable mixture of hydrogen and oxygen [8], which may compromise safety. Therefore, drying operation such as vacuum drying is employed, during which, the canister internal pressure is reduced in steps to as low as 67 Pa (0.5 Torr) to promote evaporation of water molecules [8]. Since the canister pressure becomes low, the average distance that gas molecules travel between collisions becomes comparable to the spacing between fuel rods and other small gaps inside the canister which is on the order of a few millimeters. In many locations inside the canister, the gas may become rarefied and will cause jumps in temperature at gas-solid interfaces. The US Nuclear Regulatory Commission has imposed a 400°C temperature limit on the cladding materials to ensure the cladding material has a high likelihood of remaining intact throughout vacuum drying [9].

The second application of rarefied gas comes from the aerospace industry. Re-entry vehicles are designed to quickly reduce gravitational potential energy as well as kinetic energy while combatting high physical stresses. During re-entry, there are interesting rarefied gas dynamics based on the geometry, height, and speed of the re-entry vehicle. Above 100 km, the flow can be categorized as free molecular and the vehicle will experience little to no friction with the atmosphere [10]. As the vehicle gets lower into Earth’s atmosphere, the density of atoms increases. The vehicle will experience a change in flow regimes from free molecular to transitional, slip, and finally, the continuum. At the start of the reentry, the vehicle begins to heat up due to atmospheric frictional heating. In the slip, transitional, and free molecular regimes, the vehicle experiences a large temperature jump on the heat shield, which drastically increases its temperature. This is why re-entry vehicles are designed with high material strength as well as thermal loading by rarefaction effects [10].
The third application comes from porous media, which is a material that has small pores with diameters that sometimes can be comparable to the average distance that gas particles will travel before successive collisions. Common examples of porous media are sponges, cement, and wood. But one example of porous media that takes advantage of rarefaction effects is Aerogel, which has low densities that range from 0.003 - 0.15 kg/m$^3$, and are comprised of over 99% empty space [11]. The pore size can be tailored based on application. Aerogel has an average pore diameter comparable to the mean free path of gas molecules. Aerogel takes advantage of gas rarefaction since there are small temperature jumps within each pore, which makes them as a super insulator. Although Aerogel is effective at minimizing heat transfer and great material to use for insulation, one downside of Aerogel is that it is very brittle. Since the material is mostly made of empty space, it has very little structural integrity and can only be used in low-stress environments.

The fourth and final application of rarefied gas comes from microelectromechanical systems (MEMS). MEMS devices are extremely small devices with sensing capabilities, some kind of mechanical functionality, or a combination of both [12]. MEMS devices are especially useful in smartphones and personal computers since they have small dimensions and can be manufactured to have a wide array of sensing capabilities. The dimensions of these devices can be comparable to the mean free path, and sometimes the devices are exposed to low pressures which further the level of gas rarefaction [13]. Since these devices can be often be used in temperature-driven microflows, it is necessary to understand the rarefaction effects that are occurring.

1.5. Objective of the Thesis

The objective of this thesis is to use a concentric cylinder experimental apparatus to measure heat transfer through rarefied argon, helium, and nitrogen gases and compare the results to a one-dimensional (1D) analytical solution of heat transfer to determine the coefficients to
determine the temperature jump and thermal accommodation coefficients for those gases on a stainless steel surface. The uncertainty on these coefficients is assessed and they are compared to known values in the literature to validate the experiment’s ability to measure those gases. The experiment is then used to determine the temperature jump and thermal accommodation coefficients for water vapor on stainless steel. Furthermore, computational fluid dynamics (CFD) simulations are conducted for geometry similar to the experiment, and results are compared to the measurements.
Chapter 2: Literature Review

To experimentally determine the thermal accommodation coefficient of gas on a solid surface, characteristics such as geometry, temperature, pressure, and materials must all be accurately controlled and measured. Then, using those measurements, mathematical models of rarefied gas heat transfer can be used to determine the thermal accommodation coefficient. There have been many experiments designed and mathematical models developed to determine the thermal accommodation coefficient, and this section will discuss the different approaches.

2.1. Experimental Approaches for Thermal Accommodation Coefficient Determination

2.1.1. Parallel Plate Geometry

Rader et al. [14] determined the thermal accommodation coefficient of helium, argon, and nitrogen between two parallel stainless steel 304 plates. An inter-plate gap of 10 mm was maintained for helium and argon while a 5mm gap was used for nitrogen. Hot and cold water baths were used to keep the stainless steel plates at constant temperatures and thermocouples were instrumented in the center of the stainless steel plates to measure the temperature difference. The average temperature of the hot and cold water baths was about room temperature, which resulted in a low-temperature range. The pressure of the system was varied between 0.13 and 900 Pa, which spans the transitional, slip, and into the continuum regimes. The heat transfer due to radiation was subtracted from the temperature different measurements between the parallel plates by calibrating the system at 0 Pa. As the pressure decreased in the system, the temperature difference increased. The temperature difference and pressure readings were then fitted using a first-order line of best-fit, where the value of the thermal accommodation coefficient was extracted. The surface finish of the stainless steel plates showed no effect for argon and nitrogen but the polished plate showed a
slightly higher thermal accommodation coefficient than the machined plate for helium. The thermal accommodation coefficient of helium on machined stainless steel was determined to be ~0.36, while on polish stainless steel it was determined to be ~0.39. The thermal accommodation coefficient of nitrogen on both machined and polished surfaces of stainless steel was determined to be ~0.80. And lastly, the thermal accommodation coefficient of argon on both machined and polished surfaces of stainless steel was determined to be ~0.87. The 95% uncertainty on these calculations was determined to be 0.02.

Song and Yovanovich [15] also used the parallel plate geometry approach for determining the thermal accommodation coefficients of helium, argon, and nitrogen on a stainless steel surface as well as a nickel surface. Disks with a diameter of 2.5 cm and a length of 4.5 cm were made of stainless steel 304 and nickel 200. The bottom cylinder contained a heating element and six thermocouples, while the top cylinder contained six thermocouples to measure temperature difference. Heat generation was adjusted to keep the average gas temperature at about 170°C. Experiments using the test column were conducted inside of a Pyrex bell jar and pressures were ranged from 1333 to 100,000 Pa. The gap length was not discussed in the paper therefore the rarefaction regimes could not be estimated. A least-squares regression method based on an analytical solution of heat transfer between parallel plates was used to determine the thermal accommodation coefficient. The measurements of the thermal accommodation coefficient of helium on stainless steel ranged from 0.67 to 0.58. For argon, it was determined to range from 0.96 to 0.88. And lastly, for nitrogen, it was determined to range from 0.87 to 0.79. The results for helium were substantially different compared to that of other works, while the argon and nitrogen results were in agreement with other results. An analysis of the large discrepancy in the helium result was not conducted.
2.1.2. Concentric Cylinder Geometry

Ho [16] measured the thermal accommodation coefficient of air on a stainless steel surface using a concentric cylinder geometry experiment. At the center of the inner cylinder was a 23 cm long heating element and a stainless steel sheath around the element. Moving radially outward, there was a 1 mm gap then another stainless steel cylinder which makes up the outer cylinder. Thermocouples were instrumented on the inner and outer cylinders to measure the temperature difference across the gap. The outer stainless steel cylinder was kept at a constant temperature by copper tubing containing circulating water. The temperature differences ranged between 24°C and 94°C and pressures were chosen such that the rarefied dry air was categorized as a free molecular flow. The experiments operated at low heat generation on the order of a few watts because of the large temperature jumps at gas-solid interfaces, which can be achieved in the free molecular regime. An analytical solution of heat transfer between concentric cylinders was used to determine the thermal accommodation coefficient. The average value of the thermal accommodation coefficient of dry air on stainless steel was determined to be 0.963 for a temperature range of 24°C to 94°C. The effects of surface roughness were analyzed on emissivity and thermal accommodation coefficient. Both increased with increasing roughness.

2.1.3. Cooling Sphere

Jun et al. [17] studied the cooling rates of a stainless steel sphere surrounded by inert gas mixtures to determine the thermal accommodation coefficient. By knowing the heat capacity, net cooling rate, the temperature of the sphere, and ambient gas temperature in the vessel, as well as the pressure, the thermal accommodation coefficient was determined. The net cooling rate was calculated by subtracting the cooling rate in hard vacuum from the cooling rate at the desired pressure. Temperatures and pressures used in this experiment were not reported, but the Knudsen
number ranged from 0.5 to 5.5, which place the experiments in the transitional regime. The thermal accommodation coefficients were analyzed for helium, argon, and various mixture ratios of the two. The thermal accommodation coefficients for helium and argon were determined to be 0.395 and 0.913, respectively, and serve as bounding values for the mixtures. The thermal accommodation coefficients for 75% helium-25% argon mixture, 50% helium-50% argon mixture, and 25% helium-75% argon mixture were determined to be 0.620, 0.718, and 0.819 respectively. The thermal accommodation coefficient values for helium and argon on a stainless steel surface compare well to other experiments discussed previously. The works provided in this paper provide interesting information on the thermal accommodation coefficients of gas mixtures. The values of the thermal accommodation coefficient of mixtures vary almost linearly from the lowest unmixed to the highest unmixed thermal accommodation coefficient.

Thomas and Loyalka [18] also used the approach of studying the cooling rates of a sphere surrounded by rarefied inert gases to determine thermal accommodation coefficients. The experiment consisted of a zirconium alloy-2 sphere of diameter 0.635 cm suspended by two dissimilar metal wires to create a thermocouple. The zirconium alloy-2 sphere was spaced by a small distance away from a tungsten heating element and the experiment was enclosed in a Pyrex envelope, which was connected to a vacuum system. The experiment was conducted by turning on the tungsten heating element and allowing the zirconium alloy-2 sphere to reach a temperature of 40°C or higher. The pressures used in the experiment categorized the gases to be in the transitional regime. An analytical solution of rarefied heat transfer for a sphere was used to determine the thermal accommodation coefficient. The researchers determined the value of the thermal accommodation coefficient of helium, argon, and xenon. The thermal accommodation coefficient of helium, argon, and xenon on zirconium alloy-2 were determined to be 0.34, 0.91, and 1.08
respectively. Since the value of thermal accommodation coefficient is bounded by 0 and 1, a thermal accommodation coefficient higher than one suggests that the molecules bouncing off a solid surface were exchanging more thermal energy than possible. The researchers provided some possible explanations for error; a non-uniform temperature on the sphere, use of constant thermal properties for calculations, as well as thermal expansion from the ideal spherical geometry caused by the tungsten heater. The values of thermal accommodation coefficient on zirconium alloy-2 for helium and argon compare well against the values for stainless steel.

2.2. Mathematical and Computational Models for Thermal Accommodation Coefficient Determination Based on Temperature Jump Coefficient

To determine the thermal accommodation coefficient of gas or relevant rarefied gas parameters, an accurate physics model must be chosen. The Navier-Stokes equations provide a conventional model of gas fluid properties at continuum pressures, but the model is not valid in rarefied gas conditions. The most commonly used rarefied gas physics model is the Boltzmann equation and is used to describe the statistical behavior of a thermodynamic transport system. Simplifications must be made to the Boltzmann equation for it to be solved. Simplifications that were used by researchers are outlined by Kennard [19] and Bird [1].

Other research has been conducted using computational approaches for determining the thermal accommodation coefficient of gases on surfaces. By modeling rarefied gas dynamics using accurate intermolecular potentials, a representation of a real-world system can be achieved. There are a few aspects of modeling that pose challenges, such as determining what intermolecular potential is accurate for the desired use case, as well as computational restrictions which limit the domain size.
Loyalka [20] formulated a temperature jump equation based on thermal accommodation coefficients that were not diffuse. Previous works did not investigate thermal accommodation coefficient values lower than 1, and thus the calculated temperature jump coefficients did not yield accurate results. Loyalka solved the non-dimensional linearized Boltzmann equation using a vibrational method and simplified the collision model by using the model created by Bhatnagar-Gross-Krook (BGK) model [21]. The geometry selected for this problem consisted of a cooled or heated plate embedded in a gas. Loyalka found that previous equations used to determine temperature jump can still be used, but slight corrections to the equations were necessary to yield appropriate temperature jump values. With this modification, Loyalka suggested that continuum equations like Navier-Stokes can be used for higher Knudsen number flows than previously used when using the corrected equations.

Lin and Willis [22] analytically investigated a temperature difference model between parallel plates filled with polyatomic gases and compared it to that of the monoatomic temperature difference model. The Boltzmann equation was selected with the simplifications to the collision term by using the BGK model. The top and bottom plates were fixed at different reference temperatures. The temperature contours were analyzed to extract temperature jump and temperature jump coefficients. The results show that the model developed for polyatomic gases had a relative error of less than 0.5% when compared to the model developed to be used for monoatomic gases. The researchers suggest that using the monoatomic gas model for calculating temperature jumps between parallel plates was a good simplification over the polyatomic model since it yields similar results.

Sharipov et al [23] calculated the temperature jump coefficient for mixtures of noble gases between a parallel plate geometry. The top and bottom plates were held at fixed dissimilar
temperatures, which created a small temperature gradient through the gas. Mixtures used in the
calculation included Ne-Ar, He-Ar, and He-Xe. The researchers started with the Boltzmann
equation for solving rarefied gas-particle dynamics and used the McCormack model kinetic
equation [24], which was described as a realistic potential, and was a simplification to the collision
term of the Boltzmann equation. The temperature jump coefficients of each gas mixture using the
McCormack model were compared against the hard-sphere model. The researchers found that the
temperature jump coefficient was strongly sensitive to intermolecular potentials, which the
collision potential was directly responsible for controlling. The McCormack model was suggested
to provide a more accurate temperature jump calculation for mixture ratios of gases since the hard-
sphere model does not accurately model intermolecular collisions. For gas mixtures where the
molecular mass ratios are low, the temperature jump coefficient does not vary significantly from
that of a monoatomic gas. But for high molecular mass ratio gases like He-Xe, the temperature
jump coefficient increases by about 20% based on the mixture ratio.

Sha et al [25] developed a computational model that used a popular molecular dynamics
code called LAMMPS. The code simulated helium and argon gas molecules interacting with a
constant temperature aluminum substrate. The intermolecular potential selected for these
simulations is the well-known Lennard-Jones potential. The incident gas molecule temperatures
were reported and the simulation was repeated many times to develop an average thermal
accommodation coefficient value. Thermal accommodation coefficient values of helium on
aluminum as well as argon on aluminum were found to range from -1.52 to 2.41 and 0.81 to 1.40
respectively, based on the equation used. While the thermal accommodation coefficient for a real
system should remain between 0 and 1, this indicates that some equations and combining rules
used to calculate thermal accommodation coefficient were leading to inconsistent results. The
Fender-Halsey combining rule in conjunction with the third equation from their findings of thermal accommodation coefficient yielded the most realistic results, where helium on aluminum had a thermal accommodation coefficient of 0.37 and a thermal accommodation coefficient of 0.86 for argon on aluminum. While these results seem promising, the simulations had to be repeated 10,440 times for helium and 8,184 times for argon to yield converged solutions. These computational efforts suggest that molecular dynamics codes could be benchmarked against experimental results and with more computational power, larger systems could be modeled.

Experimental, analytical, and computational research has been performed to determine the thermal accommodation coefficient of gases on solid surfaces. A summary of these results is given in Table 2.1. All research carefully analyzed geometry, temperature, and pressure effects when determining thermal accommodation coefficients. Accurate mathematical models that captured relevant physics were derived. Lastly, advancements in computational resources will allow thermal accommodation coefficient determination using molecular dynamics codes to be computationally feasible.
Table 2.1: Summary of thermal accommodation coefficients obtained from the literature for stainless steel.

<table>
<thead>
<tr>
<th>Surface Metal</th>
<th>Temperature Range</th>
<th>Geometry</th>
<th>Gas Type</th>
<th>Thermal Accommodation Coefficient</th>
<th>Surface Finish</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stainless Steel 304 [14]</td>
<td>22°C</td>
<td>Parallel Plate</td>
<td>Helium</td>
<td>0.36</td>
<td>Machined</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.39</td>
<td>Polished</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Nitrogen</td>
<td>0.80</td>
<td>Machined and Polished</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Argon</td>
<td>0.87</td>
<td>Machined and Polished</td>
</tr>
<tr>
<td>Stainless Steel 304 [15]</td>
<td>170°C</td>
<td>Parallel Plate</td>
<td>Helium</td>
<td>0.67-0.58</td>
<td>Not specified</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.87-0.79</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Argon</td>
<td>0.96-0.88</td>
<td></td>
</tr>
<tr>
<td>Stainless Steel 304 [17]</td>
<td>No specified</td>
<td>Cooling Sphere</td>
<td>Helium</td>
<td>0.395</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>75% Helium</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>25% Argon</td>
<td>0.62</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>50% Helium</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>50% Argon</td>
<td>0.718</td>
<td></td>
</tr>
<tr>
<td></td>
<td>25% Helium</td>
<td>75% Argon</td>
<td>0.819</td>
<td></td>
<td></td>
</tr>
<tr>
<td>-----</td>
<td>------------</td>
<td>-----------</td>
<td>-------</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Argon</td>
<td></td>
<td></td>
<td>0.913</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Chapter 3: Experimental Setup

The objective of this thesis is to determine the temperature jump and thermal accommodation coefficients of different gases on a stainless steel surface. Previous experimental efforts for determining these coefficients were outlined in Chapter 2. The experiment was designed with a few parameters in mind; accurate measurement of temperatures and pressure over a wide range (~40 Pa to ~102,000 Pa), small diameter-to-length aspect ratio that minimizes heat losses through the ends, and adaptability for use with different gases. The experimental apparatus consisted of three sections, 1) a cylindrical section which consists of the inner and outer cylinders as well as water jacket and pass-through connections, 2) a vacuum tree where vacuum pump and pressure gauges are connected, and 3) a water evaporation section. In this section, a brief description of the experiment is given, for more details, please refer to [26]

3.1. Geometry and Dimensions

The cylindrical section maintains a concentric geometry with a gap of 2mm between the cylinders. Figure 3.1 shows a bottom view of the inner cylinder. A Nexthermal 12800 high watt cartridge heater is surrounded by a thick 6061 aluminum cylinder, which is also surrounded by a thin sheath made of 304 stainless steel. The aluminum cylinder was inserted in the stainless steel sheath by heat shrink fit process; the aluminum cylinder was cooled to -150°C while the stainless steel sheath was placed in a furnace and heated up to 427°C, the aluminum cylinder was then allowed to slide into position. After the aluminum cylinder had been shrink fit to the stainless steel sheath, the cartridge heater was bonded to the aluminum cylinder using Loctite 7088, which filled any gaps between the aluminum cylinder and cartridge heater and provides good thermal contact.
Figure 3.1: Bottom view of the inner cylinder geometry comprised of a cartridge heater, a thick aluminum cylinder, and a thin stainless steel sheath.

Figure 3.1 also shows twelve slits and three holes in the aluminum cylinder. The slits were created along the aluminum cylinder to allow for thermocouple wire placement. The three holes in the aluminum cylinder in a triangular pattern allow the support system to be attached to the inner cylinder. The dimensions of the inner cylinder components are outlined in Table 3.1.

Figure 3.2: Bottom view of the inner cylinder with the support piece attached.
Table 3.1: Inner cylinder component dimensions of the cylindrical section.

<table>
<thead>
<tr>
<th>Component</th>
<th>Outer Diameter [cm]</th>
<th>Inner Diameter [cm]</th>
<th>Length [cm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nexthermal 12800 High Watt Cartridge Heater</td>
<td>1.240 ± 0.003</td>
<td>-</td>
<td>105.4 ± 0.1</td>
</tr>
<tr>
<td>Aluminum Cylinder</td>
<td>4.448 ± 0.003</td>
<td>1.27 ± 0.02</td>
<td>103.1 ± 0.1</td>
</tr>
<tr>
<td>Stainless Steel Sheath</td>
<td>4.548 ± 0.003</td>
<td>4.45 ± 0.02</td>
<td>103.1 ± 0.1</td>
</tr>
</tbody>
</table>

Figure 3.2 shows the 304 stainless steel support system that is used to support the inner cylinder and maintains a 2 mm annular gap between the inner and outer cylinders. It was also designed to minimize the heat losses from the ends by minimizing the contact area.

Figure 3.3: Inner cylinder attached to the outer cylinder system, with thermocouple and cartridge heater power wires.
The outer cylinder system was designed for two important functions; 1) to act as a pressure vessel and 2) to maintain a constant outer temperature. The innermost cylinder of the outer cylinder system was made of 304 stainless steel and acts as the pressure vessel. Grooves were machined on the outer surface of the stainless steel pressure vessel similar to the inner cylinder and to place twelve thermocouples to measure the temperature of the outer cylinder. Around the pressure vessel is an annular gap and is surrounded by a stainless steel cylinder. The gap is filled with water using inlet and outlet connections which are connected to a Lytron Recirculating chiller with hoses. The outer cylinder thermocouples exit the experiment through a silicone covered port in the water jacket cylinder. The dimensions of the outer cylinder components are outlined in Table 3.2.

Table 3.2: Outer cylinder component dimensions.

<table>
<thead>
<tr>
<th>Component</th>
<th>Inner Diameter [cm]</th>
<th>Outer Diameter [cm]</th>
<th>Length [cm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure Vessel</td>
<td>4.75 ± 0.02</td>
<td>5.071 ± 0.003</td>
<td>147.3 ± 0.1</td>
</tr>
<tr>
<td>Water Jacket</td>
<td>6.02 ± 0.02</td>
<td>6.350 ± 0.003</td>
<td>113.2 ± 0.1</td>
</tr>
</tbody>
</table>

At both ends of the pressure vessel were large open sections sealed using conflat flanges. The top end has a vacuum pass-through connection for the inner cylinder thermocouples and cartridge heater power wires. The bottom end is connected to the vacuum tree section. Figure 3.4 depicts a longitudinal cross-section view of the cylindrical section of the experimental apparatus. Conflat flange pass-through and vacuum connections on top and bottom were not represented for simplicity.
Figure 3.4: (a) Full cross-section and (b) zoomed views of the cylindrical section of the experimental apparatus schematic.

Figure 3.5 shows a picture of the vacuum tree section with all the hardware attached to it. Attached to the vacuum tree are a gas inlet valve, capacitance manometers, water evaporation section, and a Bellows valve which connects to the turbo vacuum pump. Table 3.3 lists the components and specifications of vacuum tree components.
Figure 3.5: Vacuum tree section with capacitance manometers, gas inlet, vacuum pump, and water evaporation section.

Table 3.3: Vacuum tree valve component and gas specifications.

<table>
<thead>
<tr>
<th>Component</th>
<th>Quantity</th>
<th>Range</th>
<th>Leak Rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pfiffer HiCube 80 Vacuum Pump</td>
<td>1</td>
<td>&lt;7.5x10^{-8} Torr</td>
<td>-</td>
</tr>
<tr>
<td>Manual Bellows Sealed SS Inline Valve</td>
<td>1</td>
<td>1x10^{-10} Torr</td>
<td>2.9 x 10^{-9} scc/sec</td>
</tr>
<tr>
<td>Fujikin Diaphragm Soft-Seal Valve</td>
<td>1</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Swagelok Metering Valve</td>
<td>1</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>High Purity Argon 99.995%</td>
<td>21C</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Ultra High Purity Nitrogen 99.999%</td>
<td>300C</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Ultra High Purity Helium 99.999%</td>
<td>300C</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>
3.2. Instrumentation

The two types of data collected from the experiment were temperature and pressure data. Thermocouples were instrumented along the length of the experiment at different radial locations to measure the temperature data while capacitance manometers measured pressure data. Twelve type-K thermocouples were instrumented on the inner cylinder, twelve on the outer cylinder, one at the top support, one at the top flange of the outer cylinder, one measuring the ambient room temperature, and six on the outer surface of the outer cylinder. For the experiments conducted with argon, nitrogen, and water vapor, two thermocouples were not operational, one thermocouple located on the inner cylinder and the other on the outer cylinder.

The capacitance manometers used in the experiment were selected for their accuracy and full-scale ranges. Connected to the vacuum tree were two pressure gauges; a 100 Torr and 1000 Torr gauges. The 100 Torr gauge was used to measure low-pressures and the 1000 Torr was used to measure high-pressures. Additionally, there was a 20 Torr gauge attached to the water evaporation section but was not used in data interpretation. The specifications of the thermocouples and capacitance manometers used in the experiment are outlined in Table 3.4.

The thermocouples and capacitance manometers were connected to data acquisition systems which feed data to a computer running LabView VI where the data was stored. Table 3.5 shows the specifications of the data acquisition system as well as components which were controlled electronically.
Table 3.4: Specifications of thermocouples and capacitance manometers.

<table>
<thead>
<tr>
<th>Component</th>
<th>Quantity</th>
<th>Range</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>OMEGA 36 AWG K-Type Glass Braid</td>
<td>34</td>
<td>-200 - 482°C</td>
<td>± 1.1°C</td>
</tr>
<tr>
<td>Insulated SLE TC Wire</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MKS Baratron Absolute Capacitance</td>
<td>1</td>
<td>0 - 1000 Torr</td>
<td>± 0.25% of reading</td>
</tr>
<tr>
<td>Manometer 626C</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>MKS Baratron Absolute Capacitance</td>
<td>1</td>
<td>0 - 20 Torr</td>
<td>± 0.25% of reading</td>
</tr>
<tr>
<td>Manometer 622C</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>INFICON CDG025D Digital Capacitance Diaphragm</td>
<td>1</td>
<td>0 - 100 Torr</td>
<td>± 0.20% of reading</td>
</tr>
<tr>
<td>Gauge</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3.5: Data acquisition, power supply, and control systems.

<table>
<thead>
<tr>
<th>Component</th>
<th>Quantity</th>
<th>Range</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Agilent 34970A Data Acquisition</td>
<td>2</td>
<td>-</td>
<td>± 1.0°C</td>
</tr>
<tr>
<td>Agilent 34970A Module, 20 Channels</td>
<td>4</td>
<td>-</td>
<td>± 1.5°C</td>
</tr>
<tr>
<td>Agilent N5770A Power Supply</td>
<td>1</td>
<td>150V, 10A, 1500W</td>
<td>0.125 – 0.20 V</td>
</tr>
<tr>
<td>Lytron Recirculating Chiller</td>
<td>1</td>
<td>5 - 35°C</td>
<td>± 0.5%</td>
</tr>
</tbody>
</table>
3.3. Water Vapor Injection System

Injecting water vapor into the experiment required a different approach compared to injecting helium, argon, or nitrogen. A water vapor injection system was developed to evaporate water and inject it into the experiment. Figure 3.6 shows the water vapor injection system that connects to the experiment through the vacuum tree. The water vapor injection system has a water storage volume where liquid water is loaded, valves controlled how much water entered the water vapor chamber. Prior to opening of the valves, the chamber is vacuumed to the lowest achievable pressure using the vacuum pump. The valves are then opened and water was allowed to enter the evaporation chamber. The pressure in the chamber is monitored using the 20 Torr pressure gauge. Small amounts of water are injected into the chamber such that the vapor pressure remained below saturation. A final diaphragm valve between the chamber and experiment is opened to release water vapor into the experiment. Table 3.6 shows a parts list for the water vapor injection section.

Figure 3.6: Picture of the water vapor injection system.
Table 3.6: Water vapor injection part list.

<table>
<thead>
<tr>
<th>Component</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mayhems Ultra Pure Water Purity 99.999%</td>
<td>1</td>
</tr>
<tr>
<td>Kurt J. Lesker TAR4CS100QF Vacuum Trap</td>
<td>1</td>
</tr>
<tr>
<td>Swagelok VASS4HVCR Diaphragm Valve</td>
<td>2</td>
</tr>
<tr>
<td>Fujikin Diaphragm Soft-Seal Valve</td>
<td>1</td>
</tr>
</tbody>
</table>

3.4. Experiment Modifications

Modifications have been made to the experiment to add some functionality. One end of the cartridge heater is capped with a stainless steel plug, while the other end containing the power wire leads did not have a plug. The end of the cartridge heater without a plug had magnesium oxide exposed to the internal volume of the experiment, and it needed to be sealed with a cap since magnesium oxide is a porous material and would absorb water. Modifications were made to remove some of the magnesium oxide material and a stainless steel plug was installed. The plug was sealed with Duraseal High-Temp Silicone and ceramic beads were used as conductive insulation for the power wires.
Chapter 4: Analytical Solution of Heat Transfer

An analytical solution of heat transfer that includes the effect of rarefaction between two long concentric cylinders is derived. This solution will be used to compare to experimental data to determine temperature jump and thermal accommodation coefficients.

4.1. Concentric Cylinder Solution of Heat Transfer

For rarefied gas flows classified in the slip regime, the convective heat transfer is valid but additional boundary conditions must be added to account for temperature jump at the gas-solid interfaces. The heat transferred due to conduction in between concentric cylinders can be analyzed using a thermal resistance model. The difference in temperature between concentric cylinders is noted by

$$\Delta T = T_{in} - T_{out}$$  \hspace{1cm} (4.1)

where $T_{in}$ is the temperature of the inner cylinder and $T_{out}$ is the temperature of the outer cylinder. The difference in temperature between solid surfaces, $\Delta T$, is a function of the heat transferred due to conduction, $Q_c$, and the sum of the thermal resistances, $R_i$ [27], as

$$\Delta T = Q_c \sum R_i$$  \hspace{1cm} (4.2)

The thermal resistance due to gas conduction, $R_c$, for concentric cylinders can be formulated as

$$R_c = \frac{\ln \left( \frac{r_{out}}{r_{in}} \right)}{2\pi L \kappa}$$  \hspace{1cm} (4.3)

where $r_{in}$ is the inner cylinder radius, $r_{out}$ is the outer cylinder radius, $L$ is the length of the cylinders, and $\kappa$ is the average thermal conductivity of the gas [27].
Using the kinetic theory [28], the thermal conductivity of a monatomic gas can be related to viscosity as

\[ \kappa = \frac{15 k_B}{4 m \mu} \]  

(4.5)

where \( k_B \) is the Boltzmann Constant \( (k_B = 1.3806 \times 10^{-23} m^2 kgs^{-2} K^{-1}) \), \( m \) is the molecular mass of gas in kilograms, and \( \mu \) is the gas viscosity. For a polyatomic gas, the thermal conductivity can be expressed as [29]

\[ \kappa = \mu c_v \frac{(9\gamma - 5)}{4} \]  

(4.6)

where \( c_v \) is the specific heat at constant volume, and \( \gamma \) is the ratio of specific heats. Appendix A gives the values for specific heat at constant volume as well as the ratio of specific heat for gases analyzed in this thesis.

The viscosity \( \mu \), of a gas can be expressed using the power law as [1]

\[ \mu = \mu_0 \left(\frac{T}{T_0}\right)^{\omega} \]  

(4.4)

where \( \mu_0 \) is the reference viscosity value at the reference temperature, \( T_0 \) \((T_0= 273.15 \text{ K})\), \( T \) is the gas temperature, and \( \omega \) is the viscosity index which is dependent on the kinetic theory approach as well as gas type [28]. The simplest approach set forth by Bird is to assume that gas molecules will have a predictable collision behavior, similar to that of hard-sphere collisions. For this approach, all momentum is transferred between gas molecules since they do not experience deformation. This approach is called the hard-sphere (HS) approach, and the viscosity index is simplified to \( \omega = 0.5 \) for all gases. The second and more realistic approach is to assume that gas molecules have a variable collisional diameter. Trajectories of gas particles after a collision can also be predicted using this approach. This approach is called the variable hard sphere (VHS)
approach and the viscosity index is modulated based on gas type. Appendix A states the reference viscosity as well as viscosity index values used for gases analyzed in this thesis.

For slip regime gas flows, an additional thermal resistance occurs at gas-solid interfaces. This thermal resistance leads to steep temperature gradients at gas-solid interfaces and is classified as a temperature jump. The thermal resistance due to temperature jump, $R_j$, can be formulated as

$$R_j = \frac{\lambda \zeta_T}{2\pi LrK}$$

(4.7)

where $\lambda$ is the mean free path, and $\zeta_T$ is the temperature jump coefficient. The extent of temperature jump is dependent on gas parameters like thermal conductivity and temperature jump coefficient, as well as mean free path, and surface area. Substituting Eqs. 4.3 and 4.7 into Eq. 4.2 leads to

$$\Delta T = Q_c \left( \frac{\lambda_{in} \zeta_{T,in}}{2\pi L r_{in} K_{in}} + \frac{\ln \frac{r_{out}}{r_{in}}}{2\pi L K} + \frac{\lambda_{out} \zeta_{T,out}}{2\pi L r_{out} K_{out}} \right)$$

(4.8)

It is important to note though, that in a concentric cylinder configuration, temperature jump occurs on both the inner and outer surfaces, which is why the following Eq. 4.8 has three thermal resistance terms. In Eq. 4.8, subscripts $in$ and $out$ are used to denote the parameters that were evaluated at the inner and outer surfaces, respectively. The other mechanism of heat transfer between concentric cylinders is the radiation heat transfer. One-dimensional heat transfer due to radiation was calculated using the following equation [27].

$$Q_R = \frac{A_{in} \sigma (T_{in}^4 - T_{out}^4)}{1 - \varepsilon_{in}} + \frac{1 - \varepsilon_{out}}{\varepsilon_{out} T_{out}^4}$$

(4.9)
4.2. Direct Determination of Temperature Jump and Thermal Accommodation Coefficients

For the sake of simplicity, the mean free path and thermal conductivity on the inner and outer surfaces are calculated as an average value. Similarly, the temperature jump coefficients on the inner and outer surfaces are assumed to be equal. Using these simplifications, Eq. 4.8 becomes,

$$\Delta T = \frac{Q_c \mu}{2\pi L_k} \zeta_T \left( \frac{1}{r_{in}} + \frac{1}{r_{out}} \right) \sqrt{\frac{2k_B T_{out}}{m \rho} \frac{1}{P} + \frac{Q_c}{2\pi L_k} \ln \left( \frac{r_{out}}{r_{in}} \right)}$$  \hspace{1cm} (4.10)

Equation 4.10 is used to calculate the temperature jump coefficient based on experiment geometry, gas species, temperature, pressure, and conduction heat transfer. It should be pointed out that the equation has two terms on the right-hand side, the first is temperature difference due to gas rarefaction and the second term is temperature difference due to gas conduction. While this equation can be solved directly for temperature jump coefficient for a given temperature, pressure, heat generation, and gas species, it can only be used in the slip regime. For the continuum regime, the temperature jump coefficient cannot be determined because the temperature jump is negligible.

To relate the temperature jump coefficient to the thermal accommodation coefficient, many models have been developed in the literature. Using the model developed by Lin & Willis [30] for monoatomic and polyatomic gases, the temperature jump coefficient can be expressed as

$$\zeta_T = \left( \frac{2 - \alpha}{\alpha} + 0.17 \right) \frac{\sqrt{\pi} \gamma}{(\gamma + 1) \Pr}$$  \hspace{1cm} (4.11)

where \( \Pr \) is the Prandtl number [27]. After the temperature jump coefficient, \( \zeta_T \), is determined using Eq. 10, the thermal accommodation coefficient, \( \alpha \), can be calculated using Eq. 4.11.
4.3. Linear Regression Determination of Temperature Jump and Thermal Accommodation Coefficients

Linear regression is an error minimization technique that is used to model a relationship between two sets of parameters which are assumed to vary linearly. From Eq. 4.10, geometry and gas type parameters remain constant while parameters like heat by conduction, temperature difference, and pressure can be varied experimentally. The experimental approach was to maintain the supplied heat generation at a specified value while pressure was varied and the temperature difference is monitored.

Combining Eqs. 4.10 and 4.11, the temperature difference can be expressed as

\[
\Delta T = \frac{Q_c}{2c_pL(y + 1)} \left( \frac{2 - \alpha}{\alpha} + 0.17 \right) \left( \frac{1}{r_{in}} + \frac{1}{r_{out}} \right) \sqrt{\frac{2k_B T_{out}}{m}} \frac{1}{P} + \frac{Q_c}{2\pi L\kappa} \ln \left( \frac{r_{out}}{r_{in}} \right)
\]  

(4.12)

It can be seen from this equation that the temperature difference is inversely proportional to the pressure. Therefore, the experimental data of temperature difference can be fitted using a linear regression expression as follows

\[
\Delta T = a \frac{1}{P} + b
\]  

(4.13)

where \( a \) is the slope and \( b \) is the intercept. The slope from Eq. 4.13 can be equated with the first term of Eq. 4.12 as

\[
a = \frac{Q_c}{2c_pL(y + 1)} \left( \frac{2 - \alpha}{\alpha} + 0.17 \right) \left( \frac{1}{r_{in}} + \frac{1}{r_{out}} \right) \sqrt{\frac{2k_B T_o}{\pi m}}
\]  

(4.14)
Since the value of $a$ can be determined from linear regression, the only unknown term in Eq. 4.14 is the thermal accommodation coefficient. From the thermal accommodation coefficient, the temperature jump coefficient can be calculated using Eq. 4.11.

### 4.4. Uncertainty Analysis on Temperature Jump and Thermal Accommodation Coefficients

The calculation of uncertainty on temperature jump and thermal accommodation coefficients is just as important as calculating the coefficients themselves. For the uncertainty analysis, the propagation of uncertainty approach is adopted [31]. The uncertainty on each parameter is determined from the uncertainty on physical measurements, uncertainty provided by manufacturers, or uncertainty on material properties. The uncertainties on each measurement is then propagated to the uncertainty on temperature jump and thermal accommodation coefficients. Using Eq. 4.14, the uncertainty on the thermal accommodation coefficient is calculated from the regression method as

$$
\left( \frac{\omega_a}{\alpha} \right)^2 = \left( \frac{\omega F}{F} \right)^2 + \left( \frac{\omega G}{G} \right)^2
$$

(4.17)

where the variable $F$ is a substitution variable and is equal to

$$
F = \frac{Q_c}{c_p L (\gamma + 1)} \left( \frac{1}{r_i} + \frac{1}{r_o} \right) \sqrt{\frac{2 k_B T_0}{\pi m}}
$$

(4.18)

and the variable $G$ is a substitution variable and is equal to

$$
G = a + 0.83 F
$$

(4.19)

The uncertainty on $F$ can be calculated using the following equation.
\[
\left( \frac{\omega_F}{F} \right)^2 = \left( \frac{\omega_{Qc}}{Q_c} \right)^2 + \left( \frac{\omega_L}{L} \right)^2 + \left( \frac{\sqrt{\left( \frac{\omega_{r_l}}{r_l^2} \right)^2 + \left( \frac{\omega_{r_o}}{r_o^2} \right)^2}}{\left( \frac{1}{r_l} + \frac{1}{r_o} \right)^2} \right)^2 + \left( \frac{1}{2} \frac{\omega_{T_o}}{T_o} \right)^2 \tag{4.20}
\]

And the last term in calculating the uncertainty on thermal accommodation coefficient is the uncertainty on G

\[
\left( \frac{\omega_G}{G} \right)^2 = \omega_a^2 + \left( \frac{0.83 Q_c}{2c_p L} \frac{\gamma}{\gamma + 1} \left( \frac{1}{r_l} + \frac{1}{r_o} \right) \sqrt{\frac{2k_B T_o}{\pi n}} \right)^2 \left( \frac{\omega_F}{F} \right)^2 \tag{4.21}
\]

From the calculation of uncertainty on the thermal accommodation coefficient, the uncertainty on the temperature jump coefficient can also be calculated.

\[
\left( \omega_{\zeta_T} \right)^2 = \left( \frac{\sqrt{\pi} \gamma}{(\gamma + 1) Pr \alpha^2} \omega_\alpha \right)^2 \tag{4.22}
\]
Chapter 5: Results and Discussion

In this chapter, the results of rarefied gas heat transfer from the concentric cylinder experimental apparatus and corresponding CFD simulations are presented. Experiments were conducted for different backfill gases, heat generation rates, and pressures. The gases selected for these experiments are helium, argon, nitrogen, and water vapor. The heat generation rates are chosen based on the experiment temperature limitations and are different for each gas. The heat generation ranged from 100 – 500W for helium, 50 – 100W for argon, and only one heat generation of 100W was used for nitrogen and water vapor. The pressures for each gas were selected such that they cover the continuum and slip regimes. For helium, the pressure ranged from 110 – 102,000 Pa. For argon and nitrogen, the pressure ranged from 40 – 102,000 Pa. For water vapor, the pressure ranged from 40 – 2,500 Pa. The maximum pressure of 2,500 Pa was used for water vapor because it is slightly below the saturation pressure of water at the temperature of 22°C. For all experiments, the outer cylinder wall was held at a constant temperature of 22 ± 1°C using a circulation water chiller.

5.1 Experimental Results

When conducting experiments, the inner cylinder temperature was set at near room temperature and the pressure was stabilized around the target pressure for that case. After turning the heating element on, the temperature and pressure increased until they reached a steady state. Steady-state is assumed to be reached when the slope of temperature and pressure variation is zero or close to zero.

Figure 5.1 shows a typical temperature and pressure temporal response for nitrogen at $Q=100\text{W}$ and $P = 1837\text{Pa}$. The temperature and pressure are then averaged over the last few hours. The pressure gauge has a resolution of ~0.5 Pa, and this can be seen with some data points in Fig.
5.1. All the experimental results presented in this chapter are the averaged steady-state temperatures and pressures.

![Graph showing temperature and pressure response over time for Nitrogen at Q = 100 W and P = 1837 Pa.]

**Figure 5.1:** Temperature and pressure response over time for Nitrogen at $Q = 100$ W and $P = 1837$ Pa.

### 5.1.1 Inner and Outer Cylinder Axial Temperature Profiles

As it was mentioned before, both inner and outer cylinders are instrumented with twelve thermocouples at three axial locations. Using these thermocouples, the axial temperature profile for the inner and outer cylinders can be generated. Figure 5.2 shows the steady-state axial temperature profiles for helium at the lowest and highest heat generations and pressures.
Figure 5.2: Temperature versus axial location for helium at the lowest and highest heat generations and pressures. The x symbols are for $Q=500\text{W}$ and open square symbols are for $Q=100\text{W}$. The dashed lines represent the average outer wall temperature, and the solid lines represent the temperature profiles from the CFD simulations.

The thermocouples are placed at three axial locations along the length, $z = -27.5$, 0, 27.5 cm. In each axial location, there are four thermocouples spaced by a 90° angle. It should be noted that in Fig. 5.2, the thermocouples at each axial location are spaced out by 1 cm such that each data point can be distinguished. The data points indicated by $T_o$ represent the steady-state outer wall temperatures, which are nearly the same for all cases. The dashed lines that pass through these points are the calculated averages. All the other data points represent the inner cylinder temperatures for two heat generations at the lowest and highest pressures. It can be seen from this figure that the temperature along the axial direction is nearly constant for each case (pressure and
heat generation) with variation less than the size of the uncertainty (error bars were omitted here for clarity). This figure also shows that at the highest pressure, $P \sim 10^5$ Pa, corresponding to the continuum regime, the temperature of the inner cylinder increases with heat generation as expected. Also, there is an increase in temperature as the pressure is decreased to $\sim 100$ Pa (corresponding to the limit between the slip and transitional regimes, $Kn = 0.1$) for each heat generation. However, this increase is more significant for the highest heat generation. This increase in inner cylinder temperature due to the decrease in pressure is due to the rarefaction effect of the gas, which creates a temperature-jump thermal resistance at the wall. This thermal resistance increases with the heat generation as shown by Eq. 4.7.

Figures 5.3, 5.4, and 5.5 show comparisons between the axial temperature profiles of helium and argon, nitrogen, and water vapor, respectively, at the same heat generation of $Q = 100$W, for the lowest and highest pressures. The highest pressures chosen for this comparison are in the continuum regime for all gases, where minimal effect of rarefaction can occur. However, the lowest pressures used in the comparisons correspond to the limit of the slip and transitional regimes at $Kn = 0.1$ for each gas.

Similar to Fig. 5.2, the steady-state outer cylinder temperatures are indicated by $T_o$ and are nearly the same for all cases shown in Figs. 5.3, 5.4, and 5.5. Since helium has a higher thermal conductivity than argon, nitrogen, and water vapor, the inner cylinder temperatures measured for helium are lower than that of argon, nitrogen, and water vapor. When the pressure was decreased from $P \sim 10^5$ Pa to the pressure corresponding to $Kn \sim 0.1$ ($P = 104$ Pa for helium, 37Pa for argon, 38 Pa for nitrogen, and 47 Pa for water vapor), the inner cylinder temperature increased due to the rarefaction effect (temperature jump). This increase is nearly the same for argon and nitrogen. However, for water vapor the increase was relatively small.
Figure 5.3: Axial temperature profiles for helium and argon at $Q = 100$ W at the lowest and highest pressures. The x symbols are for argon and open square symbols are for helium. The dashed lines represent the average outer wall temperature, and the solid lines represent the temperature profiles from the CFD simulations.
Figure 5.4: Axial temperature profiles for helium and nitrogen at $Q = 100$ W at the lowest and highest pressures. The x symbols are for nitrogen and open square symbols are for helium. The dashed lines represent the average outer wall temperature, and the solid lines represent the temperature profiles from the CFD simulations.
Figure 5.5: Axial temperature profiles for helium and water vapor at $Q = 100$ W at the lowest and highest pressures. The x symbols are for water vapor and open square symbols are for helium. The dashed lines represent the average outer wall temperature, and the solid lines represent the temperature profiles from the CFD simulations.

5.1.2 Effect of Gas Rarefaction

Figure 5.6 shows the difference between the steady-state average temperatures of the inner and outer cylinders as a function of pressure for helium at five heat generation rates in the range of 100 – 500W. The symbols represent the experimental data. From this figure, it can be seen that the temperature difference between the inner and outer cylinder walls increases with heat generation. In the continuum regime ($P > 10000$Pa), and for a given heat generation, the temperature difference remains relatively constant. This is because, in the continuum regime, the
effects of gas rarefaction is small. However, one can notice that there is a slight increase in temperature difference as the pressure decreases in the continuum regime, especially at high heat generations. This can be attributed to a slight effect of rarefaction as the pressure decreases. In the slip regime \((P < 10000\text{Pa})\), the temperature difference increases as the pressure decrease. This increase in temperature is due to rarefaction effect, and it is larger as the heat generation increases.

Figures 5.7 and 5.8, show the average temperature difference as a function of pressure for argon and nitrogen. The limit between the continuum and slip regimes for argon and nitrogen is at lower pressures than helium, because these gases are heavier than helium. Similar to helium, there is a slight increase in temperature difference in the continuum regime. However, as the pressure decreases, the temperature difference increases significantly, due to the effects of gas rarefaction.

Figure 5.9 shows the average temperature difference as a function of pressure for water vapor. The maximum pressure for water vapor was limited to \(~1700\text{Pa}\) to avoid condensation of water vapor. However, this pressure is in the continuum regime. The experiments with water vapor are more difficult to conduct compared to the other gases, due to the tendency of water vapor to condense and adsorb to the surface. Also, these experiments take significantly more time to reach steady state due to the lower thermal conductivity of water vapor (see Appendix A), and the adsorption, desorption effects. Figure 5.9 shows some preliminary results with water vapor. It shows the expected tendency of the increase of temperature difference as the pressure decreases in the slip regime. However, more data are needed to confirm these findings. Also, these experiments must be repeated several times to ensure that the data is repeatable.
Figure 5.6: Experimental steady state average temperature differences as a function of pressure for helium at five heat generation rates, \( Q = 100, 200, 300, 400, \) and \( 500 \) W. The data points with squares, diamonds, triangles, circles, and x symbols are experimental data, and the solid and dashed lines connect simulated data points.
Figure 5.7: Experimental steady state average temperature differences as a function of pressure for argon at two heat generation rates, $Q = 50$ and $100 \text{W}$. The data points with squares, and triangles are from the experimental results, and the solid and dashed lines connect simulated data points.
Figure 5.8: Experimental steady state average temperature differences as a function of pressure for nitrogen at the heat generation rate of $Q = 100\text{W}$. The square symbols represent the experimental data, and the solid line connect simulated data points.
Figure 5.9: Experimental steady state average temperature differences as a function of pressure for water vapor at the heat generation rate of $Q = 100 \text{W}$. The square symbols represent the experimental data.

To compare the effect of rarefaction between different gases, the non-dimensional temperature difference $\frac{\Delta T - \Delta T_{\text{Cont}}}{\Delta T_{\text{Cont}}}$, where $\Delta T_{\text{Cont}}$, is the average temperature difference in the continuum regime, is plotted as a function of the rarefaction parameter $\delta$ defined as

$$
\delta = \frac{\sqrt{\pi} \frac{L_c}{\lambda} \sim 1}{Kn}
$$

(5.1)

Figure 5.10 shows the non-dimensional temperature difference as a function of the gas rarefaction parameter for helium, argon, and nitrogen at $Q = 100 \text{ W}$. The non-dimensional
temperature is used to analyze the effects of gas rarefaction on the different gases at a given rarefaction level.

In the continuum regime $\delta >1000$, the non-dimensional temperature difference for all three gases is equal to zero, as expected. This is due to the minimal effect of rarefaction effect at high pressures. In the slip regime $\delta <1000$, all the three gases experience rarefaction effect. However, the effect of rarefaction is much larger on helium than argon or nitrogen. The effect of rarefaction on helium is about three times higher. This is believed to be due the smaller size of the molecule of helium compared to argon and nitrogen. Helium molecules tends to bounce off the surface in a more specular way than other gases. Therefore, it accommodates less to the surface, which reduces the energy exchange between helium molecules and the surface, leading to larger temperature jumps at the wall. One also can notice from Fig. 5.10 that both argon and nitrogen experience nearly the same relative level of rarefaction. This is also due to the size of the molecules of argon and nitrogen, which have nearly the same collisional diameter. It should be noted here that nitrogen is a diatomic molecule, while argon is a monatomic molecule.
Figure 5.10: Average temperature difference minus continuum temperature difference divided by continuum temperature difference as a function of rarefaction parameter $\delta$ for helium, argon, and nitrogen at $Q = 100$ W.

5.2 Determination of Temperature Jump and Thermal Accommodation Coefficients from Experimental Data

Using the analytical solution of heat transfer between concentric cylinders, Eqs. 4.10 and 4.11, the thermal accommodation and temperature jump coefficients can be calculated from the experimental data. Figures 5.11 – 5.14 show the experimental average temperature differences plotted as a function of the inverse of pressure for helium, argon, nitrogen, and water vapor, respectively. These data are then fitted with a linear equation using the regression method. The
slope and the intercept of the linear equations are shown on the figures. The value of the slope is then used in Eq. 4.14 to determine the temperature jump and thermal accommodation coefficients.

Figure 5.11: Helium temperature difference as a function of the inverse of pressure fitted with linear regressions and equations.
Figure 5.12: Argon temperature difference as a function of the inverse of pressure fitted with linear regressions and equations.

For $Q = 100\text{W}$, the equation is:

$$\Delta T = \frac{1062.4}{P} + 62.954$$

For $Q = 50\text{W}$, the equation is:

$$\Delta T = \frac{557.99}{P} + 32.866$$
Figure 5.13: Nitrogen temperature difference as a function of the inverse of pressure fitted with linear regressions and equations.

\[ Q = 100 \text{W}, \Delta T = 725.86/P + 45.006 \]
Figure 5.14: Water vapor temperature difference as a function of the inverse of pressure fitted with linear regressions and equations.

Table 5.1 – 5.3 gives the temperature jump and thermal accommodation coefficients calculated from experimental data for helium, argon, and nitrogen, respectively. The uncertainties on these coefficients are also included in these tables and were estimated using Eqs. 4.17 and 4.22 with a 95% confidence level. To calculate these coefficients, the portion of heat transferred by conduction $Q_c$, between the cylinders must be estimated. It must be noted that after conducting these experiments, some heat was unaccounted for. The loss in heat was attributed to slight eccentricity of the cylinders, which lowered the thermal resistance of the cylinder side with a larger gap, as well as heat lost through the ends. While it was impossible to measure the eccentricity of
the cylinders while the experiment was assembled or taken apart, estimations were made to account for these losses in heat. To estimate it, we first calculate an amount of heat transfer due to radiation (see Eq. 4.9) for all pressure at a given heat generation and gas. We then estimate the heat losses from the ends of the inner cylinder at the continuum regime by comparing the experimental data of temperature difference to the 1D analytical solution of heat transfer by conduction and radiation through the gap (see Eq. 4.3). The sum of the average radiation heat transfer and the end heat losses is then subtracted from the total heat generation and is defined as

\[ Q_c = Q - Q_L - Q_R, \]  

(5.2)

where \( Q_L \) is the sum of the end heat losses and eccentricity. It should be noted that the estimated end heat losses in the continuum regime represent the minimum loss that would occur. As the pressure in the experiment decreases and the rarefaction effect becomes significant, the heat losses from the ends of the internal cylinder increase due to the higher resistance to conduction heat transfer through the gap.

The results of the thermal accommodation coefficient show that the values obtained for helium are lower than those of argon and nitrogen as seen in Figs. 5.15, 5.16, and 5.17. This suggests that argon and nitrogen molecules accommodate better to the wall temperature than helium. This is consistent with the results from previous investigations in the literature (see Chapter 2). It was found that the thermal accommodation coefficient for lighter molecules is smaller than that of heavier molecules. Comparing Fig. 5.16 to Fig. 5.17, it can be seen that the values of the thermal accommodation coefficient for argon are slightly higher than those of nitrogen. This is also consistent with the data in the literature.
Table 5.1: The thermal accommodation ($\alpha$) and temperature-jump ($\zeta_T$) coefficients with their uncertainties for all heat generation rates of helium using regression approach.

<table>
<thead>
<tr>
<th>$Q$ [W]</th>
<th>$Q_c$ [W]</th>
<th>$T_{in,avg}$ [°C]</th>
<th>$\zeta_T$</th>
<th>$\pm w_{\zeta_T}$</th>
<th>$\alpha$</th>
<th>$\pm w_{\alpha}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>92.1</td>
<td>33.27</td>
<td>8.15</td>
<td>0.73</td>
<td>0.35</td>
<td>0.03</td>
</tr>
<tr>
<td>200</td>
<td>184.2</td>
<td>45.63</td>
<td>7.42</td>
<td>0.52</td>
<td>0.38</td>
<td>0.02</td>
</tr>
<tr>
<td>300</td>
<td>276.1</td>
<td>57.26</td>
<td>8.15</td>
<td>0.62</td>
<td>0.35</td>
<td>0.02</td>
</tr>
<tr>
<td>400</td>
<td>367.9</td>
<td>68.67</td>
<td>7.96</td>
<td>0.60</td>
<td>0.36</td>
<td>0.02</td>
</tr>
<tr>
<td>500</td>
<td>459.5</td>
<td>80.92</td>
<td>7.32</td>
<td>0.70</td>
<td>0.38</td>
<td>0.03</td>
</tr>
</tbody>
</table>

Table 5.2: The thermal accommodation ($\alpha$) and temperature-jump ($\zeta_T$) coefficients with their uncertainties for all heat generation rates of argon using regression approach.

<table>
<thead>
<tr>
<th>$Q$ [W]</th>
<th>$Q_c$ [W]</th>
<th>$T_{in,avg}$ [°C]</th>
<th>$\zeta_T$</th>
<th>$\pm w_{\zeta_T}$</th>
<th>$\alpha$</th>
<th>$\pm w_{\alpha}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>43.2</td>
<td>58.56</td>
<td>2.13</td>
<td>0.42</td>
<td>0.94</td>
<td>0.11</td>
</tr>
<tr>
<td>100</td>
<td>85.6</td>
<td>91.93</td>
<td>2.05</td>
<td>0.22</td>
<td>0.97</td>
<td>0.06</td>
</tr>
</tbody>
</table>

Table 5.3: The thermal accommodation ($\alpha$) and temperature-jump ($\zeta_T$) coefficients with their uncertainties for all heat generation rates of nitrogen using regression approach.

<table>
<thead>
<tr>
<th>$Q$ [W]</th>
<th>$Q_c$ [W]</th>
<th>$T_{in,avg}$ [°C]</th>
<th>$\zeta_T$</th>
<th>$\pm w_{\zeta_T}$</th>
<th>$\alpha$</th>
<th>$\pm w_{\alpha}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>88.2</td>
<td>72.59</td>
<td>2.26</td>
<td>0.22</td>
<td>0.88</td>
<td>0.05</td>
</tr>
</tbody>
</table>

Table 5.4: The thermal accommodation ($\alpha$) and temperature-jump ($\zeta_T$) coefficients with their uncertainties for all heat generation rates of water vapor using regression approach.

<table>
<thead>
<tr>
<th>$Q$ [W]</th>
<th>$Q_c$ [W]</th>
<th>$T_{in,avg}$ [°C]</th>
<th>$\zeta_T$</th>
<th>$\pm w_{\zeta_T}$</th>
<th>$\alpha$</th>
<th>$\pm w_{\alpha}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>86.66</td>
<td>84.32</td>
<td>1.74</td>
<td>0.26</td>
<td>1.01</td>
<td>0.09</td>
</tr>
</tbody>
</table>
Figures 5.15, 5.16, and 5.17 show that the individual approach produced thermal accommodation coefficient values that do not bracket the regression method values, this is because the individual method was performed using data from individual experiments, while the regression method used 10-12 experiments. The experimental results shown in Fig. 5.15 show regression determinations of thermal accommodation coefficient are bracketed by literature results. Though the results in Figs. 5.16 and 5.17 show that the thermal accommodation coefficients for argon and nitrogen are larger ion compared to literature results. These results suggest that the experimental apparatus is sensible to the thermal accommodation of the different gases, and it might be suitable for measuring the thermal accommodation coefficient of water vapor. Table 5.4 shows the preliminary results of the temperature jump and thermal accommodation coefficient for water vapor. The value of the thermal accommodation coefficient obtained for water vapor is slightly larger than unity. However, considering the uncertainty, the determination of thermal accommodation coefficient of water is within the acceptable range of values. From this result, it seems that water vapor molecules fully accommodate the temperature of the wall. However, more data are needed to confirm this preliminary result.
Figure 5.15: Helium thermal accommodation coefficients versus temperature. The dashed and solid lines represent the thermal accommodation coefficient value obtained using the regression method. The open square, diamond, triangle, circle, and x symbols are the thermal accommodation coefficient values for the five lowest pressures in a heat generation and the closed symbols are data collected from literature sources: Trott [14], Byung [17], Loyalka [18], and Song & Yovanovich [31].
Figure 5.16: Argon thermal accommodation coefficients vs. temperature. The dashed and solid lines represent the thermal accommodation coefficient value using the regression method. The open box and diamond symbols are the thermal accommodation coefficient values for the five lowest pressures in a heat generation and the closed symbols are data collected from literature sources: Trott [14], Byung [17], Loyalka [18], and Song & Yovanovich [31].
Figure 5.17: Nitrogen thermal accommodation coefficients vs. temperature. The dashed and solid lines represent the thermal accommodation coefficient value using the regression method. The open box and diamond symbols are the thermal accommodation coefficient values for the five lowest pressures in a heat generation and the closed symbols are data collected from literature source: Trott [14].

5.3 Comparison of Computational Fluid Dynamics Simulations with Experimental Results

A computational fluid dynamics (CFD) model of the experimental apparatus was created in ANSYS package and simulations were conducted in ANSYS Fluent for comparison with experimental data. During creation of this model, a few assumptions and simplifications were made based on geometry, materials, and boundary conditions. Only one third of the experiment was modeled in 3D space based on symmetry of the experiment. Figure 5.18 shows an isometric view of the one-third 3D model.
The thermocouple wires and grooves as well as the thermocouple vacuum pass-through fittings were not included in the model. Small geometry features in the support structures were modified to a simplistic geometry with similar volume and surface area. The model was then discretized with a total of 562,212 mesh elements. Some material properties like the thermal conductivity of 304 stainless steel, were obtained from the literature, while other properties, like emissivity, were determined from experiments conducted by Sandia National Laboratory. The material properties used in the simulations are outlined in Appendix A. The boundary conditions used in the model included a uniform heat generation along the heating element and a constant outer wall temperature. Small variations in heat generation from the heating element as well as temperature variations on the outer cylinder wall were assumed to be negligible.
Figure 5.19 shows typical temperature contours of the simulated region. The temperature gradient across the solid region are very small. The largest temperature difference is obtained across the gap region. The simulation results showed that the estimated heat losses through the ends is of the order of 1 – 2% of the total heat generation. However, this percentage does not account for heat losses due to eccentricity of the experimental apparatus. It was estimated that up to 6% of heat loss may be induced by the non-perfect concentricity of the inner and outer cylinders by comparing the experimental data to an analytical solution of one-dimensional heat transfer between concentric cylinders.
Figure 5.19: (a) Cross-sectional temperature contour at the axial center, $z = 0$ cm, and (b) isometric temperature contour plot obtained from the ANSYS Fluent CFD simulations for Argon at $Q=100$ W and $P=37$ Pa.

Experimental steady state outer cylinder temperature, pressure, and thermal accommodation coefficient were used as inputs for simulations. Each steady state simulation was allowed to run until the solution reached convergence.

The solid lines in Fig. 5.2 show the inner cylinder axial temperature profiles obtained from the simulation results for helium at the highest and lowest heat generations and pressures. These profiles exhibit a nearly flat (constant) temperature in the central region. However, a larger gradient
is obtained near the ends of the inner cylinder, which can be attributed to the end heat losses. Comparing the simulations and experimental results, one can notice that there is a good agreement between them. Similar observation can be made for Fig. 5.3, except that for argon at $P = 37$ Pa, the simulations slightly under predicted the measured temperatures. This may be due a slight over prediction of the thermal accommodation coefficient for argon. From Fig. 5.4, simulated axial temperature profile for nitrogen is lower than experimental results at $P = 38$ Pa. This result can be attributed to an over prediction of the thermal accommodation coefficient for nitrogen. And lastly, Fig. 5.5 shows simulated axial temperature profiles for water vapor. The simulated water vapor profile at $P = 1710$ Pa is below the experimental results, and this indicates that the thermal conductivity of water vapor used in the simulation was too high. More work will be completed to understand the effects of temperature and pressure on thermal conductivity of water vapor.

Figures 5.6 – 5.9 show the simulated temperature difference as a function of pressure for helium, argon, nitrogen, and water vapor respectively. Similar to the experimental results, the simulated temperature difference for all gases exhibit a nearly constant temperature in the continuum regime. However, the temperature difference increases as the pressure decreases in the slip regime. From these figures, it can also be seen that the simulations results compare well with the experimental results for helium and argon for all heat generation rates and pressure. However, they under predict the temperature difference in the case of nitrogen in the slip regime. It seems that the thermal accommodation coefficient for nitrogen was over predicted using the method described in the previous section. Contrary to nitrogen, the simulation results in water vapor under predict temperature difference in the continuum regime. More investigation is needed to understand these discrepancies.
Chapter 6: Conclusion and Future Work

Temperature jump becomes significant when the mean free path becomes comparable to characteristic length of the system. Temperature jump and thermal accommodation coefficients describe how much temperature jump exists at gas-solid interfaces and how much thermal energy is exchanged between the gas and the solid surface. Determination of the temperature jump and thermal accommodation coefficients are important for several applications, including, nuclear fuel storage, vehicle reentry from space, Aerogels, and fabrication of MEMS.

Previous experimental determinations of the temperature jump and thermal accommodation coefficients were conducted in the free molecular and transitional regimes where the Boltzmann equation was used to directly solve for these coefficients. The value of these coefficients for helium, argon, and nitrogen have been determined by previous experimental works, but only a few measurements have been taken in the slip regime. The objective of this thesis was to experimentally determine the temperature jump and thermal accommodation coefficients of helium, argon, nitrogen, and water vapor in the slip regime.

A concentric cylinder vacuum experimental apparatus was used to collect temperature and pressure measurements of heat transfer through a rarefied gas. The cylinders were spaced by a 2 mm annular gap along their lengths. In separate experiments, the gap was filled with either helium, argon, nitrogen, or water vapor. Heat was supplied to the inner cylinder by a heating element, and thermocouples were placed on the inner as well as outer cylinders. The outer cylinder was maintained at room temperature using a water jacket with a recirculating chiller. Pressure of the gas between the cylinders was measured using capacitance manometers. The experiments were conducted until steady state was achieved. Experiments were conducted with helium, argon,
nitrogen, and water vapor at pressures ranging from ~40 to 102,000 Pa and heat generations ranging from 50 to 500W.

A one-dimensional analytical solution of heat transfer between concentric cylinders was derived and used to determine the effect of gas rarefaction for helium, argon, nitrogen, and water vapor on a stainless steel surface. Difference in temperature between the inner and outer cylinders showed an inverse linear relationship with pressure, which was used to create a linear regression model.

By comparing the experimental data to the analytical solution of heat transfer, the thermal accommodation coefficient values were determined to be 0.36±0.03 for helium, 0.96±0.06 for argon, 0.88±0.05 for nitrogen, and 1.01±0.09 for water vapor. These values were in close agreement with previous works from the literature. Since no literature has investigated water vapor, this thesis provides a preliminary determination of temperature jump and thermal accommodation coefficient for water vapor.

A 3D CFD model of the experimental apparatus was created in ANSYS package and simulations were carried out in ANSYS Fluent. The results from simulations were compared to the experimental measurements. The values of the thermal accommodation coefficients determined form the experimental data were used in the simulations. The comparison showed a good agreement with the experimental measurement for helium, nitrogen, and argon. However, for water vapor some discrepancies were found that need some further investigations to discern the reasons.
Future Work

Further experiments will be conducted with pure water vapor to decrease the uncertainty on temperature jump and thermal accommodation coefficients. The determinations from this thesis can be used for rough estimations of gas mixtures for research purposes. Some modifications will be made to the experiment to allow for future experimental determinations of gas mixtures like: helium-water vapor, argon-water vapor, and nitrogen-water vapor.
Appendix A: Material Properties

Table A.1: Material properties of Magnesium Oxide

<table>
<thead>
<tr>
<th>Material Properties</th>
<th>Magnesium Oxide Cement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density [kg/m³]</td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>5380</td>
</tr>
<tr>
<td>Specific Heat [J/kg*K]</td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>877</td>
</tr>
<tr>
<td>Thermal Conductivity [W/m*K]</td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>2.07</td>
</tr>
</tbody>
</table>

Table A.2: Material properties of Stainless Steel

<table>
<thead>
<tr>
<th>Material Properties</th>
<th>304 Stainless Steel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density [kg/m³]</td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>8030</td>
</tr>
<tr>
<td>Specific Heat [J/kg*K]</td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>502.48</td>
</tr>
<tr>
<td>Thermal Conductivity [W/m*K]</td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>16.27</td>
</tr>
<tr>
<td>Emissivity of Inner Cylinder</td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>0.152</td>
</tr>
<tr>
<td>Emissivity of Outer Cylinder</td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>0.149</td>
</tr>
<tr>
<td>Emissivity of Support Structures</td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>0.3</td>
</tr>
</tbody>
</table>

Table A.3: Material properties of Aluminum

<table>
<thead>
<tr>
<th>Material Properties</th>
<th>6061 Aluminum</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density [kg/m³]</td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>2719</td>
</tr>
<tr>
<td>Specific Heat [J/kg*K]</td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>871</td>
</tr>
<tr>
<td>Thermal Conductivity [W/m*K]</td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>202.4</td>
</tr>
<tr>
<td>Emissivity</td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>0.2</td>
</tr>
</tbody>
</table>
### Table A.4: Material properties of Helium

<table>
<thead>
<tr>
<th></th>
<th>Ultra-Pure Helium</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Density [kg/m³]</strong></td>
<td>Incompressible Ideal Gas</td>
</tr>
<tr>
<td><strong>Specific Heat [J/kg*K]</strong></td>
<td>Constant</td>
</tr>
<tr>
<td><strong>Thermal Conductivity [W/m*K]</strong></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C1 0.02688713</td>
</tr>
<tr>
<td></td>
<td>C2 0.0005454916</td>
</tr>
<tr>
<td></td>
<td>C3 -5.376243e-07</td>
</tr>
<tr>
<td></td>
<td>C4 5.268244e-10</td>
</tr>
<tr>
<td></td>
<td>C5 -2.348067e-13</td>
</tr>
<tr>
<td><strong>Viscosity [kg/m*s]</strong></td>
<td>Temperature Power Law</td>
</tr>
<tr>
<td></td>
<td>µ₀ 1.865e-5</td>
</tr>
<tr>
<td></td>
<td>T₀ 273.15</td>
</tr>
<tr>
<td></td>
<td>ω 0.66</td>
</tr>
<tr>
<td><strong>Molecular Weight [kg/kmol]</strong></td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>4.002602</td>
</tr>
</tbody>
</table>

### Table A.5: Material properties of Argon

<table>
<thead>
<tr>
<th></th>
<th>Ultra-Pure Argon</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Density [kg/m³]</strong></td>
<td>Incompressible Ideal Gas</td>
</tr>
<tr>
<td><strong>Specific Heat [J/kg*K]</strong></td>
<td>Constant</td>
</tr>
<tr>
<td><strong>Thermal Conductivity [W/m*K]</strong></td>
<td></td>
</tr>
<tr>
<td></td>
<td>C1 0.006141397</td>
</tr>
<tr>
<td></td>
<td>C2 4.760566e-5</td>
</tr>
<tr>
<td></td>
<td>C3 -4.057822e-8</td>
</tr>
<tr>
<td></td>
<td>C4 2.054317e-11</td>
</tr>
<tr>
<td><strong>Viscosity [kg/m*s]</strong></td>
<td>Temperature Power Law</td>
</tr>
<tr>
<td></td>
<td>µ₀ 2.117e-5</td>
</tr>
<tr>
<td></td>
<td>T₀ 273.15</td>
</tr>
<tr>
<td></td>
<td>ω 0.81</td>
</tr>
<tr>
<td><strong>Molecular Weight [kg/kmol]</strong></td>
<td>Constant</td>
</tr>
<tr>
<td></td>
<td>39.948</td>
</tr>
</tbody>
</table>
Table A.6: Material properties of Nitrogen

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Incompressible Ideal Gas</td>
<td>Constant</td>
<td>1040.67</td>
<td>Temperature Polynomial</td>
<td>C1</td>
<td>0.003436044</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C2</td>
<td>8.473207e-5</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C3</td>
<td>-5.028743e-8</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C4</td>
<td>2.67857e-11</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Temperature Polynomial</td>
<td>C1</td>
<td>1.983588e-6</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C2</td>
<td>6.524964e-8</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C3</td>
<td>-5.817077e-11</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C4</td>
<td>6.207832e-11</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C5</td>
<td>-3.079481e-11</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Temperature Polynomial</td>
<td>C1</td>
<td>2.36775e-5</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C2</td>
<td>-1.92927e-7</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C3</td>
<td>7.96924e-10</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C4</td>
<td>-1.23828e-12</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C5</td>
<td>7.19844e-16</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Constant</td>
<td>28.0134</td>
<td></td>
</tr>
</tbody>
</table>

Table A.7: Material properties of Water Vapor

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Incompressible Ideal Gas</td>
<td>Temperature Piecewise-Polynomial</td>
<td>2014</td>
<td>Temperature Polynomial</td>
<td>C1</td>
<td>0.0013216</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C2</td>
<td>0.000128117</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C3</td>
<td>-5.46981e-7</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C4</td>
<td>1.20238e-9</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C5</td>
<td>-5.33411e-13</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Temperature Power Law</td>
<td>C1</td>
<td>2.36775e-5</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C2</td>
<td>-1.92927e-7</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C3</td>
<td>7.96924e-10</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C4</td>
<td>-1.23828e-12</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>C5</td>
<td>7.19844e-16</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Constant</td>
<td>18.01534</td>
<td></td>
</tr>
</tbody>
</table>
References


