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Heat Pipes with Arbitrary Boundary Conditions

A Dissertation Presented for the
Doctor of Philosophy
Degree
The University of Tennessee, Knoxville

Katrina Marie Sweetland

August 2022

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ABSTRACT

Heat pipes passively transfer heat in numerous applications. Traditionally one side of the heat pipe is coupled to a heat source (evaporator) while the opposite side is coupled to a heat sink (condenser). This configuration has working fluid stagnation points at each end of the heat pipe. Other configurations may also prove useful, such as heat pipes with multiple evaporators or multiple condensers. In such heat pipes, additional working fluid stagnation points form at locations dependent on the configuration of the thermal boundary conditions. These stagnation points divide the heat pipe into multiple cells that each have an evaporator and condenser. The total thermal power input and output distributes between these cells. This distribution decreases the length that the working fluid must travel as well as the working fluid velocity. This reduces the pressure drop of the vapor and liquid circulating through the heat pipe allowing it to operate at higher power densities with the same spatial footprint.

To characterize the behavior of stagnation points in such heat pipes, a copper water heat pipe was built and tested with arbitrary boundary conditions. In addition, a non-condensable gas was added at the heat pipe ends to allow insight into the vapor flow direction. The amount of non-condensable gas was either constant or controlled with a pressure controller. Throughout testing the heat pipe behaved in a stable and repeatable manner so long as the number of evaporator (hot) stagnation points was less than or equal to those in the previous state. Forming an additional hot stagnation point during a passive gas loaded heat pipe test led to most of the gas remaining on one side of the hot stagnation point. The results of these tests show stagnation points appear to form rapidly when thermal boundary conditions change. Small fluctuations in temperatures near the working fluid vapor and non-condensable gas interface were observed. These changes were caused by changes in the ambient temperature and the feedback system in the pressure controllers and were not observed when the condenser conditions were better controlled such as in the passive gas-loaded calorimeter tests.

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CHAPTER ONE

INTRODUCTION AND GENERAL INFORMATION

The document consists of six chapters starting with the introduction. In the introduction general information on the topic presented in this document is introduced and the scope of the work is discussed. The second chapter consists of a Theory chapter that discusses the operation of the heat pipes as well as behavior predictions for multiple evaporator or multiple condenser heat pipes and gas loaded heat pipes. A literature review follows the theory chapter to establish previous work and the state of the art of this technology. The fourth chapter discusses the methodology behind the heat pipe design, fabrication, test set-up. This chapter includes uncertainty analysis for the uncertainties presented in the following results chapter. The results chapter presents the key data collected during testing. It is divided into six sections depending on the type of cooling used on the heat pipe as well as the type of heat pipe test that was conducted. The last chapter consists of conclusions and future work. Here the conclusions are summarized, and a list of future work is compiled.

General Information

Heat pipes passively remove heat from systems where active heat removal methods may not be the best option. There are many types of heat pipe designs, but each design typically consists of an outer sealed tube, a wick, and a working fluid.¹ The wick returns the liquid working fluid from the condenser to the evaporator using capillary pressure. Wicks allow heat pipes to operate in horizontal and low gravity environments unlike thermosyphons, which operate similarly to a heat pipe, but use gravity to return the liquid to the evaporator. Wicks may be constructed in numerous ways, including incorporation of

mesh and grooves. Heat is transferred from one side of the heat pipe to the other side of the heat pipe by evaporating and condensing the working fluid. The working fluid for a heat pipe is chosen based on properties such as surface tension and latent heat of vaporization. To ensure the heat pipe is capable of long-term operation, the outer casing and wick material are often chemically compatible with the working fluid.

Figure 1.1 depicts operation of a fixed conductance heat pipe (FCHP) with a single evaporator and a single condenser. Operating a heat pipe requires a heat input region, the evaporator, and a heat output region, the condenser. Traditionally heat pipes have one evaporator and one condenser on opposite sides of the pipe. During operation, the wick transfers the liquid working fluid from the condenser section to the evaporator section by capillary forces. Once in the evaporator section, the liquid working fluid evaporates. The vapor in the evaporator moves along the interior of the heat pipe towards the condenser section. At the condenser section, the vapor condenses into a liquid and saturates the wick.

The position of the condenser controls the direction of the vapor and liquid flow as well as the distance they must travel in the heat pipe. If the number and positions of the condenser(s) change, the vapor and liquid flows change. Another way to alter the flow of the working fluid is to control the condenser's ability to condense vapor. In the case of a fixed conductance heat pipe, the length of the condenser that is actively condensing vapor remains close to the same for all temperatures and thermal loads after the heat pipe has reached steady state. However, the addition of a non-condensable gas or extra working fluid into the heat pipe leads to the active length of the condenser to be coupled with the temperature of the heat pipe and the thermal load that has been applied. This type of heat pipe is often referred to as a variable conductance heat pipe (VCHP). In this work non-condensable gas is introduced into the heat pipe to achieve this.

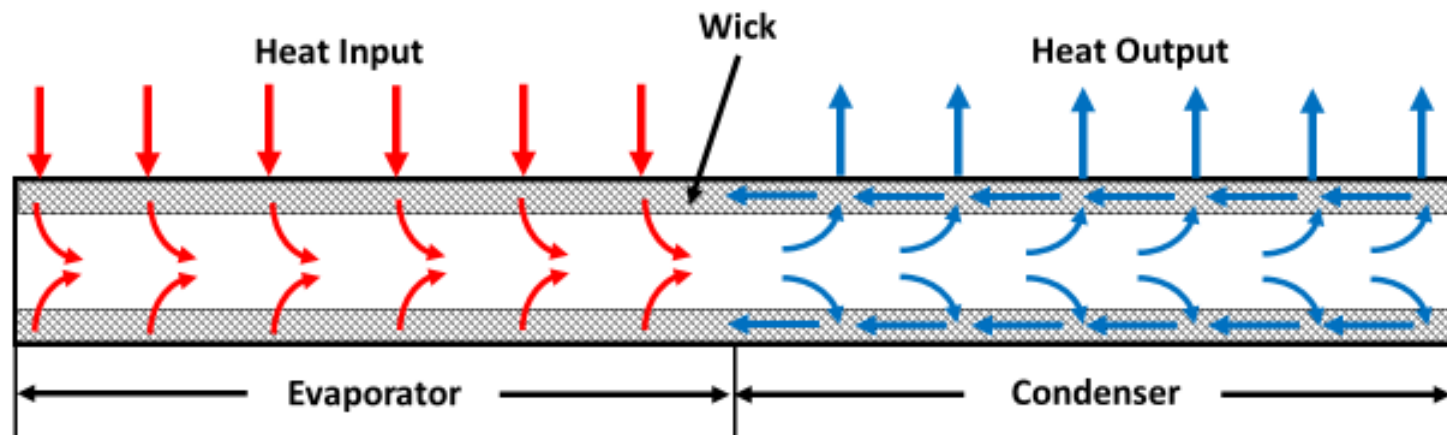


Figure 1.1: Operation of a single evaporator and a single condenser fixed conductance heat pipe.

Scope of Work

Although the electronics industry uses heat pipes with multiple condensers, there are literature on heat pipes with multiple condensers. Other than instances in the electronics industry, multiple condenser heat pipes have not yet been widely implemented. When compared to heat pipes that have a single evaporator and condenser, heat pipes with multiple condensers use space more efficiently and have improved thermal capacity. When additional condensers (heat sinks) or evaporators (heat sources) are acting on a heat pipe, counter rotating cells form. An example of this is shown in Figure 1.2.

In Figure 1.2, two counter rotating cells are surrounding a point where the velocity of the vapor flow is zero (stagnation point). This stagnation point in the vapor flow can be used to help determine the behavior of the heat pipe. The number of stagnation points, n_ξ , in a heat pipe equals the sum of separate condensers and evaporators. In a simple analysis the working fluid pressure drops associated with a traditional heat pipe were compared to the pressure drops for a heat pipe with multiple evaporators or multiple condensers. This simple analysis found that the working fluid pressure losses decreased by a maximum of $(n_\xi - 1)^2$ and $(n_\xi - 1)^{2.75}$ for laminar and turbulent flow, respectively. Although these are significant, it is important to note that in practice the increased capillary limit may be less due to radial effects such as injection and suction through the wick.

The goal of this work is to use non-condensable gas to provide insight into the formation and dissolution of counter rotating cells in a heat pipe. Additionally, the behavior of the addition of non-condensable gas to multiple evaporator or multiple condenser heat pipes is investigated. The addition of non-condensable gas into the heat pipe allows for a heat pipe to have a variable active condenser length and a lower temperature increase with increasing thermal power loads. Under lower thermal power conditions, the active condenser length is typically shorter than when the heat pipe is under higher thermal power conditions. This type of heat pipe is often referred to as a VCHP, while a heat pipe without non-condensable gas is referred to as a FCHP.

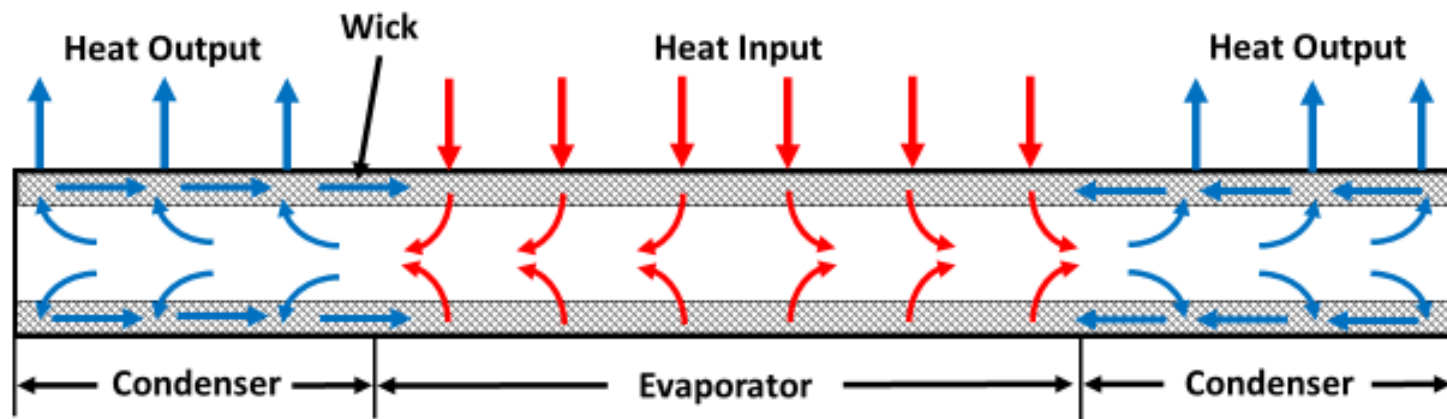


Figure 1.2: Operation of a dual condenser and single evaporator fixed conductance heat pipe.

This work examines the influence thermal power inputs and outputs (thermal boundary conditions) on heat pipes with a varying amount of non-condensable gas. Specifically, the influence of these thermal boundary condition changes on changes on the distribution of non-condensable gas and heat pipe working fluid is observed.

Tests on FCHPs observe the influence of thermal boundary condition such as constant heat flux, and a combination boundary of convection and radiation on a heat pipe. Passive VCHP tests explore the effect of the thermal input and output on the distribution of the non-condensable gas. Finally, active VCHP tests are intended to observe the influence of independent introduction and removal of non-condensable gas in heat pipes. The comparison of each of these tests provides further insight on interactions between thermal loading, stagnation point formation, and non-condensable gas distribution in heat pipes.

CHAPTER TWO

THEORY

This chapter presents the theory associated with each aspect of variable conductance heat pipes with multiple evaporators or multiple condensers. The first section discusses heat pipe operating principles to provide a knowledge base for terminology used here. The second section presents a theory on the operation and potential advantages of multiple evaporator and multiple condenser heat pipes. This theory was expanded from an unpublished document, shown in the Appendix, written by Dr. Robert S. Reid between 2013 and 2015. The last section addresses the operation and potential advantages of variable conductance heat pipes.

Heat Pipe Operation

Most heat pipes are closed systems (i.e., fixed conductance heat pipes) used to efficiently transfer heat, but they can be altered to be open systems (i.e., active controlled variable conductance heat pipes). For ease of understanding only fixed conductance heat pipes are discussed in this section and variable conductance heat pipes are discussed in the subsequent section.

Heat pipes transfer heat by evaporating and condensing a working fluid and typically consist of an outer sealed casing, a wick, and a working fluid.¹ Operating a heat pipe requires a heat input region, the evaporator, and a heat output region, the condenser. Traditionally heat pipes have one evaporator and one condenser on opposite sides of the pipe. During operation, the wick transfers the liquid working fluid from the condenser section to the evaporator section by capillary forces. Once in the evaporator section, the liquid working fluid evaporates. The vapor in the evaporator moves along the interior of

the heat pipe toward the lower pressure condenser section. At the condenser section, the vapor condenses into a liquid and refills the wick.

The wick returns the liquid working fluid from the condenser to the evaporator by producing a capillary pressure. Wicks can be constructed from several types of capillary structures, including mesh and grooves. The working fluid for a heat pipe is chosen based on properties such as surface tension and latent heat of vaporization. To ensure the heat pipe is capable of long-term operation, the outer casing and wick material must be chemically compatible with the working fluid. Additionally, the capillary pressure produced by the wick allows for operation in a horizontal orientation as well as a counter gravity orientation (where the force of gravity pulls the liquid towards the condenser). This gives the heat pipe an advantage over a thermosyphon, which operates similarly to a heat pipe, but uses gravity to return the liquid to the evaporator and cannot operate in a horizontal or counter gravity orientation.

During operation, the circulating working fluid experiences pressure losses from friction, inertia, phase changes, and body forces. To operate, the capillary pressure produced by the wick must overcome the sum off all of pressure losses experienced by the working fluid,¹ shown in Equation (2.1).

$$\Delta P_{cap,max} \geq \Delta P_v + \Delta P_l + \Delta P_b \quad (2.1)$$

In Equation (2.1), $\Delta P_{cap,max}$ is the maximum capillary pressure drop, ΔP_v is the vapor working fluid pressure drop, ΔP_l is the liquid working fluid pressure drop, and ΔP_b is the pressure drop associated with the bulk forces (e.g., gravity forces). The capillary pressure produced by the wick can be calculated using the Laplace-Young equation, shown below.¹

$$\Delta P_{cap,max} = \frac{2\sigma}{r_{eff}} \cos(\theta) \quad (2.2)$$

The losses associated with gravitational forces can be avoided by operating the heat pipe in a horizontal orientation. The phase change pressure losses are typically negligible unless there is a high rate of vaporization and condensation occurring in the heat pipe. The differences in the liquid and vapor pressure present in the heat pipe create a meniscus at the liquid-vapor interface in the wick. The meniscus has maximum curvature where the

capillary pressure is maximum (dry point), and the curvature is at a minimum where the liquid and vapor pressure is equal (wet point). It is common for a liquid pool to form at the end of the condenser after the wet point. Liquid pool formation occurs when there is excess liquid in the pipe, often from additional working fluid introduced into the heat pipe while it is being filled. As the heat pipe temperature increases, liquid density decreases leading the liquid pool to grow. However, for simplicity in the rest of this section a perfect fill is assumed, meaning there is no liquid pool formation. Figure 2.1 shows an example of the axial pressure profiles of the liquid and vapor in a heat pipe compared to the meniscus shape along the axis.

The vapor flow can be characterized by mass addition in the evaporator and mass subtraction in the condenser caused by vaporization and condensation, respectively. If gravitational and phase change losses can be neglected, the dominating vapor pressure losses can easily be identified for each of the sections in a heat pipe. The vapor pressure losses in the evaporator consist of friction losses and inertial losses from acceleration caused by the mass inflow. The vapor in the condenser section has friction losses and the ability to recover the inertial losses experienced in the evaporator. This pressure recovery only occurs under certain circumstances and can change the location of the wet point.¹

In heat pipes with an adiabatic section between the evaporator and condenser, pressure losses in the adiabatic section are friction losses. For the liquid flow, mass addition occurs at the condenser and mass subtraction occurs in the evaporator. The liquid pressure losses are primarily dominated by the frictional losses because the liquids higher density makes the influence of acceleration and deceleration comparably small.

When the pressure losses exceed the capillary pressure, the wick is unable to return the liquid phase working fluid to the evaporator region, and the wick in that region begins to dry out leading to the heat pipe no longer operating. This acts as an operational limitation for the heat pipe. The thermal power throughput at which this occurs is called the capillary limitation. These pressure losses also play a significant role in determining other operational constraints for the heat pipe.

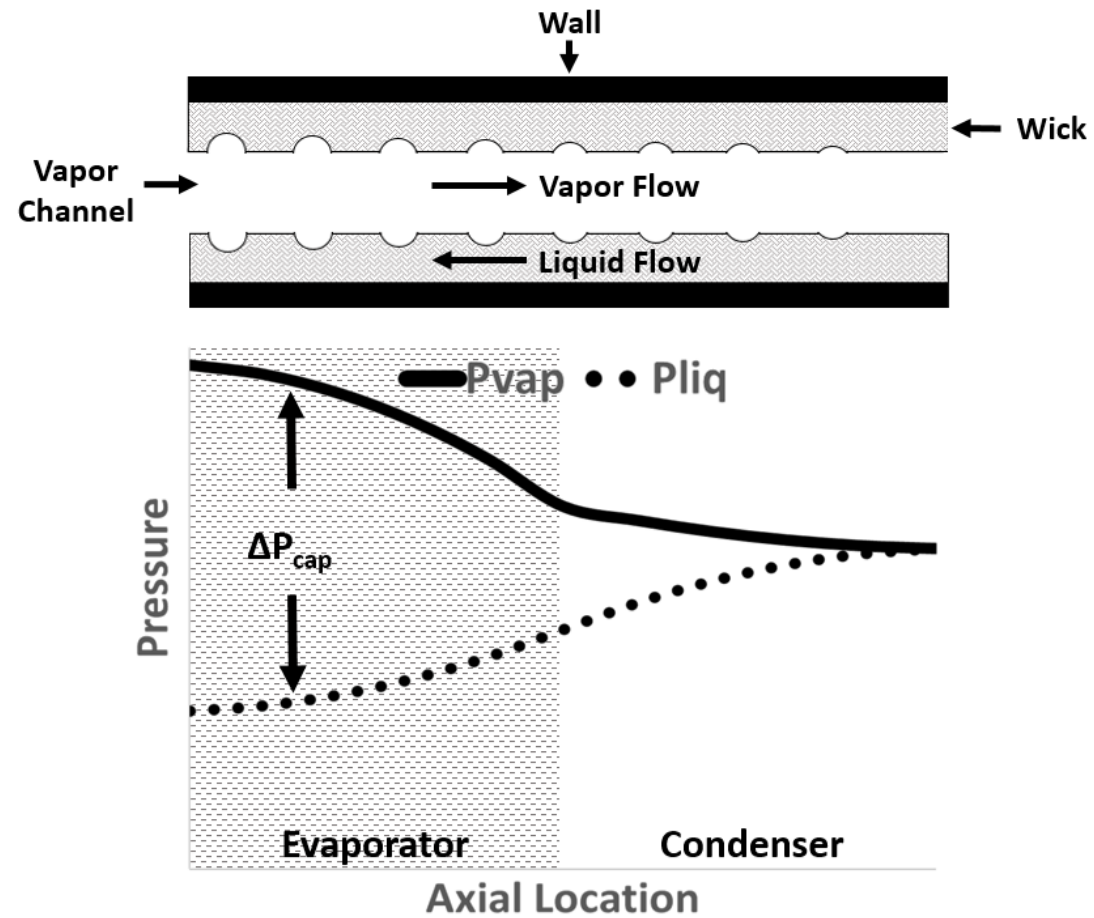


Figure 2.1: Example heat pipe working fluid pressure profile compared to meniscus curvature.

Common operational constraints of heat pipes are the viscous limit, the sonic limit, the capillary limitation, the entrainment limitation, and the boiling limitation. For clarification, the conventional terminology uses the term “limit” to describe phenomena that do not make the heat pipe lose its ability to operate, while using the term “limitation” to describe phenomena where the heat pipe is no longer able to operate.¹ Operation is limited by whichever constraint has the lowest associated thermal power. Operational heat pipes operate on a condenser (heat sink) coupling curve that is typically located below most of the power constraints. Figure 2.2 shows an example of constraints for an arbitrary heat pipe calculated using HTPIPE.⁵

The viscous limit occurs when friction acting on working fluid vapor limits working fluid circulation. This typically occurs at low temperatures during start up. It is common for high temperature heat pipes to operate with condenser coupling curves along this limit. This limit is overcome as the heat pipe starts up and reaches normal operating region. Equation (2.3) describes the viscous limit presented by Busse in 1972.⁸

$$Q_{viscous} = \frac{d_v^2 h_{fg} P_0 \rho_0}{64 \mu_v L_{eff}} \quad (2.3)$$

In Equation (2.3), $Q_{viscous}$ is the amount of thermal power applied to the pipe when the viscous limit occurs, d_v is the diameter of the vapor channel, h_{fg} is the latent heat of vaporization, P_0 is the pressure at the beginning of the evaporator, ρ_0 is the density of the vapor at the beginning of the evaporator, μ_v is the dynamic viscosity of the vapor, and L_{eff} is the effective pipe length.

For uniform heating, Ivanovskii presents L_{eff} as shown Equation (2.4).¹ In Equation (2.4) L_{ad} is the length of the adiabatic zone, L_e is the length of the evaporator, and L_c is the length of the condenser.

$$L_{eff} = L_{ad} + \frac{L_e + L_c}{2} \quad (2.4)$$

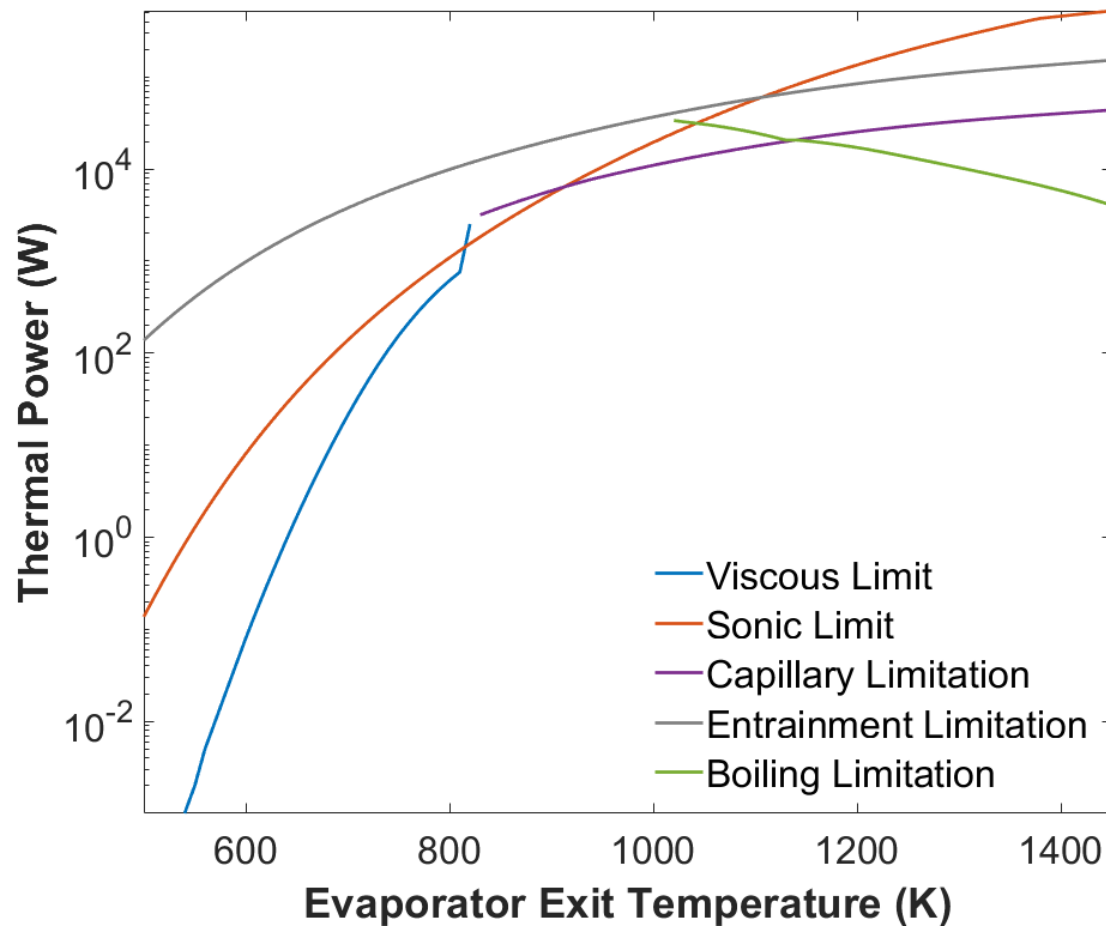


Figure 2.2: Example operational constraints for an arbitrary annular heat pipe.

Another limit that occurs at lower temperatures is the sonic limit. The sonic limit refers to the conditions where the working fluid vapor reaches the speed of sound at the evaporator exit. This chokes the flow meaning the pipe cannot transfer any additional thermal power.⁹ The sonic limit can be described by finding the thermal power associated with mass flow rate of the vapor traveling at the speed of sound when evaluated at the evaporator exit. Like the viscous limit, in high temperature heat pipes it is common to operate with condenser coupling curves along this limit as the heat pipe starts up. Ivanovskii presents this relation is shown in Equation (2.5).¹ In this equation Q_{sonic} is the amount of thermal power applied to the pipe when the sonic limit occurs, ρ_v is the density of the working fluid vapor at the evaporator exit, a is the speed of sound of the vapor at the evaporator exit, h_{fg} is the latent heat of vaporization evaluated at the evaporator exit, and A_v is the cross-sectional area of the vapor channel at the evaporator exit.

$$Q_{sonic} = \rho_v h_{fg} A_v a \quad (2.5)$$

In entrainment limitation, the vapor sweeps liquid droplets off the wick and back to the condenser, which may lead to evaporator drying. This limitation can be described by a Weber number equal to one, which occurs when the inertial forces of the vapor are equal to the liquids surface tension. The entrainment limit can be avoided by using wick geometries that shield the liquid flow from the vapor flow. Some examples of wick geometries that accomplish this are annular wick, artery wick, and screen covered grooves. The Heat Pipe Design Handbook presents the following equation for the entrainment limitation by applying the Weber number to the heat pipe.¹⁰ In Equation (2.6), σ_l is the surface tension of the liquid, and z is the characteristic dimension of the wick.

$$\left(\frac{Q}{A}\right)_{Entrainment} = \sqrt{\frac{\rho_v \sigma_l h_{fg}^2}{z}} \quad (2.6)$$

The last limitation is the boiling limitation. Typically, this limit is defined as the heat flux required for the onset of nucleate boiling in the wick or on the inner surface of the heat pipe walls.^{5,10,11} When boiling occurs within the liquid flow space, vapor bubbles reduce the heat pipes ability to transport liquid to the evaporator. This may lead to local wick dry out and hot spots in the evaporator. The boiling limit can be calculated by determining the conduction across the wick and liquid that has the temperature drop corresponding to superheat. Equation (2.7) is a general equation for boiling limit.¹⁰ More specific equations can be found using information about the heat pipe geometry.

$$Q_{boiling} = \frac{k_{eff} \Delta T_{critical}}{t_{wick}} \quad (2.7)$$

In Equation (2.7) k_{eff} is the effective thermal conductivity across the liquid and wick, $\Delta T_{critical}$ is the temperature drop corresponding to superheat, and t_{wick} is the thickness of the wick. An expression for $\Delta T_{critical}$ is presented in equation (2.8) where r_n is the radius of the largest nucleation site.¹²

$$\Delta T_{critical} = \frac{T_{sat}}{h_{fg} \rho_v} \left(\frac{2\sigma}{r_n} - \Delta P_{cap,max} \right) \quad (2.8)$$

Multiple Condenser and Multiple Evaporator Heat Pipe Theory

Heat pipes function by coupling heat sinks and heat sources to the heat pipe at various locations. The heat sources add thermal power and heat sinks remove thermal power, which leads to evaporation and condensation, respectively. Since the evaporation and condensation is driven by the thermal power (external thermal boundary conditions) applied to the pipe, the vapor flow inside of the pipe may also be controlled by the external thermal boundary conditions. For the vapor, mass is added to the flow in the evaporator or

heat source region by evaporation. Mass is subtracted from the vapor flow in the condenser or heat sink region by condensation. The opposite occurs for the liquid. The mass flow rate, $\dot{m}$, created by evaporation and condensation for both the vapor phase working fluid and liquid phase working fluid can be described using Equation (2.9).

$$\dot{Q} = \dot{m}h_{fg} \quad (2.9)$$

A heat pipe has a heat rate per unit length of $Q'(z)$ along the entire length of the heat pipe ($0 \leq z \leq L$). In the heat source (evaporator) regions the heat rate is positive ($Q'(z) > 0$), in the insulated regions (adiabatic) the heat rate is zero ($Q'(z) = 0$), and finally in regions coupled to a heat sink (condenser) the heat rate is negative ($Q'(z) < 0$). At steady state the total thermal power throughput of the heat pipe can be calculated using Equation (2.10).

$$\dot{Q}_T = \frac{1}{2} \int_0^L |Q'(z)| dz \quad (2.10)$$

By examining the flow created by the evaporation and condensation occurring in the heat pipe, in cases where the heat pipe has multiple sperate evaporators or numerous condensers counter rotating cells begin to emerge. These cells are separated by a stagnation point, ξ_i , where the velocity is zero. Figure 2.3 shows an example of various boundary conditions and their influence on the vapor flow in heat pipes.

For Figure 2.3 the flow towards the right of the stagnation point, ξ_i , is considered positive while flow to the left of the partition is negative. Thermal power above the axis is considered positive (thermal power added to the pipe), and thermal power below the axis is considered negative (thermal power removed from the pipe). The stagnation points are related to both the heat sink and heat source.

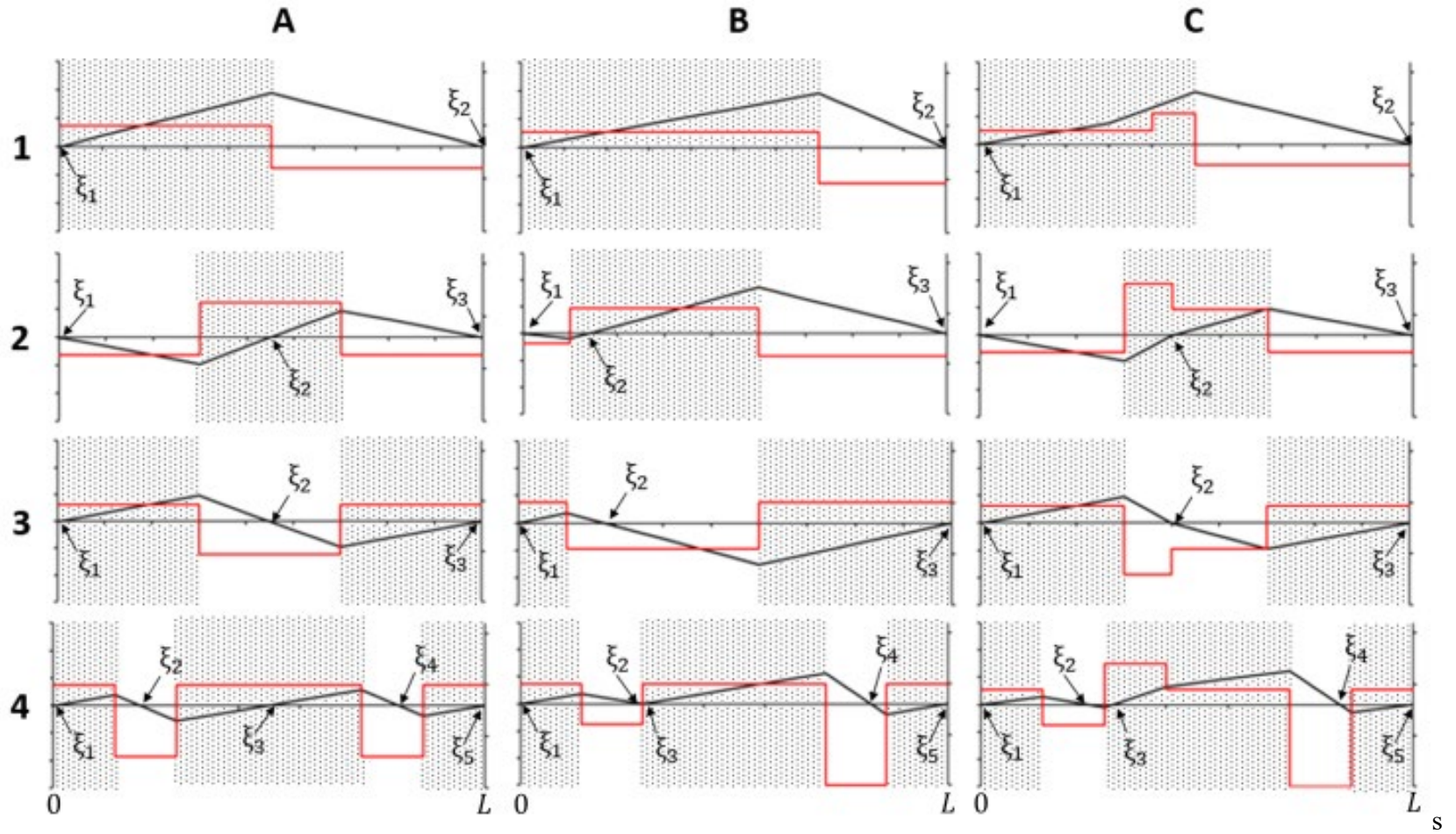


Figure 2.3: Example axial thermal power profiles (red) and corresponding axial vapor velocity profiles (black). The dotted regions represent the evaporator while the white regions represent the condenser and the ξ_i labels the stagnation points.

The total number of stagnation points may be calculated as the sum of the number of condensers and number of separate evaporators acting on the heat pipe.

$$n_{\xi} = n_c + n_e \quad (2.11)$$

In Equation (2.11), n_{ξ} is the number of stagnation points in the heat pipe, n_c is the number of separate condensers acting on the heat pipe, and n_e is the number of separate evaporators acting on the heat pipe. As we go down the columns in Figure 2.3, the number of separate evaporators and condensers increase and so do the number of stagnation points. Row 1 only has two stagnation points, while rows 2 and 3 have three stagnation points. Finally row 4 has five stagnation points. Only evaporators and condensers acting separately, meaning they are surrounded by their opposites produce stagnation points (i.e., condenser surrounded by evaporators or evaporators surrounded by condensers). This can be seen in column C of Figure 2.3. Further examination of the locations of these stagnation points and their relation to external thermal boundary conditions, leads to an equation to identify these points. These stagnation points must satisfy Equation (2.12).

$$\int_{\xi_{i-1}}^{\xi_i} Q'(z) dz = 0, \quad i = 1 \dots n_{\xi} \quad (2.12)$$

For example, a single evaporator and single condenser heat pipe, such as the one shown in row 1 in Figure 2.3, there are two stagnation points. One partition point is located at $z = 0$ while the other is located at $z = L$. Using information about the locations of the formed stagnation points, the thermal power throughput, and effective lengths for each of the counter rotating cells may be determined. The axial thermal power throughput associated with each counter rotating can be calculated using the Equation (2.13).

$$\dot{Q}_i = \frac{1}{2} \int_{\xi_{i-1}}^{\xi_i} |Q'(z)| dz \quad (2.13)$$

The location of these stagnation points and information about the boundary conditions applied to each cell can be used to determine an effective length of each cell. Equation (2.14) shows an equation to calculate these effective lengths.

$$L_{eff,i} = \frac{1}{\dot{Q}_i} \int_{\xi_{i-1}}^{\xi_i} \dot{Q}(z) dz \quad (2.14)$$

Given the locations of the stagnation points and the effective lengths of each counter rotating cell, the vapor and liquid pressure drops in each cell may be calculated. These pressure drops provide insight into the advantages of heat pipes with multiple condensers or multiple evaporators. These pressure drops may be smaller than those experienced in heat pipes in the traditional configuration, but the pressure drop largely depends on the boundary conditions imposed on the heat pipe. A simplified analysis of a heat pipe divided into various cells provides insight into the maximum pressure drop reduction that can be expected. For these purposes radial effects are neglected during this analysis.

For simplicity, the same heat pipe is used in each configuration and the thermal boundary conditions consist of uniform thermal power addition and removal. In the multiple condenser and evaporator configurations, the thermal loads are uniform and distributed evenly over each stagnation point. Additionally, this heat pipe is assumed to have no adiabatic regions and flow is assumed to be fully developed without radial flow components such as injection or suction. Figure 2.4 shows the configurations explored by this simple analysis. In Figure 2.4, the red region of the heat pipe represents an evaporator while the white represents a condenser.

Both the velocity of the working fluid and the effective length decrease as the number of stagnation points, n_ξ , increases for both the vapor and liquid. Relating the decrease in effective length and working fluid velocity to pressure drop may provide an estimated maximum performance augmentation factor.

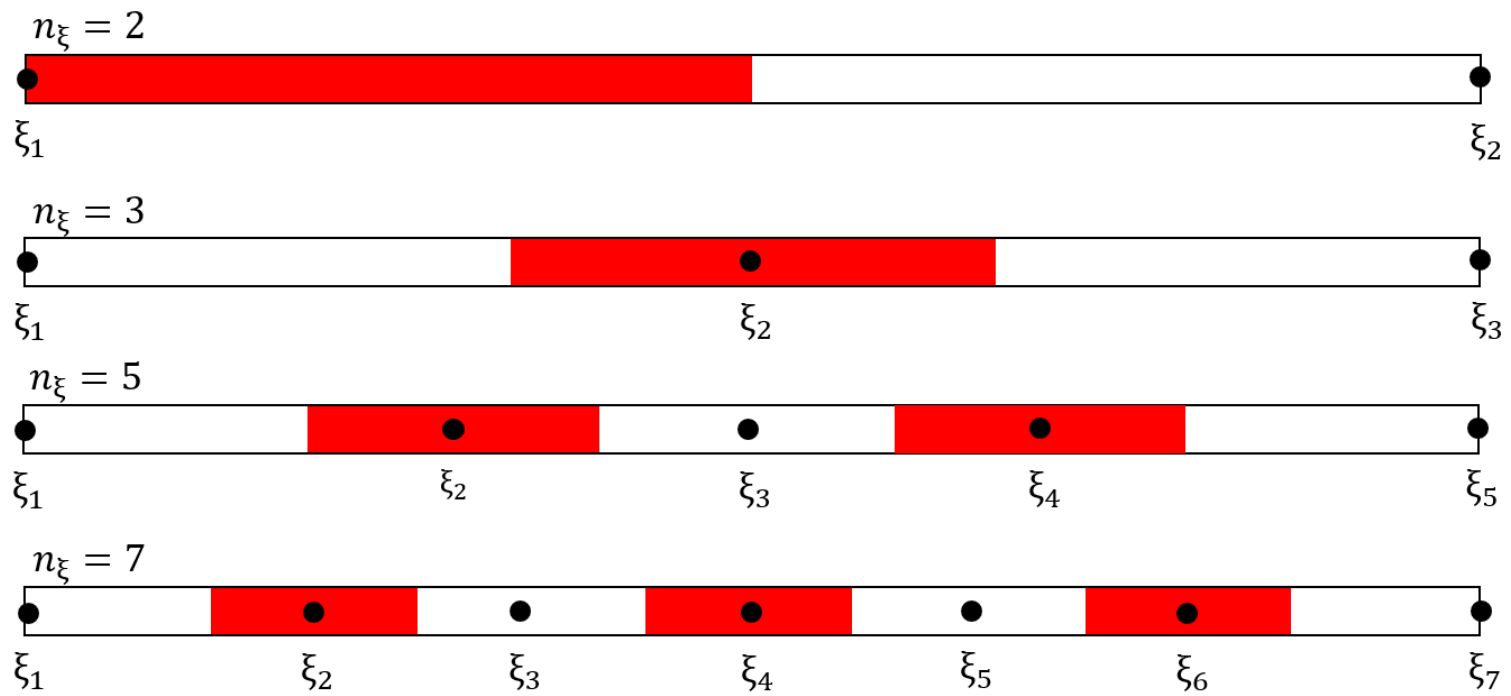


Figure 2.4: Multiple evaporator and multiple condenser example configurations with stagnation points.

Using the Darcy Friction Factor⁶¹ and the Blasius correlation⁶¹ with Hagen–Poiseuille formula⁶¹ an estimated pressure drop may be calculated.

$$\Delta P = \begin{cases} \frac{32\mu v L}{D^2}, & Re, < 2300 \\ \frac{0.1582 \mu^{0.25} \rho^{0.75} v^{1.75} L}{D^{1.25}}, & Re, axial > 2300 \end{cases} \quad (2.15)$$

In compound wicks, such as annular wicks, the wick is designed to allow for a relatively low liquid pressure drop while maintaining a high capillary potential. For these wicks, the liquid pressure drop is negligible when compared to the vapor pressure drop. For such a heat pipe comparing the estimated vapor pressure drops from heat pipes with multiple evaporators and condensers to the estimated pressure drop of a traditionally configured heat pipe provides an augmentation factor. This augmentation describes the potential increase in the heat pipe's capillary limitation by increasing the number of evaporators and condensers. From equations (2.9) and (2.14) the velocity and effective length decrease at a rate of $1/(n_\xi - 1)$. Substituting this into Equation (2.15) provides maximum potential augmentation factors of $(n_\xi - 1)^2$ and $(n_\xi - 1)^{2.75}$ for laminar and turbulent flow, respectively. When calculated for the configurations shown in Figure 2.4, the maximum potential augmentation factors are shown in Table 2.1.

As n_ξ approaches infinity, the effective length and velocity approach zero. This leads to an infinite maximum potential augmentation factor; however, there are practical and theoretical considerations that may reduce this augmentation factor. Practical considerations include the ability to remove and add heat from finite surface areas. To evaluate these theoretical considerations, more complex multidimensional effects need to be incorporated. These effects include compensating radial mass injection and radial mass suction of the vapor and liquid spaces.

In the evaporator suction acts on the liquid flow and injection acts on the vapor flow while the opposite occurs in the condenser. It is common practice to describe this flow using a Radial Reynolds number where a positive Radial Reynolds number describes injection, and a negative radial Reynolds number describes suction. Equation (2.17) shows expressions for the radial Reynolds numbers, Re_r , in the evaporator and condenser.

$$Re_r = \begin{cases} \frac{\dot{m}}{2 \pi L_e \mu_v}, & \text{Evaporator} \\ \frac{-\dot{m}}{2 \pi L_c \mu_v}, & \text{Condenser} \end{cases} \quad (2.16)$$

These multi-dimensional effects are complex and have not been fully characterized over the upper operational range of heat pipes (radial Reynolds numbers that have an absolute value greater than 2.2 in the condenser). Additionally, the radial mass injection and radial mass suction along the evaporator and condenser respectively means the flow in the heat pipe is likely not fully developed.

To show some of the limitations of this estimated augmentation factor and provide a more accurate estimate HTPIPE was run to investigate the capillary limit for a copper water heat pipe in three different configurations. HTPIPE is a code that analytically calculates the heat pipe limits and uses a combination of analytical relations and curve fitting to calculate vapor and liquid pressure drops and a temperature at the liquid vapor interface in the heat pipe.⁵ HTPIPE incorporates a pressure drop correlation that includes mass injection and suction up to a Radial Reynolds number in the condenser of approximately -2 presented by Busse in 1967.^{4,13} After a radial Reynolds number of -2 in the condenser, Busse says the solution to the approximation diverges. Results from HTPIPE were compared to experimental results by Reid et al. in 2000 and Lawdinsky in 2021.^{6,7} Reid et al. showed that a heat pipe followed the viscous limit curve reasonably well and Lawdinsky showed that HTPIPE calculated the temperature of the active region of the heat pipe within the thermocouple uncertainty.^{6,7}

Table 2.1: Maximum potential augmentation factors for example configurations shown in Figure 2.4.

n_ξ	Max. Augmentation Factor for Laminar	Max. Augmentation Factor for Turbulent
	Flow	Flow
2	1.0	1.0
3	4.0	6.7
5	16.0	45.3
7	36.0	138.0

The configurations that were run in HTPIPE include a traditionally configured water heat pipe with two stagnation points, the same water heat pipe with three stagnation points, and lastly the same heat pipe with 5 stagnation points. Figure 2.5 shows that for this heat pipe, the dual condenser and dual evaporator configurations do not reach the maximum augmentation factor and the augmentation factor is not constant with temperature. This is due to the incorporation of radial effects by HTPIPE. The augmentation factor calculated with HTPIPE ranges from approximately 1.5 to 1.9 for the $n_\xi = 3$ and from approximately 1.8 to 2.8 for the $n_\xi = 5$ case which is significantly less than the maximum augmentation factors shown in Table 2.1.

The augmentation factors may differ if the lengths of the evaporators and condensers are not equal. This leads to the vapor and liquid experiencing a different pressure drop depending on the distance they travel. For example, a heat pipe segment with a shorter evaporator and longer condenser may experience a larger vapor pressure drop than one with a longer evaporator and shorter condenser. This shows that radial effects have a significant influence on the heat pipe performance improvements made by using multiple evaporators or multiple condensers. It also shows the limitations of the multiple condenser or multiple condenser heat pipe analysis presented in this section.

Variable Conductance Heat Pipe Theory

Variable conductance heat pipes remain in a smaller temperature range over a wider range of evaporator and condenser thermal power levels. There are several methods used to make variable conductance heat pipes including the use of excess liquid or the use of non-condensable gas to flood the condenser.⁴³ Tests completed in this dissertation use non-condensable gas to flood the condenser and create a region where the condenser was shut off. Figure 2.6 shows a schematic of a variable conductance heat pipe using non-condensable gas condenser flooding. In the figure, the word gas refers to non-condensable gas.

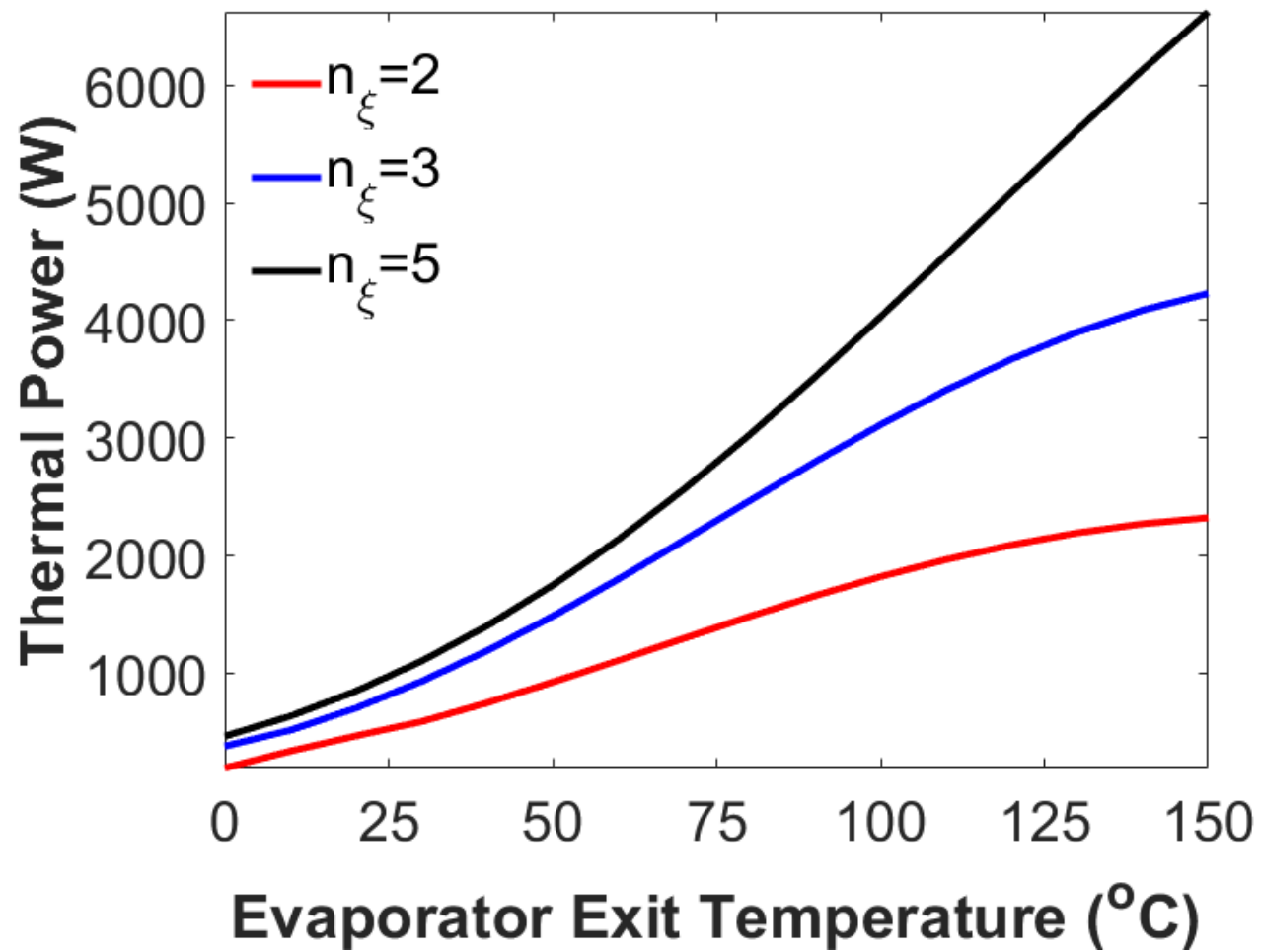


Figure 2.5: HTPIPE capillary limit for a heat pipe with various quantities of stagnation points and equal areas in the evaporator and condenser.

During operation, working fluid vapor sweeps the non-condensable gas from the evaporator to the condenser. Once at the condenser the working fluid vapor condenses, while the non-condensable gas remains causing the gas to accumulate at the condenser. The region with the non-condensable gas blocks vapor flow, creating an inactive region of the condenser. The non-condensable gas expands, or compresses based on the changes in the working fluid vapor pressure. The vapor pressure changes are often caused by alterations in thermal power input to the outside of the pipe, which changes the length of condenser capable of removing heat.

Figure 2.6 also shows a liquid pool present in the heat pipe. The liquid pool was added to the figure because it is common to have a small amount of excess liquid that pools at the end of the condenser as the liquid expands with increasing temperature. This liquid pool also prevents heat transfer and similar to the non-condensable gas shuts off the condenser in that region. It is important to incorporate the liquid pool when considering the length of the shut off condenser region. When a constant amount of non-condensable gas is present in the heat pipe, the heat pipe's thermal boundary conditions passively control the length of the inactive condenser region. Gas addition and removal actively controls the length of the inactive condenser region by changing the amount of non-condensable gas in the system.

The length of the inactive region can be estimated by assuming the saturation pressure of the working fluid and the pressure of the non-condensable gas are equal at their interface. An equation may be found by assuming that the gas front is flat, that no diffusion occurs between the working fluid and non-condensable gas, and that no liquid pool is present. Using the Clausius Clapeyron equation to describe the working fluid vapor saturation pressure and the ideal gas law to describe the non-condensable gas pressure, Equation (2.17) can be derived.

$$L_g = m_g R_g T_g \left[A_v P_{ref} e^{\left(\frac{-h_{fg}}{R_v} \left(\frac{1}{T_v} - \frac{1}{T_{ref}} \right) \right)} \right]^{-1} \quad (2.17)$$

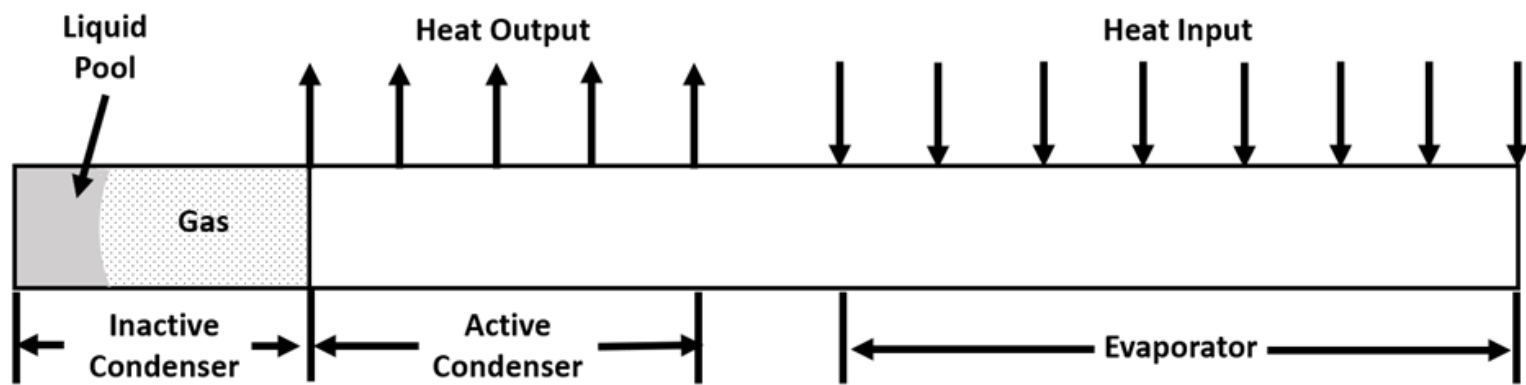


Figure 2.6: Non-condensable gas variable conductance heat pipe schematic.

In this equation, the subscript g refers to the non-condensable gas and the subscript v refers to the working fluid vapor. P_{ref} and T_{ref} are a set of reference saturation pressure and reference saturation temperature. The notation m is mass, the R is the specific gas constant, and T is the temperature. While A_v is the cross-sectional area of the vapor channel in the center of the pipe, h_{fg} is the latent heat of vaporization of the working fluid, and L_g is the length of the vapor channel containing non-condensable gas. The above equation was used to create an example of variable conductance heat pipe operation compared to fixed conductance heat pipe operation, shown in Figure 2.7. The heat pipe used to produce Figure 2.7 has a 1 m long evaporator, 1 m long condenser, and is an annular design. The working fluid used in the heat pipe is sodium. In this example variable conductance heat pipe, the initial temperature corresponds to the non-condensable gas being approximately 0.4 m long.

As shown in Figure 2.7, the variable conductance heat pipe can transfer a wider range of power over the same temperature range than a fixed conductance heat pipe. At lower temperatures, the length of the inactive condenser region is larger because of lower working fluid vapor pressure. This makes the surface area of the condenser smaller, which decreases its ability to output power.

At higher temperatures, the working fluid saturation pressure increases and compresses the non-condensable gas. The compressed non-condensable gas occupies less space in the pipe, which decreases the length of the inactive condenser region. The decrease in inactive condenser region length results in a larger condenser surface area, which increases the condenser's ability to output power. This effect is more noticeable in high temperature heat pipes because of their wider thermal power capabilities, but it also occurs in lower temperature heat pipes.

Like fixed conductance heat pipes, variable conductance heat pipes can operate with multiple evaporators and condensers. Figure 2.8 shows a schematic of the operation of a dual condenser variable conductance heat pipe. In the figure, the non-condensable gas is referred to as gas to save space. In this case, the working fluid vapor sweeps non-condensable gas to both condensers.

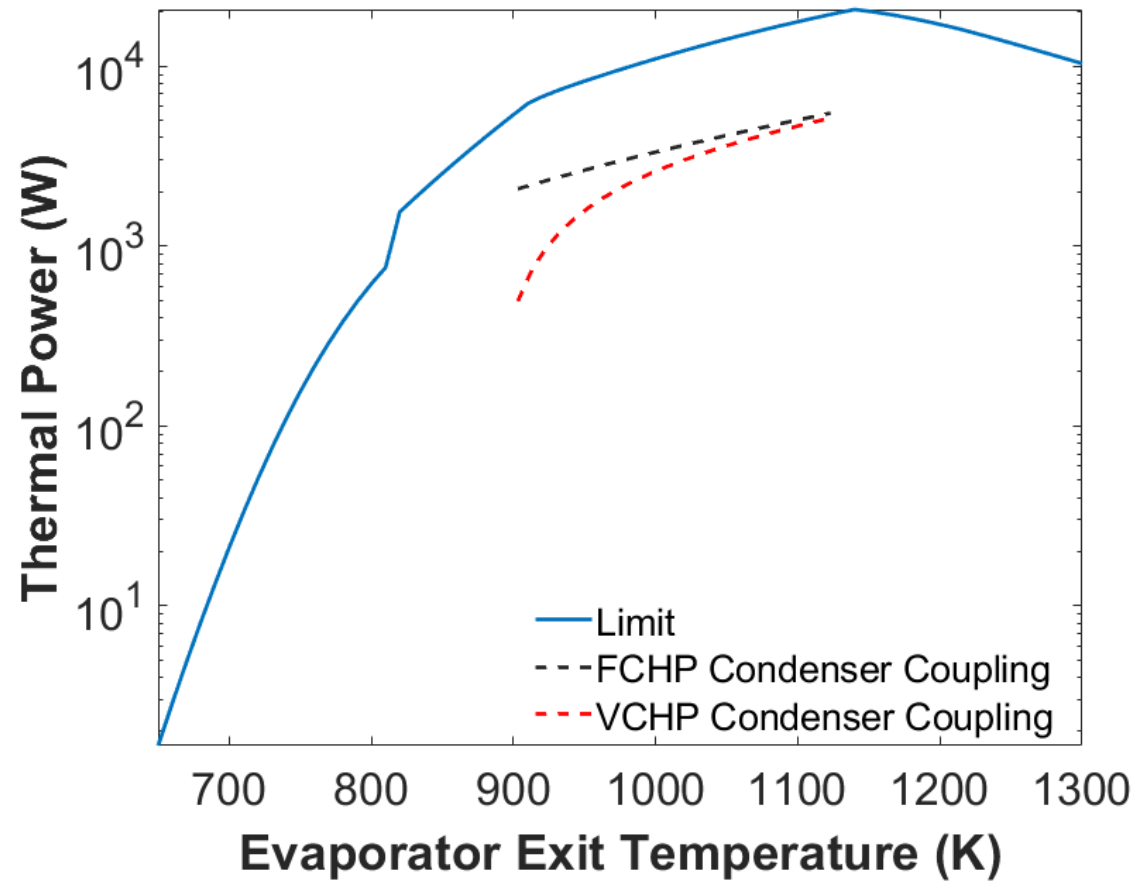


Figure 2.7: Power output of a fixed conductance heat pipe compared to variable conductance heat pipe with the same condenser length (1 m evaporator and 1 m overall condenser).

For variable conductance heat pipes with multiple evaporators or multiple condensers there are two scenarios. The first is a passive variable conductance heat pipe where the mass of gas in the heat pipe is constant throughout the operation. The second is active variable conductance heat pipe where the operator changes the mass of gas throughout operation. During startup of a passive variable conductance heat pipe, the gas is swept to cold stagnation points by the vapor flow. Once the gas is swept towards the initial stagnation points in the condenser, ξ_i , new active cold stagnation points are formed, ξ'_i by the change in boundary locations caused by the inactive region of the condenser. While the gas gathers around the initial cold stagnation points, performance can be calculated using the location of the active cold stagnation points. Figure 2.9 shows an example of non-condensable gas location with respect to stagnation point location in heat pipes. If the heat input and heat output are identical for each counter rotating cell, an equal amount of non-condensable gas may be present on each side of initial cold stagnation points. The location of the active cold stagnation point may be found by completing an iterative thermal power balance and gas length calculation using Equation (2.17). When the external thermal boundary conditions are not the same for each counter rotating cell, the length of gas surrounding each cold stagnation point may differ.

The length of non-condensable gas surrounding cold stagnation points may differ on each side because one side may be experiencing more or less compression depending on the pressure of the working fluid vapor and gas interface. Another way to get an uneven amount of non-condensable gas surrounding condenser stagnation points is to alter the thermal boundary conditions to create additional condenser stagnation points, i.e., the number of stagnation points in the new configuration is greater than the number of condenser stagnation points in the previous configuration. If there is a sufficient condenser pool surrounding the initial cold stagnation point, the liquid pool may act as a wall and prevent the non-condensable gas to pass through it. For active variable conductance heat pipes, non-condensable gas is often injected near the condenser stagnation points. In this case, the non-condensable gas remains on the same side of the condenser stagnation point that it injected. This is true in cases where the impulse created by the gas injection does not overcome the vapor that is continually sweeping it towards the condenser stagnation point.

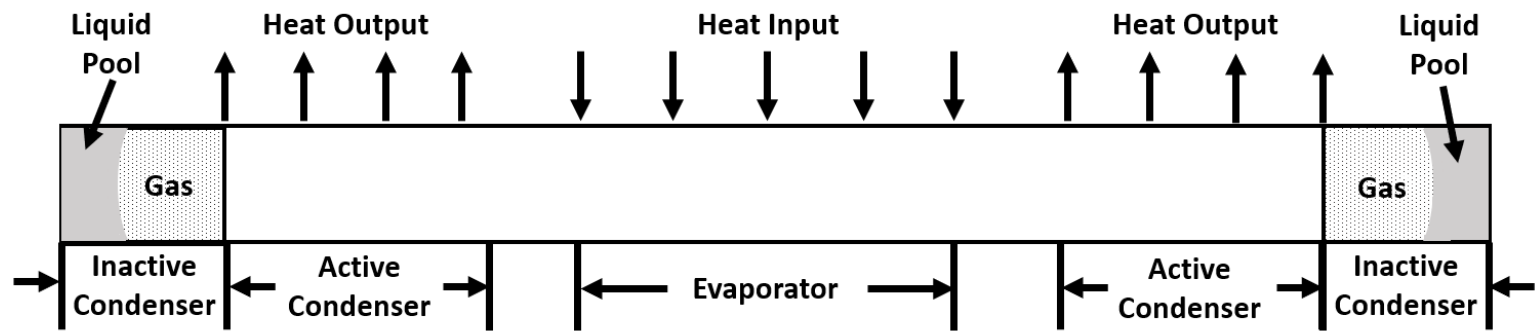


Figure 2.8: Dual condenser non-condensable gas variable conductance heat pipe schematic.

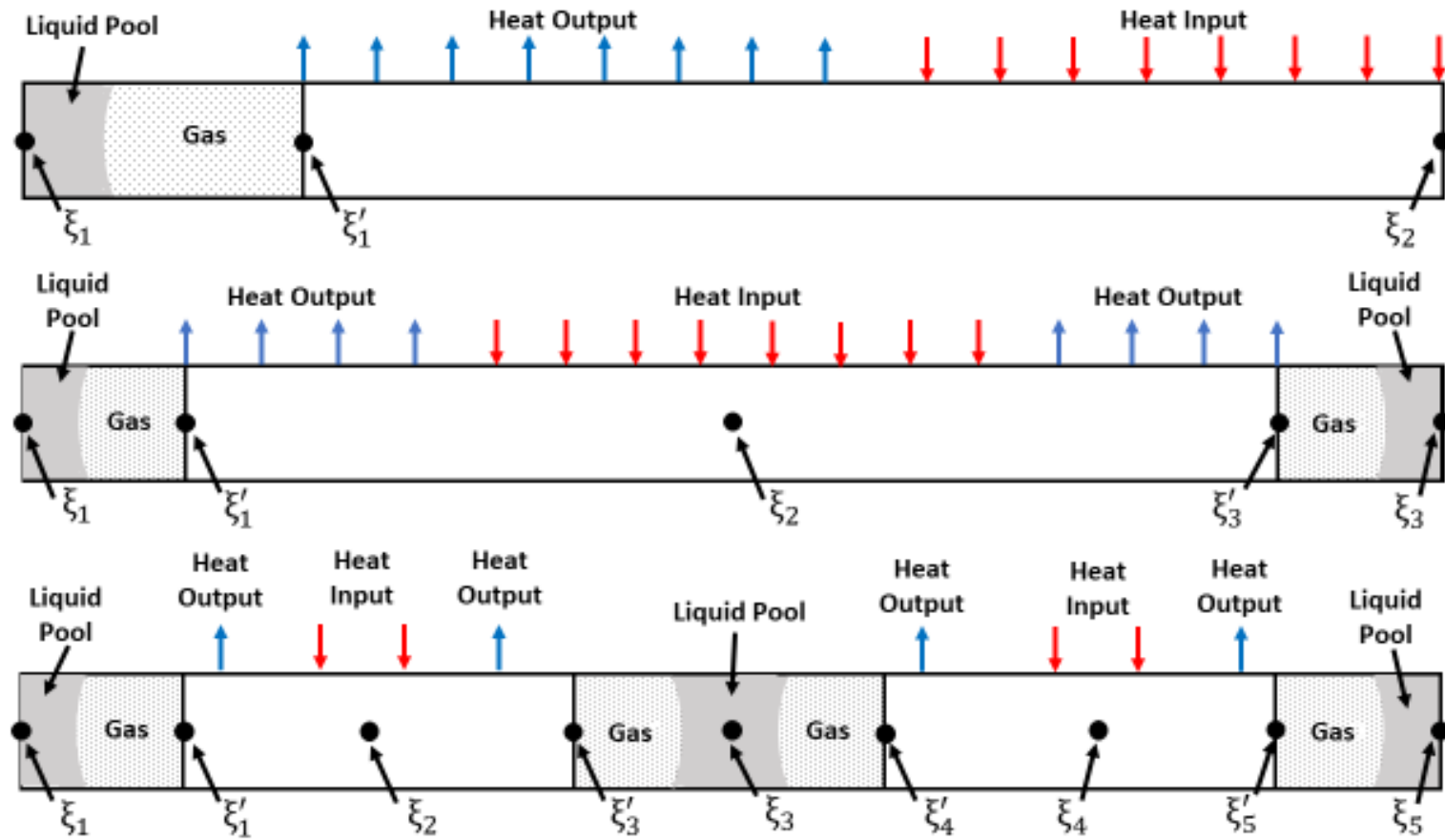


Figure 2.9: Example of non-condensable gas locations for various heat pipe thermal boundary condition configurations with initial and active cold stagnation points.

As mentioned in the literature review, for a heat pipe with walls that have a large thermal conductivity, the gas region may have an irregular front. To compensate for this a flat front model with a conduction correction may be used to inspect the gas region. A similar approach was taken by Faghri et al in 1994.⁵⁷ In this dissertation, the gas region is treated as a fin while the active region of the heat pipe is treated as the fin base. To further simplify the problem, the ends of the inactive condenser region can be treated as adiabatic fin tips either when it is next to another inactive condenser region (such is the case with a centered condenser), or when the ends of the heat pipe are insulated (in the standard configuration or in dual condenser configuration). This approach is limited to use only after the heat pipe has reached a steady or quasi-steady-state condition. Incorporating transient effects would require a more complex thermal equation that would need to couple with a transient equation describing the state of the heat pipe including any additional storage. Since the heat pipe temperature is dependent on the length of the non-condensable gas region and the non-condensable gas is also dependent on the temperature of the heat pipe, an iterative solver would be needed.

The adiabatic fin tip equation sets the origin at the base. For the case of a gas loaded heat pipe, $x = 0$, would be the point where the heat pipes condenser transitions from being active to inactive. This case is shown in Figure 2.10. Completing a thermal power balance and solving the resulting ODE produces the adiabatic fin tip equation as shown in a heat and mass transfer textbook.⁶² This is shown in Equation (2.18).

$$\frac{\theta(x)}{\theta_b} = \frac{\cosh(m(L-x))}{\cosh(mL)} \quad (2.18)$$

In Equation (2.18) $\theta(x) = (T(x) - T_\infty)$, $\theta_b = (T_b - T_\infty)$, and $m = \sqrt{(hP)/(kA_c)}$ where $T(x)$ is the temperature at location x , T_∞ is the temperature of the surroundings, h is the heat transfer coefficient, P is the perimeter of the fin, and A_c is the cross-sectional area of the fin.

Applying Equation (2.18) to a gas loaded heat pipe is challenging base location is not easily found. To avoid this issue, the coordinates of the fin can be switched so the origin,

$x = 0$, is located at the fin tip. The fin tip location can be found by solving for the locations of heat pipe stagnation points using the external thermal boundary conditions applied to the pipe. Finding these points was outlined in the previous sections. Figure 2.11 shows a schematic of this fin arrangement.

The general solution to this problem is the same as it is for the problem shown in Figure 2.10. In this case the external thermal boundary conditions used to solve the problem differ. These conditions are shown in Equations (2.19) and (2.20).

$$\left. \frac{d\theta}{dX} \right|_{x=0} = 0 \quad (2.19)$$

$$\theta(L) = T_b - T_\infty = \theta_b \quad (2.20)$$

Using these boundary conditions produces the adiabatic fin tip equation when the origin is located at the fin tip. This is shown in Equation (2.21).

$$\frac{\theta(x)}{\theta_b} = \frac{\cosh(mx)}{\cosh(mL)} \quad (2.21)$$

For a fin of identical geometry and base temperature, Equation (2.18) and (2.21) produce identical results. Neither (2.18) nor (2.21) accurately capture effects at locations for $x \geq |L|$. Table 2.2 shows a comparison of these results for a 0.5 m long cylindrical copper fin with a 2 cm diameter attached to a base at 60°C in a room at 20°C. The film coefficient, h , was assumed to be 5 W/ (m² K). Figure 2.12 shows a comparison of the coordinates and x-values for each case.

Table 2.2 shows using Equation (2.21) produces the same results as Equation (2.18) and may be suitable for calculating the temperature of the gas region and estimating the length of the gas region from experimental data when using a heat pipe with walls that have large thermal conductivities. From these temperatures the heat removed from the gas region, commonly referred to as heat leak, may be calculated.

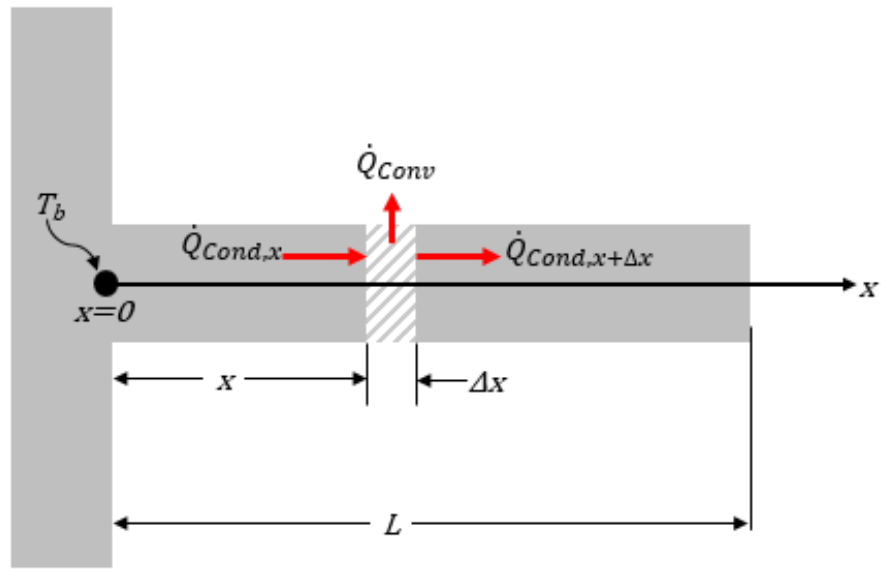


Figure 2.10: Adiabatic fin tip schematic with origin located at the base.

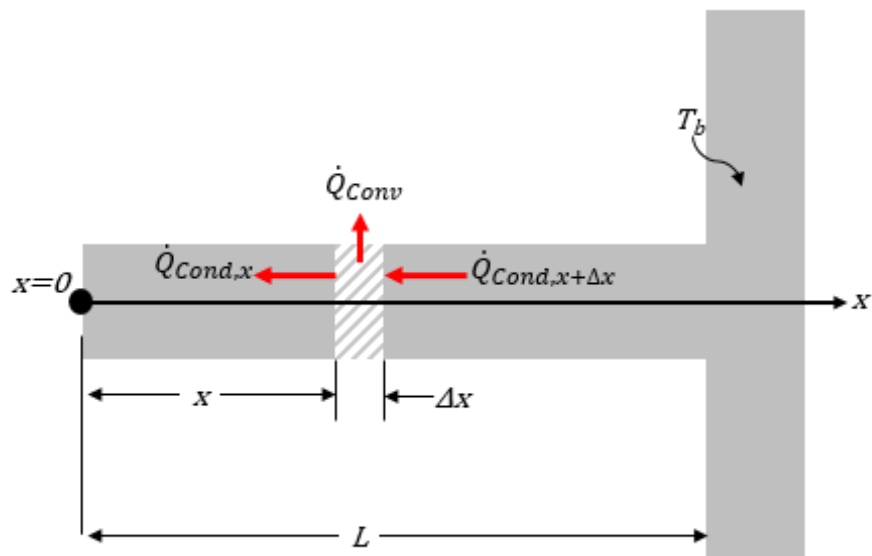


Figure 2.11: Adiabatic fin tip schematic with origin located at the fin tip

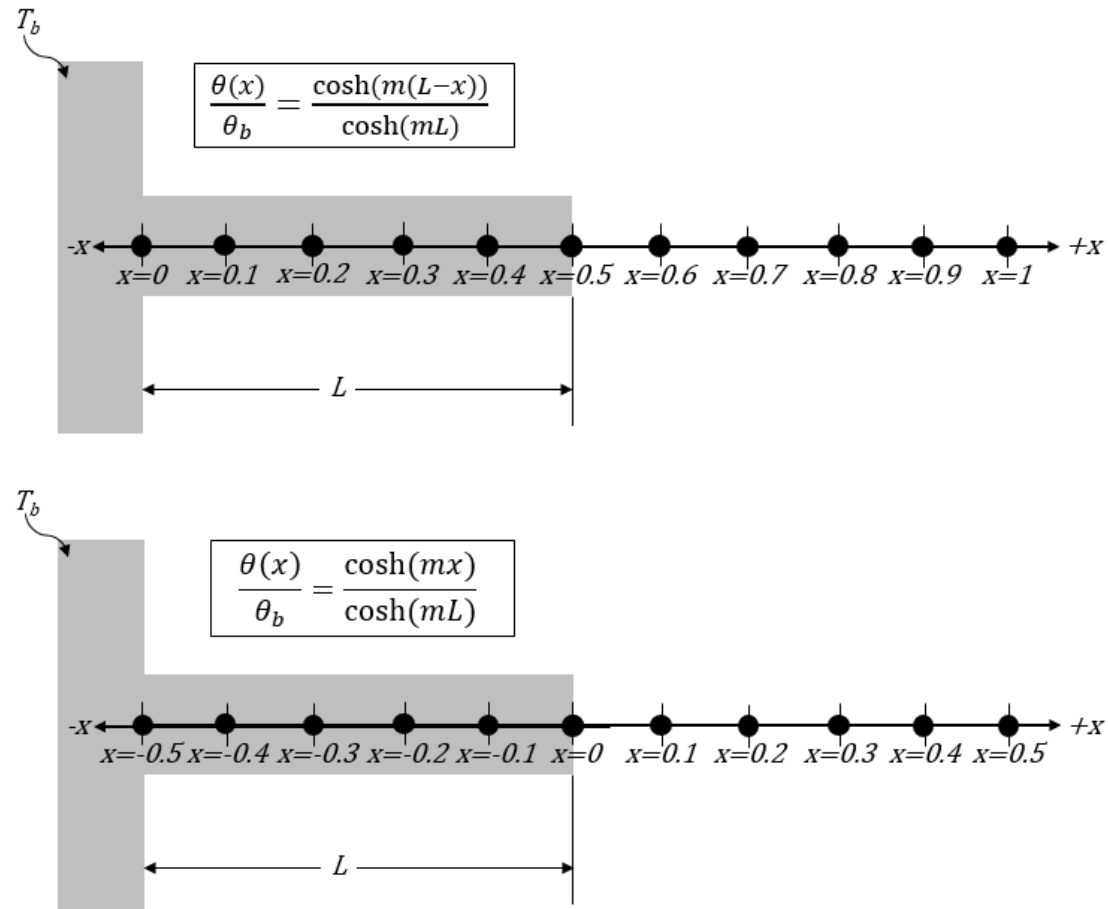


Figure 2.12: Comparison of adiabatic fin tip problem with an origin at the fin base to adiabatic fin tip problem with origin at fin tip.

Table 2.2: Results from Equation (2.18) compared to results from Equation (2.21)

x with origin at fin base (m)	Equation (2.18) Temperature (°C)	x with origin at fin tip (m)	Equation (2.21) Temperature (°C)
0.0	50.0	-0.5	50.0
0.1	47.2	-0.4	47.2
0.2	45.0	-0.3	45.0
0.3	43.5	-0.2	43.5
0.4	42.6	-0.1	42.6
0.5	42.3	0.0	42.3
0.6	42.6	0.1	42.6
0.7	43.5	0.2	43.5
0.8	45.0	0.3	45.0
0.9	47.2	0.4	47.2
1.0	50.0	0.5	50.0

CHAPTER THREE

LITERATURE REVIEW

In this chapter, previous work on heat pipes is presented in two sections after a brief introduction on the applications of heat pipes. The first section discusses work related to heat pipes with multiple evaporators or multiple condensers, while the second section discusses previous work on variable conductance heat pipes. A third section presents the state of the art in heat pipe technology.

Although the heat pipe concept existed earlier, it was first named and demonstrated by Grover and his coworkers in the 1960's.¹ Heat pipes are of technical interest because of their ability to maintain close to isothermal walls and their ability to transfer heat, in some cases with heat fluxes as high as 10^7 W/m² down its axis.¹ In addition to these capabilities, heat pipes can operate as a closed system and do not require additional equipment to operate. Heat pipes have been used in many applications including heat exchangers, solar power technologies, electronic components, HVAC systems, bakeries, metal forges, automobiles, spacecraft, and power plants.^{2-4,45-49}

With the widespread use of heat pipes there has been a significant effort to numerically and analytically model their behavior. Heat pipe limitations have been widely studied and numerous analytical and numerical models have been used to characterize them. Starting in the 1960's C. A. Busse and J.E. Kemme made a significant effort in studying and mathematically modelling the limitations and behavior of heat pipes.^{8, 15-17} After this work, a multitude of numerical models, mathematical models, analytical solutions, and approximations of heat pipe behavior under both steady-state and transient conditions were published.^{5, 18-30} Some of the models developed are still used to model and design heat pipes in present day.

Heat Pipes with Multiple Evaporators and Multiple Condensers

Starting in the mid-1980s multiple investigators studied heat pipes in nontraditional configurations. One of the first studies took place in 1986 by Gernet.³¹ Here a low temperature heat pipe with five evaporators in series was studied. It was found when each of the five evaporators were supplied with the same power, dry out occurred at the evaporator that was farthest from the condenser.³¹ Gernet also adapted an existing heat pipe model to handle a heat pipe with multiple evaporators. The model over predicted the dry out power level compared to the measured power level, but it could calculate the evaporator wall temperature within 2°C and the condenser temperature within 6°C.³¹

In the late 1980s Faghri conducted several studies with heat pipes having multiple evaporators and a single condenser.³²⁻³⁵ Cheng and Faghri numerically studied heat pipes with single and multiple evaporators.³² The model was compared to two experimental cases for validation. The first case was a high temperature sodium heat pipe with an annular wick. For this case, the model predicted the temperature within 3°C in the evaporator and 6°C in the condenser.³² The second case examined the heat pipe studied by Gernet in 1986.³¹ The model predicted the temperature of the water vapor within 1°C but had a 6°C difference between the measured and predicted temperature in the condenser region. Faghri and Chen attributed the larger discrepancy in measured to predicted condenser temperature to thermocouple measurement error.³²

In 1991, Faghri et al. studied the startup behavior of a high temperature heat pipe with single and multiple evaporators in two environments consisting of both vacuum and air.³³ After a multitude of tests, it was found that the behavior during frozen startup, when the working fluid is solid when the heat pipe is not in operation, depended on the heat removal rate from the condenser and two way vapor flow was observed during the two evaporator startup.³³ The results of this study showed that the vapor flow in heat pipes is strongly influenced by the position of its evaporators and condensers. A second paper observed the steady-state behavior of the pipe as well as implemented a numerical model and compared it to the experimental results.³⁴ The model predicted temperature within 1% or 0.5 % of the measured temperature depending on their grid sizing.³⁴

In 2011, Shabgard and Faghri developed a steady-state analytical model for cylindrical heat pipes with multiple evaporators.³⁵ Results from the model were compared to experimental results from two different copper water heat pipes and results from a previous numerical model. Both copper water heat pipes were about 1 m long and had a two-layer copper screen wick. An analytical model, updated from a previous one, had good agreement with the previous numerical model and experimental results.³⁵ The new model was more computationally efficient than the previous model and showed for this heat pipe axial conduction had a large influence on the calculated liquid pressure.

Park and Boo studied a heat pipe with two dissimilar condensers in series with an adiabatic section between them.³⁶ The authors completed a series of experiments where the amount of fluid in the pipe, the heat input rate, and the coolant inlet temperature varied. In addition to these parameters the authors varied whether the first condenser was insulated or not. From the experiments, the authors concluded that when both condensers were active, the effective thermal conductivity was higher than when only one condenser was active.³⁶ The results of this study suggest that heat pipes with multiple condensers tend to have higher effective thermal conductivities and may be more efficient.

In 2017, Tang et al. conducted a study on a 300 mm long grooved copper water heat pipe with multiple evaporators.³⁷ The evaporators consisted of two block heaters located at the ends of the heat pipe. The condenser was cooled with a water jacket located between the two evaporators. The location of the condenser, heat input, and heat pipe orientation varied during the experiments. Tests were conducted on a heat pipe with double ended heating and the condenser in the middle, with the condenser to the right of the middle, and with the condenser to the left of the middle. Tests with single ended heating with the condenser positioned on the opposite end of the evaporator were conducted for comparison. The authors found that dual ended heating middle cooled heat pipe showed enhanced performance in both inclined and horizontal orientation.³⁷

In 2018, Tang et al. published a study directly relevant to the work presented in this dissertation. A configuration was examined with a varying number of centered evaporators and condensers on each end of the heat pipe. The stated purpose of the work was to determine the suitability of this configuration for cooling multiple heat sources in

spacecraft. The authors examined a single condenser and single evaporator configuration, a dual condenser configuration with a single small evaporator in the center, and a dual condenser configuration with multiple evaporators in the middle.³⁸ The authors tested various heat input and output rates. Both the power supplied to each heater and the amount of thermal power removed from each condenser were equal. From the multiple condenser theory this set up allowed the heat pipe to behave symmetrically about the center of the pipe. During testing, the authors experienced dry out for the single evaporator and single condenser configuration as well as the dual condenser and single evaporator configuration during a 15 W and 25 W thermal power input.³⁸ The dual condenser and single evaporator configuration did not experience dry out in any of the testing from 10 W to 60 W. From these results the authors concluded that the dual condenser configurations have improved thermal performance when compared to the single condenser configuration.³⁸ The data also showed that the dual condenser configuration with multiple evaporators had significantly increased performance. This effect is likely caused by the decreased length the vapor must travel and thus a decrease in vapor pressure losses. A sufficiently long evaporator may reproduce the results of the dual condenser and multiple evaporator tests. The influence of various cooling water flow rates in the condenser on the heat pipe performance was also examined. The authors found that differences in cooling water flow rates had little effect on the heat pipes performance. From the data the authors concluded that a dual condenser configuration heat pipe offers a high heat flux heat pipe that efficiently uses space.³⁸

Variable Conductance Heat Pipes

Similar to fixed conductance heat pipes, there has been a significant effort studying variable conductance heat pipes. Some of the early research involving variable conductance heat pipes involved laser cells. In 1978, Deverall and York tested two variable conductance heat pipes to determine the feasibility of using heat pipes as metal laser cells.³⁹ This study is important because the heat pipe is configured with an evaporator in the center and two condensers on each side. The first variable conductance heat pipe was

approximately 8 cm in diameter and 92 cm long. Water and mercury were used as working fluids. In this test, the non-condensable gas front was relatively vertical and the non-condensable gas buffer kept the laser cell transmission windows clean.³⁹ The results determined that using variable conductance heat pipes as laser cells is plausible. The authors continued their study by fabricating and testing a sodium variable conductance heat pipe cell. The heat pipe used an annular wick and neon as the non-condensable gas. The sodium heat pipe cell was about 10 cm in diameter and 94 cm long. As with the first test, the windows were kept clean. Unfortunately, the authors did not note the shape of the gas front during the second test.

The aerospace industry has conducted a significant amount of research into variable conductance heat pipes with a single evaporator and single condenser. In 1971, Marcus and Fleischman studied transient and steady-state operation of a passive controlled hot internal reservoir variable conductance heat pipe.⁴¹ In this study, a resistance wire was used as the heat input mode and natural convection and radiation into the ambient surroundings as the mode of heat removal. From the results, the authors concluded that diffusion has a large effect on this heat pipe and that both the gas front length and sharpness depend on the axial thermal conductivity of the pipe.⁴¹ The working fluid and non-condensable gas combination, the heat removal rate per unit length in the condenser, and the pipe diameter all influences the sharpness of the gas front.⁴¹ Also in 1971, Edwards and Marcus completed a parametric study of a passive controlled non-condensable gas variable conductance heat pipe.⁴² The authors investigated the effects of working fluid, axial wall conductivity, and heat pipe operating temperature on the non-condensable gas front behavior using a numerical model they developed. The axial wall conductance had the strongest effect on the gas front behavior, the operating temperature had a small effect on the gas front behavior, and the working fluid had the least effect on the gas front behavior.⁴² The results of these two studies suggest that during the testing of a heat pipe made from a material with a larger thermal conductivity, a correction factor for thermal conduction may improve the numerical models results.

In the early 1970s both Bienert et al. and Depew et al. studied variable conductance heat pipes.^{40,44} Bienert et al. studied a feedback controlled variable conductance heat pipe

to determine whether an electrical feedback or a mechanical feedback system provided better thermal control in 1971.⁴⁰ The mechanical feedback system worked by using a incompressible liquid heated by the heat pipe's heat source to change the volume of a non-condensable gas reservoir. In the electrical feedback system, an electrical heater was used to heat the non-condensable gas to change its volume. The mass of non-condensable gas in each heat pipe system remained constant throughout the test. The analysis and tests found that the electrical feedback control allowed for greater control of the heat pipe's temperature and that this method allows for a much broader range of thermal power input and output conditions.⁴⁰ In 1973, Depew et al. studied a variable conductance heat pipe controlled mechanically with bellows and an auxiliary fluid similar to Bienert et al.^{40,44} The difference is the auxiliary fluid was controlled independently of the heat pipes heat source. Depew et al. found that the variable conductance heat pipe stabilized the temperature of the heat source with varying power better than a fixed conductance pipe.⁴⁴ Both studies show that variable conductance heat pipes can stabilize temperature over a range of temperatures. Additionally, Bienert et al. showed that independent control of the amount of non-condensable gas in variable conductance heat pipes allows them to operate in a wider range of thermal power conditions.

Several studies took advantage of VCHP's ability to operate over a wide temperature range. Some of these studies include using VCHPs to control the temperature of electronics, to act as potential radiators for fission reactors in Lunar a Martian missions, to thermally link systems in spacecraft, and to control the heating-cooling cycle of oil during a cold start of an engine.⁴⁵⁻⁴⁹

In addition to the studies completed by the aerospace industry, there has been a significant advance in modeling and understanding the physical mechanisms occurring in variable conductance heat pipes. C.L. Tien has conducted multiple studies involving both variable conductance heat pipe experiments and modeling. In 1973, A.R. Rohani and C.L. Tien performed a two-dimensional steady-state numerical analysis of heat and mass transfer occurring in the vapor-gas region of a variable conductance heat pipe.⁵⁰ The analysis assumed that axial conduction through the heat pipe wall and liquid filled wick were negligible. The authors introduced two dimensionless parameters. The first parameter

describes the magnitude of the axial thermal energy transfer rate diffusing into the non-condensable gas region compared to the energy transfer rate caused by the axial conduction.⁵⁰ The second parameter compares the magnitude of the energy transferred by diffusion to the energy transferred by conduction.⁵⁰ The numerical analysis tested six cases presented in papers published by other authors. The variables observed were the vapor and gas mixture, heat pipe diameter, initial vapor temperature, and ambient temperature. The cases resulted in various values for the non-dimensional parameters. The authors concluded that the physical parameters of the heat pipe have an influence on the performance of the variable conductance heat pipe. They also concluded that energy and mass transfer between the gas and vapor might have a large influence on heat pipe operation.⁵⁰

In 1984, K. Hijikata, S.J. Chen, and C.L. Tien studied the influence of non-condensable gas on vapor condensation in a thermosyphon.⁵¹ Thermosyphons operate in a similar manner to heat pipes, except thermosyphons use gravity to return liquid to the condenser while heat pipes use capillary forces produced by a wick. To describe the significance of radial diffusion the authors used two dimensionless parameters like those presented by Rohani and Tien.⁵⁰ The parameters describe the amount of non-condensable gas in the pipe and the radial diffusion rate compared to the condensation rate.⁵¹ To investigate the effect of the non-condensable gas on the vapor condensation, several cases using steam and air were examined with an analytical model. The results of the model showed that the flat front model does not predict the length of the non-condensable gas sufficiently.⁵¹ Hong et al. continued this work by adding wall conduction effects in 1991.⁵⁰ The authors found that the wall conduction effect increased the diffusion length, but larger diffusion parameters minimize both the wall conduction effect and radial diffusion.⁵⁰

R.P. Bobco presented a first order analytical model for variable conductance heat pipes in a 1987 paper.⁵² Similar to works by Tien,^{51,53,54} Bobco also used two parameters in the analytical model. The parameters used in this model are the location where condensation begins and a heat flux parameter.⁵² These parameters can be determined from test data by conducting multiple tests at both minimal and excessive heat loads.⁵² The analytical model was tested using data from the 1971 Marcus and Edwards report,⁴¹ and it was found that the analytical model was in good agreement with the experimental data.⁵²

In 1988, P.F. Peterson and C.L. Tien used a wet/dry bulb to make measurements of gas concentrations in a variable conductance thermosyphon.⁵³ The effects of thermosyphon orientation as well as gas and vapor combinations were investigated. The authors observed that during operation the condenser was divided into three regions. The first region is the region at the end of the thermosyphons where no condensation takes place due to the prevalence of non-condensable gas in this region. The second region has limited condensation occurring from a region with vapor mixed with non-condensable gas. In this region, the amount of condensation varied axially as the amount of vapor changes in the mixture.⁵³ The last region is the farthest away from the end of the pipe. Here the vapor is dominant and fully condenses. In a case where air was used as the non-condensable gas and the thermosyphon was vertical, cooler air would incrementally slip down causing rapid condensation and a pressure fluctuation.⁵³ In addition, it was found that non-condensable gas mixtures behave differently than single gases.⁵³ All the phenomena observed by the authors can influence the performance of thermosyphons and heat pipes.

In 1989, P.F. Peterson and C.L. Tien implemented a new model using integral analysis.⁵⁴ The integral analysis simplified equations and allowed inclusion of axial conduction. Results from this model were in good agreement with both the experimental results and the exact numerical results.⁵⁴ A parametric study was conducted to determine the influence of two-dimensional effects and temperature dependencies. The parametric study found that two-dimensional effects become important when the wall conduction parameter is less than ten and the diffusion parameter is less than one.⁴⁹ In addition, when natural convection and radiation are used as the cooling mechanism for the condenser, the temperature dependence of the heat transfer coefficient can increase the length of non-condensable gas in the condenser.⁵⁴ This effect is most prominent when the walls have higher thermal conductivity.

In 1991, Kobayashi et al. conducted an experimental and analytical study of the flow field in variable conductance heat pipes.⁵⁶ The focus of the study was to determine the influence of gravity on the vapor and gas interface.⁵⁶ The experiments took place on a thermosyphon to eliminate any interactions between a colder wick and the working fluid vapor. The thermosyphon was investigated in a vertical orientation and a two-degree tilt

from horizontal orientation, called the “almost horizontal” orientation.⁵⁶ The analytical and experimental results showed that gravity had a large influence on the vapor and gas interface in a vertical thermosyphon. It was observed that gravity had a stronger effect on the temperature gradient than on the concentration gradient, but clear concentration layers were still present in the interface.⁵⁶ In the almost horizontal case the layers were unclear, likely because in this orientation gravity does suppress axial diffusion.⁵⁶ Based on the results the authors suggest that the flat front model may not be applicable in horizontal variable conductance thermosyphons and heat pipes.

In 1994, Faghri and Harley applied a lumped capacitance heat transfer model to analyze both fixed conductance and variable conductance heat pipes.⁵⁷ In an effort to increase computational efficiency, the authors assumed a flat gas front and incorporated axial conduction to obtain more accurate condenser temperatures. This analysis had good agreement with experimental data, but the results showed a slightly sharper temperature drop than the experimental data.⁵⁷ The authors determined this effect was caused by the analysis neglecting mass diffusion. Further analysis found that although the condenser temperature agreement in cases with significant diffusion affects is not as good, the predicted active condenser length and evaporator temperatures are the same when compared to a more computationally expensive model.⁵⁷

Zhou and Collins studied the spatial distribution of heat flux in a variable conductance thermosyphon in 1995.⁵⁸ The authors measured the thickness of condensed working fluid film to determine the heat flux. A copper walled condenser and glass walled condenser were investigated experimentally. When using the glass condenser at high power levels instabilities were observed. These instabilities were believed to be caused by local heat flux variations.⁵⁸

In 2012, Saad et al. conducted experiments and developed a thermal resistive network model to investigate the transient performance of a variable conductance heat pipe.⁵⁹ The model used multiple elements in both axial and radial directions to incorporate thermal conduction and the ability to use non-uniform boundary conditions.⁵⁹ This network model was combined with a one-dimensional flat front model for the non-condensable gas behavior. The model and experimental results agreed in both steady-state and transient

cases. During start up, the axial conduction increased the vapor and gas interface, but did not affect the startup time.⁵⁹ However, the non-condensable gas was found to increase the shutdown time.⁵⁹

State of the Art

Although there have been numerous studies relating to a wide variety of heat pipes in the past couple years, the author was unable to find recent research closely related to the work completed in this dissertation. The studies that closely relate to the current work were conducted by Tang et. al in 2017, Tang et. al in 2018, Deverall et. al in 1978; as well as work presented by Bienert et. al in 1971.^{37,38,39,40} In the 2017 and 2018 papers by Tang et al, independently acting evaporators and condensers were used unlike other multiple condenser or multiple evaporator papers where the evaporators and condensers were broken into several parts.^{37,38} Deverall studied the first variation of a gas loaded variable conductance heat pipe with multiple condensers.³⁹ Beinart studied a both passive and active controlled variable conductance heat pipe by changing the temperature of the non-condensable gas reservoir.⁴⁰

The work presented in this dissertation uses separate evaporators and condensers as well as active and passive variable conductance heat pipes. Unlike Beinart, the non-condensable pressure is directly controlled in this work. Although the work done by Deverall in 1978 is closely related to this work, the objectives of that work and this dissertation are different. Deverall's work was aimed at maintaining window cleanliness using this type of heat pipe,³⁹ while the work in this dissertation seeks to observe the behavior of this type of heat pipe. To show the relationship between the research in this survey and the work in this document, a chart was produced where the term separate evaporators or condenser refers to evaporators and condenser that act alone. For example, a separate evaporator only has a condenser neighboring it and not another evaporator. Connected evaporators have another evaporator neighboring them.

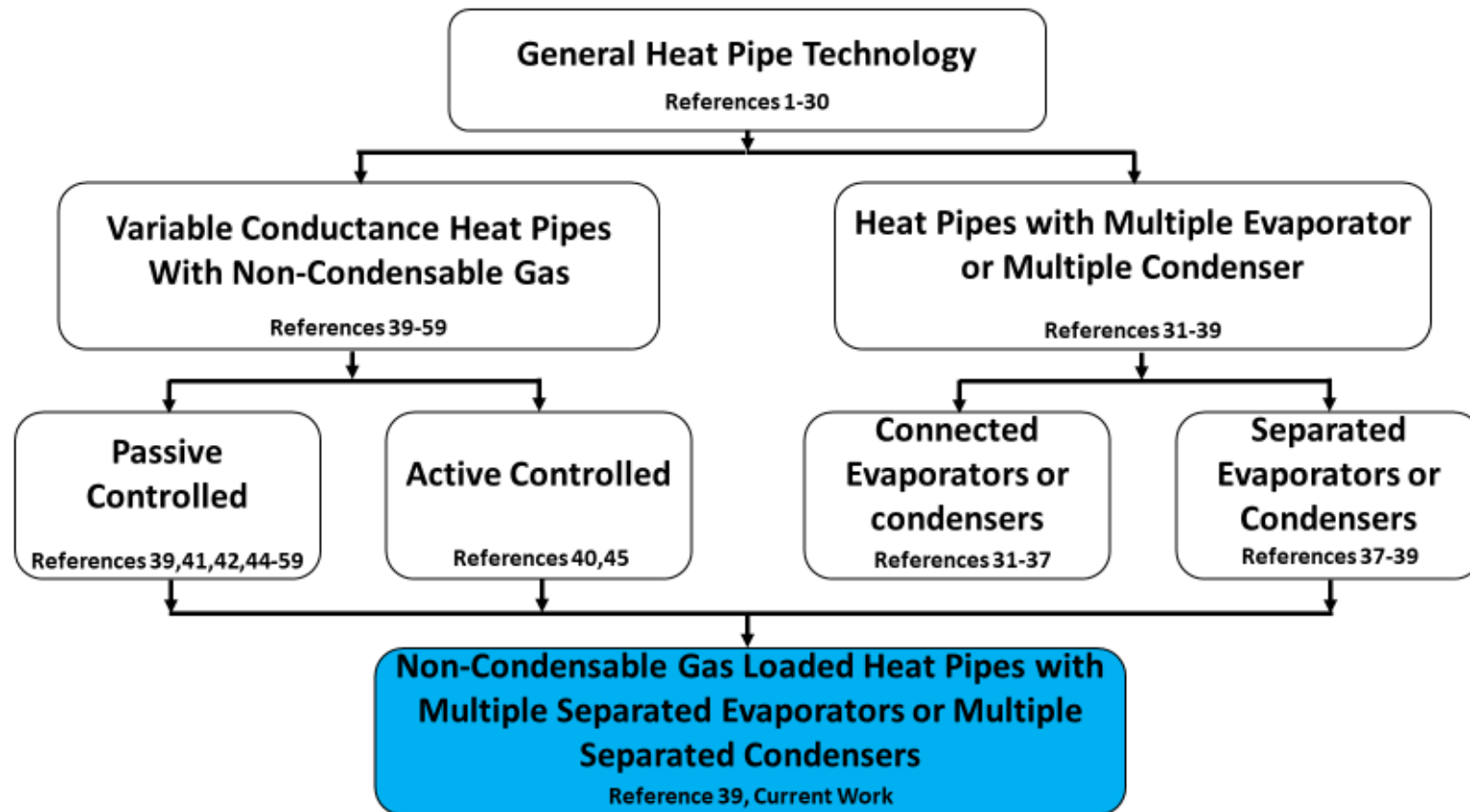


Figure 3.1: Scope of work and previous work.

CHAPTER FOUR

METHODOLOGY

In this chapter the methodology used behind heat pipe design, fabrication, and testing is discussed. The chapter is divided into three sections. The first section discusses the design and fabrication aspects of the heat pipe. The second section describes the test setup and methods used during the tests cooled using the air in the surroundings. This section includes details on active gas control and RGA set up. The final section discusses the test set-up for tests using calorimeters to cool the heat pipe. Details about calorimeter design and assembly are included along with details on calorimeter specific instrumentation and insulation.

Heat Pipe Design and Construction

Multiple tests were conducted on a heat pipe to determine the influence of the thermal boundary conditions on heat pipe behavior. To examine these effects, a copper water heat pipe was fabricated to ensure known heat pipe properties. An annular wick geometry was used to improve the heat pipes performance. The wick was made from 100 X 100 copper mesh material, shown in Figure 4.1. To ensure a suitable wick was produced, two duplicate wicks were constructed by rolling four layers of the mesh material around a 14 mm copper mandrel. These dimensions would allow for a radial annular gap of approximately 0.7 mm. To ensure the material was rolled tightly the initial layer was adhered to the mandrel using Cyanoacrylate. After the wick was rolled, it was secured with a solder seam. To remove it from the mandrel it was soaked in acetone until the adhesive dissolved. Figure 4.2 shows the fabricated copper wicks.



Figure 4.1: 100 X 100 copper mesh material.



Figure 4.2: Fabricated copper wicks.

A microscope was used to survey a sample of the wick material to observe overall pore size. Figure 4.3 shows the resulting image. From the photograph in Figure 4.3 it was determined that the average effective pore radius of this sample was close to 50 μm . In addition to the microscope imaging, the wick underwent a bubble point test, similar to the bubble point tests described in Adkins et. al,⁶⁰ in water to determine the maximum effective pore radius. During this bubble point test the wick was submerged under water and argon was allowed to flow into the wick. The argon pressure was increased until bubbles escaped from the wick. At this point a water manometer was used to record the pressure. Once the pressure was recorded at the for the initial and largest bubble points, the pressure was increased to find additional points. Figure 4.4 shows the bubble point test being conducted. Figure 4.5 shows the effective pore radius and location of the wick gaps. The axial location uncertainty is ± 0.32 cm and is not visible in Figure 4.5. The effective pore size uncertainty is related to the uncertainty of the pressure measurement which was ± 31.1 Pa. Using the methodology presented by Kline and McClintock in 1953,⁶³ this uncertainty was used to calculate the uncertainty of the effective pore radius. Since the error associated with the pressure measurement is constant, as the pressure measurement gets larger the effective pore size error gets smaller. The effective pore radius takes the contact angle into account to provide a better estimate of the wick's capabilities. The bubble point test revealed the largest effective pore radius in the wick is approximately 550 μm , while the pore radius of the bulk material is rated to 75 μm . This indicates that there were either gaps in the wick material or the seam. The region with the 550 μm gap located at approximately 235 cm was severed from the rest of the wick allowing a maximum effective pore radius of 320 μm in the evaporator region. The wicks were inserted into copper tube with an outer diameter of approximately 1.91 cm and inner diameter of approximately 1.65cm. Two end fittings were used to seal the ends of the heat pipe. These end fittings were designed using quick connect fittings welded to a stainless-steel disk from Kurt J. Lesker to allow for the heat pipe to be emptied and refilled when necessary. The end fittings have a fill stem with an adjustable length and attached shut off valves. The fill stems also allow for non-condensable gas to be added or removed from the heat pipe when the shut off valves are open. Figure 4.6 shows the heat pipe outer shell with the wick and the heat pipe end-fittings.

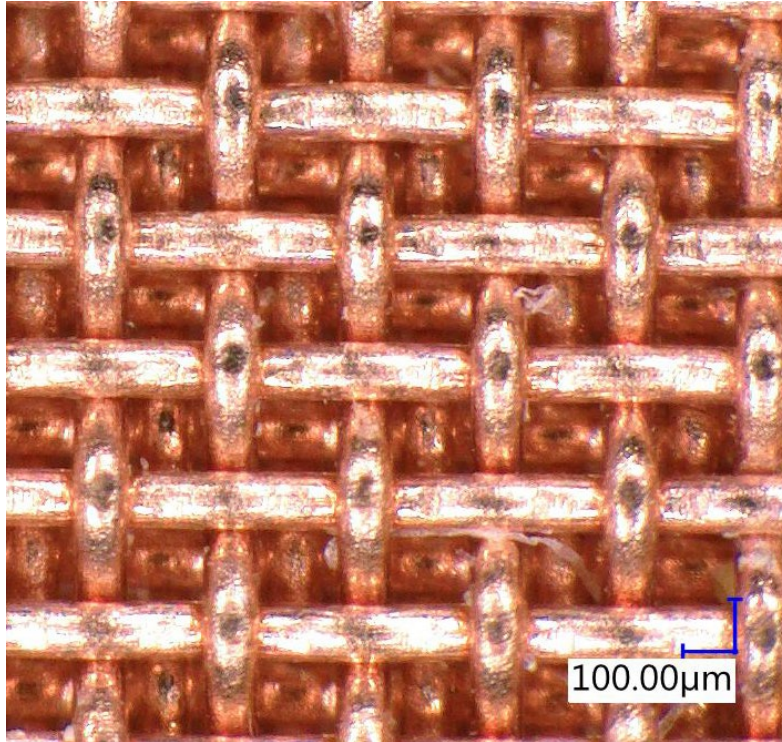


Figure 4.3: Microscope image of wick material.



Figure 4.4: Bubble point test in water.

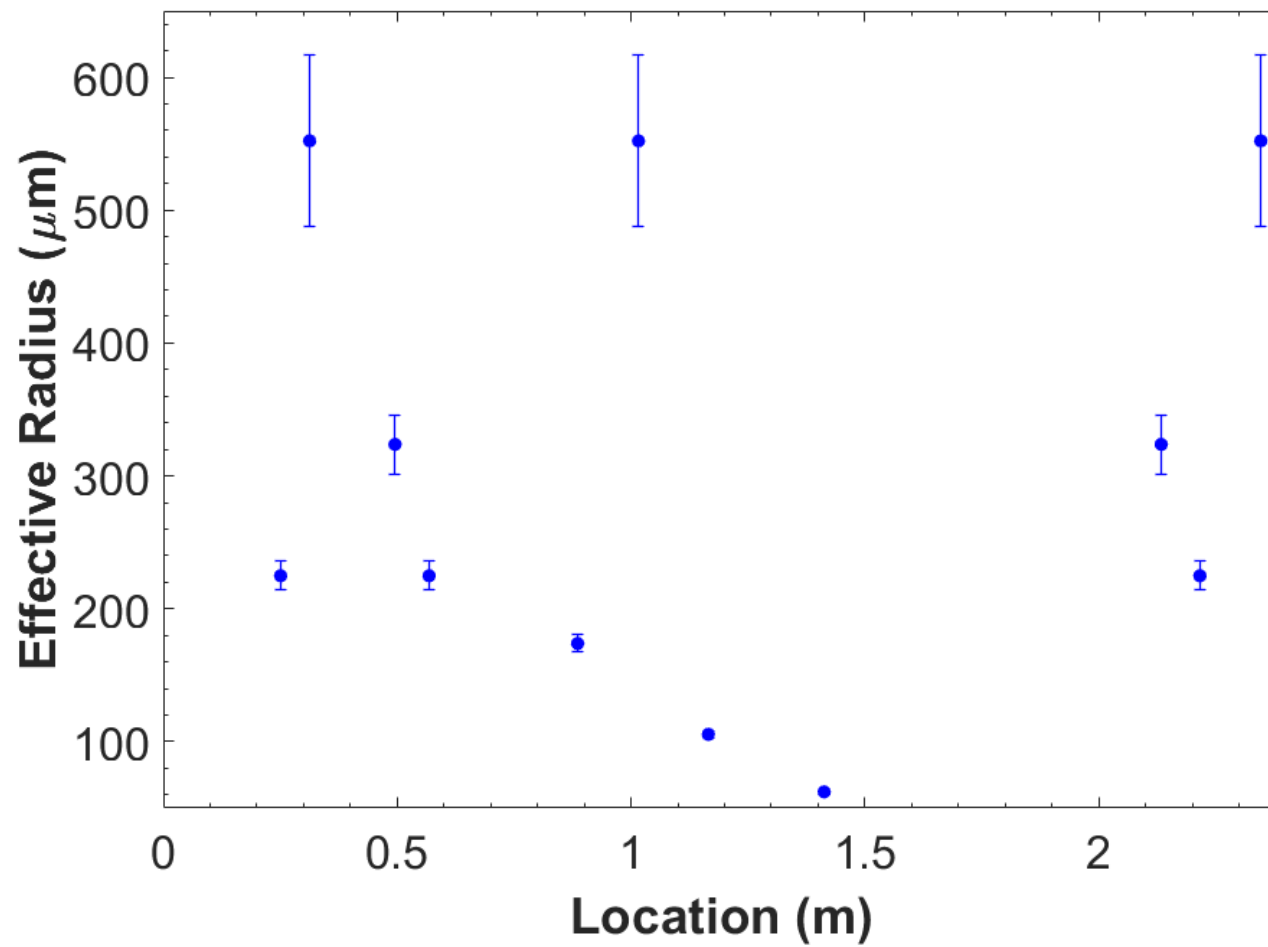


Figure 4.5: Measured wick effective pore radius by axial location.

The heat pipe was assembled and filled with 110 ± 15 mL of water. During fill, a slight amount of air remained in the pipe. A permanent marker indicated the evaporator side of the heat pipe ensuring that pore with an effective radius of $550 \mu\text{m}$ is located on the condenser side of the pipe to limit its the effect on heat pipes performance.

Using information about the wick and heat pipe shell as well as the contact angle of water on copper the de-rated limits of the pipe were calculated using HTPIPE.⁵ Figure 4.7 shows the limits for a single evaporator and single condenser of this pipe.

The power component of the operating range was determined from estimated maximum thermal power that could be applied to the heat pipes using film heaters and low voltage DC power supplies. The maximum temperature component was selected to ensure the water vapor pressure remained less than 101 kPa to prevent potential positive pressure leaks out of the vacuum fittings on the end of the heat pipe. The minimum temperature and thermal power were estimated as the minimum power and temperature where the heat pipe would operate.

Air-Cooled Tests

Test Setup

A test stand was designed and fabricated to use during the testing of this heat pipe. The test stand was constructed from T slotted aluminum frame and was used to house the heat pipe, pressure controller power supply, vacuum pump, pressure controllers, and an infrared background. Figure 4.8 shows the test stand.

The heat pipe was mounted to the stand using insulated wires to minimize thermal losses. Single wires were used to suspend the heat pipe on the ends while in the middle two wire were used to secure the heat pipe. Figure 4.9 shows the mounting method for the ends of the heat pipe as well as the middle.

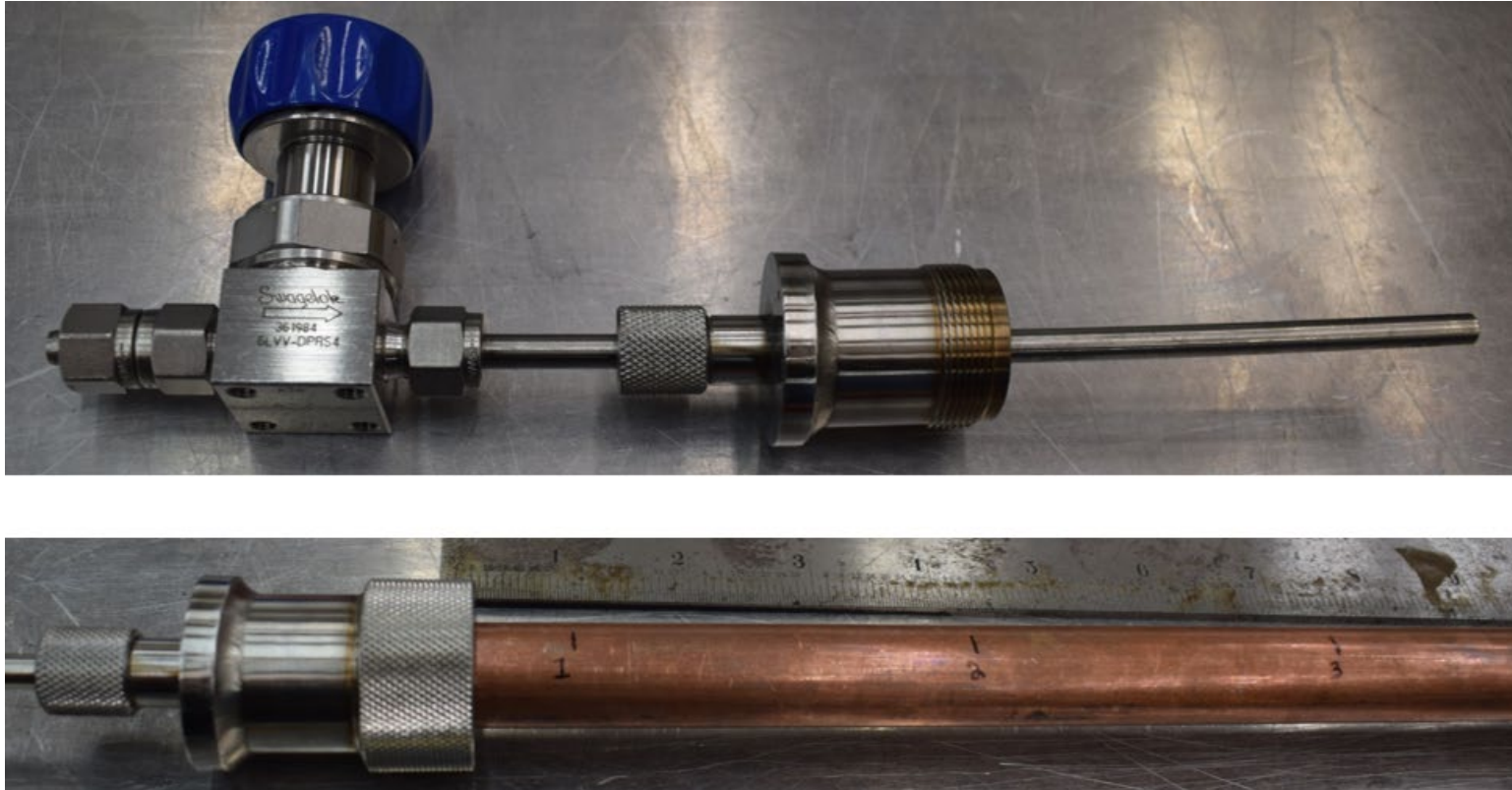


Figure 4.6: (Top) Heat pipe end fitting with installed valve and (Bottom) heat pipe outer shell with end fitting installed.

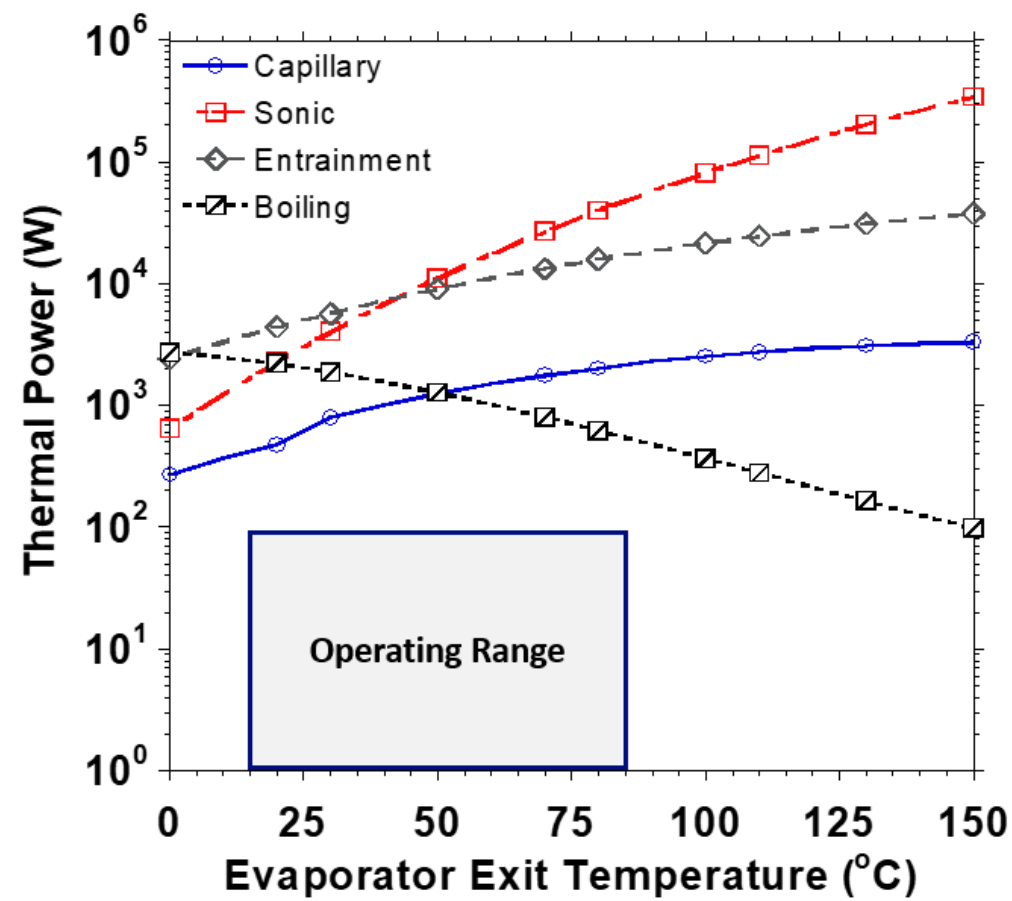


Figure 4.7: Operational limits for single condenser and dual condenser heat pipe configurations.

The evaporator of the heat pipe consisted of three zones of three rectangular 22.8 cm by 2.5 cm polyimide film heaters spaced 3.2 ± 0.16 cm from the end fittings and 0.635 ± 0.16 cm from each other. The heaters were adhered to the surface of the heat pipe using pressure sensitive adhesive. Each of the heaters was wired in parallel to ensure the same voltage was applied to each heater. Figure 4.10 shows the film heaters installed on the heat pipe. Each heater has a resistance of $23.8 \pm 0.09 \Omega$. The heaters were controlled using a custom switch box and they were powered with a 30 Volt variable DC power supply. The heaters wired in parallel so they would have an equal supply voltage. Figure 4.11 shows the custom switch box for the heaters and Figure 4.12 a circuit diagram for the switch box.

Proportion-Air model QB1X pressure controllers with a range of vacuum to 170 kPa controlled gas pressure in the heat pipe. The right image in Figure 4.13 shows this pressure controller installed in the system. The Proportion-Air pressure controllers are a closed loop system with two valves; one valve is connected to a high-pressure reservoir and acts as the inlet into the system while the other valve is connected to a low-pressure reservoir and acts as the exhaust of the system. The pressure controllers are equipped with an internal pressure transducer which provides a signal that is compared to the command signal. The difference in signals allows the control system to open and close the appropriate valves to maintain the set system pressure. The manufacturer calibrated the QB1X pressure controllers with a Druck Pace 1000 and a Fluke 525 B before delivery. The calibration specified the QB1X pressure controllers maintain pressures within $\pm 0.2\%$ of the full-scale value which is equivalent ± 338 Pa.⁶⁴⁻⁶⁶ The command signal was controlled using a 0-10V potentiometer. The potentiometer shown in the left image of Figure 4.13 controls voltage at increments of 0.001V, which equated to a pressure increment of 20 Pa. From this it can be seen that the pressure controller's uncertainty contributes most of the error in non-condensable gas pressure in the heat pipe.

For the pressure controllers to operate, a minimum volume of approximately 16 mL was required. Each controller was attached to an empty Kurt J. Lesker KF25 straight, part number QF25-100-N, to ensure each pressure controller had an adequate volume when the heat pipes isolation valves were closed. Figure 4.14 shows the KF25 straight installed onto the outlet of the pressure controller.

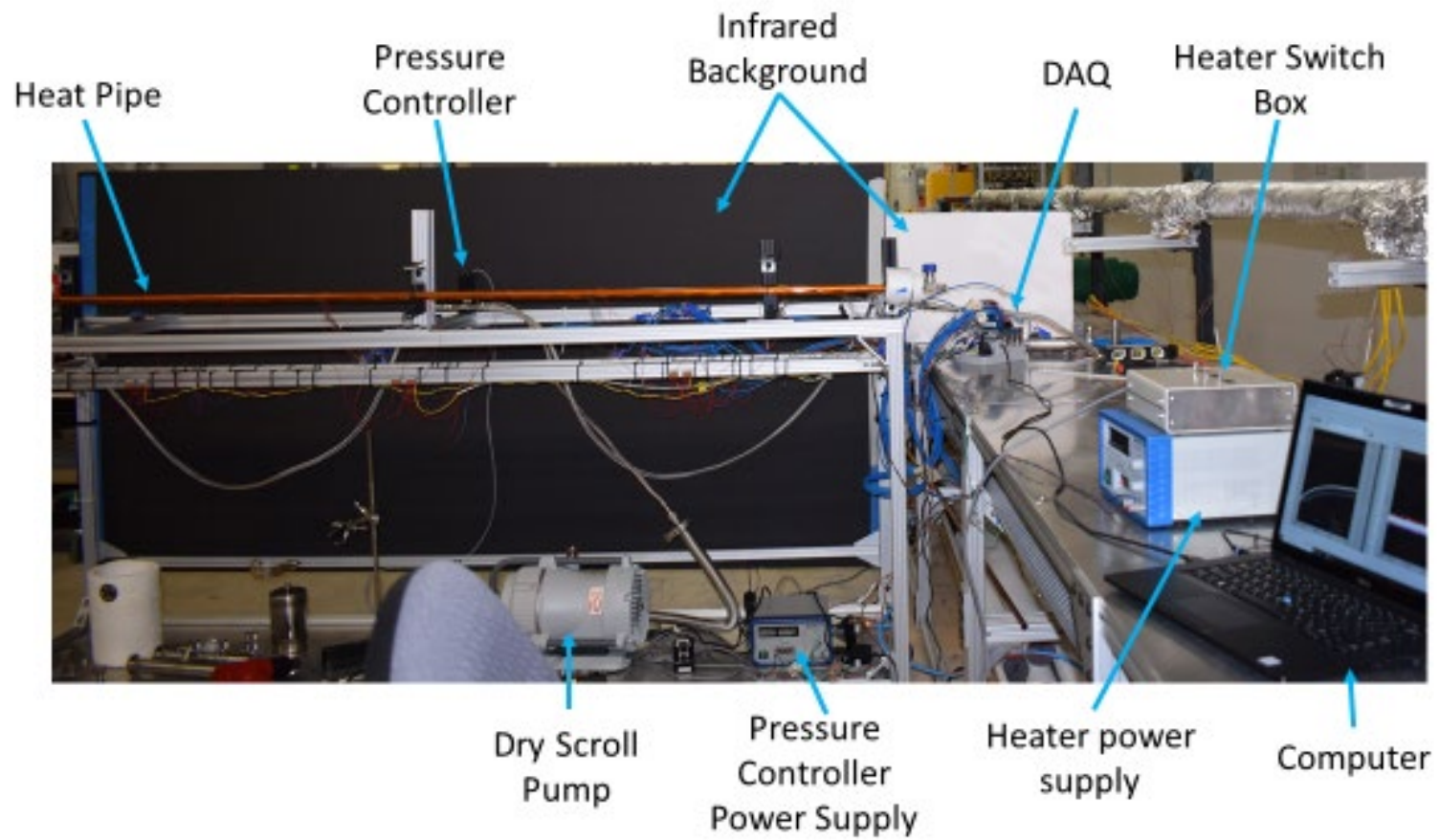


Figure 4.8: Heat pipe test stand.

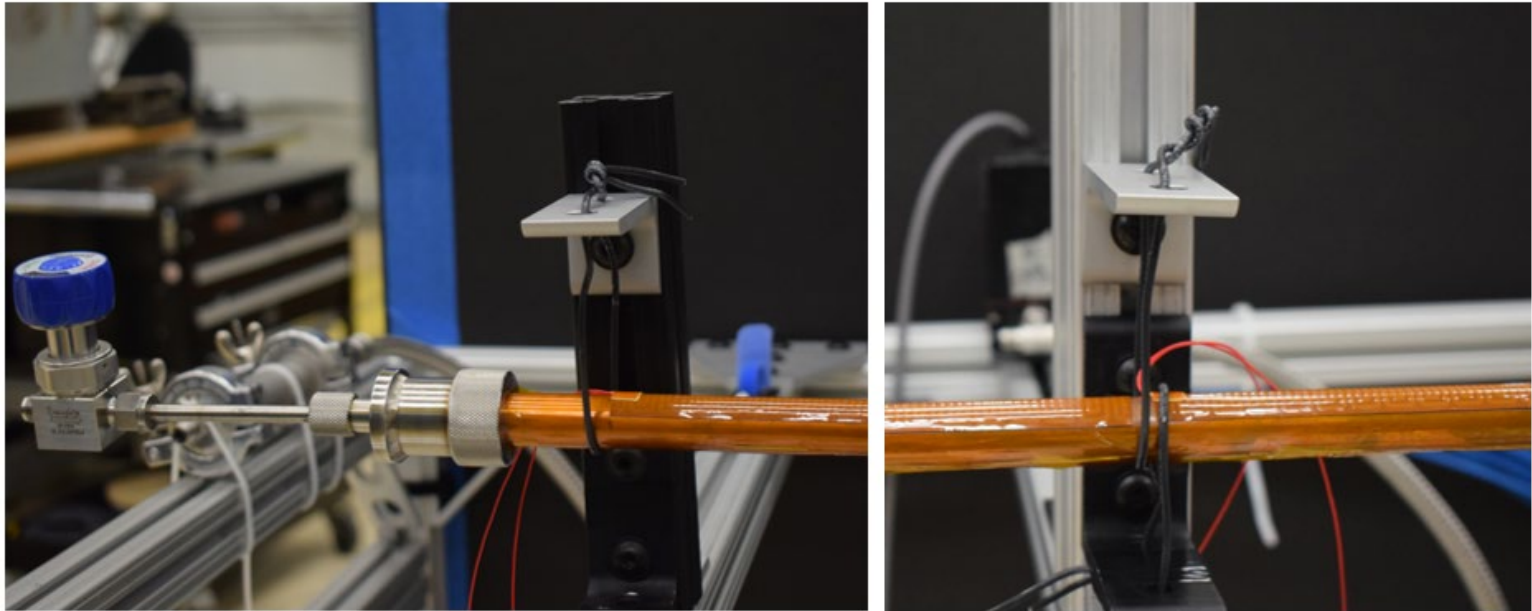


Figure 4.9: (Left) Heat pipe end mounting method and (Right) middle mounting method.

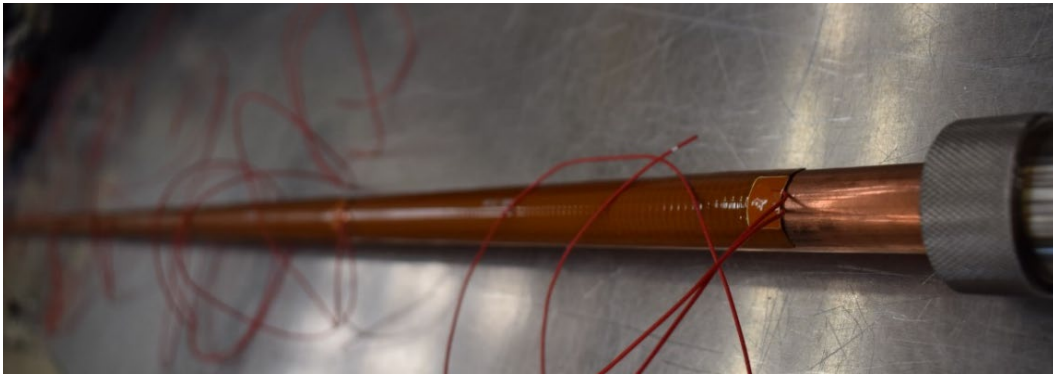


Figure 4.10: Polyimide film heaters installed on heat pipe.

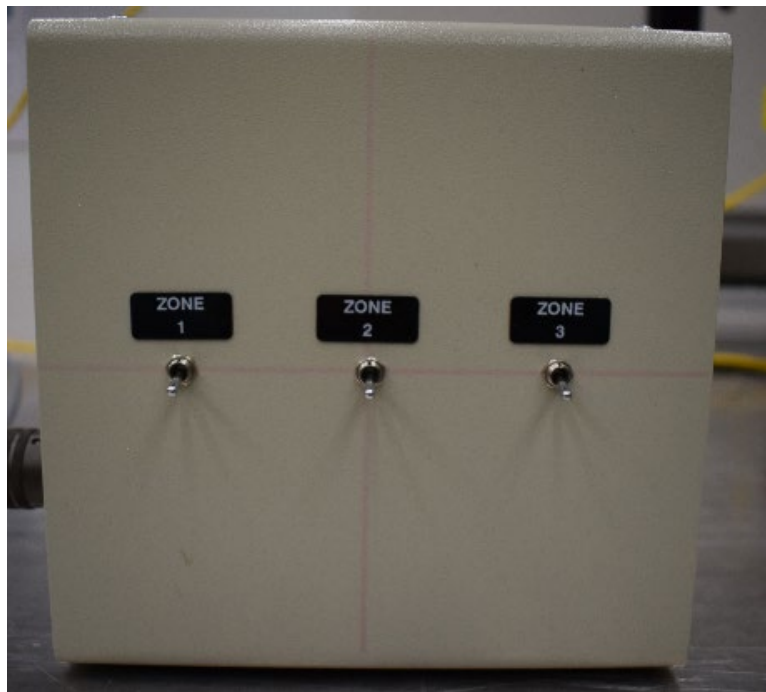


Figure 4.11: Custom heater switch box.

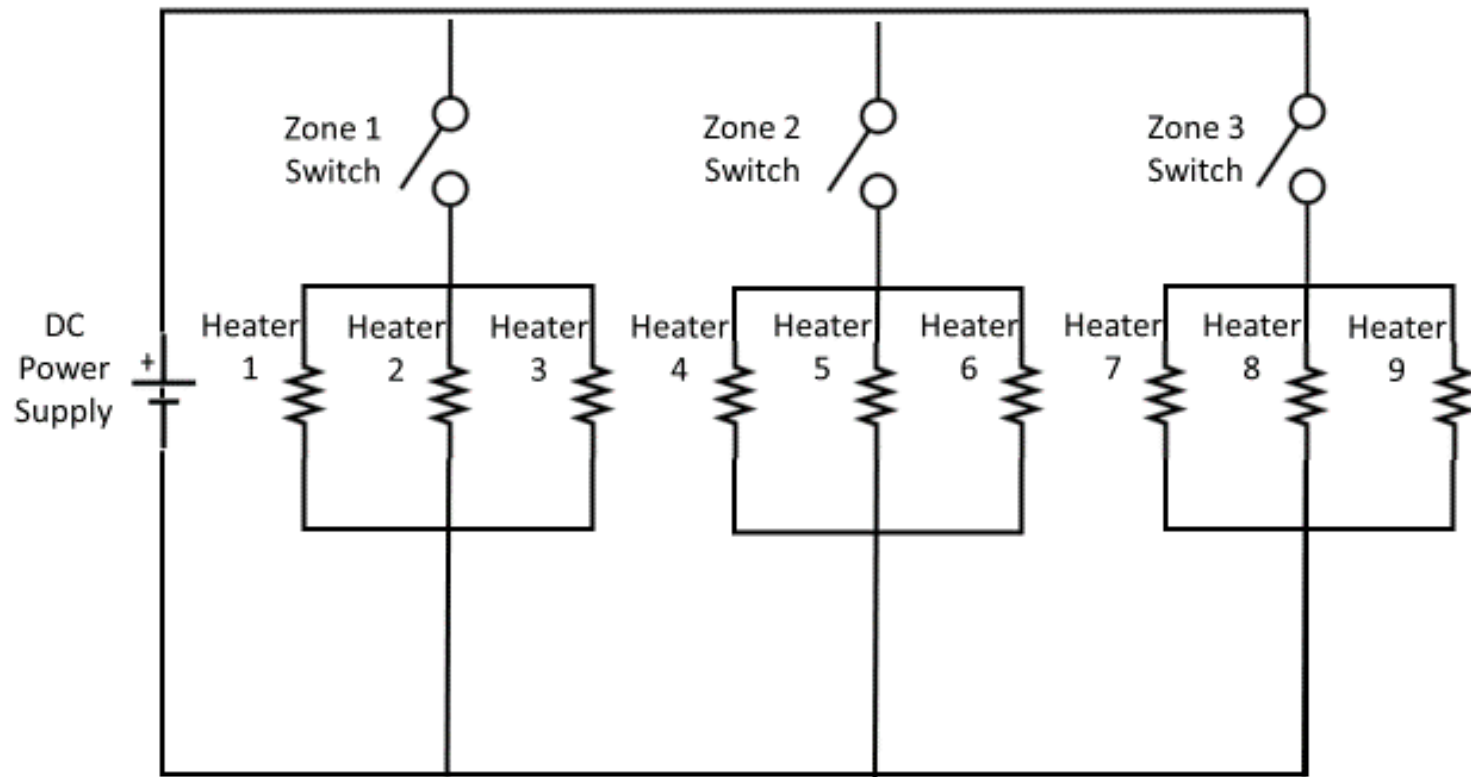


Figure 4.12: Heater switch box circuit diagram.



Figure 4.13: (Left) 0-10 VDC potentiometer and (Right) QB1X pressure controller.

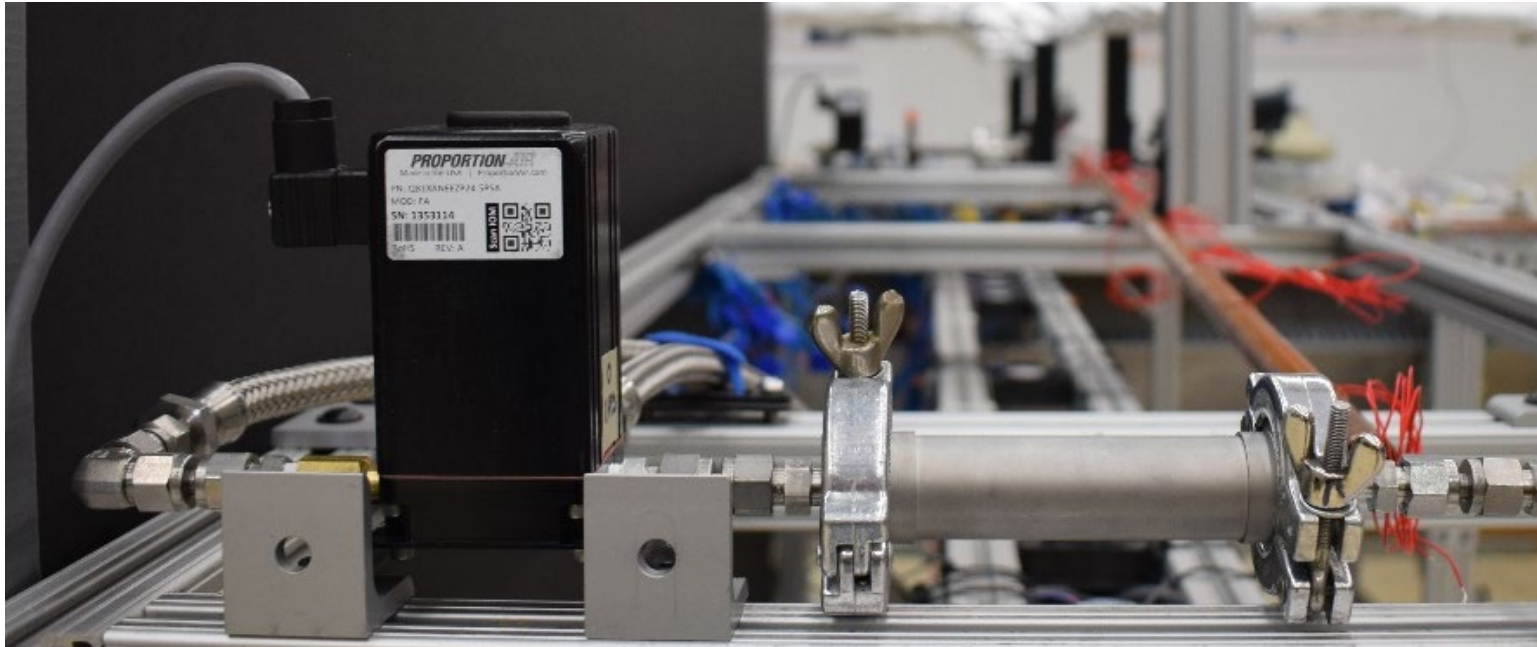


Figure 4.14: KF25 volume for pressure controllers.

Data Acquisition, Instrumentation, and Uncertainty

The heat pipe was instrumented with thirty-one model SA1-T-SRTC surface mount T-type thermocouples from Omega Engineering to provide spatial resolution during testing. An additional thermocouple was used to monitor room temperature. Table 4.1 shows the location of the thermocouples measured from the left end fixture. The thermocouple wires were taped down to the pipe with polyimide tape for at least 4 cm after the junction to reduce lead wire losses. The T-type thermocouples have a temperature uncertainty of $\pm 1.0^{\circ}\text{C}$ ⁶⁷ and a positional uncertainty of ± 0.15 cm. Temperature data was collected using two NI 9213 thermocouple modules plugged into a CDAQ-9174 chassis. The sampling rate was set to 10 samples/second to ensure the transient heat pipe response was captured.

A FLIR A655sc infrared camera was used to observe and record additional temperature gradients on the surface of the pipe. The A655sc has a temperature accuracy of $\pm 2^{\circ}\text{C}$.⁶⁸ To provide uniform emissivity the entire heat pipe was covered in polyimide tape. The normal emissivity of both the polyimide tape adhered to a flattened copper tube and the polyimide heaters adhered to a flattened copper tube was measured using a Gier Dunkle DB-100. Before measurements were taken the instrument was calibrated using AZ technologies gold and ML-85SB standards. The gold standard had an emissivity of 0.019 and the ML-85SB standard had an emissivity of 0.922. After calibration, twelve measurements of both the polyimide tape on a flattened copper tube and polyimide heaters on the flattened copper tube were taken. The mean normal emissivity was measured to be 0.838 with a standard deviation of 0.005. To be conservative, the error was assumed to be equal to ± 0.01 . The measured normal emissivity was used to calculate the hemispherical emissivity using the charts presented by Kauder in 2005.⁷⁴ The calculated hemispherical emissivity is to 0.796 ± 0.01 .

In addition to the thermocouple cards, an NI 9221 was used to record both the voltage output signal from the pressure controllers and the voltage from the heaters. The manufacturer states the pressure controller signal has an accuracy of $\pm 0.2\%$ of the full value. This equates to ± 0.02 VDC or a pressure of ± 0.34 kPa.

Table 4.1: Nominal thermocouple locations.

TC #	Location (cm)	TC #	Location (cm)
1	-108±0.15	17	7±0.15
2	-101±0.15	18	14±0.15
3	-94±0.15	19	22±0.15
4	-87±0.15	20	29±0.15
5	-79±0.15	21	36±0.15
6	-72±0.15	22	43±0.15
7	-65±0.15	23	51±0.15
8	-58±0.15	24	58±0.15
9	-51±0.15	25	65±0.15
10	-43±0.15	26	72±0.15
11	-36±0.15	27	79±0.15
12	-29±0.15	28	87±0.15
13	-22±0.15	29	94±0.15
14	-14±0.15	30	101±0.15
15	-7±0.15	31	108±0.15
16	0±0.15	32	Room

National Instruments provides the error of the voltage reading on the NI 9221 as the sum of the gain error and the offset error, shown in Equation (4.1).⁶⁹

$$\delta V = \pm(0.26\%(V) + 0.27) \quad (4.1)$$

The process presented by Kline and McClintock was used to determine the uncertainty of the thermal power applied by the heaters using a combination of the joule heating equation, uncertainty from the voltage, and the uncertainty of the resistance measurements.^{63,70}

$$\delta Q_{in} = \sqrt{\left(\frac{2V}{R} (0.26\%(V) + 0.27)\right)^2 + \left(\frac{-V^2}{R^2} (0.07\%(R) + 0.07)\right)^2} \quad (4.2)$$

For a few active controlled variable conductance heat pipe tests, an MKS eVision 2 RGA (residual gas analyzer) was used to further investigate the stagnation point formation. The manufacturer states that the RGA has an error of $\pm 2 \times 10^{-11}$ Torr.⁷¹ The pressure controllers were used to add and remove gas to the heat pipe. Helium was added to one side of the heat pipe while air was added to the other. The maximum operating pressure of the RGA was 10^{-6} Torr. To reach this pressure an orifice was added as well as an additional vacuum system setup. A P&ID of this vacuum system is shown in Figure 4.15, and a picture of the test set up is shown in Figure 4.16.

The isolation volume doubled as an initial sample volume. Samples were taken from the right side of the heat pipe to avoid a false positive caused by residual helium in the left sample volume. Before a sample was taken the right sample volume underwent successive dilution by cycling between pulling a vacuum on the sample volume and filling the volume with air. After the successive dilution, the sample valves were opened, and the pressure controllers were used to remove gas from the sample volume. The gas taken from the sample volume traveled to the RGA and was analyzed to ensure it does not contain helium. If the RGA shows helium is still present in the sample volume another dilution process took place. If the sample volume is clean, a sample of the non-condensable gas was taken. To sample the non-condensable gas, the pressure of the sample volume was set to slightly below the saturation pressure of the heat pipe at its current temperature. Once the sample

volume was at the set pressure, the heat pipe isolation valve was opened for a short time and then closed. The sample was then transferred to the main sample volume by evacuating the isolation volume while the valves to the bypass, V4, V6, and V7 were closed to the bypass pumps and the RGA sample valves, V5 and V8 were open. After the sample was transferred V4, was closed to the sample volume and successive dilution of the isolation volume began.

To take samples from the left side of the heat pipe the isolation volume was set to constantly pull vacuum for a considerable amount of time. Successive dilution was unable to be used on this side since the inlet gas was helium. While continuously pulling vacuum valve V4 was closed to the bypass and opened to the sample volume. Then the remaining bypass valves were closed, and the sample valves were opened. Then the isolation valve was rapidly opened and closed. V4 was then closed to the sample volume and opened to the bypass. After several minutes, the bypass valves were opened to clear the sample volume. Before further samples were taken, the volume underwent successive dilution using the pressure controller connected to the air supply. During this cycle, V4 was open to the sample volume and closed to the bypass while all the other sample and bypass valves were open.

Calorimeter Tests

Test Setup

A gas gap calorimeter was designed to allow for improved control of thermal power output during testing. The gas gap calorimeter design was chosen to avoid potential over coupling (removing more thermal power than is being added to the heat pipe) and to allow space for thermocouple attachment on the heat pipe wall. The main body of the calorimeter consists of two concentric stainless-steel tubes sealed by welds. Figure 4.17 shows the concept of the gas gap calorimeter.

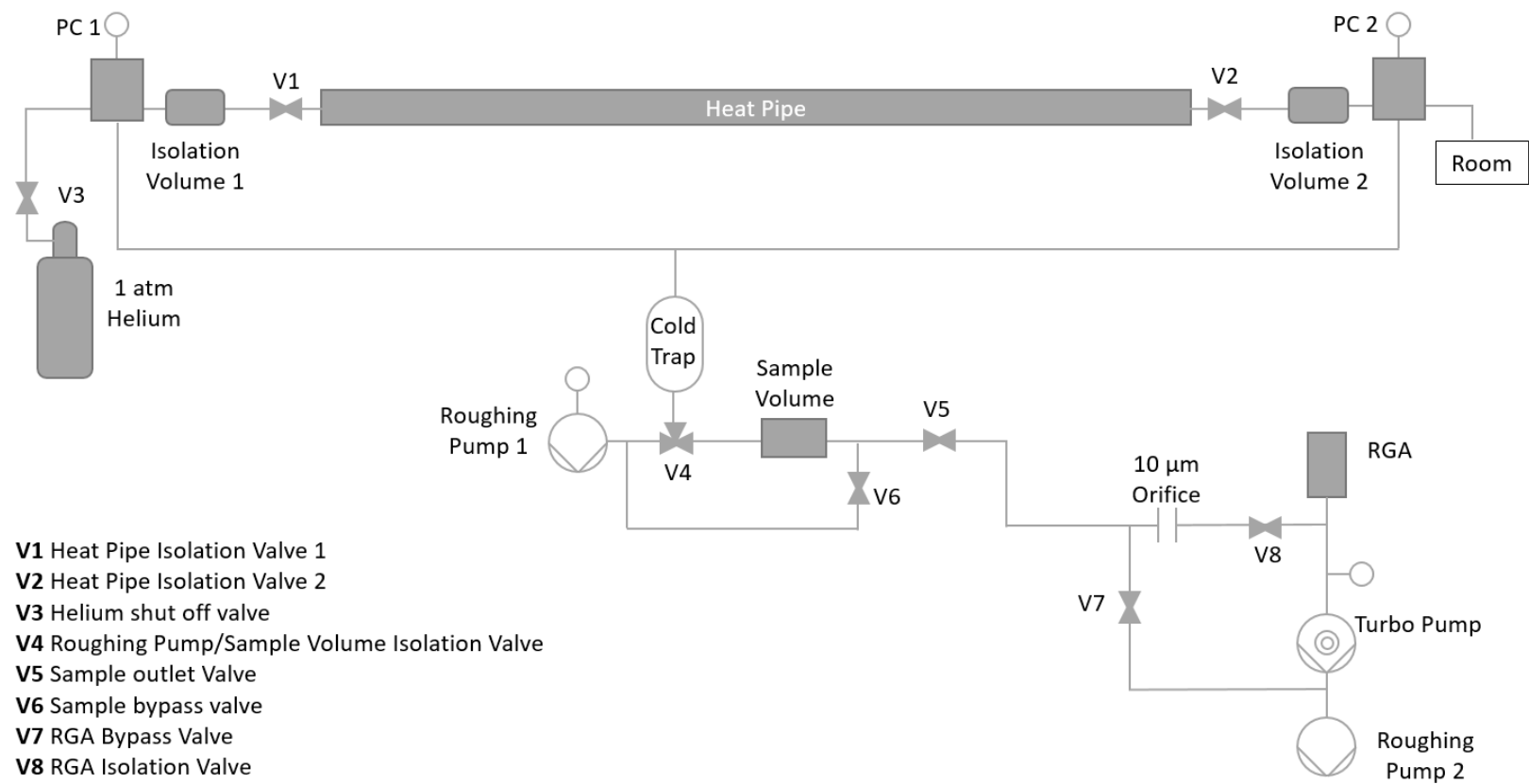


Figure 4.15: P&ID for RGA Vacuum System.

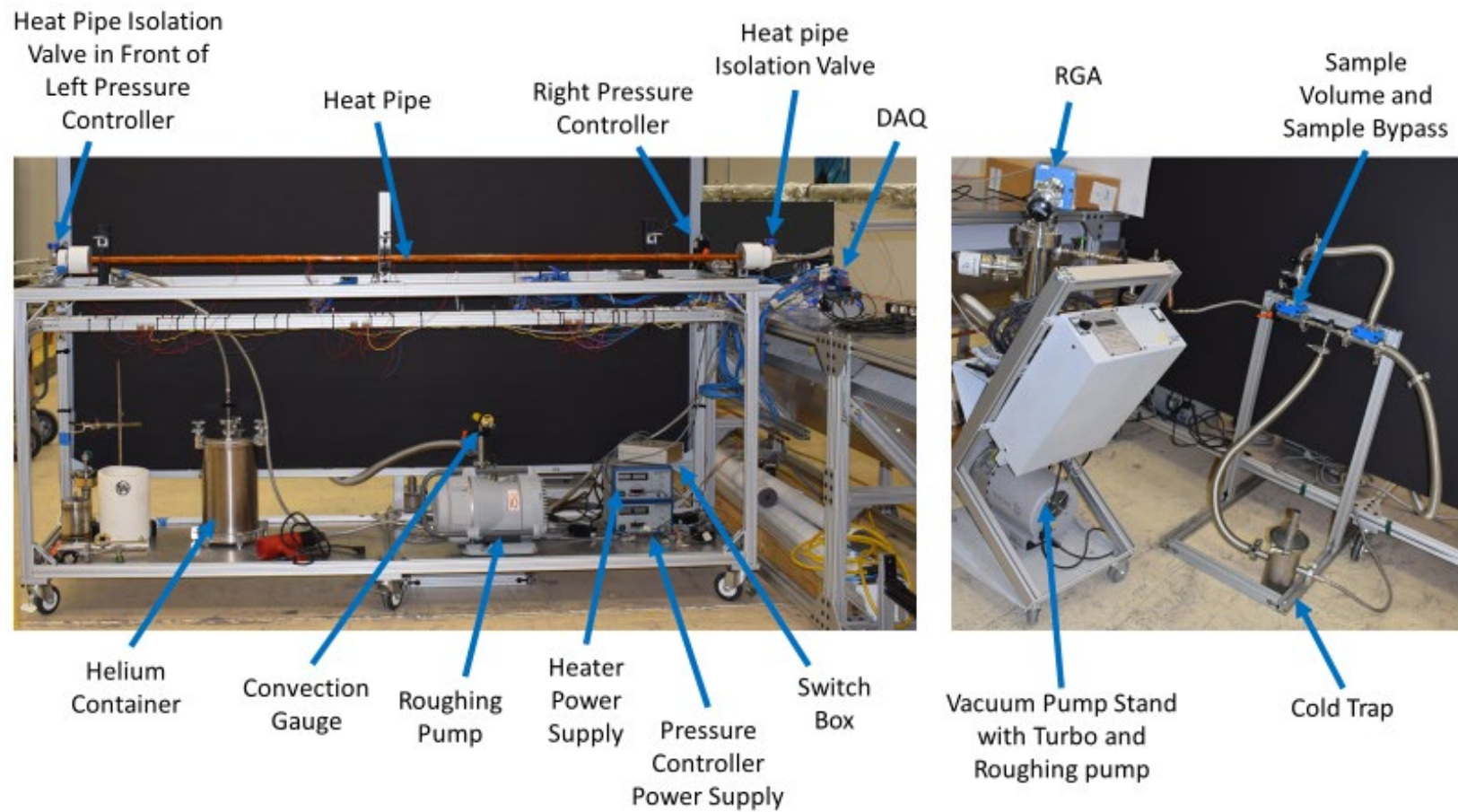


Figure 4.16: RGA test setup from the front (left) and the back (right).

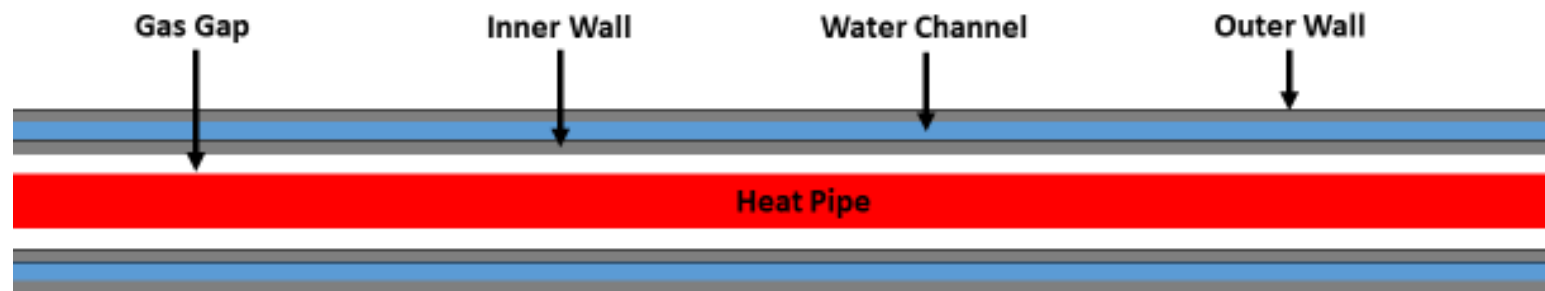


Figure 4.17: Gas gap calorimeter concept.

The thermal conductivity of the gas gap can be manipulated by mixing helium and argon. For the maximum amount of thermal conductivity pure helium would be used, while pure argon would provide the minimum thermal conductivity. To ensure that the primary modes of heat transfer from the heat pipe to the calorimeter are conduction and thermal radiation, the radial gas gap must be small enough to ensure the product of the geometric shape factor and Rayleigh number is less than 100.⁶² If the product is below 100, the natural convection currents are negligible. A maximum gas gap size of 1.6 cm was calculated from equations presented in a heat and mass transfer textbook.⁶² If the radial distance between the heat pipe and the inner wall is smaller than 1.6 cm the primary modes of heat transfer from the heat pipe to the calorimeter are thermal radiation and conduction through the gas gap.

A thermal resistive network analysis was used to determine a combination of tubes that would allow for the calorimeter to remove approximately 100 W, which is the maximum power that the heaters could provide with the attached power supply. It was found that the size of the gas gap was a critical dimension. If the gas gap were too large, the heat pipe would need to reach a much higher temperature to reach a thermal equilibrium. Additionally, the gas gap needed to be large enough to allow the thermocouples to attach to the heat pipe walls without touching the calorimeter walls. The last constraint in determining the gas gap size and needed length of the calorimeter was the size of commercially available tubing. From these constraints, the gas gap was sized to be 0.11 cm for a calorimeter 30.5 cm long. This sizing is below the maximum gas gap size, so natural convection effects are negligible. Vacuum fittings were added to the calorimeter to ensure it could be sealed around the heat pipe, evacuated, and then filled with gas. Two calorimeter assembly were manufactured to accommodate dual condenser centered evaporator testing.

Figure 4.18 shows a model of a calorimeter assembly for a dual condenser centered evaporator configuration. For a single condenser configuration, a second calorimeter and KF40 Tee are connected to the KF40 Tee of the shown calorimeter assembly. The calorimeters were made from stainless steel tubes welded to modified KF40 vacuum fittings. Each calorimeter has approximately 30.5 cm of water-cooled length. Cool water

was chilled and pumped through the system using a Fischer Scientific ISO Temp 1006S. The flow rate of the water supplied to each calorimeter was controlled by a variable Omega rotameter. The inlet and outlet water temperatures were measured using a small diameter (approximately 0.32 cm) RTDs. Before the calorimeters could be installed, the heat pipe was emptied and dried. Once the heat pipe was dried a single thermocouple was removed for the single condenser and single evaporator test and two thermocouples were removed for dual condenser centered evaporator test. These thermocouples needed to be removed to ensure proper calorimeter fit. Additionally, only one zone of heaters was adhered to the heat pipe at a time. Any distance between the heaters and calorimeter or calorimeters were designed to be an adiabatic zone.

Once the calorimeters were installed onto the heat pipe, the heat pipe was filled with 110 ± 15 mL of liquid. A small volume of air was allowed to enter the heat pipe during the fill operation to create a passive variable conductance heat pipe. After the heat pipe was filled, tube insulation with a thermal conductivity of 0.25 W/m K placed around the heat pipe and calorimeter. To reduce the thermal power losses, the heat pipe and calorimeter were further insulated using packing peanuts. Figure 4.19 shows the assembly process of the calorimeter and insulation around the heat pipe.

Data Acquisition, Instrumentation, and Uncertainty

In the calorimeter tests the T-type thermocouples had the same spacing for most thermocouples. The differences in thermocouple spacing were caused by the need to remove thermocouple number 17 in the single condenser configuration and thermocouples 7 and 23 in the dual condenser configuration, thermocouple numbers and locations are shown in Table 4.1. Like the air-cooled tests, the type T thermocouples were connected to two NI 9213 cards and had a sample rate of 10 samples/second. The IR camera was not used to monitor temperature during these tests. Water was supplied to the calorimeters by a ISOTEMP 1006D refrigerated/heated circulation bath and two FL-3804ST Omega rotameters. The ISOTEMP bath pumped water at a consistent set temperature to the calorimeters through the rotameters. Figure 4.20 shows the ISOTEMP 1006D circulator bath and the two rotameters before it was connected to the calorimeters.

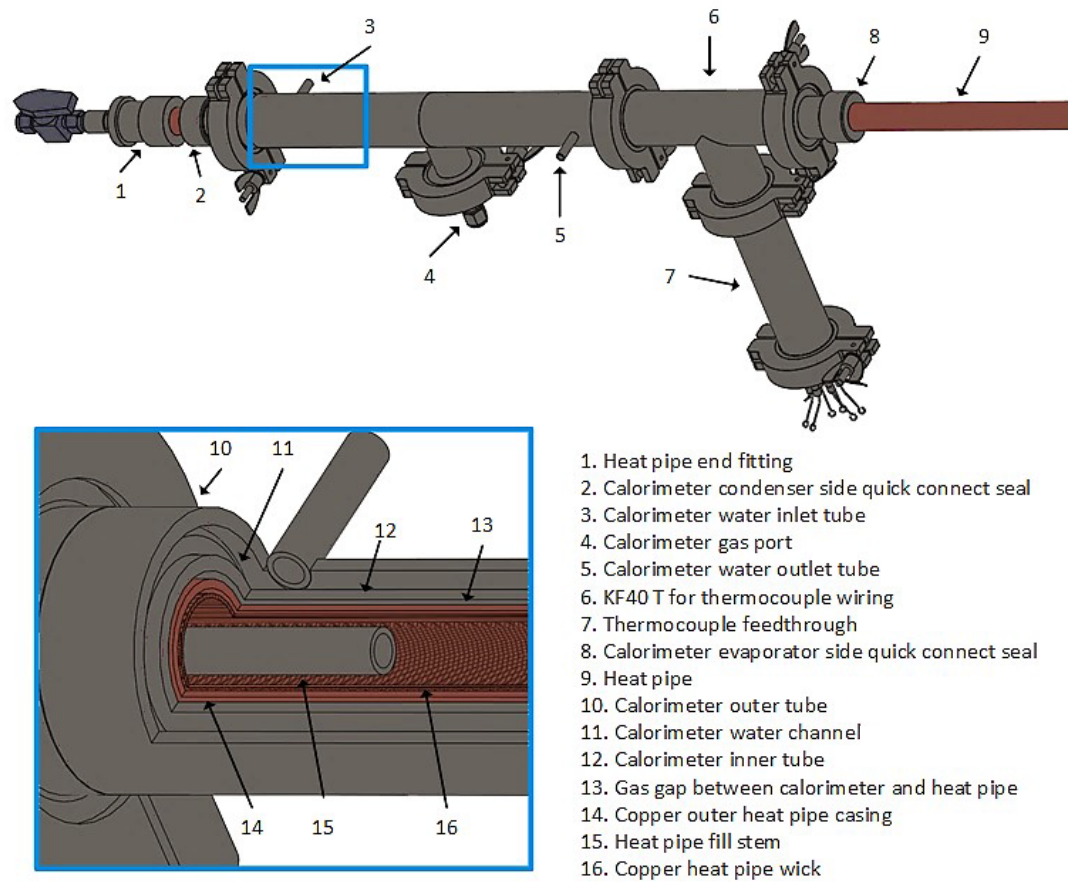
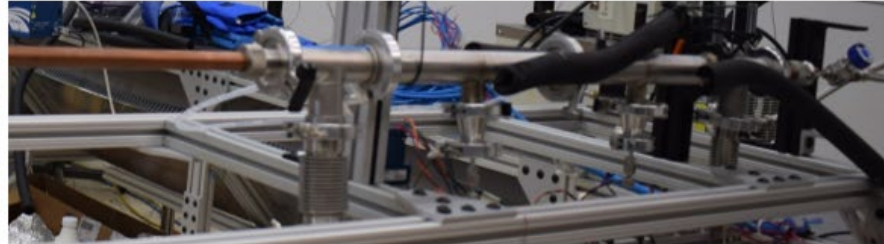
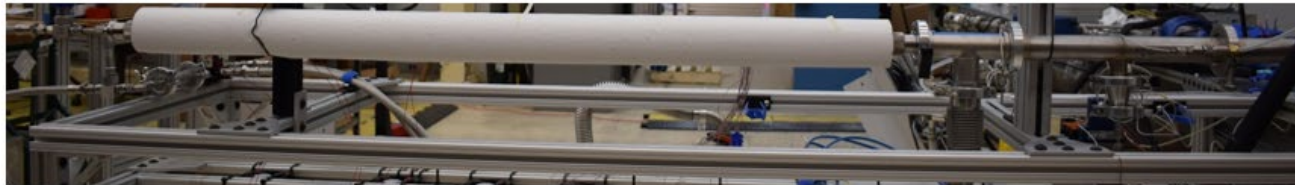


Figure 4.18: Detailed model of gas gap calorimeter assembly.

- 1.) Attach thermocouples, install calorimeters onto heat pipe, and fill heat pipe with working fluid.



- 2.) Insulate evaporator and adiabatic sections with foam tube insulation.



- 3.) Insulate calorimeter with foam tube insulation and install inner packing peanut container.



- 4.) Fill inner container with packing peanuts and add outer packing peanut insulation bags.



Figure 4.19: Calorimeter assembly process.



Figure 4.20: (Left) ISOTEMP 1006D circulator with control box, (Right) ISOTEMP 1006D with Omega rotameters.

The scale manufacturer provides uncertainty of ± 0.1 gram.⁷² The error associated with the timer was conservatively estimated to be ± 2 seconds caused by human error. Additionally, the rotameter scales have an uncertainty of $\pm 2\%$ of the full-scale value. The full scale of the rotameters is 150 mm meaning the scale has an uncertainty of ± 1 mm. Using this information and the methodology presented by Kline and McClintock in 1953, the uncertainty of the mass flow rate was calculated using (4.3).⁶³

$$\delta \dot{m} = \sqrt{2 \left(\frac{\delta m^2}{t_f^2} \right) + 2 \left(\frac{m_f^2 \delta t^2}{t_f^4} \right)} \quad (4.3)$$

In the equation above, t_f is the final time value in seconds and m_f is the mass measured at the final time. Applying Equation (4.3) to the data gathered during the calibration tests, the uncertainty of the mass flow rate was calculated to be $\pm 0.5\%$ of the measured value. Since the error associated with reading the scale, $\pm 2\text{E-}5$ kg/s, is larger than the error associated with the mass flow rate measurement, $2\text{E-}7$ kg/s to $5\text{E-}6$ kg/s, the error associated with reading the scale was used to determine an updated error in mass flow rate using the slope of the calibration curve. From this slope analysis it was found that the error in reading the rotameter scale leads to an error in mass flow rate of $\pm 2\text{E-}5$ kg/s. The mass flow rate measured during calibration compared to the rotameter scale value is shown in Figure 4.21.

The temperature of the water supplied to the calorimeters was measured using two Omega model PR-20-2-100-1/8-2-E-T Resistive Thermal Devices (RTD's) which are class A three wire RTD's with a 1/8-inch diameter. An NI 9216 temperature module was used in the chassis to collect the data. To reduce error, the RTD's were installed as close to the calorimeter as possible. The manufacturer provides Equation (4.4) as the uncertainty of the RTD's.⁷³

$$\delta T_{RTD} = \pm(0.15 + 0.002(T)) \quad (4.4)$$

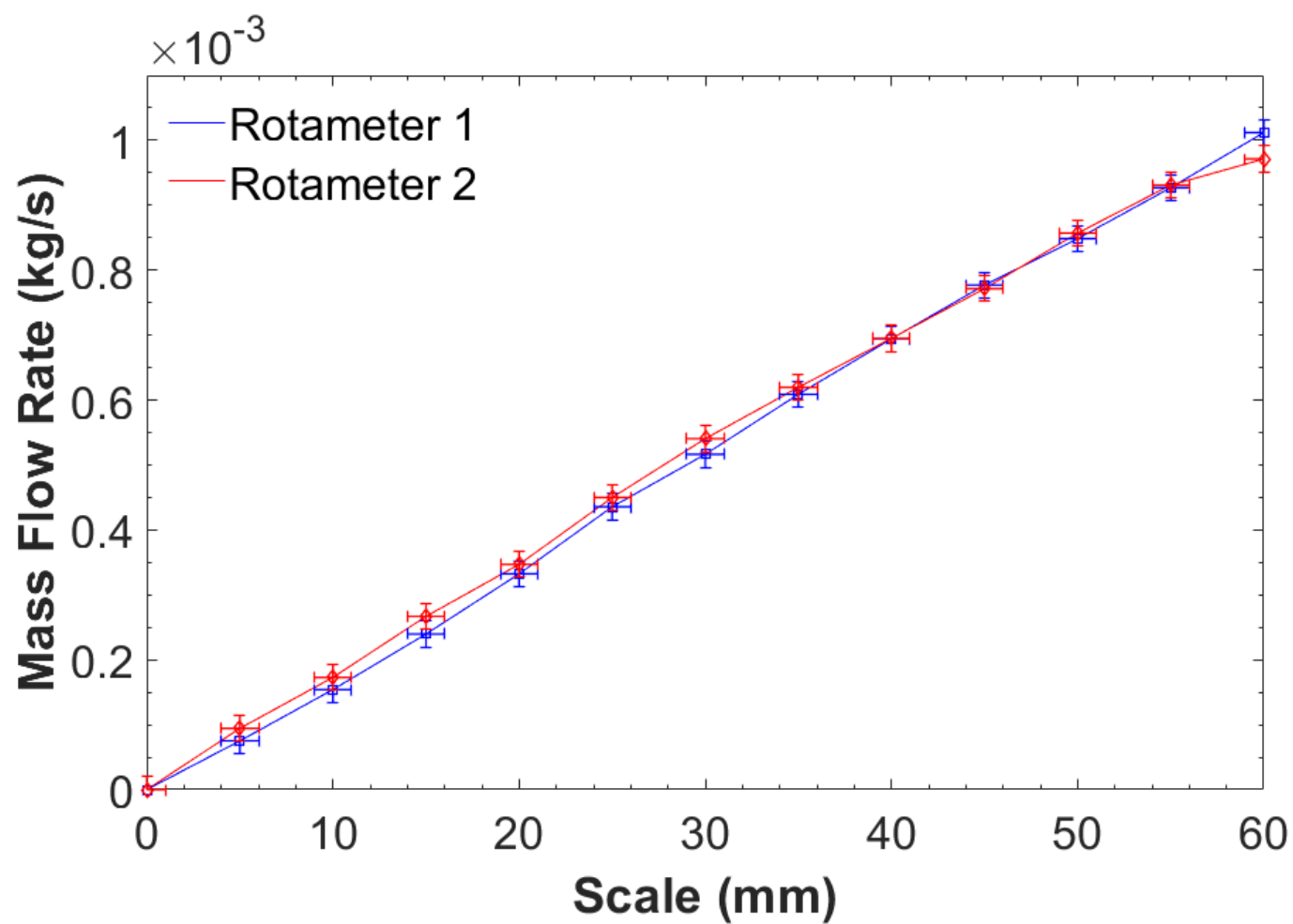


Figure 4.21: Calculated rotameter scale error and mass flow rate error.

Using the error in both mass flow rate and temperature, the error associated with thermal power removed from each calorimeter can be calculated. For consistency, the rotameters were set to a scale value of 45 mm for each of the calorimeter tests. Since the largest error in mass flow rate comes from the error in the scale (± 1 mm), the error associated with this value of $\pm 2.2\%$ is used for the error in mass flow rate. The value of specific heat c_p was assumed to be a known value with no error. Equation (4.5) shows the estimated error in thermal power removed by each individual condenser.

$$\delta Q = \sqrt{(c_p(T_o - T_i)\delta\dot{m})^2 + (\dot{m}c_p\delta T_o)^2 + (\dot{m}c_p\delta T_i)^2} \quad (4.5)$$

In Equation (4.5) T_o is the water temperature at the outlet of the calorimeter, T_i is the water temperature at the inlet of the condenser, $\delta\dot{m}$ is the error in mass flow rate, $\dot{m}$ is the mass flow rate, δT_o is the error in the outlet temperature measurement, and δT_i is the error in the inlet temperature measurement. The errors in the water temperature measurements can be found using Equation (4.4). Since δQ and both δT_{RTD} are dependent on changing values in the data, the values are presented using error bars on the data.

CHAPTER FIVE

RESULTS AND DISCUSSION

The results are divided into six sections. The sections are determined by the thermal boundary conditions applied to the heat pipe during the tests, the heat pipe mode of operation (fixed conductance, passive variable conductance, and active variable conductance), the type of stagnation point in the heat pipe during testing (static stagnation points remain and dynamic stagnation points), and lastly any special diagnostics used during testing. For the purposes of this section the term “static stagnation points” refers to a heat pipe that has a constant number of stagnation points that do not change during heat pipe operation, while the term “dynamic stagnation points” refers to a heat pipe with stagnation points that are added or removed during heat pipe operation.

Air-Cooled Fixed Conductance Heat Pipe with Static Stagnation Points

A set of fixed conductance heat pipe tests was completed to produce a set of control data. In these tests, the heat pipe was slightly underfilled from removing all the non-condensable gas present in the heat pipe after it was filled with water. A thermal power of approximately 65 W was applied to the heat pipe with the film heaters in three different locations. Locating the heaters on one end of the heat pipe generated a single evaporator and single condenser heat pipe, a centered location on the heat pipe created a centered single evaporator and dual condenser heat pipe, and a heater location that was off center by approximately 23 cm created an off centered single evaporator and dual condenser heat pipe. Figure 5.1 shows a schematic of these evaporator and condenser configurations with the evaporator being red and the condenser being white. In the single evaporator and single condenser configuration a short unheated region, L_1 , was allowed between the left end of the heat pipe and the evaporator was introduced to encourage a small liquid pool to form at this end of the wick. For this heat pipe, a small liquid pool seals the wick in this

configuration instead of a using a conventional end plug. Sealing the wick in this configuration helps maintain the heat pipes capability to transfer heat. If the wick remains unsealed, the unsealed wick may decrease the capillary limitation, which is especially important to consider in the single evaporator and single condenser case because of the increased axial pressure losses.

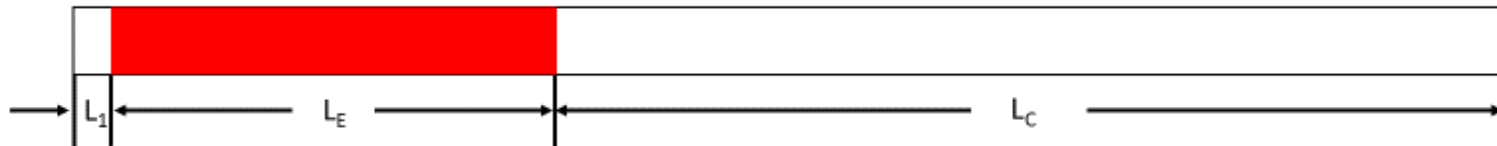
The purpose of these tests was to observe the influence of heat input boundary condition on the wall temperature of the heat pipe and any differences in heat pipe start up trends. Each test reached a quasi-steady state where temperature changes were less than 0.25 °C over a period of 10 min. These tests never reached true steady-state due to the influences of the room's HVAC system on the thermal power balance of the heat pipe.

Figure 5.2-Figure 5.4 show a comparison of results for these tests. From each of the figures the first apparent effect noted is that the evaporator entrance and exit have a larger temperature gradient in the single condenser and single evaporator test. This effect can be seen in every figure for this test. This is likely caused by a partial dry out occurring in the evaporator. The partial dry out may be caused by a low volume of liquid as well as being in the configuration that requires the greatest capillary potential from the wick.

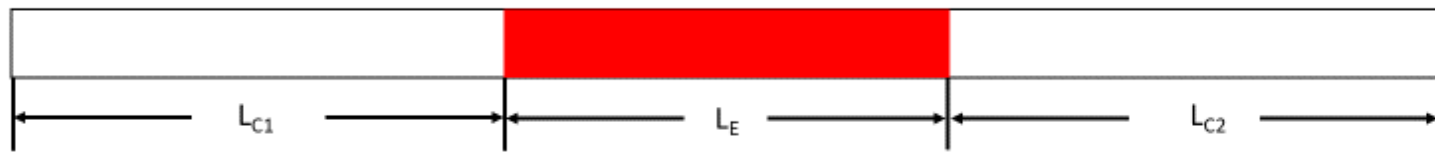
Figure 5.2 shows the thermocouple data plotted as function of time for all three configurations. For clarity in these plots the error bars are not shown and data from selected thermocouples has been removed. Figure 5.3 shows the axial temperature distribution of the during startup for all three configurations. Although these plots have error bars for both location and temperature measurements, the location error bars are very small in comparison to the length of the heat pipe and cannot be seen on the plot. Figure 5.4 shows a top view of a contour plot that includes temperature, location, and time data. Similar to Figure 5.2, the error bars have been left off this plot for clarity.

In Figure 5.2 the single centered evaporator and dual condenser case has the smallest axial temperature distribution (2.1° C) at steady state while the single evaporator and single condenser case has the largest axial temperature distribution (7.0° C). Some of this difference can be attributed to the partial dry out, but some may be caused by the performance enhancement of the centered evaporator as discussed in the theory chapter.

Single Evaporator and Single Condenser



Single Centered Evaporator and Dual Condenser



Single Off Centered Evaporator and Dual Condenser

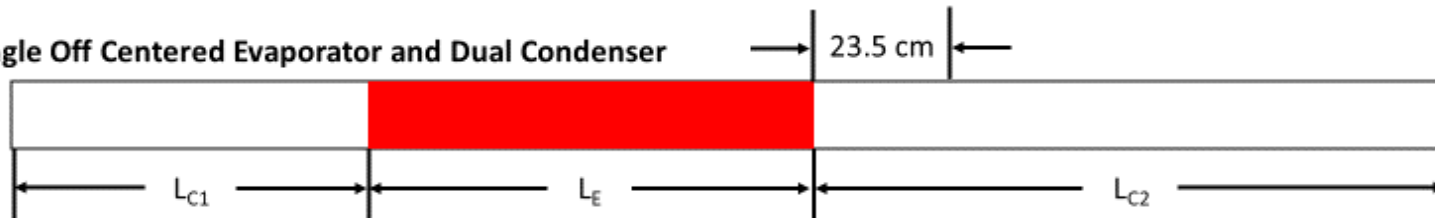


Figure 5.1: Evaporator and condenser configurations for fixed conductance heat pipe tests.

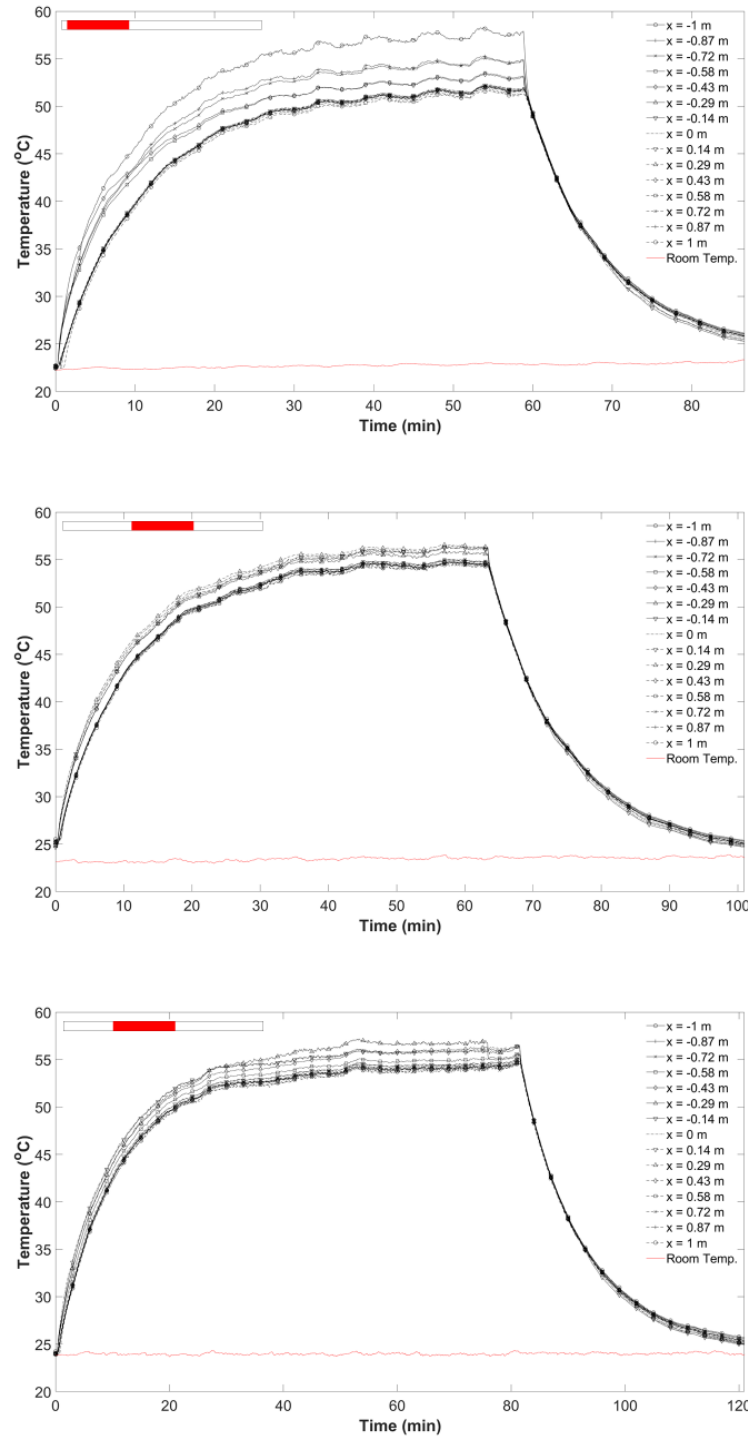


Figure 5.2: Time vs temperature data for (top) fixed single evaporator and single condenser case, (middle) fixed single centered evaporator and dual condenser case, and (bottom) fixed single off centered evaporator and dual condenser case.

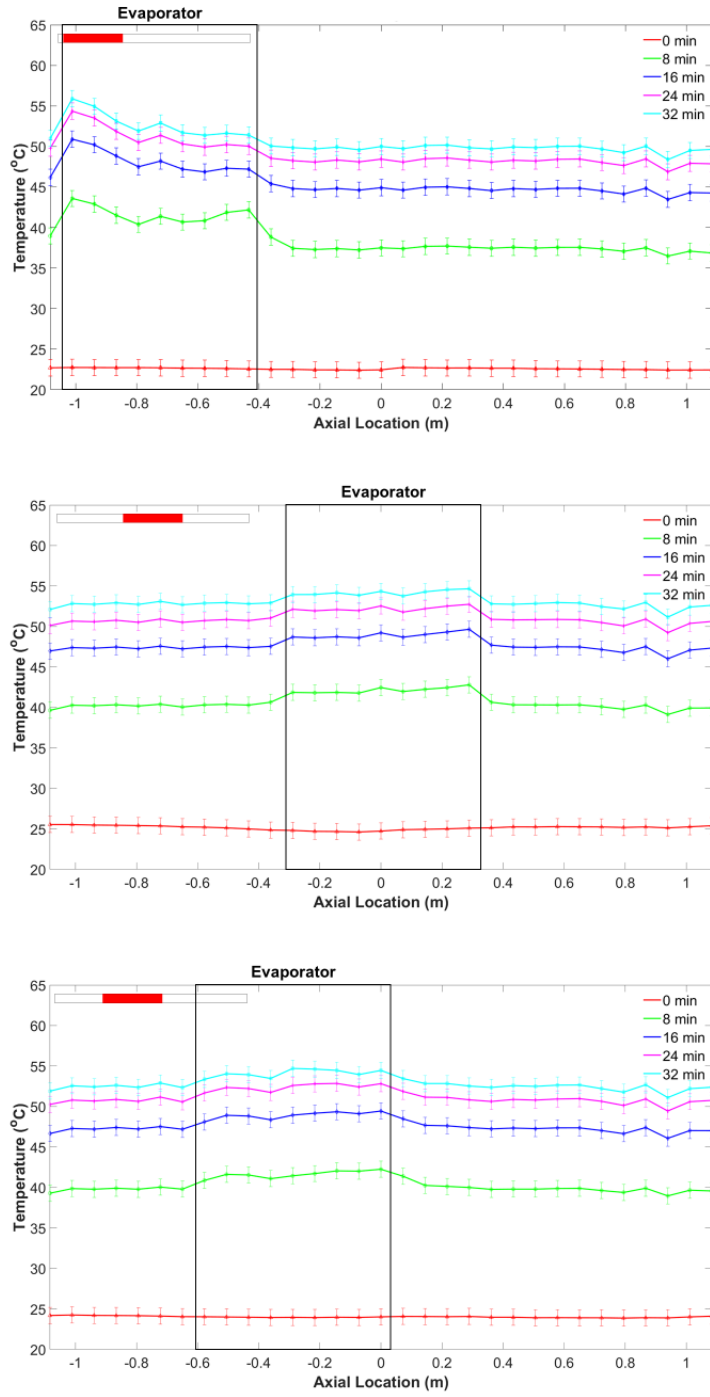


Figure 5.3: Axial location vs temperature at selected startup times for (top) fixed single evaporator and single condenser case, (middle) fixed single centered evaporator and dual condenser case, and (bottom) fixed single off centered evaporator and dual condenser case.

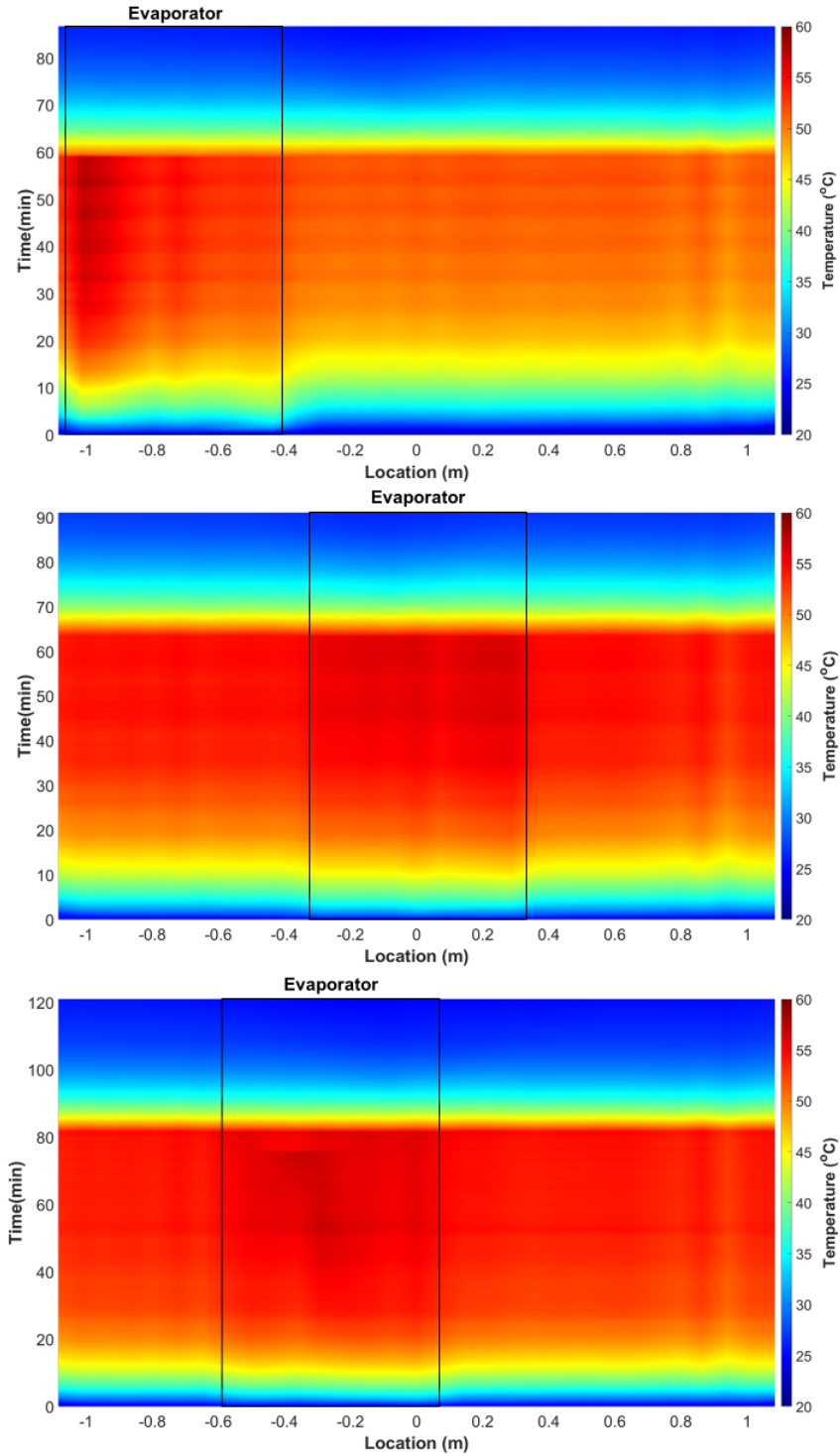


Figure 5.4: Heat pipe surface temperature with time and axial location for (top) fixed single evaporator and single condenser case, (middle) fixed single centered evaporator and dual condenser case, and (bottom) fixed single off centered evaporator and dual condenser case.

The single off centered evaporator has an axial temperature distribution similar to the single centered evaporator case; however, the heat pipe in this configuration experiences a larger distribution from approximately 35 min to 75 min. Part of the increased axial temperature distribution (3°C) is caused by two thermocouples measuring an increased wall temperature before suddenly dropping approximately 1.2°C and returning to a lower value shown in Figure 5.5.

After the neighboring thermocouples returned to a lower value the axial temperature distribution is approximately the same as the single centered evaporator and dual condenser case (2.2°C). The increase in temperature near this point may be caused by a gas or vapor bubble that became trapped in that location. Once the bubble moved the temperature rapidly decreased and the values were closer to those in the rest of the heat pipe. Another anomalous thermocouple is located at 0.87 m. In Figure 5.2-Figure 5.4, this thermocouple consistently reads slightly lower than the others. It is likely that this thermocouple is not maintaining full contact with the heat pipe or there is a defect in the thermocouple.

Figure 5.2- Figure 5.4 all show the evaporator as warmer than the condenser. This is expected as there is temperature loss going across the wall, annulus full of water, and wick in the heat pipe. A resistive network was used to estimate the portion of temperature drop associated with crossing the heat pipe. Figure 5.6 shows is a schematic of a resistive network to estimate temperature drop for this heat pipe.

The axial resistance across the vapor in the vapor channel and liquid in the annulus was calculated using an estimated ΔT from an HTPIPE simulation with corresponding thermal power and evaporator exit temperature. For this heat pipe at this power level and temperature, the minimal temperature drop between the evaporator and condenser is approximately 0.24°C . Additional sources of temperature drop may include the partial dry out in the single evaporator and single condenser configuration and other non-optimal heat pipe conditions. Some of these non-optimal heat pipe conditions include a bent or unlevel heat pipe which may increase the temperature drop associated with both the vapor and liquid flow. Another source of non-optimal heat pipe behavior may be from changes in wick position (concentric vs eccentric) or changes in the contact angle caused by oxidation occurring on wick and wall.

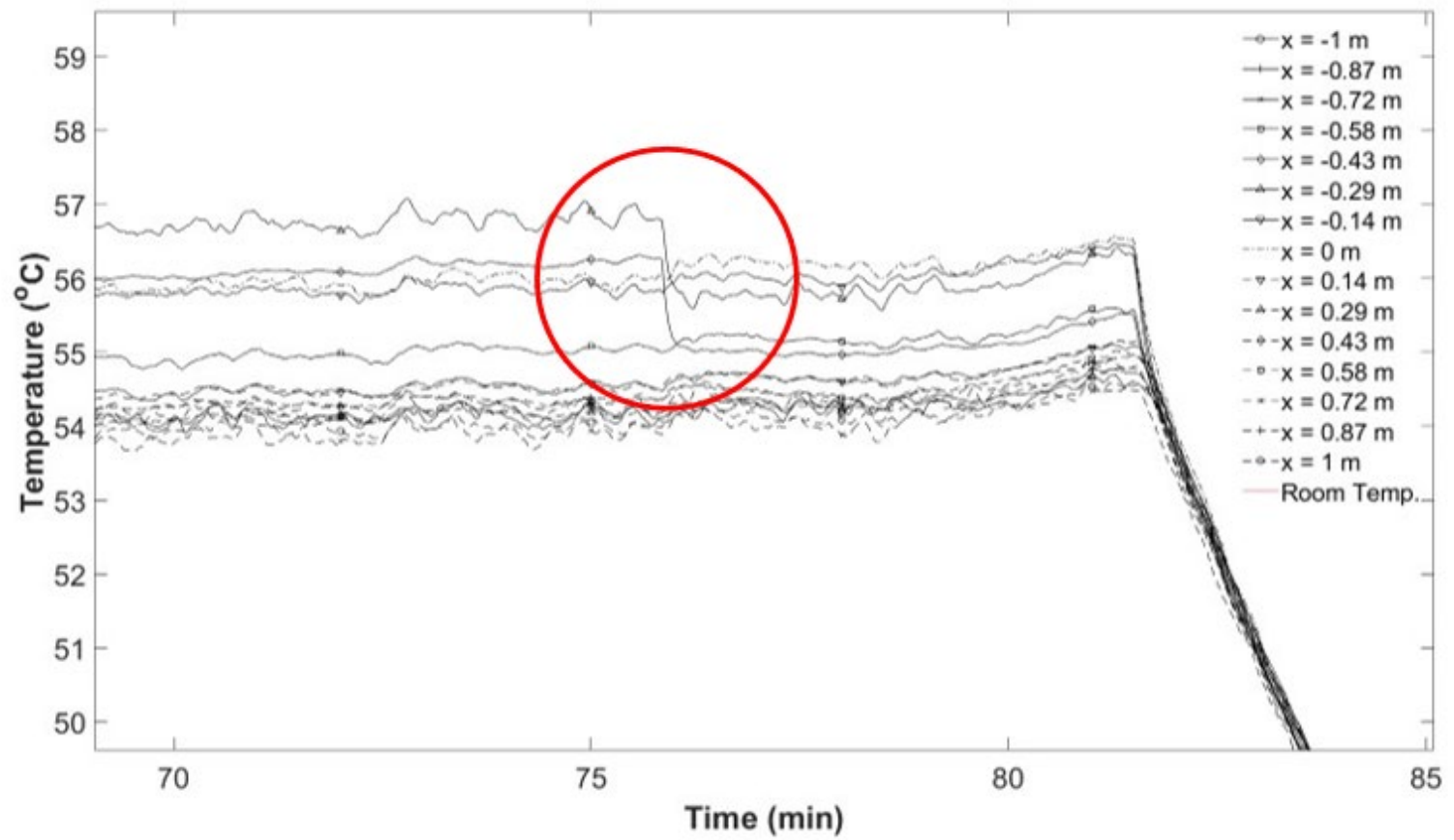


Figure 5.5: Sudden temperature change observed in fixed single off centered evaporator and dual condenser case.

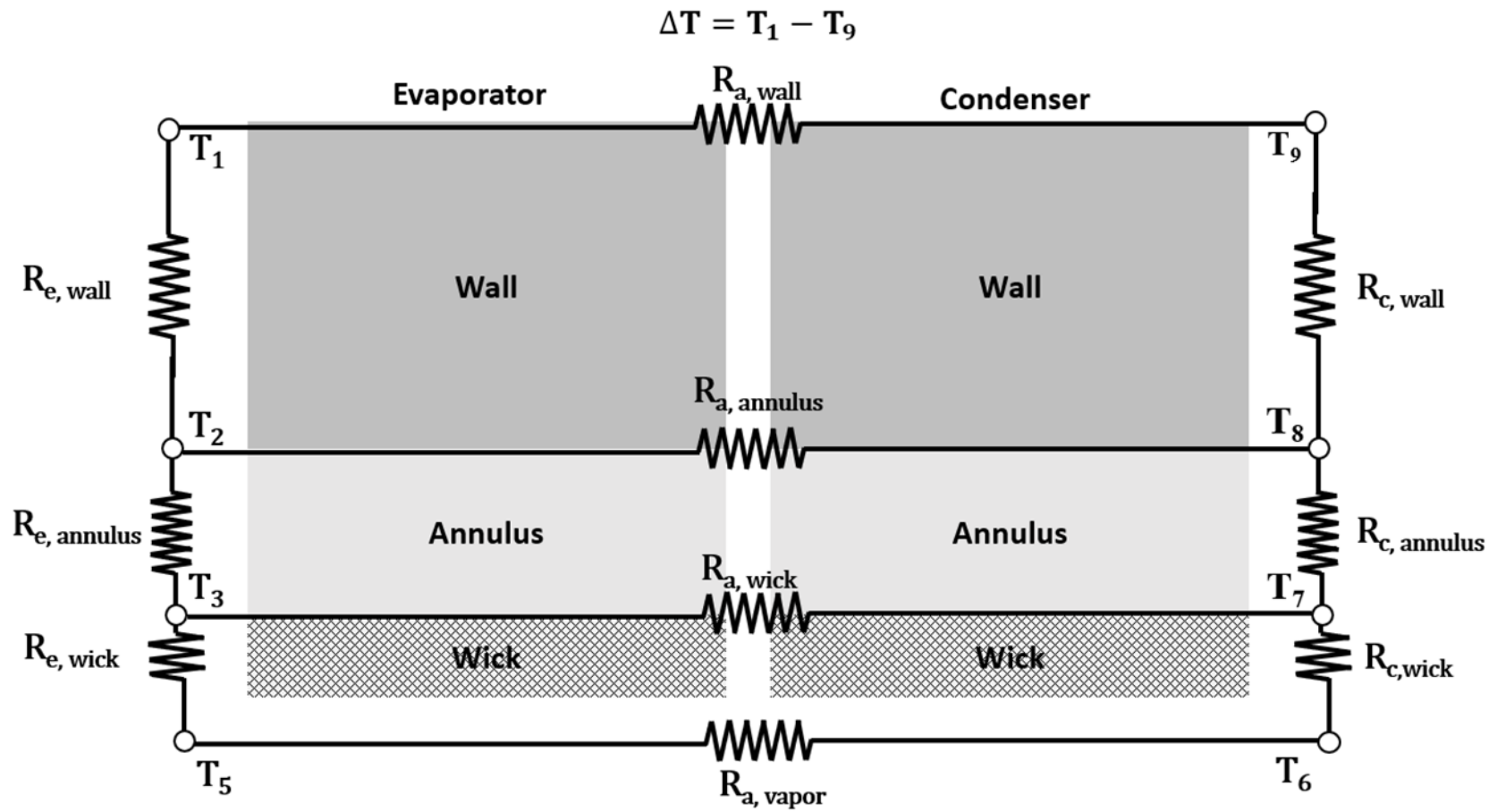


Figure 5.6: Resistive network to estimate temperature drop associated with the heat pipes wall, annulus, and wick.

Lastly in Figure 5.2 and Figure 5.4, there is a small amount of time where the evaporator begins to heat up before the condenser. This time delay is consistent with an estimated thermal time constant for the evaporator area of about 15 min. During cooling the entire pipe remains close to isothermal as it cools since the entire pipe is cooling from similar conditions. This leads to apparent uniform cooling profile.

Overall, these tests show that this heat pipe does not need any major design changes to operate in different evaporator and condenser configurations. Additionally, there were no significant differences in condenser temperatures in the dual condenser cases, suggesting that the thermal power distributes to each cell of the heat pipe in a similar way as the theory describes.

Air-Cooled Passive Controlled Variable Conductance Heat Pipe with Static Stagnation Points

These tests used the same three configurations as the fixed conductance tests: a single evaporator and single condenser configuration, a single centered evaporator and dual condenser configuration, and a single off centered evaporator and dual condenser configuration. In these tests a small amount of air was introduced into the heat pipe before operation. The goal of these tests was to determine if the non-condensable gas would surround cold stagnation points as the theory suggests. In the first three tests the heat pipe was allowed to start up to nearly steady-state conditions before it was shut down. During each of the three tests, 75 W was applied to the heat pipe in the evaporator and the heat pipe dissipated heat into the surroundings. Each test was run to a nearly steady-state condition where the temperature changed by less than 1°C in a 10-minute period. Figure 5.7 through Figure 5.13 present data resulting from these tests.

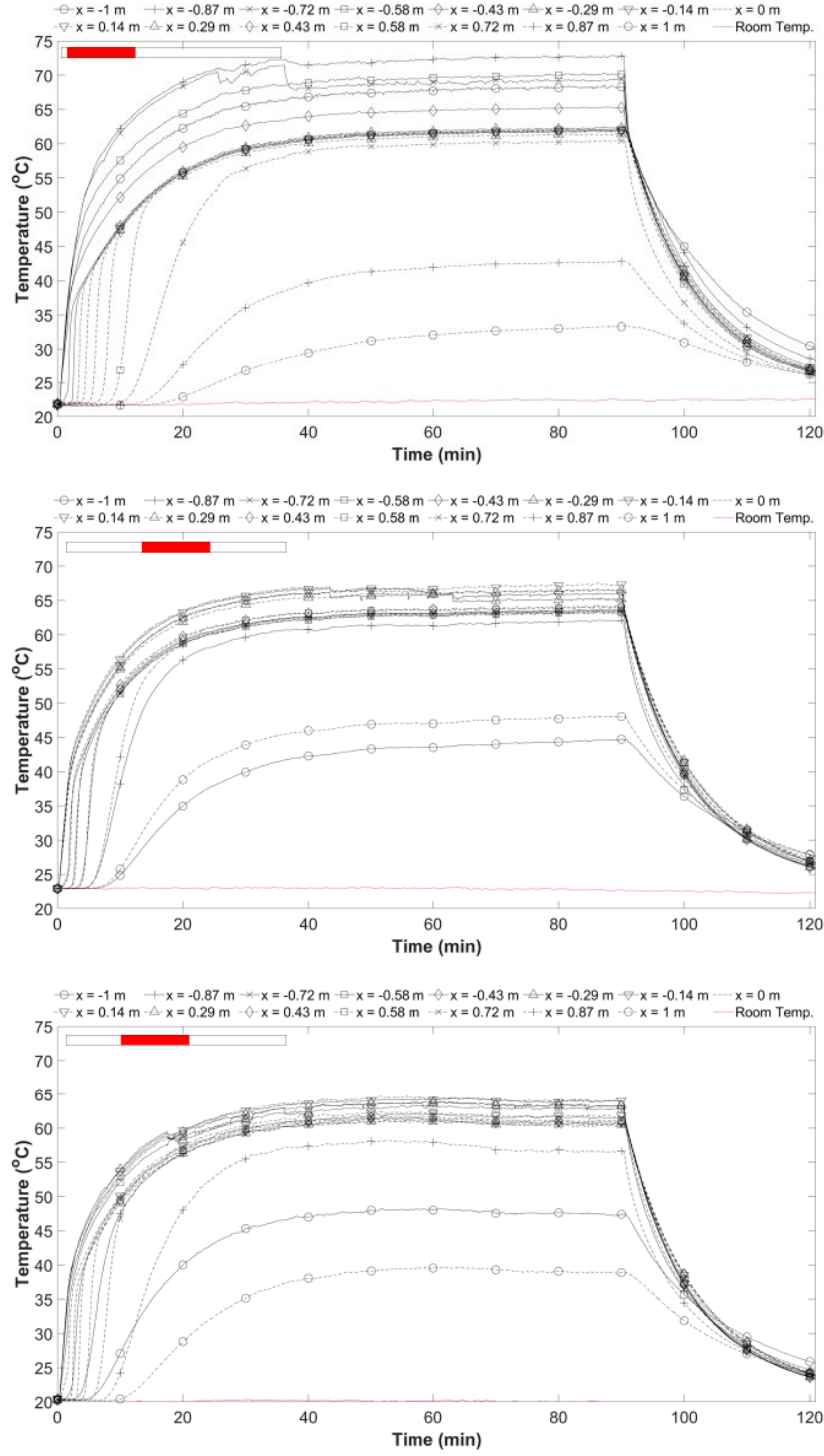


Figure 5.7: Time vs temperature data for the (top) passive single evaporator and single condenser case, (middle) passive single centered evaporator and dual condenser case, and (bottom) passive single off centered evaporator and dual condenser case.

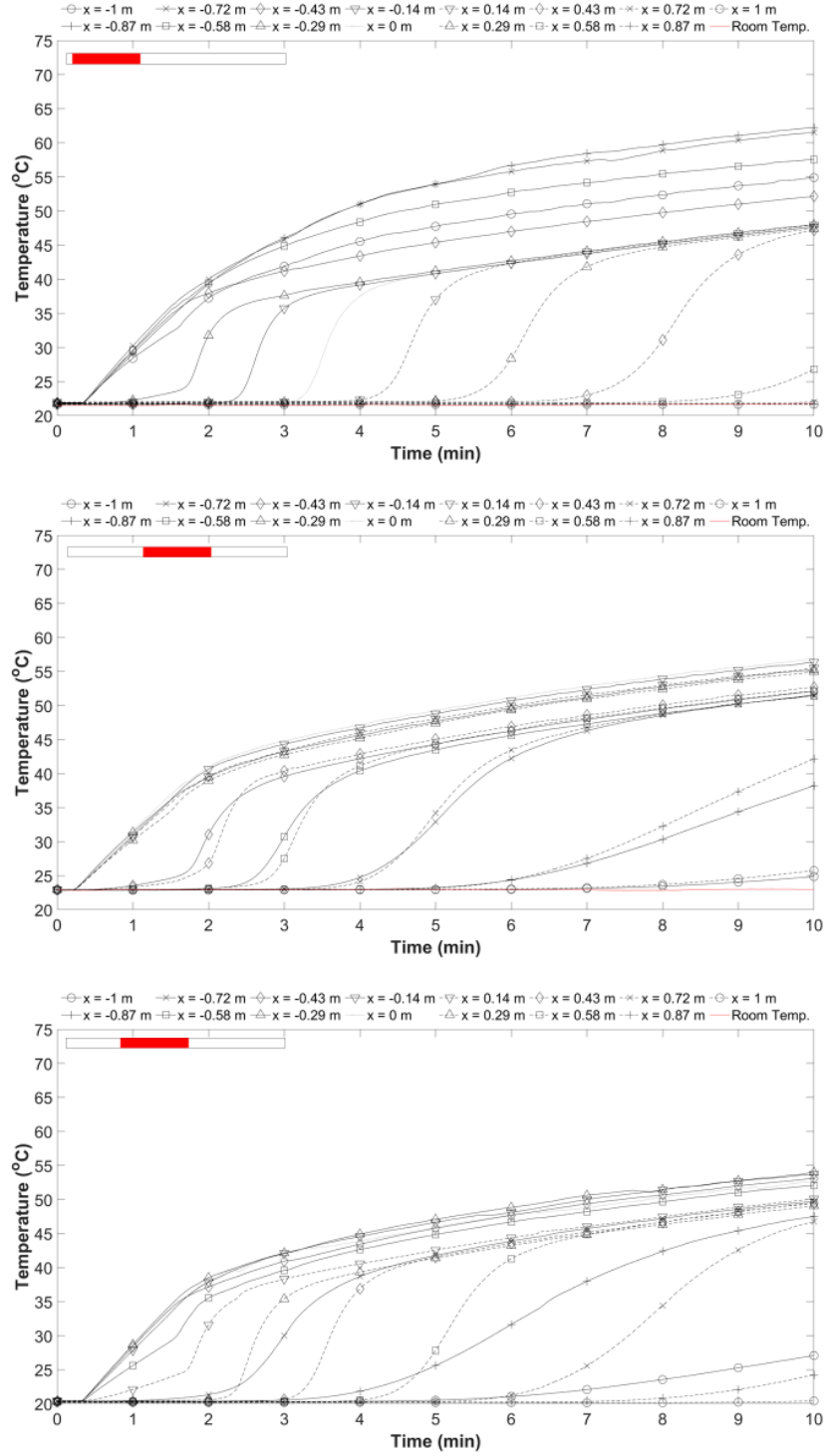


Figure 5.8: Time vs transient temperature data for the (top) passive single evaporator and single condenser case, (middle) passive single centered evaporator and dual condenser case, and (bottom) passive single off centered evaporator and dual condenser case.

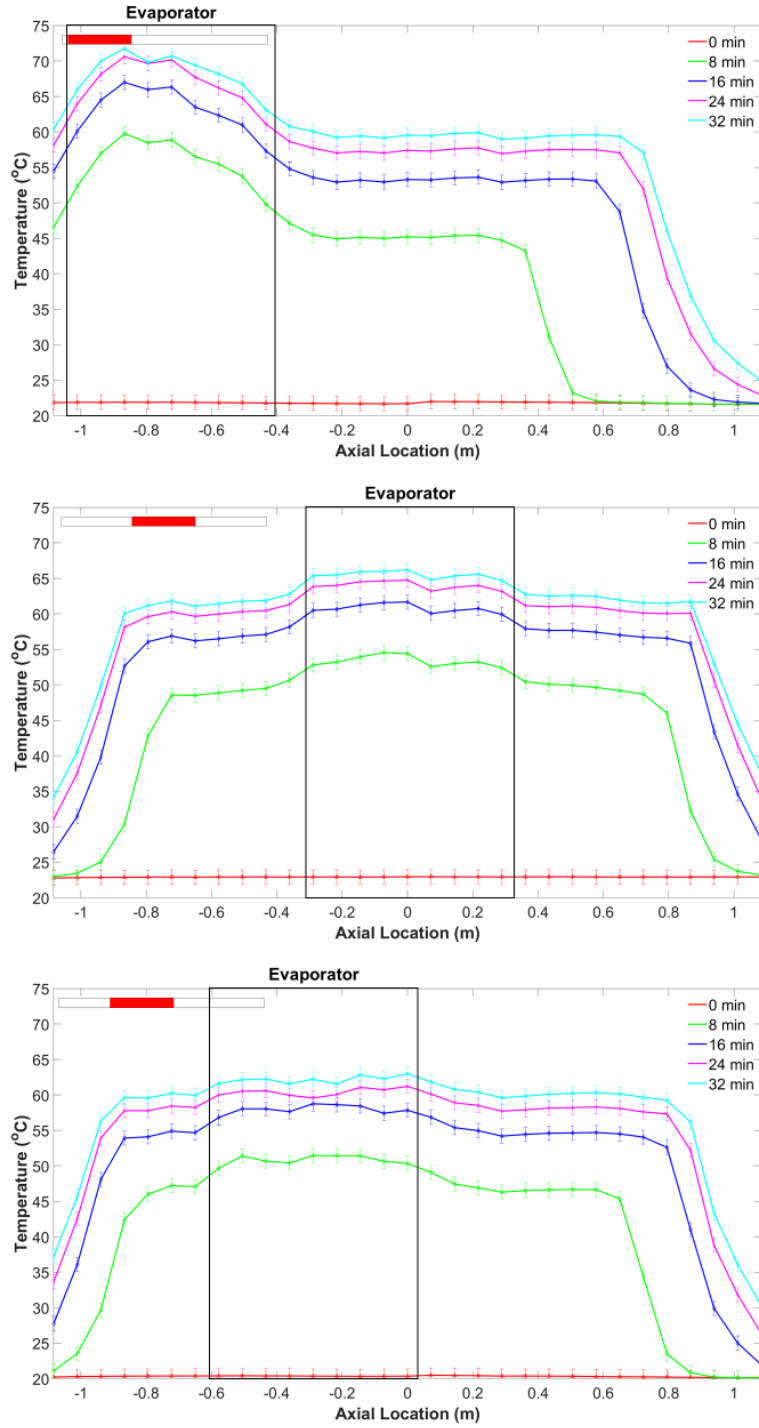


Figure 5.9: Axial location vs temperature at selected startup times for (top) passive single evaporator and single condenser case, (middle) passive single centered evaporator and dual condenser case, and (bottom) passive single off centered evaporator and dual condenser case.

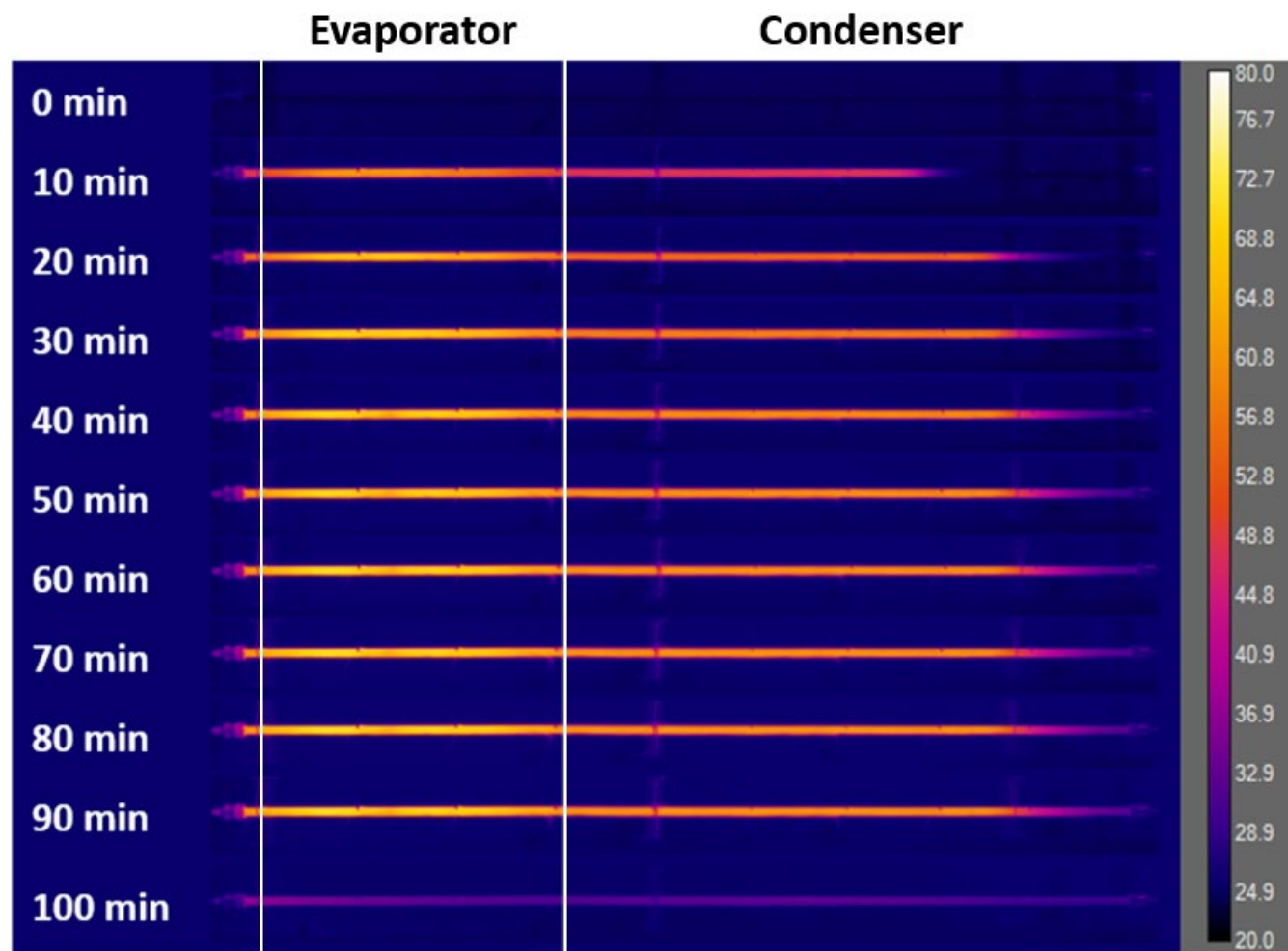


Figure 5.10: Infrared images at selected times for passive single evaporator and single condenser case.

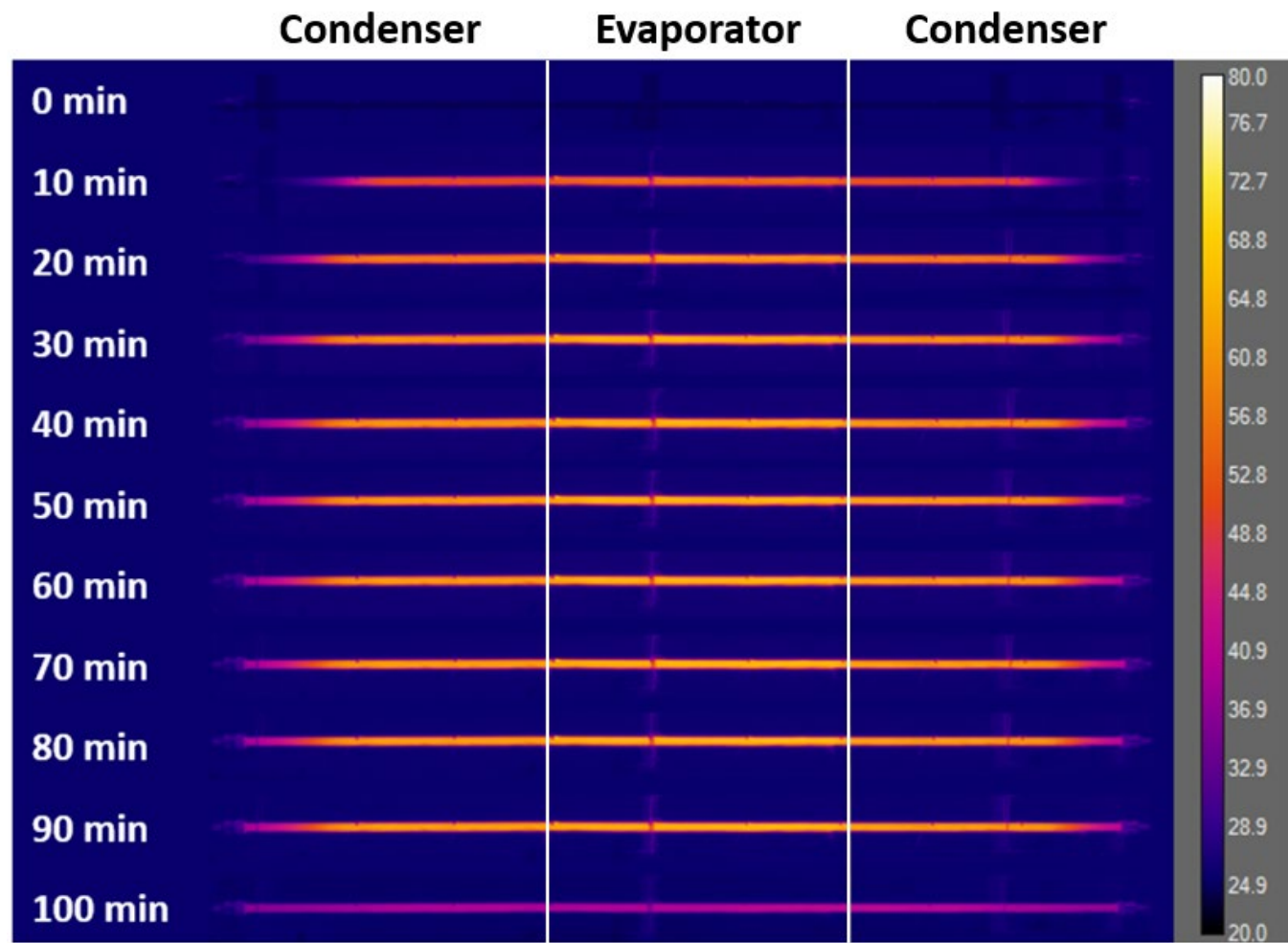


Figure 5.11: Infrared images at selected times for passive single centered evaporator and dual condenser case.

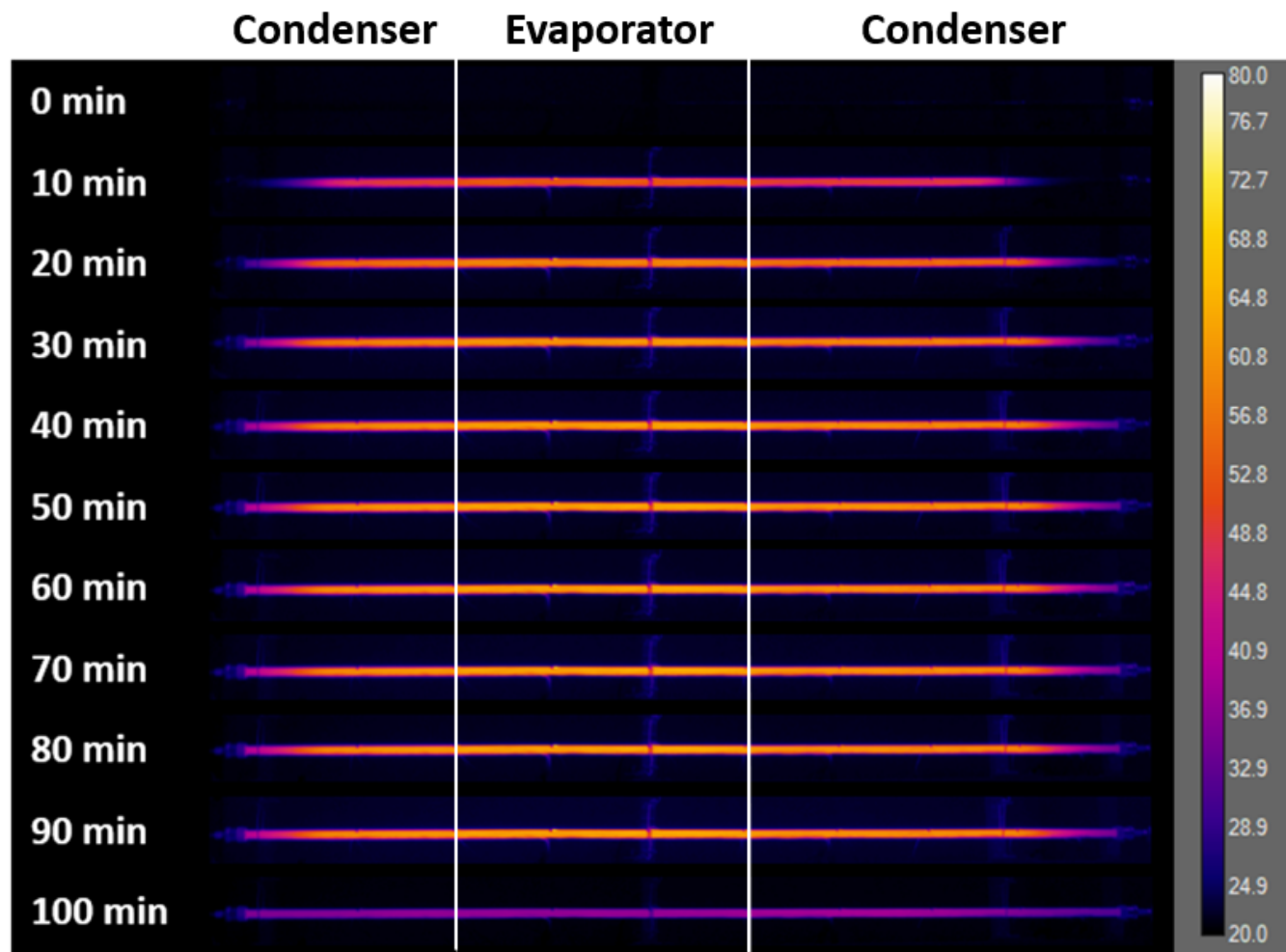


Figure 5.12: Infrared images at selected times for passive single off centered evaporator and dual condenser case.

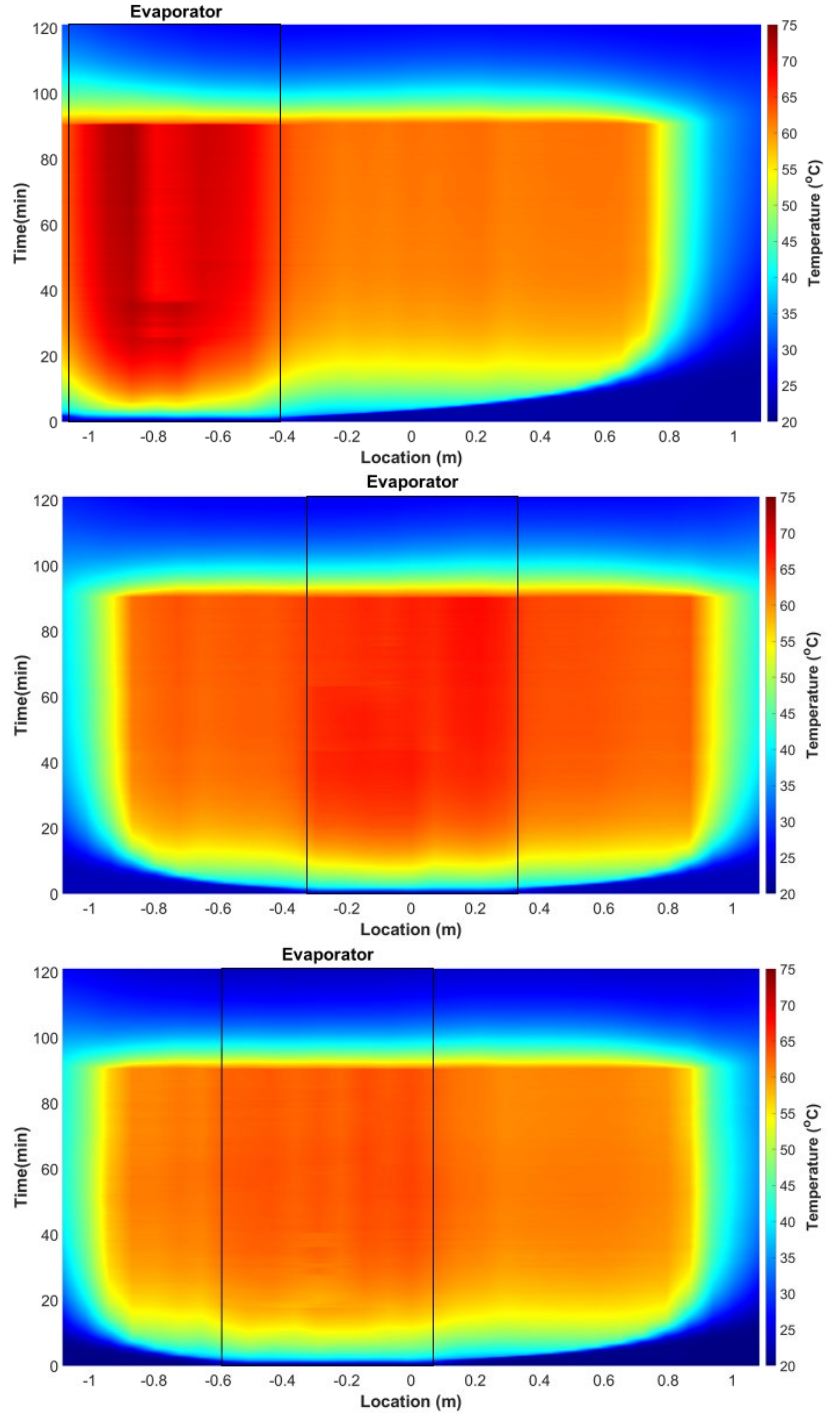


Figure 5.13: Heat pipe surface temperature with time and axial location for (top) passive single evaporator and single condenser case, (middle) passive single centered evaporator and dual condenser case, and (bottom) passive single off centered evaporator and dual condenser case.

Figure 5.7 through Figure 5.13 show that the single evaporator and single condenser case has a significantly warmer evaporator than the evaporators in the other configurations. This warmer evaporator may indicate a partial “dry out” of the wick in that region caused by limitation, undercharging, or a non-condensable gas bubble in the wick or annulus. Figure 5.8 shows the non-condensable gas front slowly getting compressed. The figure suggest that the evaporator may be slightly off center in the single centered evaporator dual condenser case. This is shown by the temperatures of locations equally spaced from the center of the heat pipe rising at slightly different times. This is further illustrated by the single off centered evaporator dual condenser case where a time lag of about 5 minutes is shown between the thermocouples located 0.7 m from the center of the heat pipe.

Figure 5.7 shows a similar trend to Figure 5.2, where when the heat pipe is nearly at steady state, the single evaporator and single condenser case has a larger temperature difference between the evaporator and active condenser regions. It should also be noted that passive single off centered evaporator and dual condenser case reaches a lower temperature than other cases. The lower temperature may be caused by a room temperature that was approximately 2-3 °C lower during this test. Figure 5.9 shows the temperature as a function of axial location for selected times. The difference in evaporator and condenser temperatures after startup is smaller in the dual condenser configurations which is consistent with the results of the fixed conductance tests. This decreased temperature drop may be caused by a lower vapor pressure drop in the dual condenser configurations. Figure 5.9 shows that during startup in the single centered evaporator and dual condenser case, the temperature on the right side (positive side) of the heat pipe is slightly higher than the temperature on the left. This is consistent with the evaporator being slightly off center. Post-test measurements found that heaters were shifted by 0.2 cm to the right of center.

Figure 5.10 through Figure 5.12 show infrared images of the heat pipe from 0 to 100 minutes in 10-minute increments. In all three cases the radial temperature of the heat pipe is close to the same in the active region of the heat pipe. Additionally, the additional spatial resolution shows that the gas region has a large temperature gradient and does not have a clear flat vapor and non-condensable gas interface. Early in the startup of all three cases, the vapor-gas interface appears to have a slight parabolic temperature gradient in Figure

5.10 through Figure 5.12. This may result from a combination of wall conduction in the heat pipe, diffusion of the vapor into the non-condensable gas, or asymmetric vapor flow. In later times this profile is not as noticeable, but it may be hidden by the end fitting.

Figure 5.13 shows a comparison of temperature surface plots of the heat pipe over the time of the test. For the single evaporator and single condenser test, the evaporator region is warmer than the other cases. At approximately 40 min, a cooler spot appears in the evaporator at around -0.8 m. When comparing this to Figure 5.5, it shares similar behavior as Figure 5.2. It is plausible that without an end plug and having a small amount of spacing between the wick and the wall of the end fitting, some air entered the annulus before operation. If air is present in the annulus, it may become trapped by the liquid flowing in the wick and annulus in the evaporator section. The air would get stuck in the evaporator as the liquid in the wick and annulus is flowing from the condenser towards the evaporator during operation. The temperature drop observed could be from the non-condensable gas being pushed back into the vapor channel as the temperature approaching the evaporator entrance remains elevated. Another explanation is that the heat pipe in this configuration is experiencing a partial dry out similar to that seen in the fixed conductance tests.

A power balance using measured temperatures and heat pipe dimensions estimated a combined heat transfer coefficient that included effects from thermal radiation, natural convection, and forced convection. Forced convection was incorporated to compensate for the air movement caused by the HVAC. For the power balance the heat pipe was broken into thirty-one isothermal control volumes. A heat transfer coefficient was independently calculated for each segment because of temperature gradients along the non-condensable gas region. The power balance determined the heat transfer coefficient was on the order of $15 \text{ W}/(\text{m}^2 \text{ K})$. To be conservative the heat transfer coefficient was assumed to be accurate within 20% of the calculated value.

For this heat pipe the inactive condenser region includes both the non-condensable gas and any liquid pool that may be present. The liquid pool behaves in the opposite way of the non-condensable gas. As the heat pipe increases in temperature the liquid pool may increase in length and the opposite occurs when the heat pipe decreases in temperature.

The length of the liquid pool may be determined iteratively with knowledge of the liquid charge. However, in the case of this heat pipe it become more difficult to determine length of the liquid pool because of uncertainty in the fill amount associated with adding and removing non-condensable gas and with the 4.1 cm long end fittings. Since water is not a solid at the room temperature, it is easy for the pressure controllers to remove water along with the non-condensable gas when approaching a desirable non-condensable gas charge. Using the recorded data, a steady state analysis was completed with the Newton Raphson method and Equation (2.21). This analysis estimated the length of inactive condenser region.

Table 5.1 shows the estimated inactive condenser region length for each of the configurations tested. In the dual condenser with a centered evaporator case the gas regions on each side are reasonably close to the same. As expected from the thermocouple data, in the off centered evaporator dual condenser case there is a smaller gas region on the side with the smaller condenser. These calculated lengths were used to estimate the axial temperature profile of the heat pipe. If the location in the model exceeded the non-condensable gas length, the temperature was assumed to be equal to the base temperature. The base temperature was calculated as the average temperature of the active condenser region.

Figure 5.14 uses these calculated lengths with Equation (2.21) to estimate the axial temperature profile of the inactive condenser region. When compared to the measured axial temperature profile the fin approximation underestimated the temperature towards the cold stagnation point. This is likely caused by additional thermal storage from the end fittings.

In a fourth test the heat pipe was set up in a single off centered evaporator and dual condenser configuration. In this test the heat pipe underwent a change in thermal power input. Additionally, this test had a lower quantity of non-condensable gas inside of the heat pipe during operation. The goal of this test was to observe the behavior of multi-condenser variable conductance heat pipes in an environment where thermal power input may not be steady and to determine if the changes in power lead to hysteresis in the length of non-condensable gas on each side. Results for this test are shown in Figure 5.15 through Figure 5.17.

Table 5.1: Estimated non-condensable gas length in condensers for air cooled passive variable conductance heat pipe tests.

Case	Length of Left Inactive Condenser Region	Length of Right Inactive Condenser Region
Single Evaporator and Single Condenser	N/A	38.7 cm
Single Centered Evaporator and Dual Condenser	24.5 cm	23.2 cm
Single Off Centered Evaporator and Dual Condenser	21.9 cm	29.0 cm

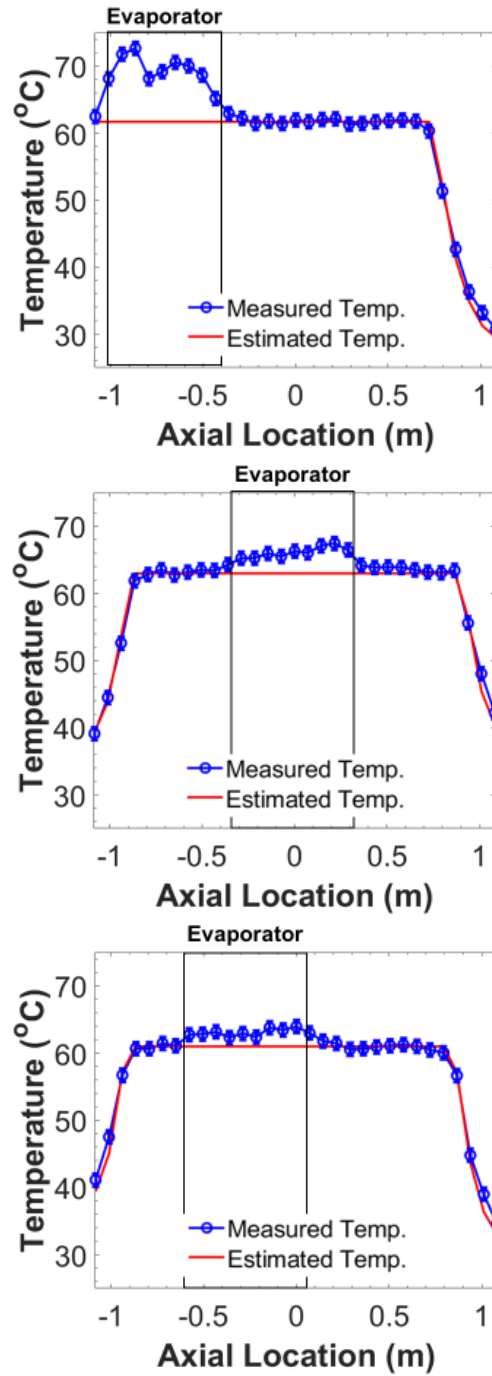


Figure 5.14: Adiabatic fin approximation compared to measured data for (top) passive single evaporator and single condenser case, (middle) passive single centered evaporator and dual condenser case, and (bottom) passive single off centered evaporator and dual condenser case.

Figure 5.15 through Figure 5.17 show that when the heat pipe experiences an increase in thermal power, the non-condensable gas is compressed as the heat pipe increases in temperature. Then when power decreases the non-condensable gas expands with the decreasing temperature of the heat pipe. From Figure 5.17 the non-condensable gas appears to be in similar locations for both instances of heating with a thermal power of 25.0 ± 1.1 W and cooling with 25.0 ± 1.1 W. However, when examining Figure 5.15 and Figure 5.16 the two sections of the data that have a thermal power of 25.0 ± 1.1 W are not at the same temperature. The active condenser regions temperature in Condition A was 0.7 °C lower than the temperature at the same point in Condition C. This difference in temperature could be caused by a combination of the heat pipe not being at steady state at the times shown as well as slight differences in the thermal power applied to the heat pipe and room temperature. When reviewing the voltage applied to the resistive heaters a small difference between the voltage applied for the first 45 minutes and the voltage applied for the time between 105 minutes and 155 minutes was observed. This leads to a thermal power of 24.7 ± 1.1 W for the first 45 minutes and a power of 25.4 ± 1.1 W between 105 minutes and 155 minutes. Additional differences in temperature may be caused by the heat pipe not being at steady state in one or both conditions. To investigate this, the temperature data was evaluated at times approaching 45 min and 135 minutes shown in Figure 5.18.

Figure 5.18 shows that the heat pipe at both 45 minutes and 135 minutes it likely still approaching an equilibrium temperature and have not reached a true steady state at those points in time. In the time leading up to 45 minutes the temperature of the nominal temperature of the heat pipe increases by 1 °C and in the time leading up to 135 minutes the nominal heat pipe temperature decreases by about 1.2 °C. Although this is a small difference in temperature especially considering the thermocouple error of ± 1 °C, the trend shows that the heat pipe was likely not at steady state at either 45 minutes or 135 minutes. This could easily account for the remaining temperature difference that was not expected from the power balance.

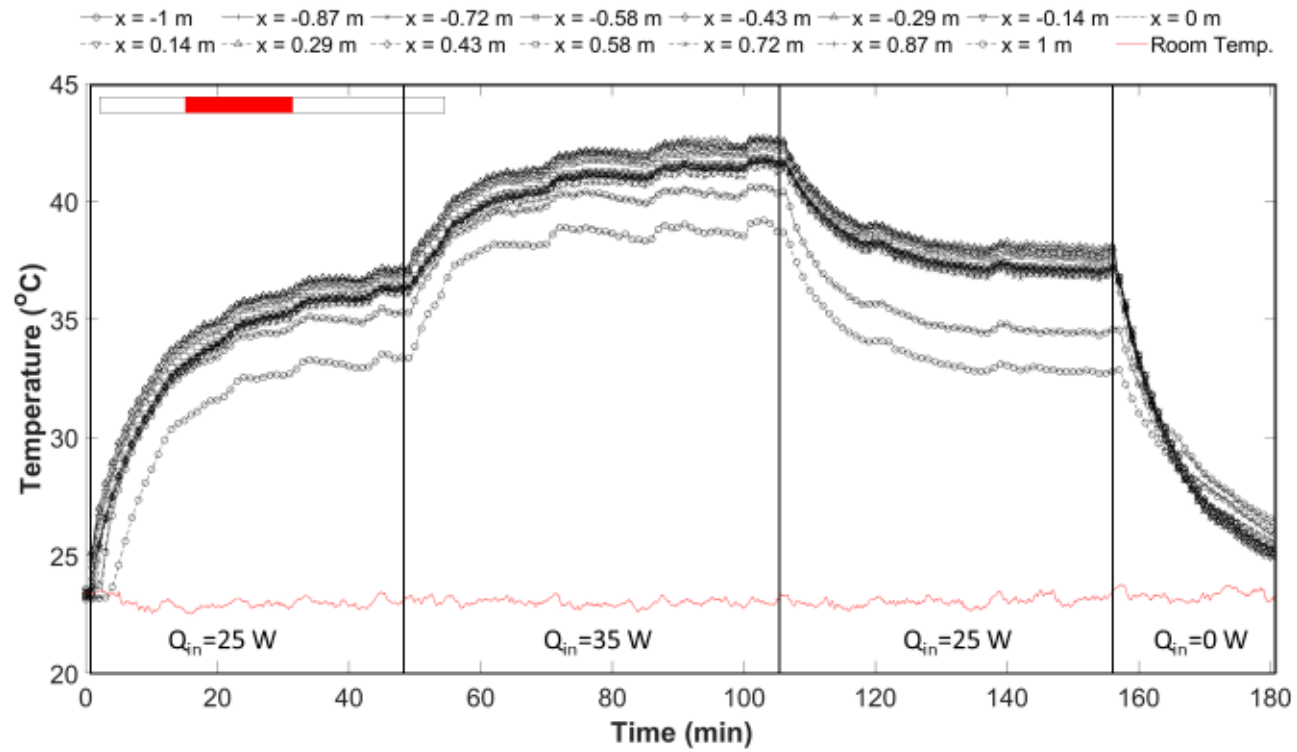


Figure 5.15: Time vs temperature data for the passive single off centered evaporator and dual condenser case with a changing thermal power applied to the heat pipe.

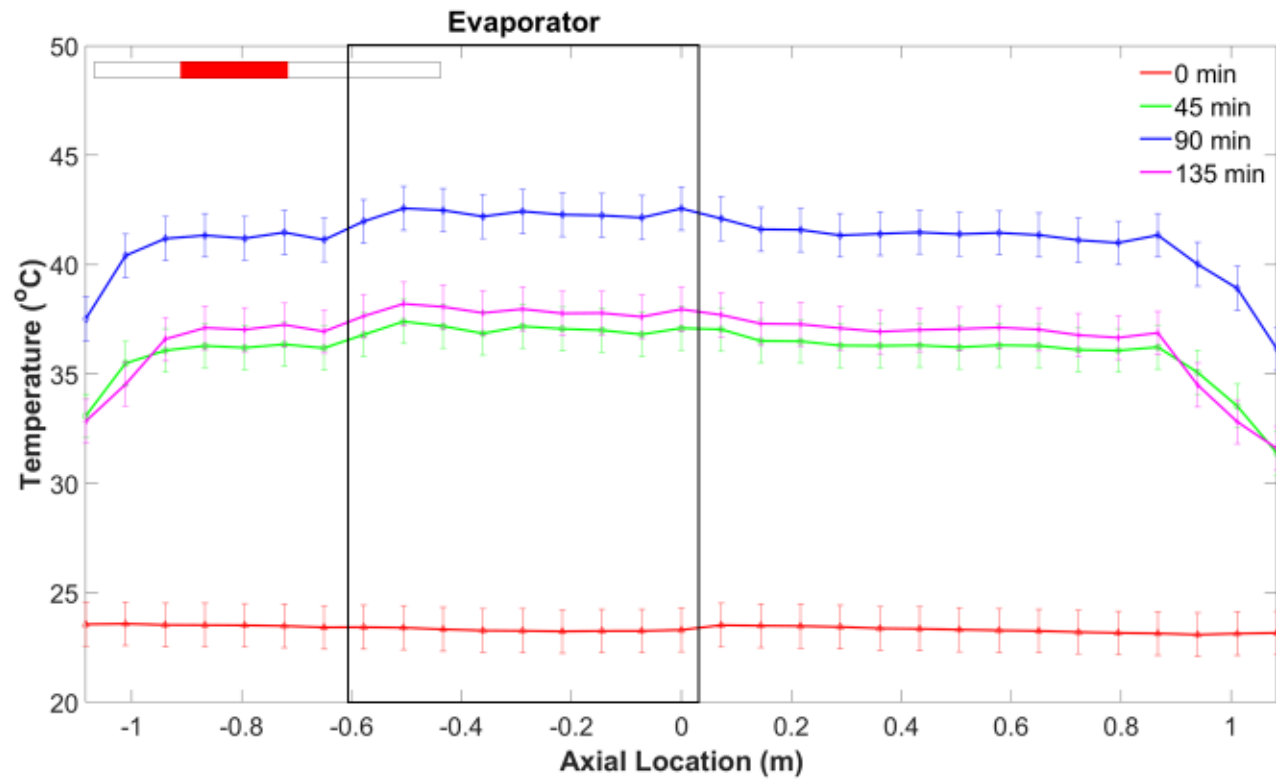


Figure 5.16: Axial location vs temperature at steady state conditions for each of the powers applied to the single off centered evaporator dual condenser case.

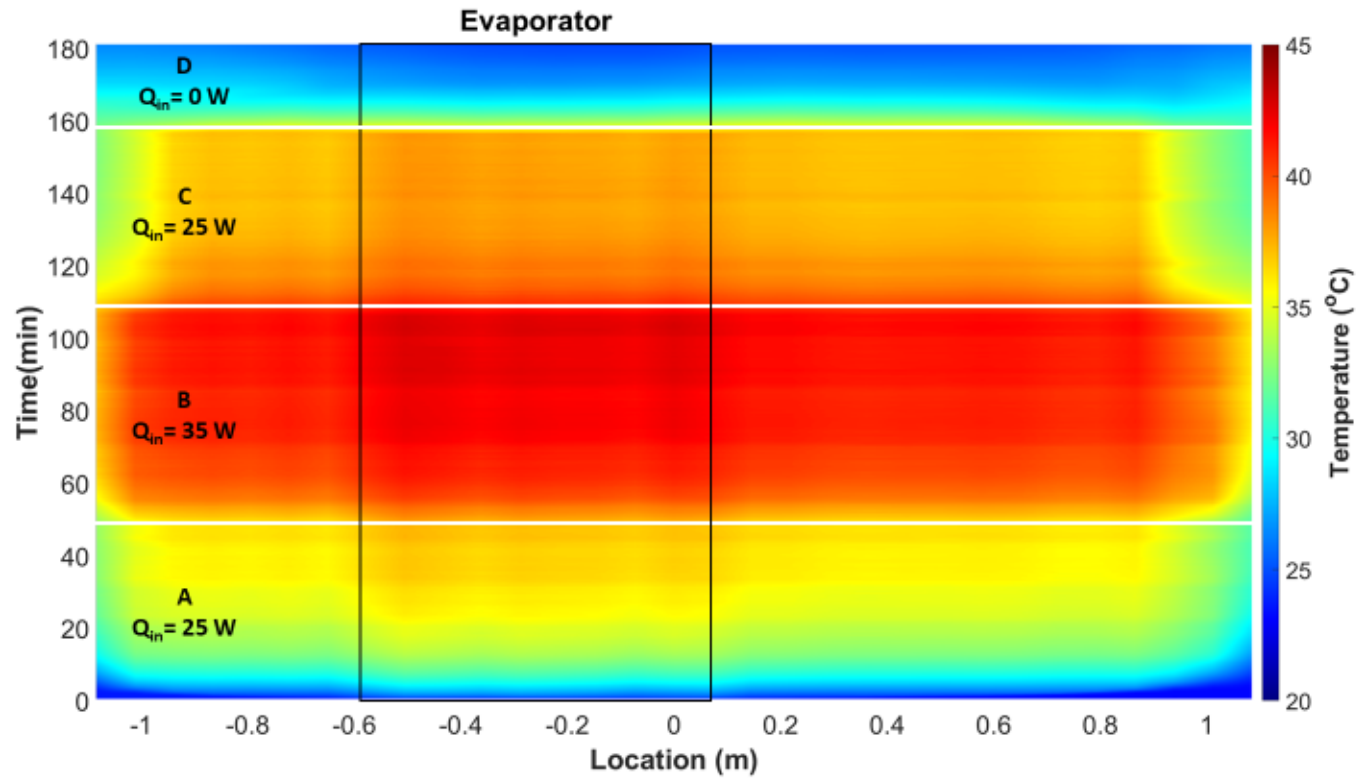


Figure 5.17: Heat pipe surface temperature with time and axial location showing the response of a variable conductance single off centered evaporator dual condenser heat pipe to changes in thermal power

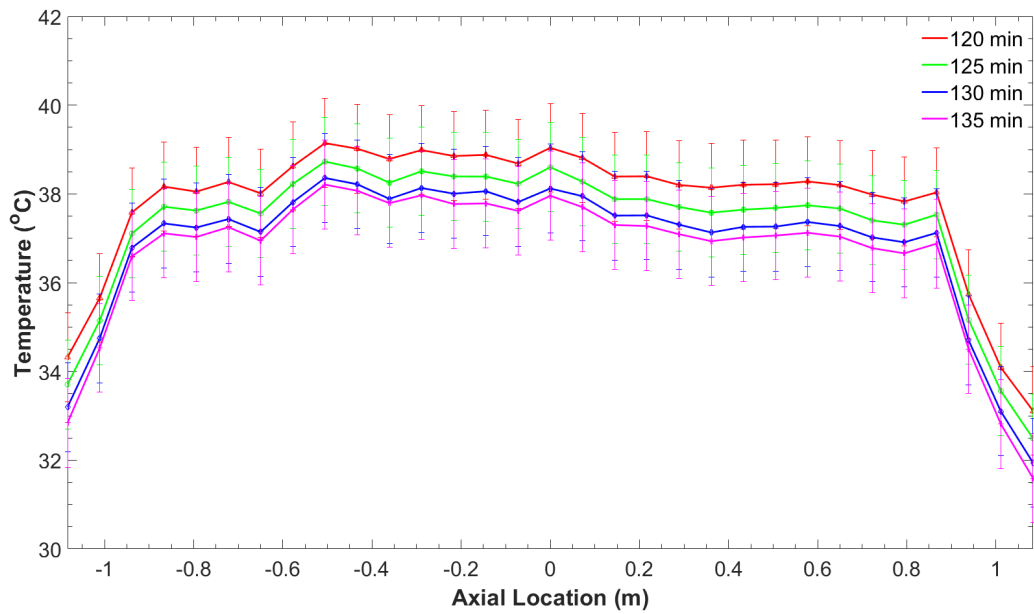
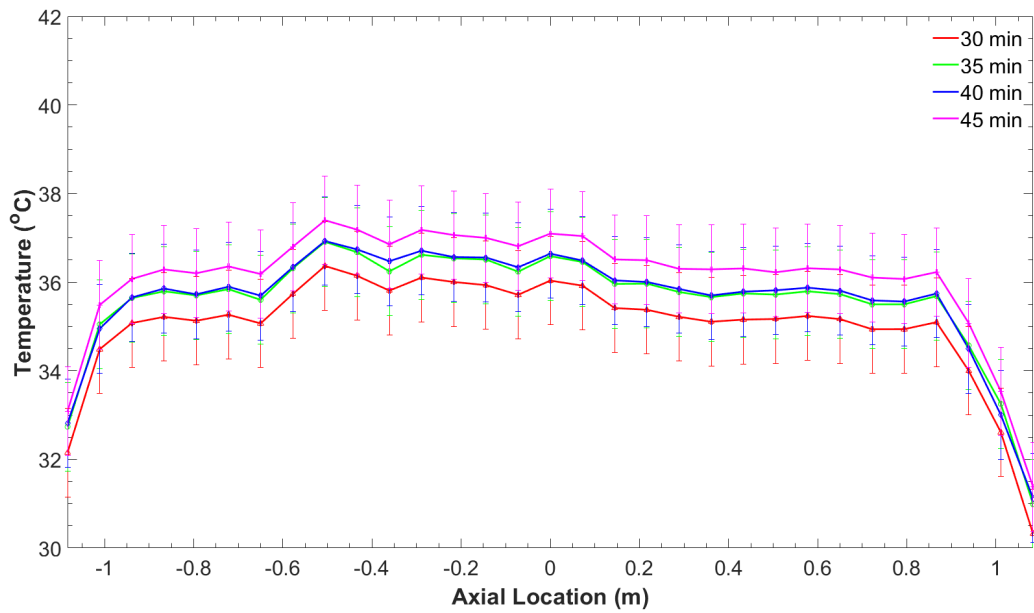


Figure 5.18: Axial location vs temperature for heat pipe approaching 45 minutes (top) and 135 minutes (bottom).

Overall, these passive variable conductance startup tests show how the non-condensable gas compresses as the heat pipe starts up and how the thermal boundary conditions influence the distribution of non-condensable gas in the heat pipe. The single off centered evaporator test shows that in a passive variable conductance heat pipe the heat pipe may return to a similar state at the same power conditions. This may indicate that the mass of non-condensable gas on each side of the heat pipe remains constant in this type of test. Further investigation of the influence of thermal boundary conditions on the non-condensable gas distribution was conducted in later tests. Additionally, it shows that it is possible to capture the temperature trends in the non-condensable gas region using the adiabatic fin tip approximation.

Air-Cooled Active Controlled Variable Conductance Heat Pipe with Static Stagnation Points

These tests incorporate both active control of the amount of gas in the heat pipe during operation and thermal power changes. The goal of these tests was to determine the influence of adding or removing non-condensable gas during heat pipe operation. The amount of gas added or removed to the heat pipe was controlled using the Proportion-Air QB1X pressure controllers. The different evaporator and condenser configuration are shown in Figure 5.1. For all tests other than the single condenser test, the heat pipe was had symmetric gas loading (the same non-condensable gas pressure applied to both sides of the heat pipe) and asymmetric gas loading (different non-condensable gas pressure applied to each side of the heat pipe). During the symmetric gas loading tests, a single pressure controller was used to avoid any asymmetric effects caused by two pressure controllers interacting with each other. In the asymmetric tests each pressure controller acted on each side of the heat pipe independently. Air was used as the non-condensable gas for each of the tests. The single evaporator and single condenser test was conducted with a single pressure controller since there is only one condenser. Figure 5.19 through Figure 5.21 show the results of an active controlled single evaporator single condenser heat pipe.

In the figures, each condition is labeled with a letter. The letter corresponds to the pressure and thermal power condition applied to the heat pipe. The thermal power and non-condensable gas pressure corresponding to each condition are shown in Table 5.2.

By comparing Figure 5.19 through Figure 5.21 to Table 5.2, it can be seen that as gas is added to the heat pipe during the transition between Condition A and Condition B, the evaporator and active condenser region rapidly increase in temperature while the inactive region of the condenser decreases in temperature. When power increases from 24.7 W to 36.3 W during the transition between Conditions B and C, the heat pipe experiences a relatively slight increase in temperature, 1.3° C, but the gas region compresses to increase the surface area of the condenser. When transitioning from C to D, the heat pipe decreases in temperature and the gas region expands decreasing the surface area of the active condenser region, as the thermal power applied to the heat pipe reduces. Then when transitioning from Condition D to Condition E the pressure of the non-condensable gas is decreased. When this occurs the heat pipe temperature drops rapidly as the gas exits the heat pipe and the inactive region of the condenser disappears. This test provides a baseline for behavior that is to be expected with this type of test.

In the next tests, a single centered evaporator dual condenser heat pipe and a single off centered evaporator dual condenser heat pipe were tested in a configuration that allowed for non-condensable gas addition that is symmetric about the center of the heat pipe. Table 5.3 and Table 5.4 describe the non-condensable gas pressure and the thermal power applied to the heat pipe throughout testing. Figure 5.22 through Figure 5.24 show results for the symmetric non-condensable gas loading tests of the dual condenser configurations. It can be seen when comparing the two cases that for the single centered evaporator dual condenser case, the heat pipe starts in a fixed conductance state and air is injected into both sides of the heat pipe. In the single off centered evaporator dual condenser heat pipe test air was introduced into the heat pipe immediately, and at the end of the test the air was removed. Other differences include differences in thermal power conditions and quantity of non-condensable gas added into the heat pipe.

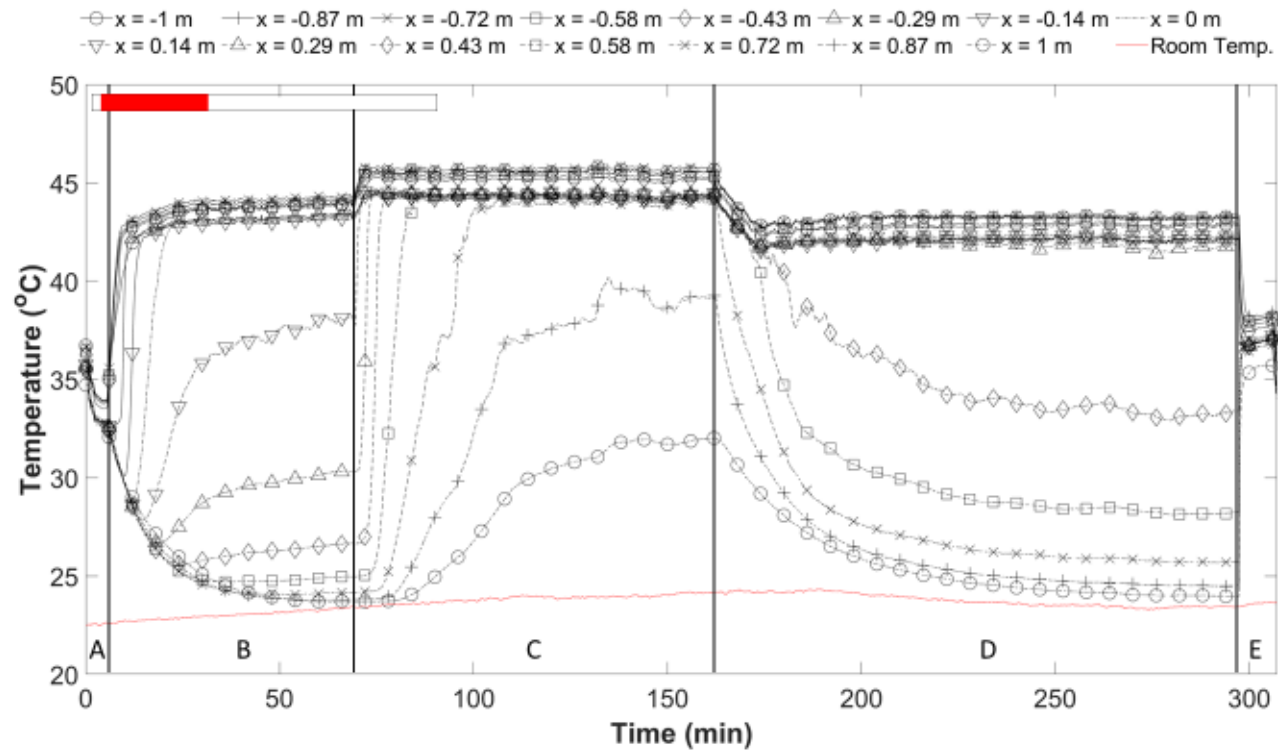


Figure 5.19: Time vs temperature data for active controlled single evaporator and single condenser case.

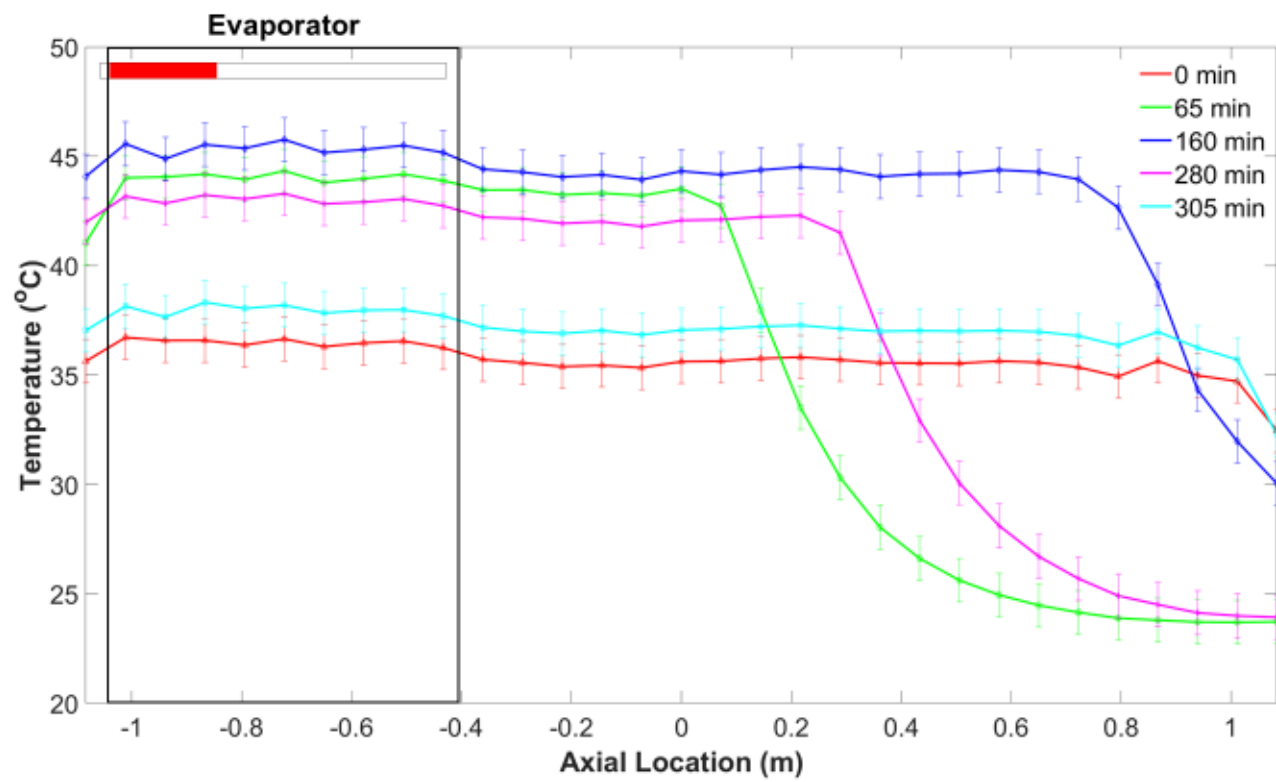


Figure 5.20: Axial location vs temperature at steady state for each of the conditions for the single evaporator and single condenser case.

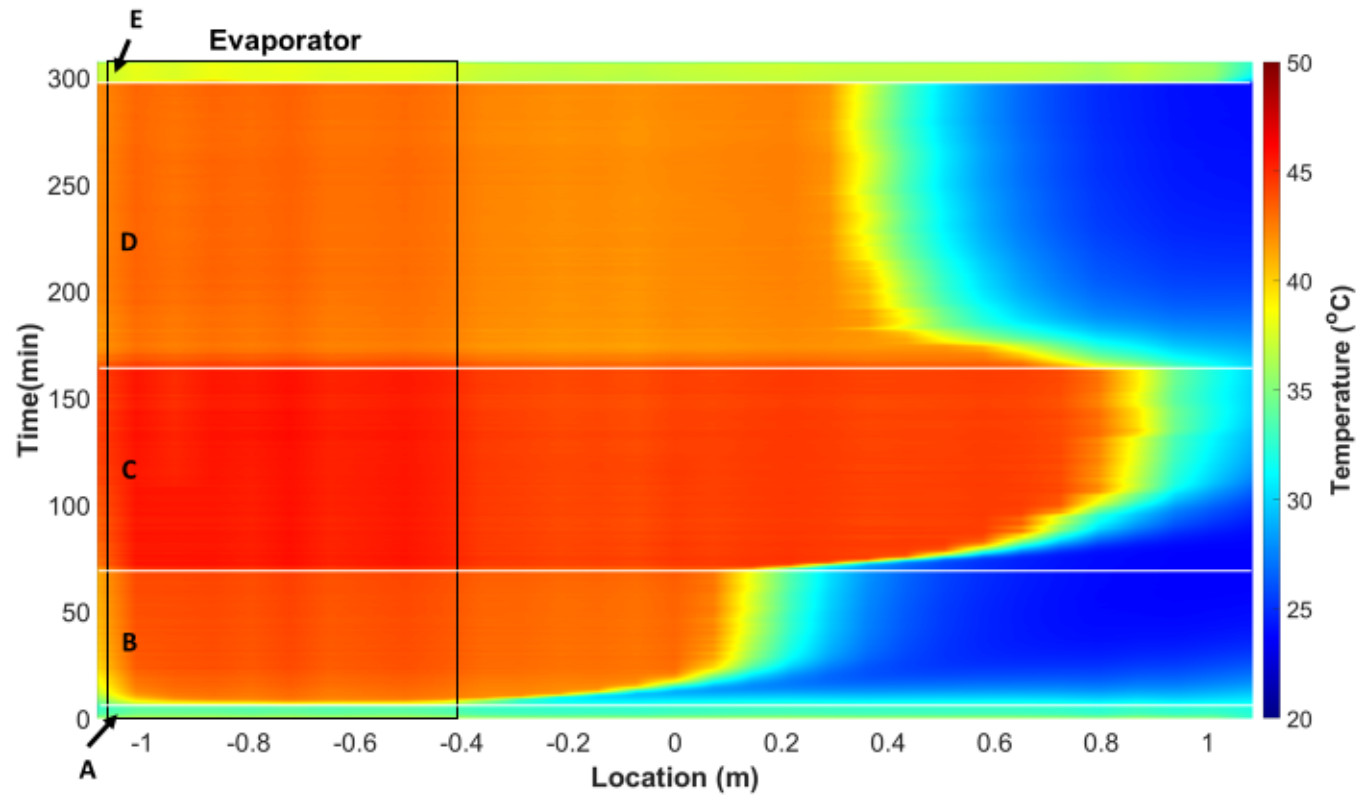


Figure 5.21: Heat pipe surface temperature with time and axial location showing the response of an active controlled variable conductance single evaporator and single condenser heat pipe.

Table 5.2: Active controlled variable conductance single evaporator single condenser heat pipe test conditions.

Condition	Pressure (KPa)	Thermal Power (W)
A	6.53±0.34	24.6±1.1
B	10.47±0.34	24.6±1.1
C	10.89±0.34	36.3±1.4
D	9.89±0.34	25.1±1.1
E	7.47±0.34	25.1±1.1

When comparing the results in Figure 5.22 through Figure 5.24, there is a noticeable difference in gas distribution. In the single off center evaporator case the left, shorter, condenser has a shorter gas section. This is like the results observed in the passive controlled test with the same configuration. It is likely that this is caused by the increased vapor pressure losses experienced in the longer condenser on the right. Additionally, in the single off centered evaporator case there is a much larger temperature change associated with changes in power such as the transition to and from Condition C. This may be caused by having too large of a volume of gas for the lower power to displace effectively. Given the gas pressure in the system it may not be able to expand enough to allow the heat pipe to remain at a higher temperature with a lower power. Figure 5.23 and Figure 5.24 show that the non-condensable gas region on the left is nearly at the evaporator exit meaning it cannot expand much more without influencing the operation of the evaporator. In this case, it may be advantageous to remove a small amount of non-condensable gas from the heat pipe before proceeding. Figure 5.22 and Figure 5.24 show slight fluctuations in the heat pipe temperature near the vapor and non-condensable gas interface. The fluctuations were attributed to the feedback system in the pressure controllers adjusting during testing and changes in room temperature. Aside from these fluctuations, the vapor and non-condensable gas remained stable during testing.

In the next tests, a different gas pressure is applied to each side of the heat pipe. Table 5.5 and Table 5.6 show the heat pipe conditions during these tests. Figure 5.25 through Figure 5.27 show the results of these tests. As shown in Table 5.5 and Table 5.6 the conditions for both the single centered evaporator dual condenser tests and the single off centered evaporator test are similar. When the heat pipe undergoes a change in non-condensable gas pressure, for example transitioning between Condition A to Condition B, the pressure on one side of the heat pipe influences the pressure on the other side of the heat pipe. This occurs because both controllers are connected to the same volume. This also imposes a limitation on the maximum pressure difference achievable in the heat pipe before the pressure controller compensate for each other. Additionally, when the heat pipe increases or decreases in temperature it strongly influences the pressure. Both these effects should be accounted for during operation of this type of heat pipe.

Table 5.3: Active controlled symmetric gas loaded variable conductance single centered evaporator dual condenser heat pipe test conditions.

Condition	Pressure (KPa)	Thermal Power (W)
A	6.27±0.34	25.1±1.1
B	9.86±0.34	25.1±1.1
C	9.86±0.34	35.9±1.4

Table 5.4: Active controlled symmetric gas loaded variable conductance single off centered evaporator dual condenser heat pipe test conditions.

Condition	Pressure (KPa)	Thermal Power (W)
A	8.32±0.34	25.0±1.1
B	8.32±0.34	11.8±0.71
C	8.32±0.34	24.7±1.1
D	6.87±0.34	24.7±1.1

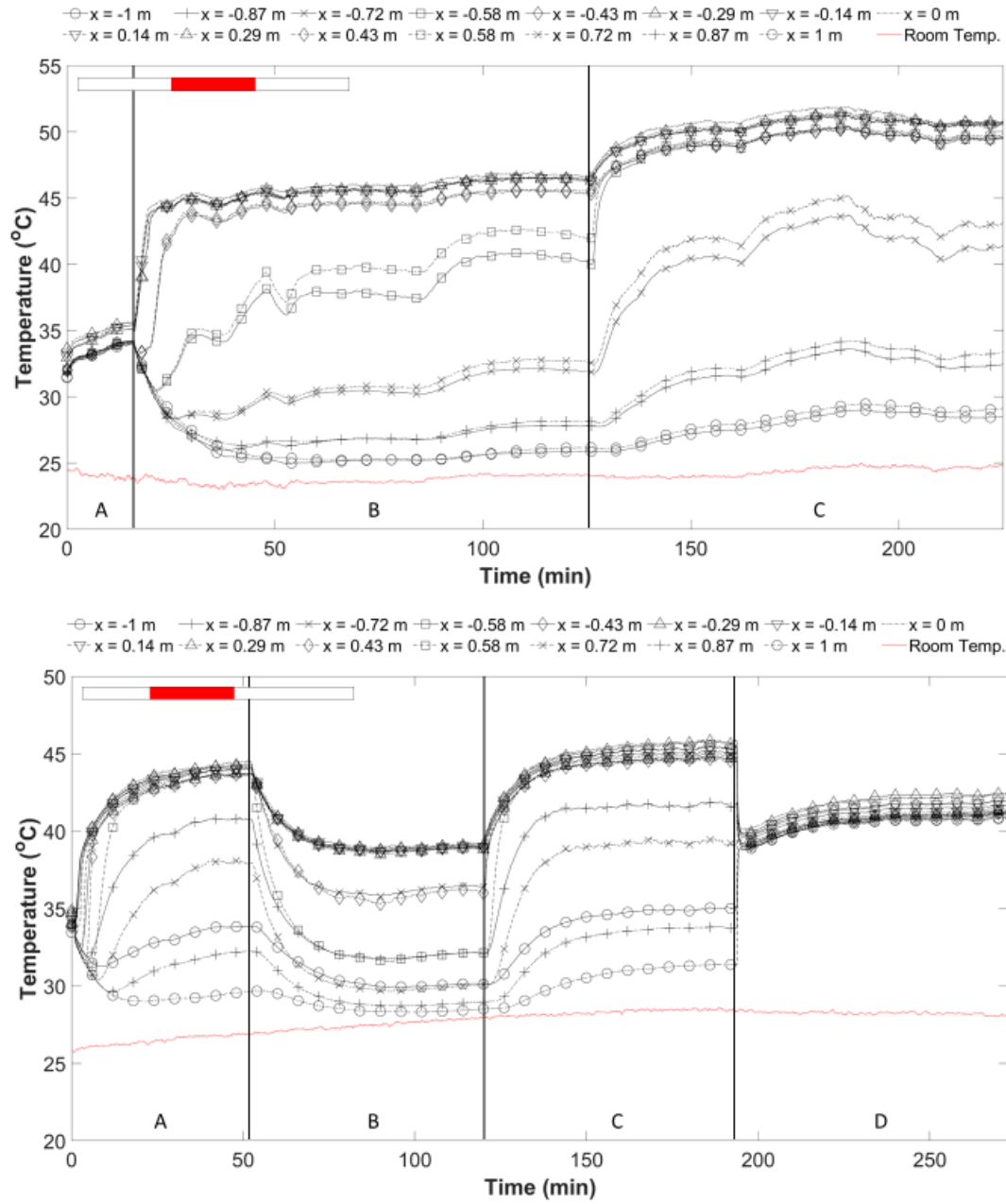


Figure 5.22: Time vs temperature data for the (top) active symmetric gas loaded variable single centered evaporator and dual condenser case, and (bottom) active symmetric gas loaded variable single off centered evaporator and dual condenser case.

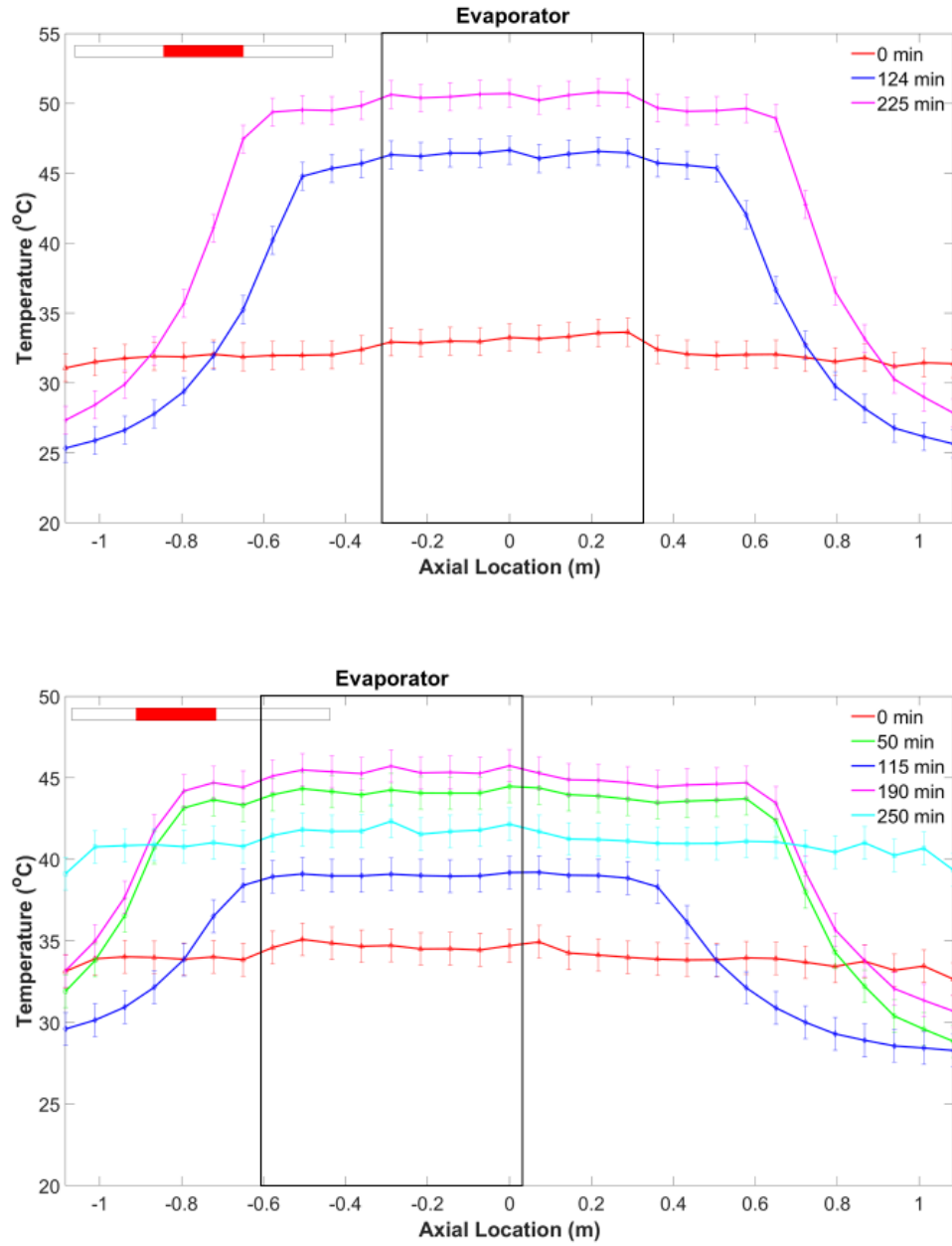


Figure 5.23: Axial location vs temperature at each pressure and thermal power condition for (top) active symmetric gas loaded variable single centered evaporator and dual condenser case, and (bottom) active symmetric gas loaded variable single off centered evaporator and dual condenser case.

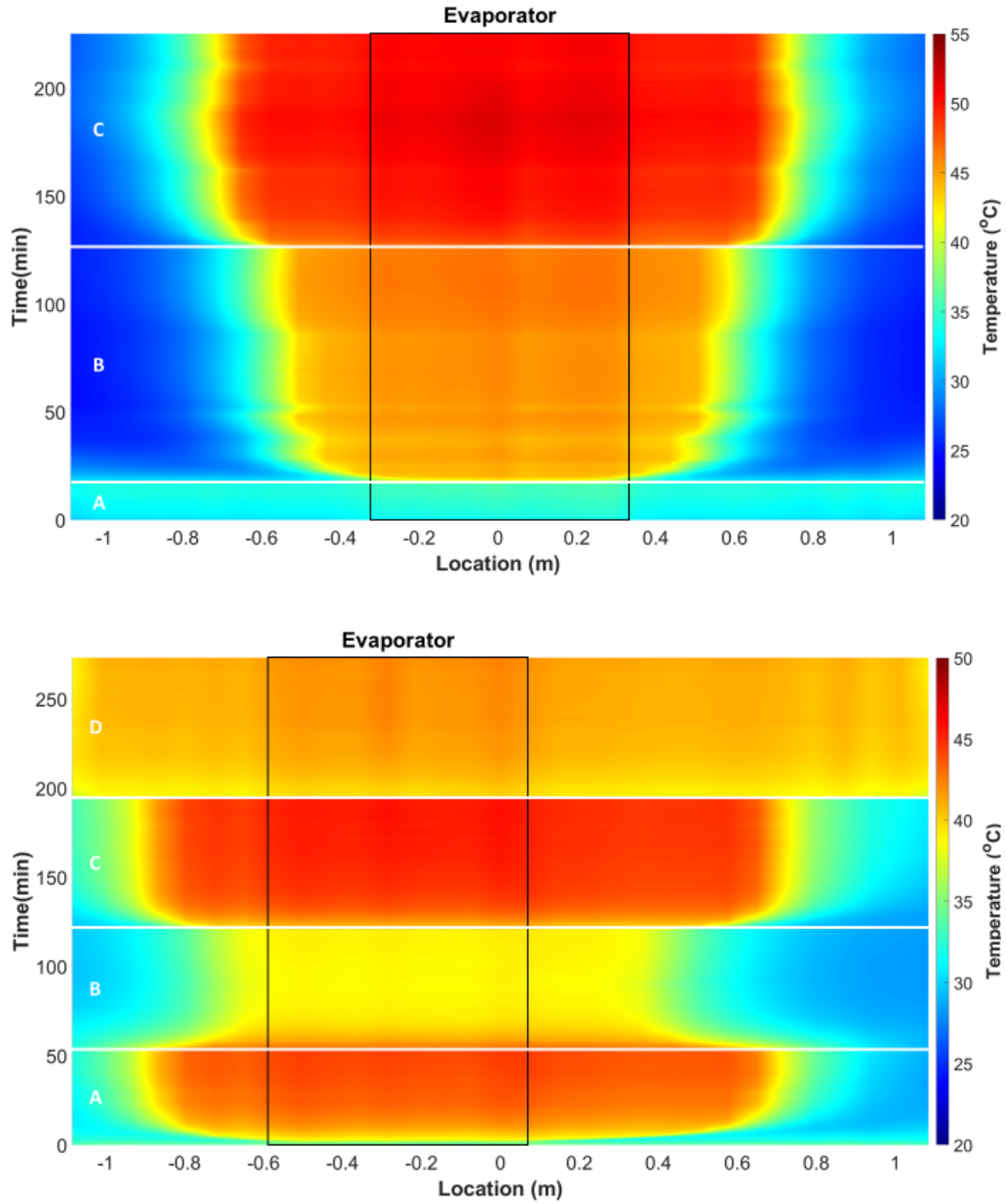


Figure 5.24: Heat pipe surface temperature with time and axial location for (top) active controlled symmetric gas loaded variable single centered evaporator and dual condenser case, and (bottom) active controlled symmetric gas loaded variable single off centered evaporator and dual condenser case.

When comparing the results in Figure 5.25 through Figure 5.27, it can be seen in the off centered evaporator configuration there is typically more gas on the right and less gas on the left when compared to the centered evaporator configuration. This gas distribution is similar to those seen in the passive controlled variable conductance heat pipe tests. Larger fluctuations in temperature near the vapor and non-condensable gas interface can be seen in Figure 5.25 and Figure 5.27. This was attributed to the interaction between the two different pressure controllers. During testing it was apparent that the controllers would begin interacting as the pressure controllers would make audible clicking noises while adjusting pressure. This interaction made these results considerably less repeatable. To avoid this interaction, it is suggested that in future testing the heat pipe be isolated from the pressure controllers after the necessary adjustments are made.

The non-condensable gas distribution may be altered in conditions where gas is intentionally added to or removed from a side such as in Condition B. At Condition B, the single centered evaporator dual condenser heat pipe has a longer non-condensable gas region on the left than on the right regardless of the centered evaporator. However, when the single off centered evaporator dual condenser configuration is at Condition B, it has similar non-condensable gas lengths in each condenser. This can be attributed to the different working fluid vapor pressure losses experienced by each configuration. In the single centered evaporator dual condenser configuration, the vapor pressure losses on each side of the heat pipe are similar, but in the single off centered evaporator dual condenser configuration, the vapor on the left experiences less pressure loss than the vapor on the right because of the differences in the length over which the vapor travels.

When transitioning from Condition A to Condition B, the non-condensable gas pressure transitions from being the same on both sides of the heat pipe to the non-condensable gas pressure of the left being greater than the non-condensable gas pressure on the right. The vapor pressures on each side of the heat pipe are comparable for the centered evaporator case; the lower non-condensable gas pressure on the right causes the gas/vapor interface to shift farther away from hot stagnation point. This also happens in the off centered evaporator configuration because the hot stagnation point is farther to the left in the off centered evaporator condition the gas/vapor interface.

Table 5.5: Active controlled asymmetric gas loaded variable single centered evaporator dual condenser heat pipe test conditions.

Condition	Pressure on Left (kPa)	Pressure on Right (kPa)	Thermal Power (W)
A	7.56±0.34	7.56±0.34	24.2±1.1
B	7.61±0.34	7.38±0.34	24.2±1.1
C	7.76±0.34	7.36±0.34	35.3±1.3
D	9.44±0.34	9.14±0.34	35.3±1.3
E	9.29±0.34	8.89±0.34	23.6±1.1
F	7.08±0.34	7.20±0.34	23.7±1.1

Table 5.6: Active asymmetric gas loaded variable single off centered evaporator dual condenser heat pipe test conditions.

Condition	Pressure on Left (kPa)	Pressure on Right (kPa)	Thermal Power (W)
A	7.45±0.34	7.45±0.34	24.8±1.1
B	7.93±0.34	7.51±0.34	24.8±1.1
C	7.93±0.34	7.51±0.34	35.6±1.3
D	10.59±0.34	10.35±0.34	35.6±1.3
E	10.27±0.34	9.95±0.34	25.3±1.1
F	7.50±0.34	7.68±0.34	25.3±1.1

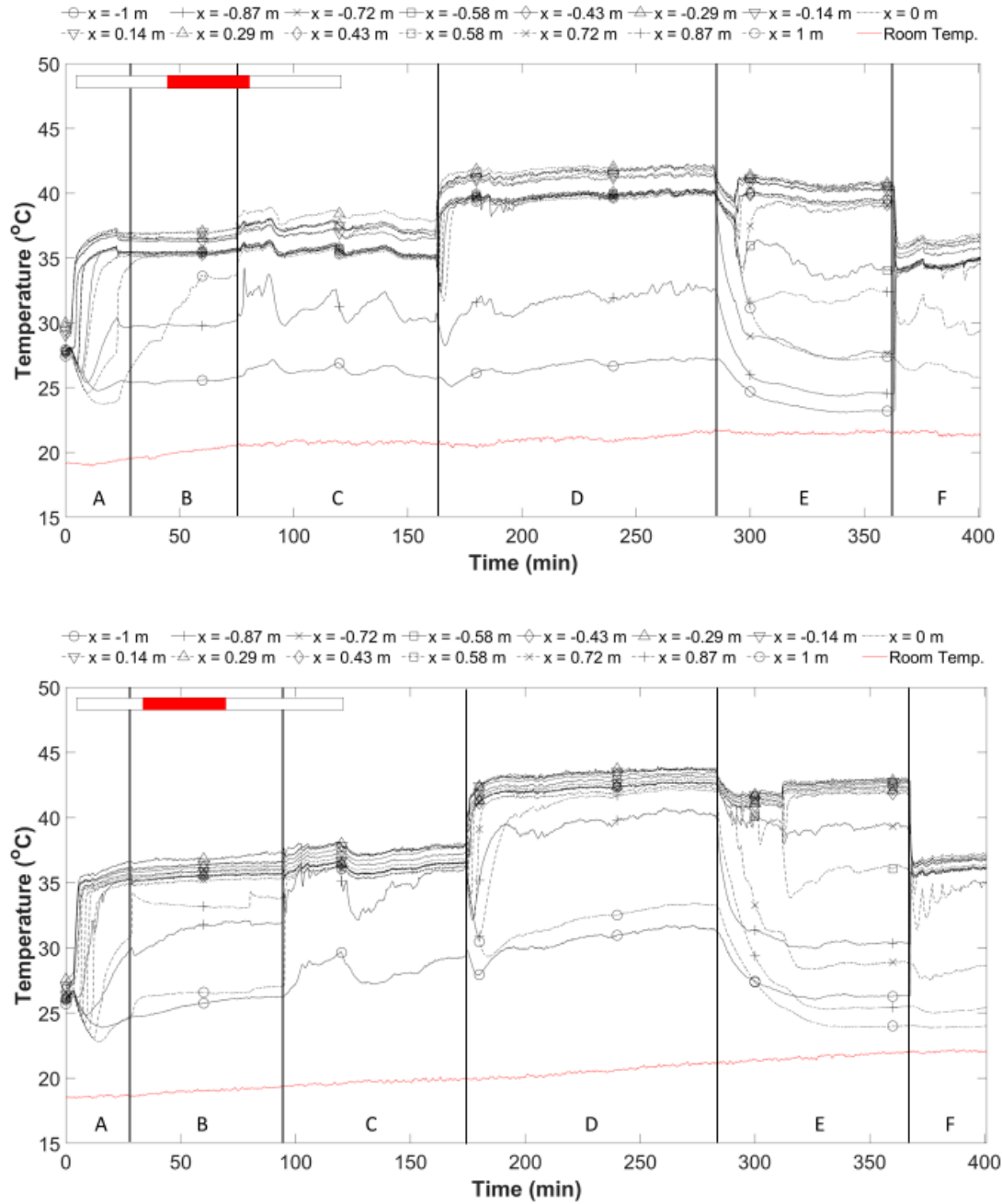


Figure 5.25: Time vs temperature data for the (top) active asymmetric gas loaded single centered evaporator and dual condenser case, and (bottom) active asymmetric gas loaded single off centered evaporator and dual condenser case.

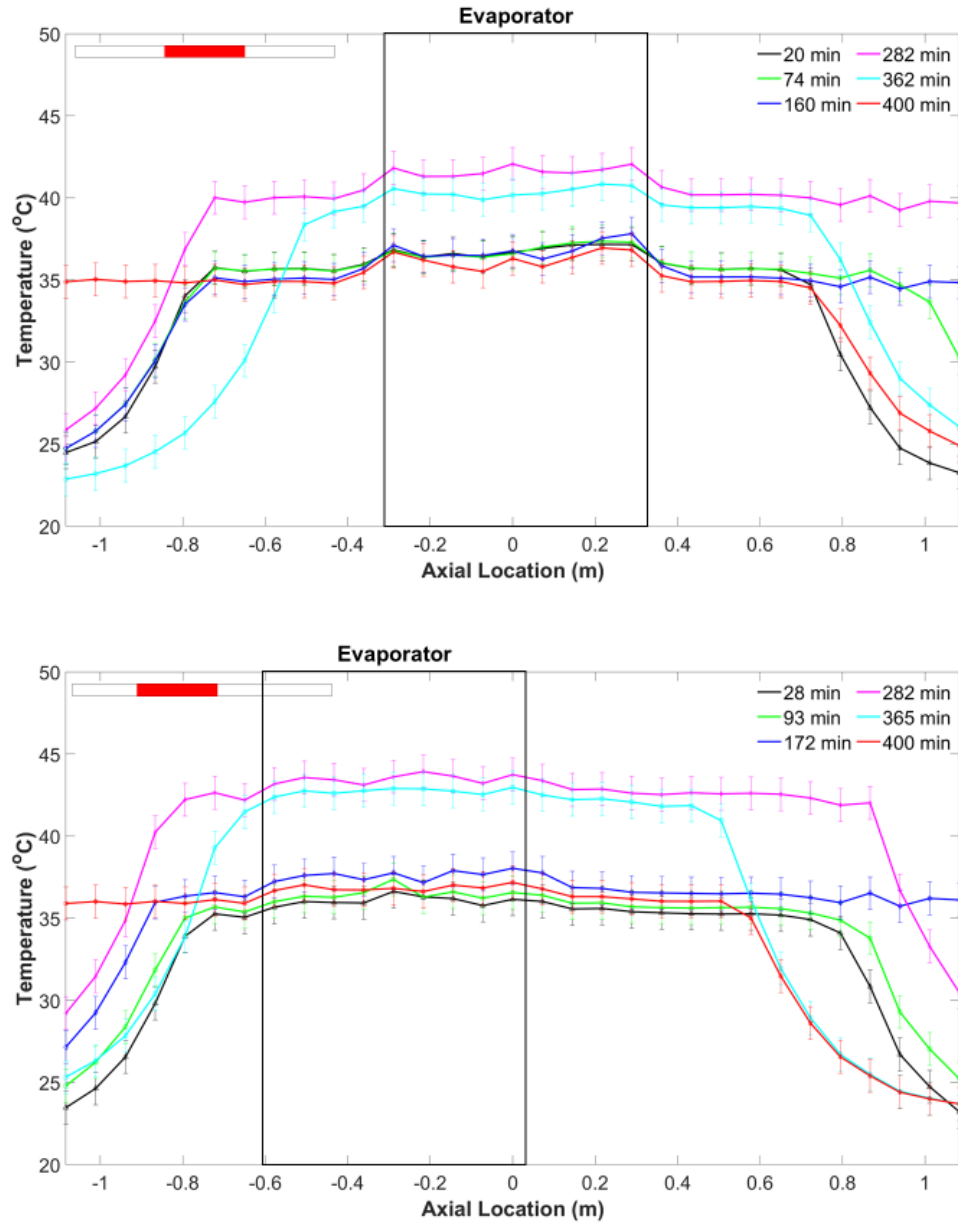


Figure 5.26: Axial location vs temperature at each pressure and thermal power condition for (top) active asymmetric gas loaded single centered evaporator and dual condenser case, and (bottom) active asymmetric gas loaded single off centered evaporator and dual condenser case.

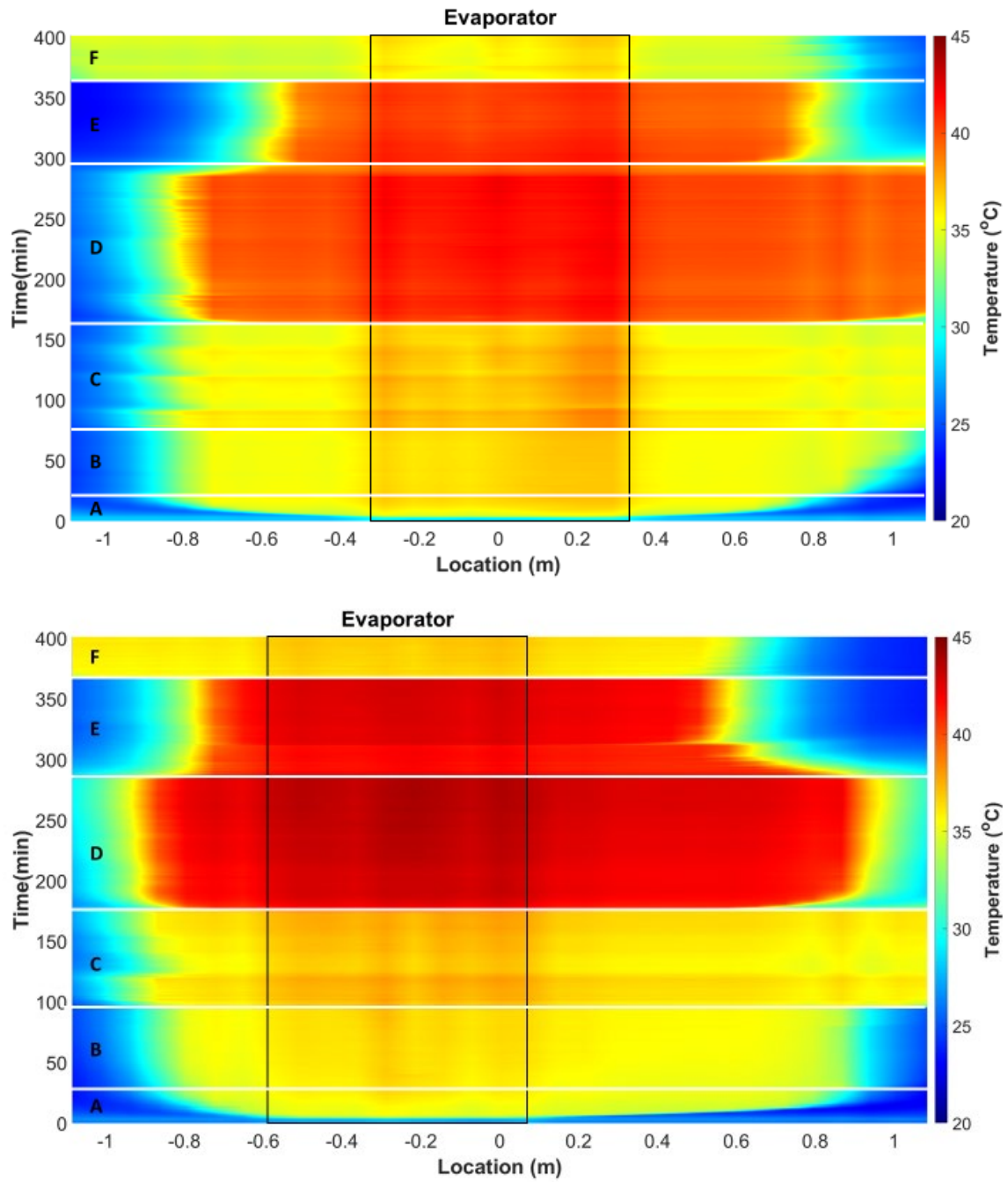


Figure 5.27: Heat pipe surface temperature with time and axial location for (top) active asymmetric gas loaded single centered evaporator and dual condenser case, and (bottom) active asymmetric gas loaded single off centered evaporator and dual condenser case.

The results of Condition B in both configurations suggests that the use of actively controlled non-condensable gas may be able to compensate for asymmetries in a system to produce a heat pipe where similar amounts of thermal power are removed from each condenser regardless of the location of the thermal boundary conditions applied in the evaporator section. Another possibility is that active controlled variable conductance heat pipes could use gas to select which condenser is operating while the other is turned off.

It can be observed that in conditions where the thermal power applied to the heat pipe is increased or decreased without a change in non-condensable gas pressure, the heat pipe experiences a relatively small change in temperature. Figure 5.25 and Figure 5.27 illustrate this when transitioning from Conditions B to C and Conditions D to E in both configurations. This indicates that this type of heat pipe may be of use in applications where equipment temperature must remain in a narrow range, but the equipment experiences a wide range of thermal powers, such as electronics or space systems.

Overall, both the symmetric and asymmetric non-condensable gas loaded variable conductance tests show that the amount of non-condensable heat pipes plays an important role in the temperature of the heat pipe. Additionally, these tests show that the non-condensable gas regions may be actively controlled to produce different results than heat pipes that are passively controlled. These results indicate that for active controlled variable conductance multiple evaporator or multiple condenser heat pipes, both the expected thermal boundary conditions and the range of non-condensable gas pressure must be considered in the design phase. Lastly, actively controlled variable conductance heat pipes may provide the advantage of being able to remove the gas before the heat pipe shuts down. This is especially useful in preventing non-condensable gas from entering the wick or annulus which was observed during the passive controlled tests.

Calorimeter-Cooled Passive Controlled Variable Conductance Heat Pipe with Dynamic Stagnation Points

To observe the influence of changes in the condenser thermal boundary conditions on heat pipe thermal power throughput, two heat pipe tests were conducted with calorimeters. The goal of these tests was to observe the power removed for the heat pipe in various configurations. More information about the calorimeters is presented in the Calorimeter Tests section of Chapter 3: Methodology.

Tests were completed on a passive controlled single evaporator single condenser and passive controlled single centered evaporator dual condenser heat pipe. In the single evaporator single condenser configuration, two 30.5 cm long condensers were attached to each other using vacuum fittings. To evenly distribute the thermal power removed by each calorimeter, the water inlets were directly connected to the ISOTEMP circulator. In the single centered evaporator dual condenser configuration, the calorimeters were separated and placed on each end of the heat pipe. The calorimeters were installed with the inlets farthest from the evaporator so the calorimeters would function as are like a parallel flow heat exchanger. The calorimeters were installed in this way to minimize the water temperature losses over the non-condensable gas region. For these tests, the gas gap inside the calorimeter was filled with helium. In each test, Calorimeter 1 was installed in the left most position. To reduce thermal power loss from the heat pipe to the environment, the water supplied to the calorimeters was kept below 20°C. By maintaining a lower water temperature, the heat pipe remains at a lower temperature reducing the temperature differential between the heat pipe and environment. For these tests, the steady state temperatures and thermal power distributions were of interest, so only the steady state heat pipe wall temperatures and contour heat pipe wall temperatures are presented. Additionally, calorimeter specific data such as water inlet and outlet temperatures and calculated thermal powers are presented. In each test, the heat pipe was evaluated under three different conditions. These three conditions are presented in Table 5.7. The times at each condition varied between each configuration.

Figure 5.28 shows the temperature of the heat pipe with both time and position. When comparing the wall temperature for the single evaporator single condenser heat pipe test to the wall temperature for the single centered evaporator dual condenser, multiple cool regions are visible in the single evaporator single condenser plots. The cool regions at 0.4 m and from 0.6 m to 0.8 m may be from the heat pipe contacting the wall. They may also be related to the calorimeter interacting with the non-condensable gas region. To determine the cause additional testing with alternate calorimeter designs would need to be completed.

In Figure 5.28, the single evaporator and single condenser heat pipe is shown to cool rapidly around 400 minutes. During the test, the valve in variable flow rotameter connected to Calorimeter 1 reopened. This allowed chilled water to flow through the calorimeter which cooled the heat pipe. After the rotameter failure, the flow to the calorimeter was stopped and the heat pipe began to increase in temperature again. A similar rotameter fluctuation can be seen just after 200 min. In this case flow decreased rapidly for a short amount of time. When comparing Figure 5.28 and Figure 5.29 a change in the water temperature is also noticeable which is consistent with a sudden increase in flow. To prevent or better observe this in the future, switching to a digital flow meter or flow controller is advised.

The wall of the single evaporator and single condenser heat pipe was warmer during testing. This may be caused by a combination of differences including insulation, differences in the thermal coupling between the heat pipe and calorimeter, differences in water inlet temperature, differences in the water mass flow rate, and differences in the amount of non-condensable gas in the heat pipe. To change between the single evaporator single condenser configuration, and the single centered evaporator dual condenser, the heat pipe had to be removed from all insulation and emptied before reconfiguration. This may have led to differences in the insulation for each case. From Figure 5.29, the Calorimeter 1 inlet temperature is close to the similar at the end of the test for both configurations meaning this is likely not the cause of the differences. The difference the thermal power input was compared to the thermal power output for each of these configurations. Figure 5.30 shows the thermal power input and output broken down by component, the total

thermal power input and thermal power output, and the thermal power losses for both configurations.

Aside from the differences caused by the water flow rate fluctuations discussed above, the power components and percent thermal power loss are very similar for both the single evaporator single condenser heat pipe and the single centered evaporator dual condenser heat pipe. For Condition C where the single evaporator single condenser heat pipe is warmer, both heat pipe configurations experience a $20\pm 4\%$ thermal power loss. This indicates the insulation used in each heat pipe configuration had a similar effectiveness.

The heat pipe had several slight bends along the length. The gas gap design of the calorimeter is sensitive to changes in the gas gap size between the heat pipe and inner wall of the calorimeter. If the heat pipe is bent in the narrow gap, this may lead to changes in thermal coupling. Another source of thermal coupling differences could come from differences in gas composition in the gas gap. If the single evaporator single condenser heat pipe configuration had a leak develop in the calorimeter, helium may have leaked out creating a gap with higher thermal resistance. This would lead to the heat pipe increasing in temperature to reach the same power output. This is the most likely cause of the temperature difference.

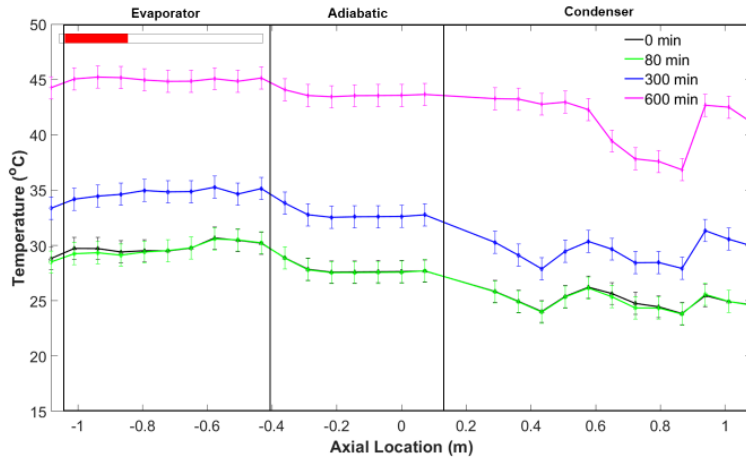
Figure 5.30 shows that for Condition A and B each calorimeter removes a similar amount of thermal power from the heat pipe. In the single evaporator single condenser case Calorimeter 1 removes slightly more thermal power. Figure 5.28 shows that for the single evaporator single condenser configuration the temperature is slightly higher under this calorimeter which would cause an increase in thermal power output. Under Conditions A and B, Calorimeter 2 removes slightly more thermal power than Calorimeter 1 for the single centered evaporator single condenser configuration. After the test, it was found that Calorimeter 2 was 1 ± 0.16 cm closer to the evaporator than Calorimeter 1. This may account for the power difference as Calorimeter 1 had shorter length non-condensable gas region under it.

These results indicate that passive variable conductance heat pipes with multiple separate condensers distribute heat to each condenser evenly, but additional testing is required to confirm this.

Table 5.7: Calorimeter test conditions.

Condition	Thermal Power (W)	Calorimeter 1 Status	Calorimeter 2 Status
A	25.1 ± 1.1	On	On
B	25.1 ± 1.1	On	On
C	35.9 ± 1.4	Off	On

Single Evaporator Single Condenser



Single Centered Evaporator and Dual Condenser

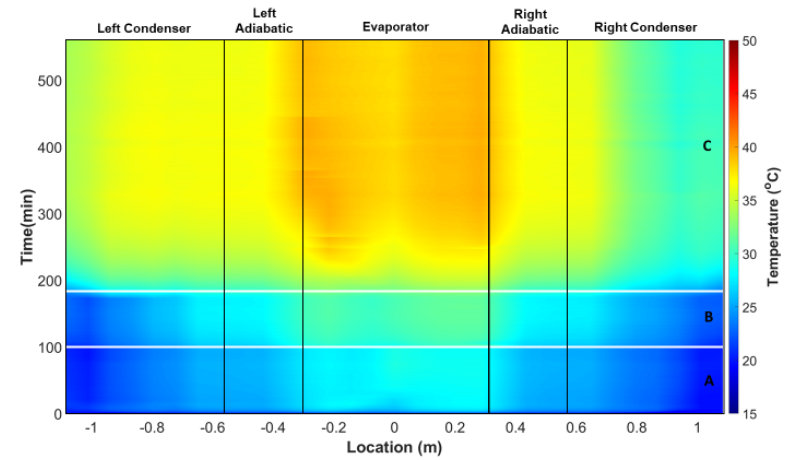
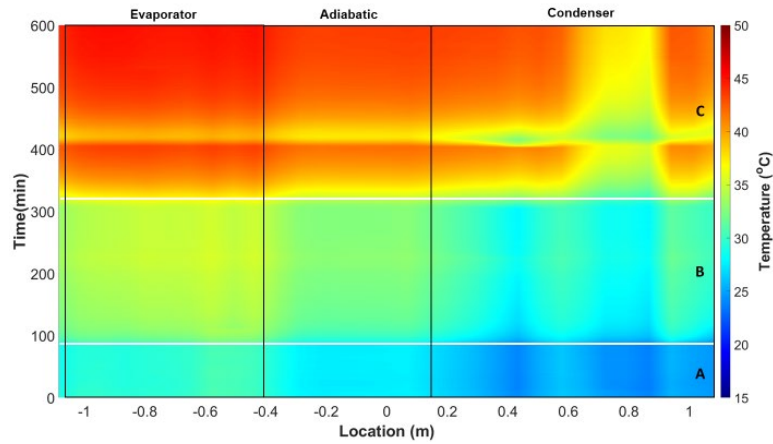
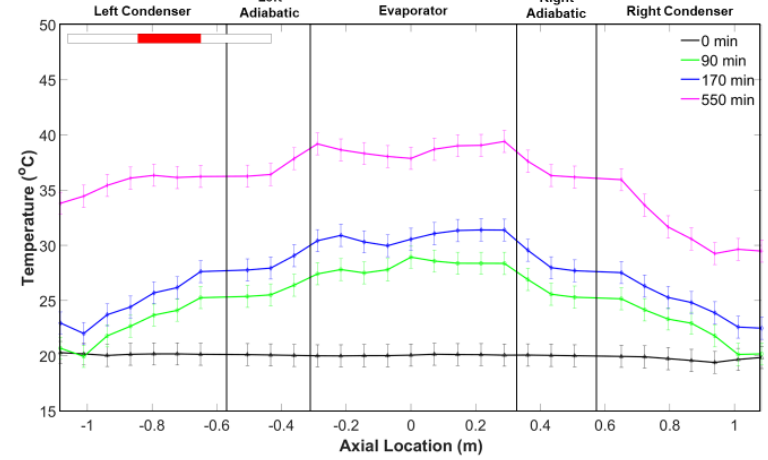


Figure 5.28: Heat pipe axial location vs wall temperature at steady state (top) and heat pipe surface temperature with time and axial location (bottom) for the calorimeter cooled test.

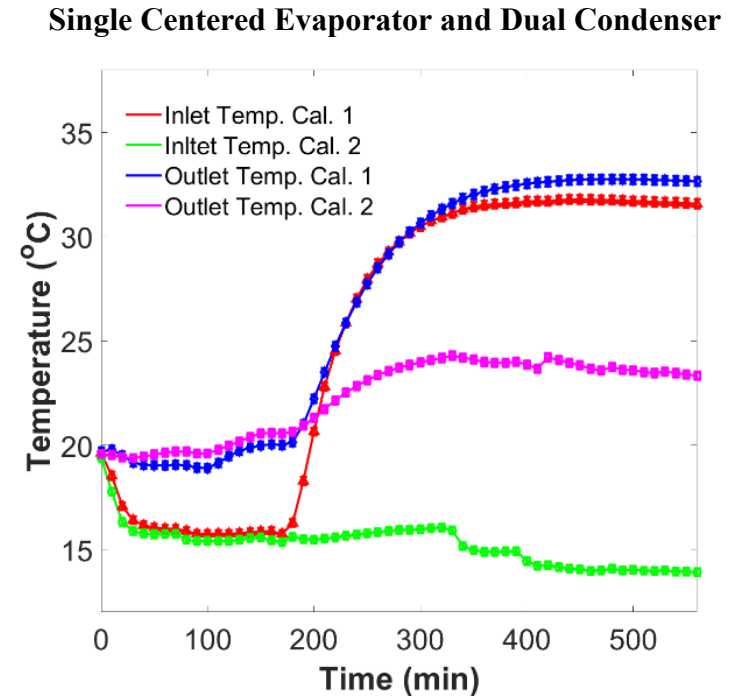
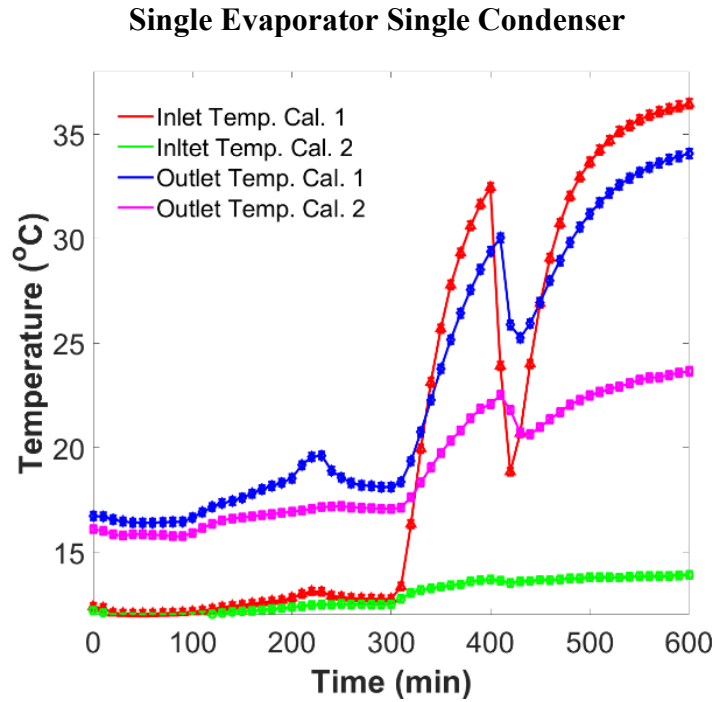


Figure 5.29: Calorimeter inlet and outlet water temperatures for the single evaporator single condenser and the single centered evaporator dual condenser tests with $\pm 0.2^\circ\text{C}$ error bars for the RTD's.

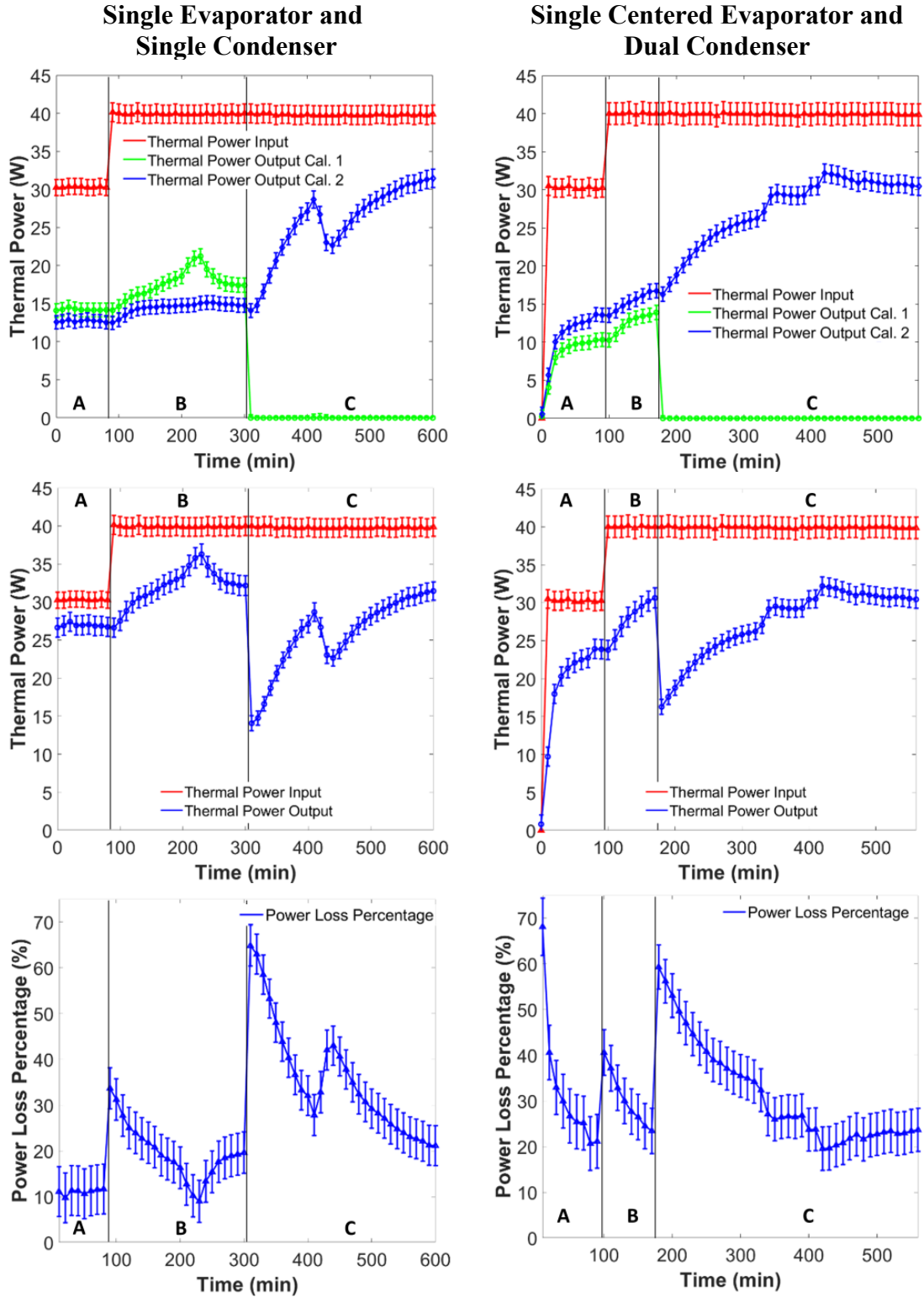


Figure 5.30: Thermal power input and output components (top), total thermal power input and output (middle) and percentage of thermal power loss from the system (bottom).

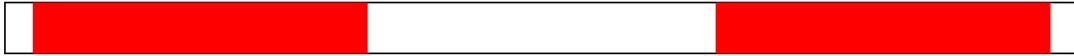
The design of the calorimeters led to leak development across the vacuum fittings when installing insulation. Additionally given the small gas gap and the bends in the heat pipe, the thermal coupling was inconsistent. Furthermore, a power loss of $20\pm 4\%$ could obscure other effects, meaning that additional tests with better insulation are required to fully investigate the influence of the non-condensable gas on thermal power output in multiple condenser heat pipes. Lastly a greater amount of time should be allowed for testing. It is apparent from Figure 5.30 that the heat pipe and calorimeters had not reached thermal equilibrium by the time the end of the allotted testing time.

Air-Cooled Passive Controlled Variable Conductance Heat Pipe with Dynamic Stagnation Points

In these tests the switch box shown in Figure 4.11 was used to change the external thermal boundary conditions acting on a passive variable conductance heat pipe during operation. The purpose of these tests was to observe the formation of cold and hot stagnation points in response to thermal boundary condition changes during heat pipe operation. The results from the first test were used to determine the feasibility of using the fin approximation to capture the trends of the non-condensable gas region was investigated.

In the first test, 23 ± 1 W was applied to the heat pipe at each of the conditions. Figure 5.31 shows the evaporator locations for each of these conditions. For this test, the heat pipe was preheated in a single evaporator and single condenser configuration with the evaporator on the left. The heat pipe was allowed to reach a quasi-steady state before transitioning to the next condition. This heat pipe was determined to be at a quasi-steady state when the thermocouple data changed by less than $0.5\text{ }^{\circ}\text{C}$ in a period of 15 minutes. To match the amount of thermal power applied to the heat pipe, the voltage supplied to the heaters was increased when transitioning from Condition A to Condition B to compensate for the decrease from six heaters in Condition A to three heaters in Condition B. Table 5.8 shows the thermal power applied by each zone of three heaters with the zones numbered from left to right. Figure 5.32 shows the power applied to each heater zone with time.

A - Dual Evaporator and Single Centered Condenser



B - Single Centered Evaporator and Dual Condenser



C - Right Single Evaporator and Single Condenser



D - Single Centered Evaporator and Dual Condenser



E - Left Single Evaporator and Single Condenser



Figure 5.31: Evaporator locations during the uniform thermal power input test.

Table 5.8: Power supplied to each heater zone in the dynamic stagnation point test with uniform total thermal power test.

Condition	Zone 1 Power	Zone 2 Power	V Zone 3 Power	Total Input Power
A	11±0.7 W	0±0.0 W	11±0.7 W	22±1 W
B	0±0.0 W	23.7±1.1 W	0±0.0 W	23.7±1.1 W
C	23.7±1.1 W	0±0.0 W	0±0.0 W	23.7±1.1 W
D	0±0.0 W	23.7±1.1 W	0±0.0 W	23.7±1.1 W
E	0±0.0 W	0±0.0 W	23.7±1.1 W	23.7±1.1 W

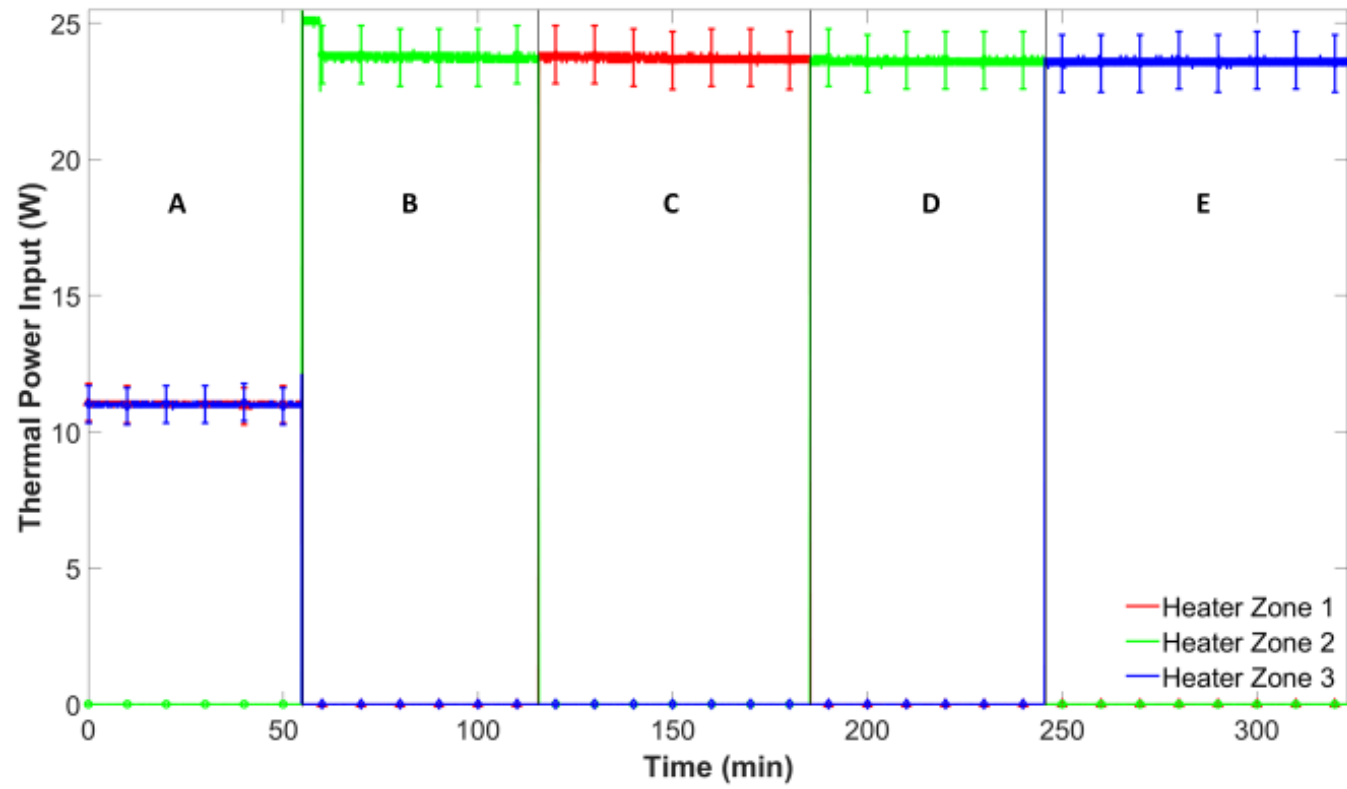


Figure 5.32: Thermal power input for the dynamic stagnation point test with uniform thermal power input test.

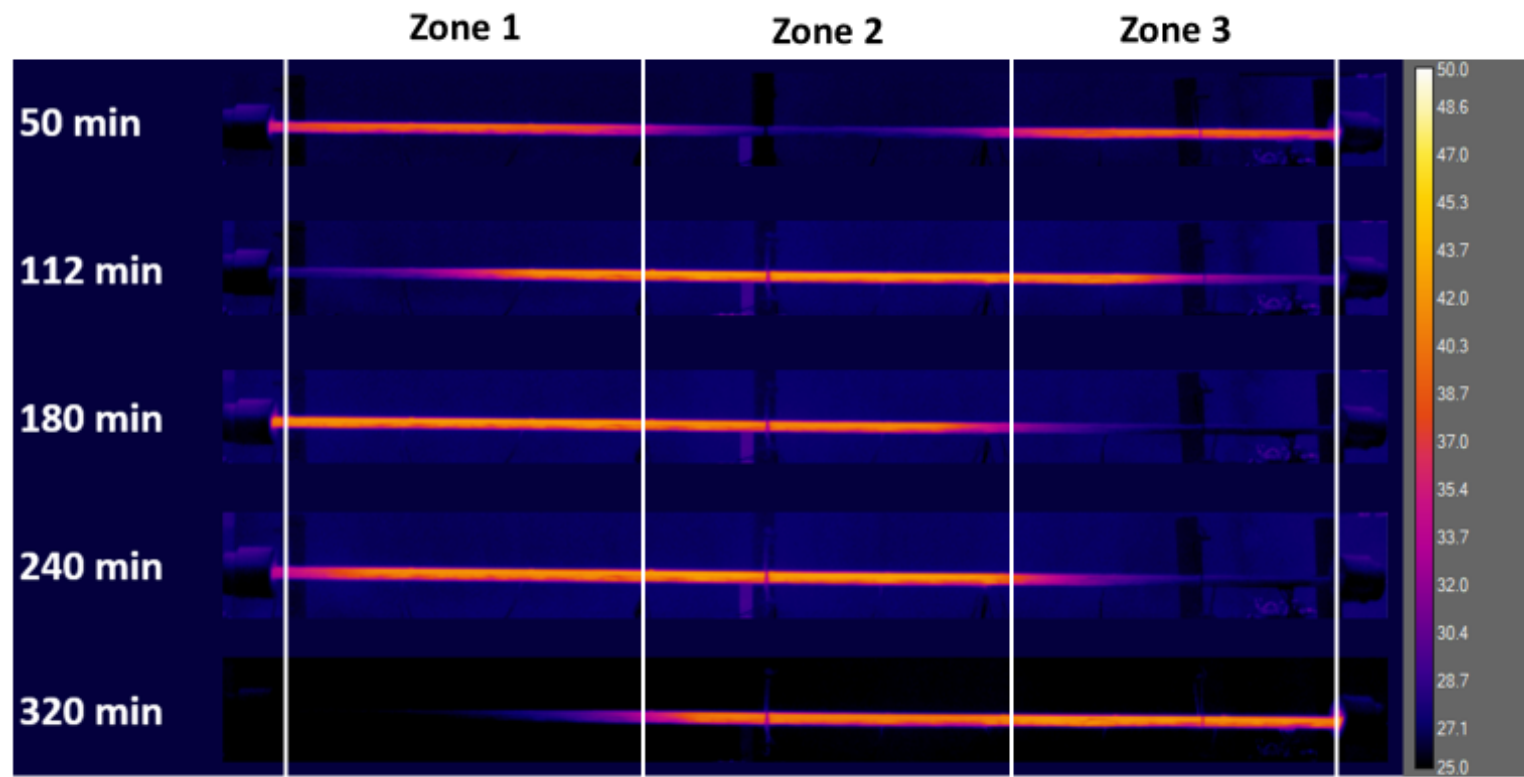


Figure 5.33: Selected infrared images for the dynamic stagnation point test with uniform thermal power input test

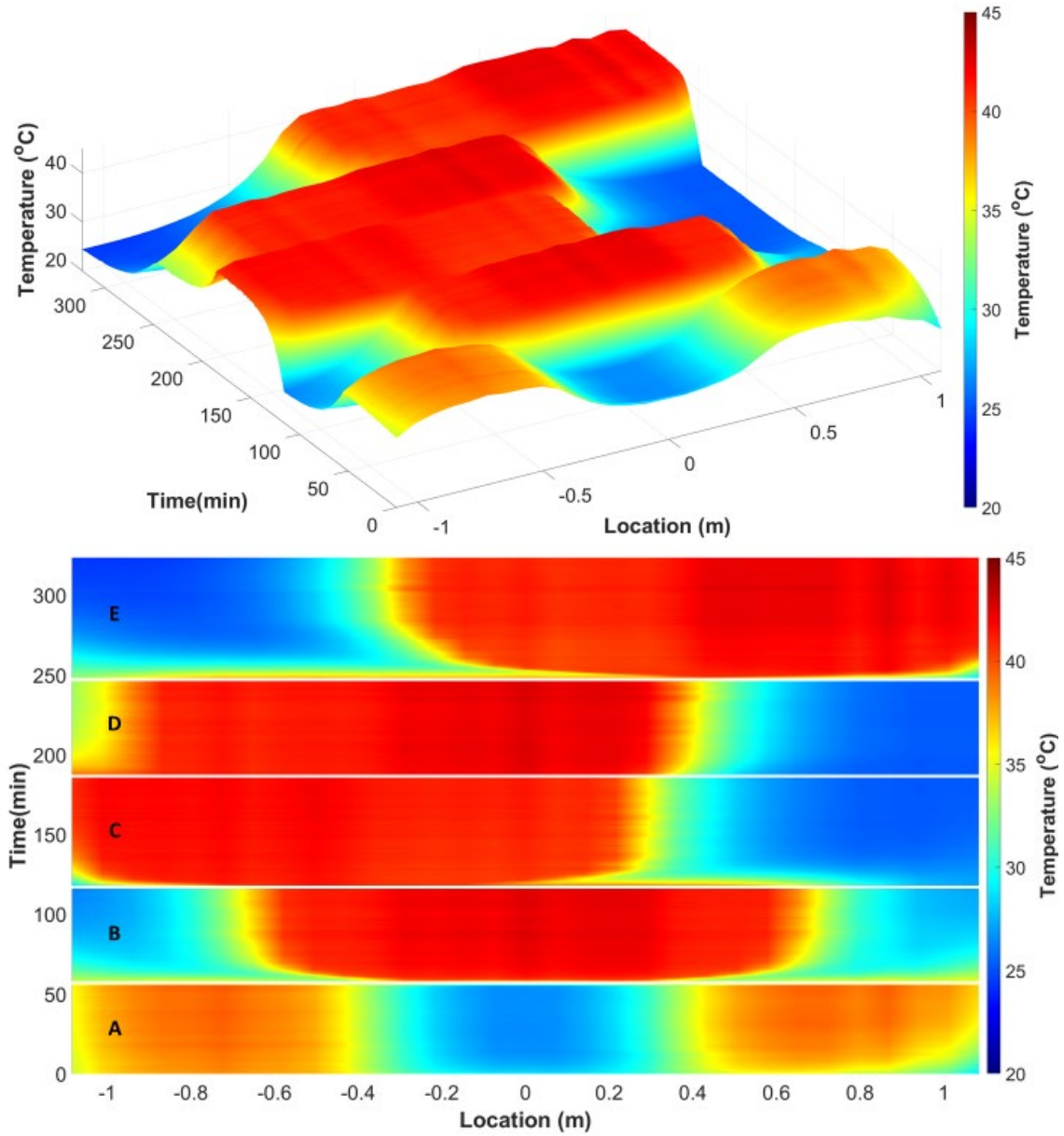


Figure 5.34: Three-dimensional contour plot of heat pipe surface temperature with time and axial temperature (top) and corresponding two-dimensional heat pipe surface temperature with time and axial location (bottom) for the dynamic stagnation point test with uniform total thermal power test.

Figure 5.33 and Figure 5.34 show the wall temperature of the heat pipe with time and axial location. In Figure 5.33 white lines were added to identify the corresponding location of the heater zones. As with the previous infrared images of the gas loaded heat pipe tests, there is not an easily identified vapor gas interface location. Figure 5.33 shows that the background temperature at 320 min which occurs at the end of Condition E is significantly lower than in the other conditions. Both Figure 5.33 and Figure 5.34 show that the inactive gas region may be slightly longer in Condition E than it is in Condition C, which explains why the temperature is similar in both conditions. In the Figure 5.34 the white lines were added to separate each condition. The three-dimensional contour plot shows that during the transition between each condition, the heat pipe never shuts down completely.

In Figure 5.33 and Figure 5.34, the heat pipe is cooler in Condition A. This is caused by the heaters supplying 1.7 W less than in the other conditions as shown in Table 5.8. Early in condition A, a cooler region is disappearing. This is caused by this regions transition from the preheat configuration (right single evaporator and single condenser) to its new configuration. Although the thermal power applied in Conditions B and D are the same, the temperatures are very different. In these transitions the order of the transitions matter. When transitioning between Conditions B and C, the middle hot stagnation point is dissolved, making the heat pipe decrease in the number of stagnation points. Since the vapor flows away from hot stagnation points, it sweeps the vapor away from the stagnation points as well.

When transitioning from C to D, a new hot stagnation point is formed in the middle of the pipe. The vapor flowing from this stagnation point sweeps the gas away from the hot stagnation point and towards the cold stagnation point. Figure 5.33 and Figure 5.34 show that in Condition D little non-condensable gas is located on the left side of the heat pipe. This may be caused by additional hot stagnation point forming between the location of the gas in Condition D and the left side of the heat pipe. This can be seen clearly by comparing the lines at 112 minutes and 240 minutes in Figure 5.35. To return the heat pipe to a state where an even amount of non-condensable gas is on each side of the heat pipe, another step is necessary, such as the dual evaporator and single centered condenser configuration.

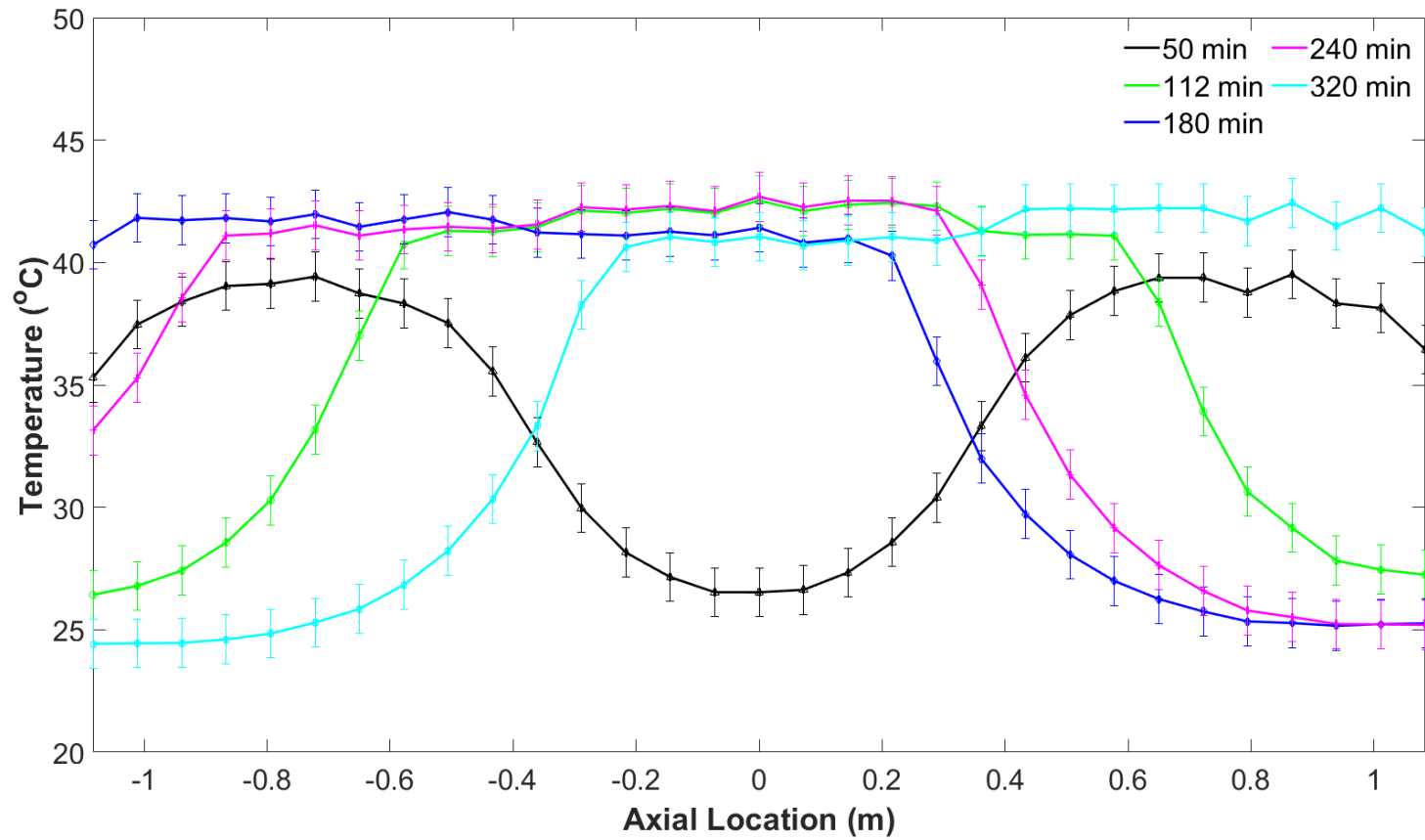


Figure 5.35: Steady-state heat pipe wall temperatures for each condition in the dynamic stagnation point test with uniform total thermal power test.

Figure 5.35 shows the single evaporator tests have similar evaporator and condenser temperatures. The temperature data at these times were used for a segmented thermal power balance calculation. During this calculation, the heat pipe was broken into thirty-one nearly isothermal segments. The temperatures of these segments were determined by the thermocouple in the segment. The thermal power leaving each segment was calculated in a thermal power balance. From these powers, a combined heat transfer coefficient including a linearized radiation component, natural convection component, and forced convection component was calculated for each segment. This combined heat transfer coefficient was used with the adiabatic fin equation shown in Equation (2.21) to show the predicted the axial temperature profile of the non-condensable gas region.

During the analysis, a point in the middle of the gradient was chosen to estimate the length of non-condensable gas in the system. This estimated length and the segmented combined heat transfer coefficients were used to estimate the temperature at each location. If the axial location along in the heat pipe passed beyond the length of non-condensable gas region, the temperature was assumed to be equal to the base temperature. The cross-sectional area used in the approximation was a combination of the cross-sectional area of the heat pipe outer wall and the heat pipe wick. The thermal conductivity for the copper was set to 400 W/(m K) .⁶² Figure 5.36 shows a comparison of the measured heat pipe wall temperature to the heat pipe wall temperature that was estimated using the adiabatic fin equation.

In Figure 5.36, the estimated temperature tends to predict a lower temperature than the measured temperature at the end of the fin. During the analysis it was observed that the solution is very sensitive to the heat transfer coefficient as well as the location and corresponding temperature used in the initial non-condensable gas length estimate. It was found that the best fit results occurred when selecting the temperature where the slope transitions from sharp to more gradual. In each of the configurations, the approximation estimates a lower temperature towards the cold stagnation point. This may be caused by the errors in the heat transfer coefficient and any residual heat in the liquid pool or liquid in the annulus. Further testing with variations in heat pipe wick geometry, fill amounts, and heat transfer coefficients may be useful in providing an explanation.

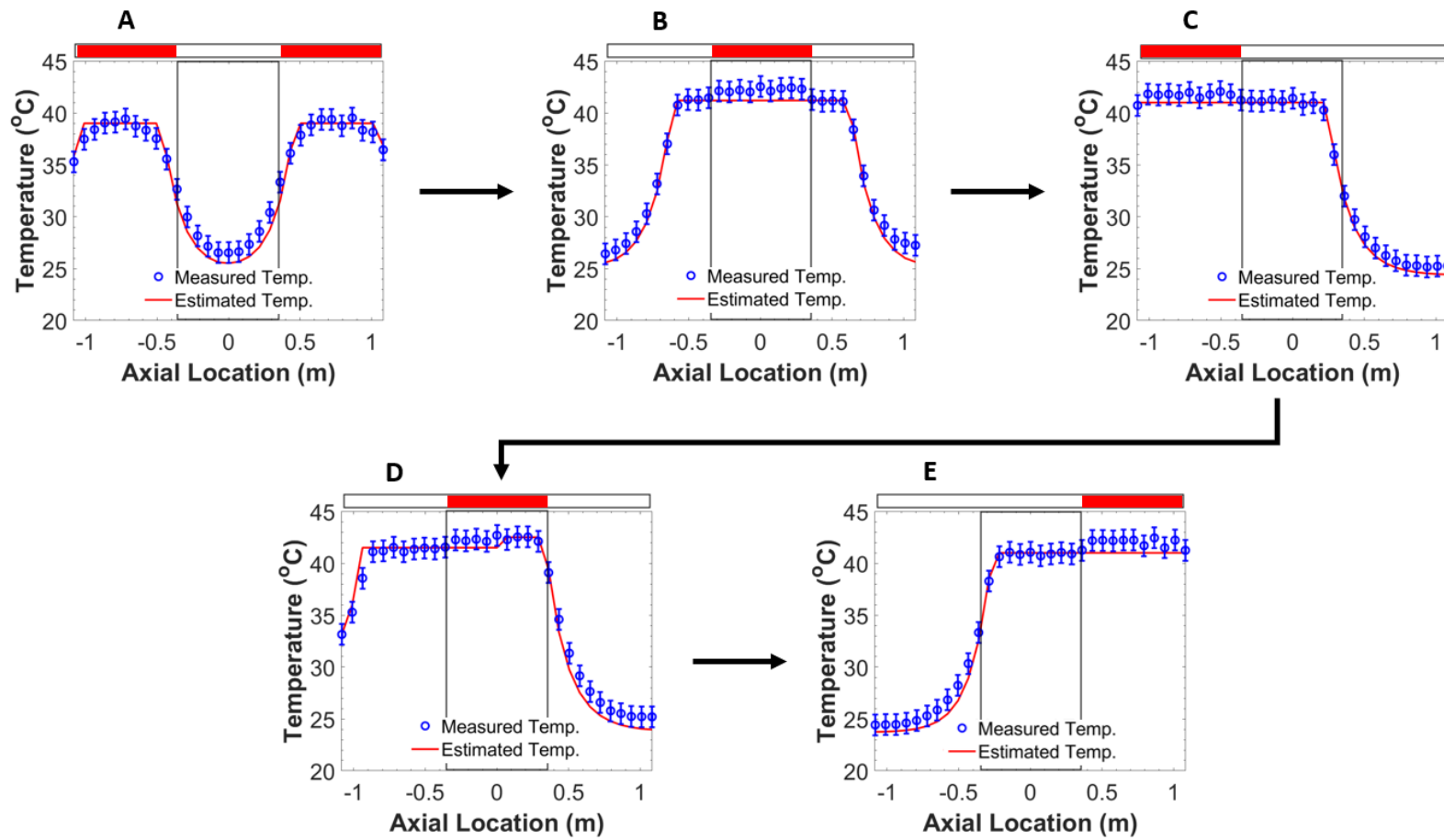


Figure 5.36: Measured heat pipe wall temperature at close to steady state compared to estimated wall temperature for dynamic stagnation point test with uniform total thermal power test.

A second test was conducted to verify the shifts in cold or hot stagnation points caused by differences in the thermal power applied to the heat pipe. Before this test, the pressure controllers were used to reduce the amount of gas present in the heat pipe. In this test multiple variable DC power supplies were used with the switch box to vary the power applied by each zone of heaters. The heat pipe switched between a dual evaporator and single centered condenser configuration and a single evaporator and single condenser configuration. The heater zones are labeled from left to right. Figure 5.37 shows the power applied to the heat pipe by each zone of heaters. The spikes in power occur when the desired voltage is overshoot and is brought back down to the desired voltage. The heat pipe starts in a dual evaporator centered condenser condition with heater zone 1 and heater zone 3 applying 12.3 ± 0.7 W. The total thermal power input heat pipe Condition A, is 24.6 ± 1.0 W. In Condition B the thermal power supplied by heater zone 3 is increased to 23.0 ± 1.0 W for a total thermal power throughput of 35.3 ± 1.3 W. Condition C is right single evaporator and single condenser configuration with total thermal power throughput of the heat pipe is 24.6 ± 1.0 W with all thermal power applied by zone 3. In Condition D the heat pipe returns to a dual evaporator and single centered condenser configuration with heater zone 1 and heater zone 3 providing a total thermal power of 24.6 ± 1.0 W. Condition E is the opposite of Condition B with heater zone 1 applying 23.2 ± 1.0 W and heater zone 3 providing 12.3 ± 0.7 W. Here a total thermal power of 35.5 ± 1.3 W is applied to the heat pipe. Finally, the heat pipe transitions to a left single evaporator and single condenser configuration with a thermal power input of 23.2 ± 1.0 W.

Figure 5.39 shows that the gas region travels almost 0.3 m to the left. Although the gas has also undergone compression, the center point and left most edge of the visible gas region has moved to the left. This is consistent with the cold stagnation point shifting to the left with the increased power input on the right. The opposite can be seen when transitioning from Condition D to E. This test also confirms that it is important to consider what types of stagnation points, hot or cold, are created when predicting non-condensable gas location during stagnation point addition.

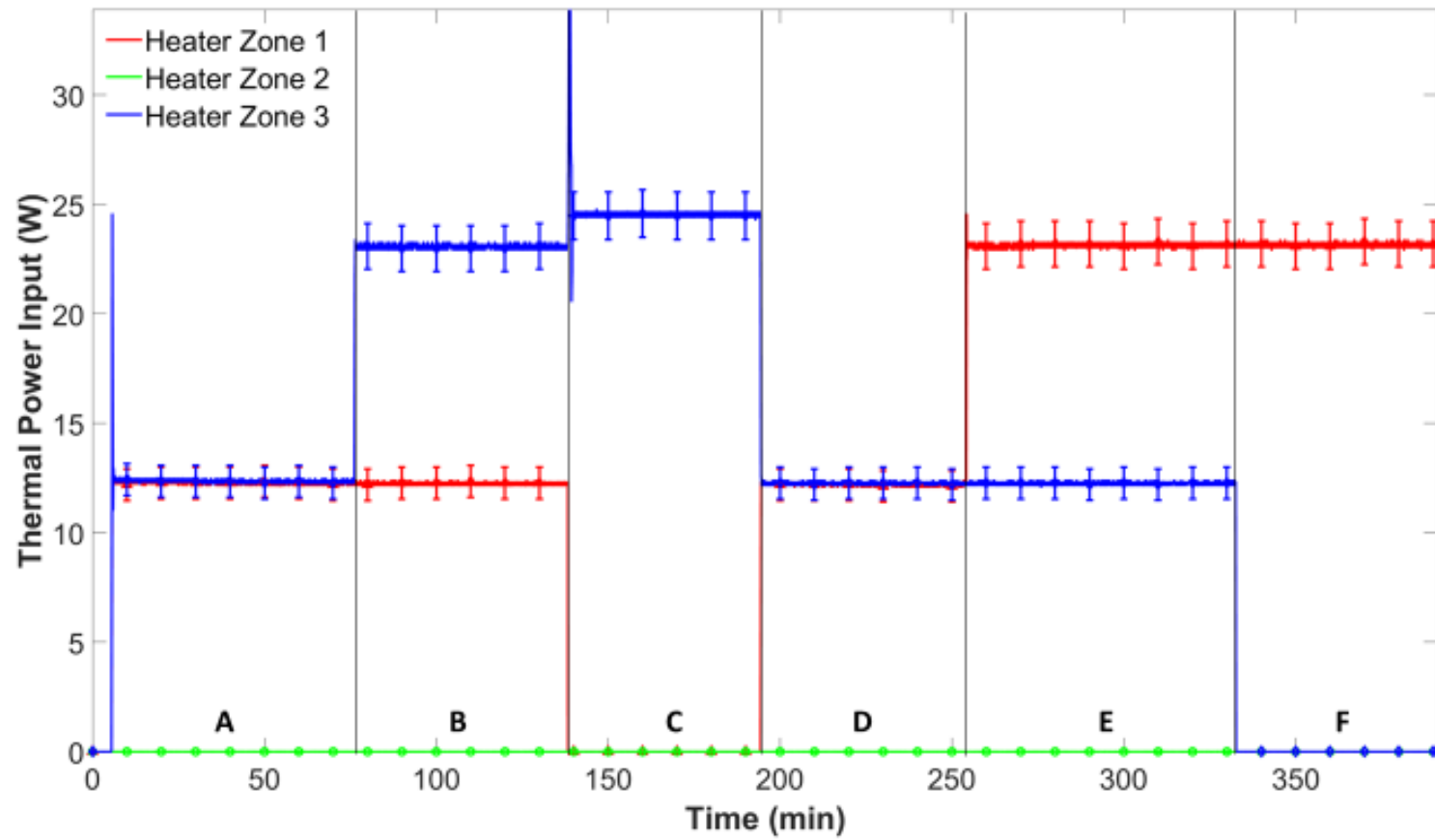


Figure 5.37: Thermal power input for dynamic stagnation point test with non-uniform thermal power input test.

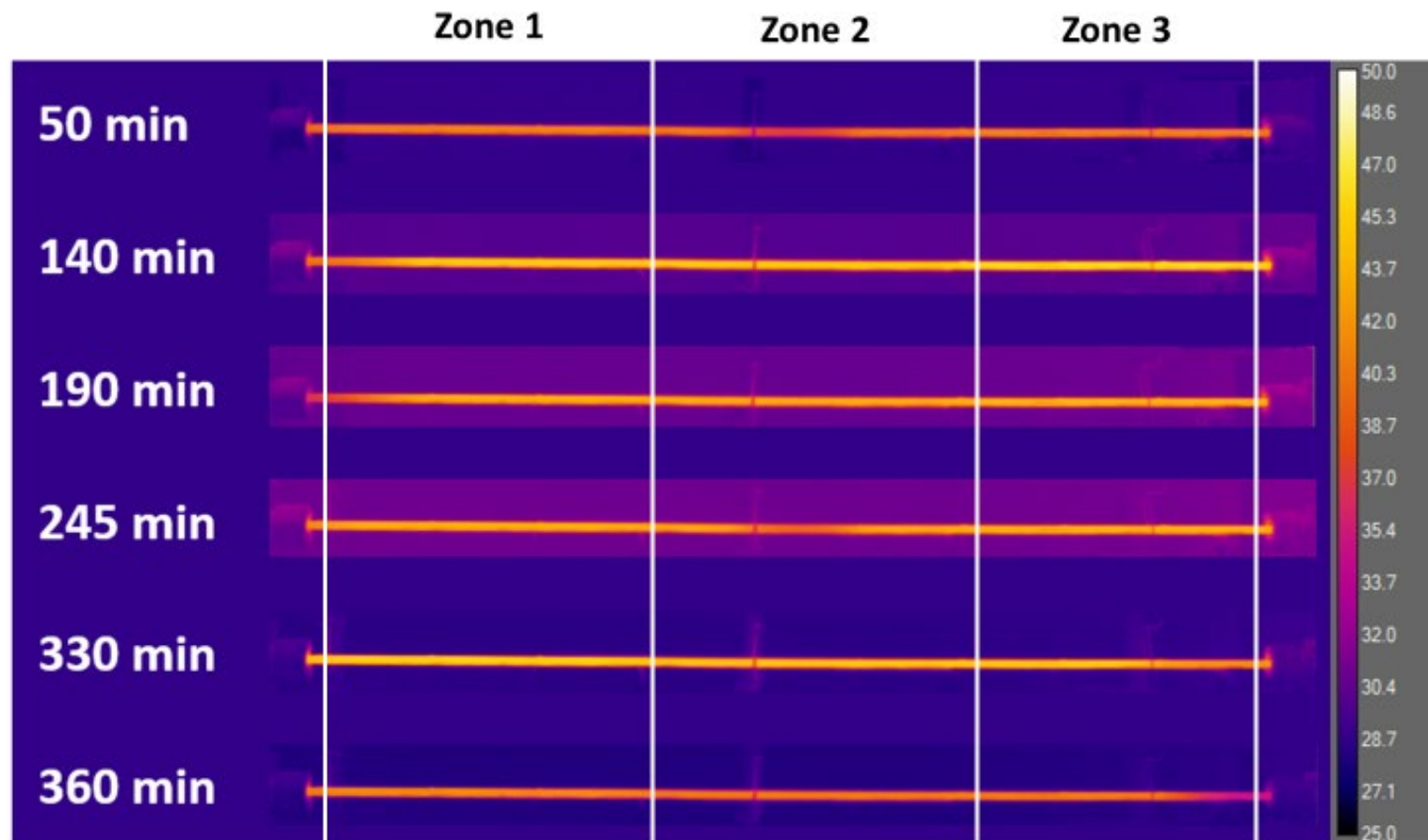


Figure 5.38: selected infrared images for the dynamic stagnation point test with non-uniform thermal power input test.

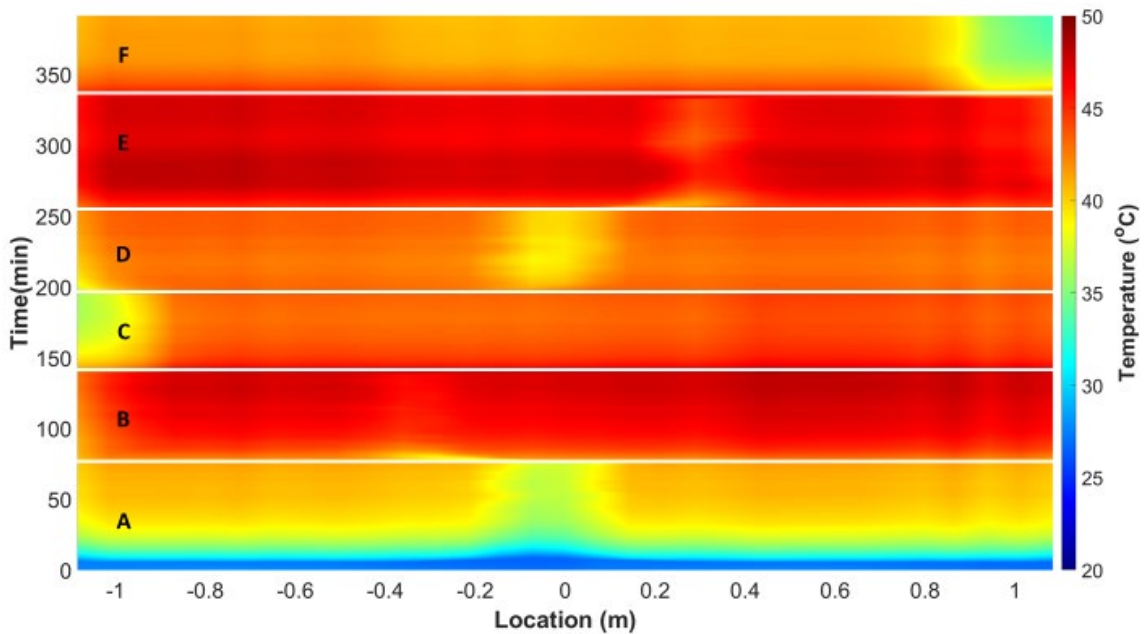
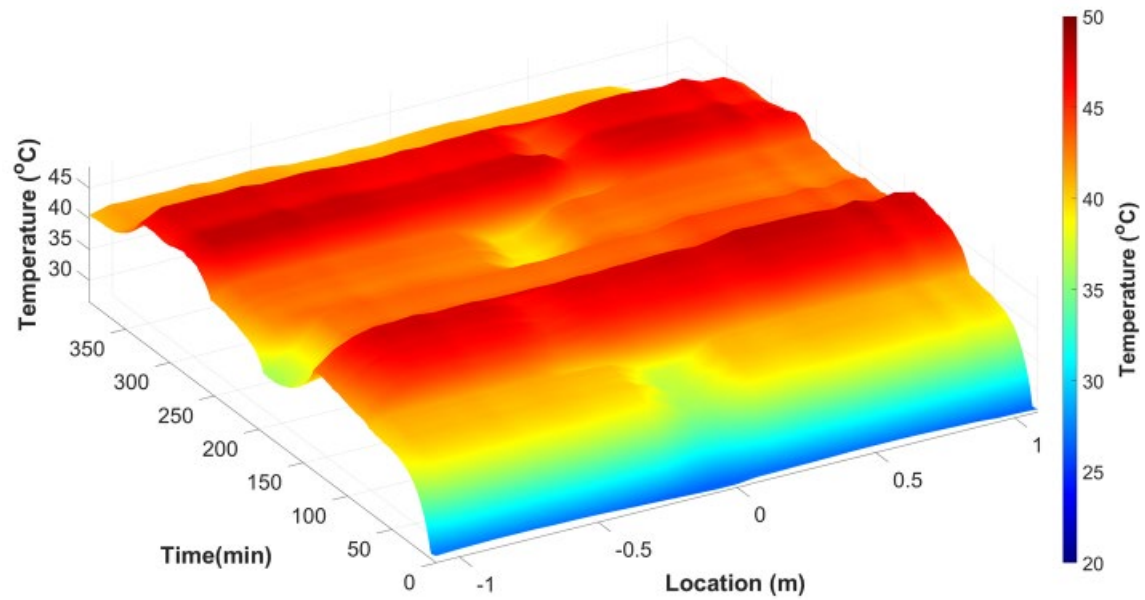


Figure 5.39: Three-dimensional contour plot of heat pipe surface temperature (top) and two-dimensional surface temperature (bottom) for the dynamic stagnation point test with non-uniform thermal power input test.

Unlike the previous test, the non-condensable gas freely moves to the center of the heat pipe when transitioning from two to three stagnation points. This is because the additional stagnation point is a cold stagnation point in the center and the direction of the vapor flow does not impede non-condensable gas travel.

These tests show the influence thermal boundary conditions have on stagnation point locations. The first test showed it is possible for an uneven non-condensable gas distribution to occur with symmetric boundary conditions, as shown in Condition D in Figure 5.34 and Figure 5.35. This indicates that when determining the distribution of non-condensable gas, the previous state of the heat pipe and what types of stagnation points are added by the new thermal boundary conditions must be considered. The first test also showed that the fin approximation can estimate the axial temperature profile of the non-condensable gas region in multiple configurations. The approximation is very sensitive to the heat transfer coefficient and the location and temperature used to estimate the gas length. In each of the approximations the temperature closest to the cold stagnation point was below the measured value. Further testing is suggested to determine the cause of this underestimate.

Air-Cooled Active Controlled Variable Conductance Heat Pipe with Dynamic Stagnation Points and RGA Sampling

In these tests and RGA was used to track helium in an active controlled heat pipe with dynamic stagnation point. Helium was used to determine whether the two gas regions surrounding a middle cold stagnation point would remain separate or mix. This information is needed to determine if a heat pipe can return to its previous state when middle stagnation points are switched between cold and hot. Figure 4.15 shows a diagram of the heat pipe connected to an MKS eVision2 RGA.

In all RGA tests, the heat pipe was allowed to start up in the single centered evaporator and dual condenser configuration. The QB1X pressure controllers injected helium into the left side of the heat pipe and air into the right side of the heat pipe. The helium container

was filled to 1 atm after being evacuated to 300 mTorr. Since air was in the container before helium, it is likely that some residual air remained in the helium container. The environment was used as the air supply to the right pressure controller.

During initial testing helium was measured on the right side of the heat pipe after injection. This was thought to have been caused by the rapid impulse of higher pressure non-condensable gas entering the heat pipe and overwhelming the vapor flow. To test this theory the pressure differential was decreased by incorporating a needle valve before the heat pipe. Before the next test, the right side of the heat pipe underwent successive dilution to remove the helium. Once the right side was sampled and no helium was measured the method restarted. When helium was slowly injected into the left side of the heat pipe with the needle valve cracked open, no helium was detectable on the right side of the heat pipe.

In the first test conducted with the RGA the heat pipe started in fixed conductance single off centered evaporator dual condenser configuration. This test is divided into seven conditions that are labeled in order with letters. At the start of Condition A, helium and air were injected into the left and the right sides of the heat pipe, respectively. Condition B starts with taking an initial sample to determine if any helium mixed with the air during injection. After this sample, the pressure in the isolation volumes was reduced and the sample volume was successively diluted. In Condition C, the heat pipe transitions to a dual evaporator off centered condenser configuration. Concerns regarding the redistribution of gas in changing back to the dual condenser configuration led to the heat pipe being temporarily shut down, rewired, and restarted to have a dual evaporator and single centered condenser during Condition D. Condition E is the heat pipe operating in a dual evaporator and single centered condenser configuration. Condition F is the heat pipe operating in the single centered evaporator and dual condenser configuration while Condition F is the pressure controllers taking a final sample.

Figure 5.40 shows the wall temperature of the heat pipe during Conditions A through Condition G. The horizontal white lines divide each condition. In Condition D the heat pipe can be seen cooling briefly while the heaters were rewired. In addition, the evaporator in Condition F is warmer than the evaporator in Condition B. Figure 5.41 shows the thermal power applied to the heat pipe is similar in both conditions, $35.9 \pm 1.3\text{W}$ in condition B and

35.4 $\pm$ 1.3 W in Condition F. The heat pipe in Condition F is likely warmer due to the increase in room temperature during the test. Thermocouple data shows that between Condition B and condition F, the room temperature increased from 28.5 $\pm$ 1.0°C to 29.7 $\pm$ 1.0 °C.

Figure 5.41 shows the heater power data, pressure controller data, and RGA data measured throughout the duration of the test. To reduce the baseline noise of 1.8×10^{-10} Torr present in the helium data, a moving average filter was applied using 10-point subsets. It can be seen that the initial sample taken after condition A contained a small amount of helium. The RGA measured a helium partial pressure of $5 \times 10^{-10} \pm 2 \times 10^{-11}$ Torr. When successively diluting the sample volume, a helium was detected with a partial pressure on the order of $1 \times 10^{-8} \pm 2 \times 10^{-11}$ Torr was initially measured. During the last measurement only components of air were measured shown in Figure 5.42. This shows that the sample and right isolation volume are free of detectable helium at this point. At the end of the test the heat pipe was once again sampled from the right. This sample showed a higher partial pressure of helium than measured initially. The partial pressure measured was $6 \times 10^{-10} \pm 2 \times 10^{-11}$ Torr. This change is relatively small but may indicate that some mixing took place during the test. The concentration of helium in the helium air mixture was calculated using the partial pressures. Figure 5.43 shows the concentration of helium in the helium-air mixture on the right side of the heat pipe during testing. In agreements with the partial pressure numbers, Figure 5.43 shows that the greatest amount of helium is measured during Condition G. This further supports that mixing occurred after Condition B. The mixing is likely a result of shutting down and restarting the heat pipe during the test.

A second test active controlled gas loaded heat pipe test was conducted with the RGA. Before the test began, 35 mL of water was added into the heat pipe to prevent dry out during testing. For Condition A the heat pipe was in a fixed conductance single centered evaporator dual condenser configuration with a thermal power input of 35.7 $\pm$ 1.35 W. After startup, helium was added to the left side of the heat pipe followed by the addition of air to the right side of the heat pipe.

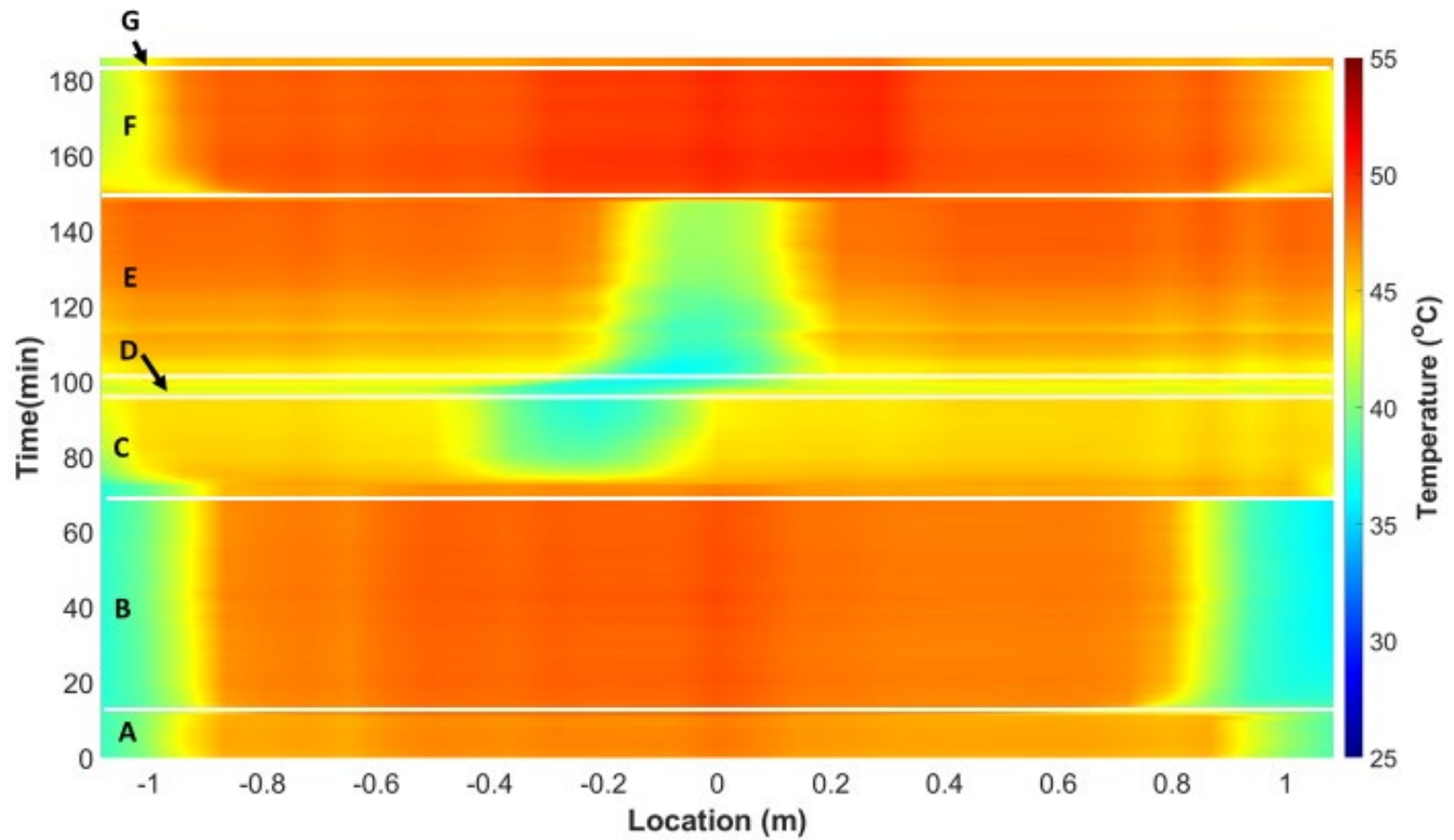


Figure 5.40: Two-dimensional contour plot of heat pipe wall temperature with time and axial location of the initial active controlled gas loaded heat pipe test with an RGA.

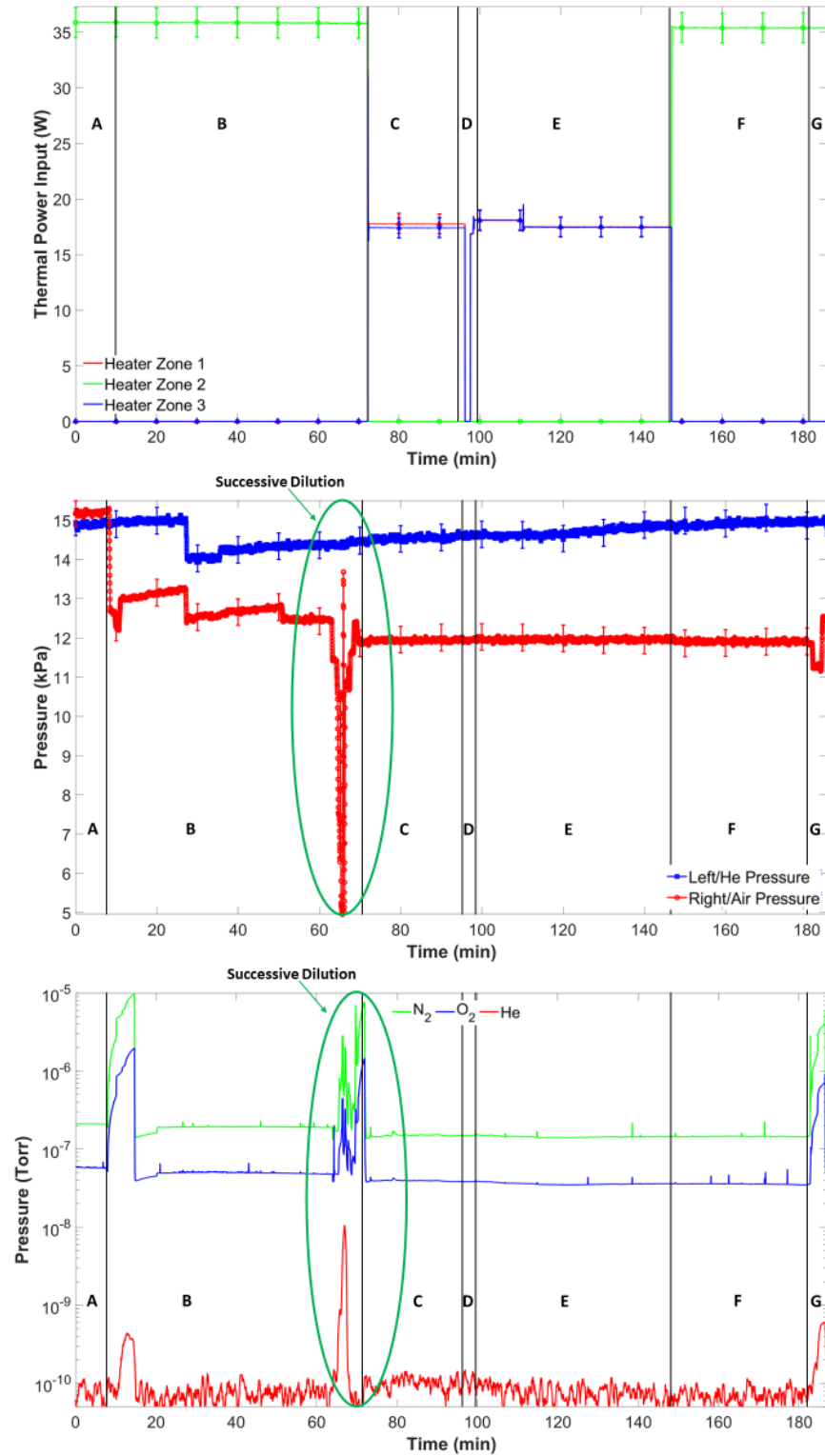


Figure 5.41: Heat power data (top), pressure controller data (middle), and RGA data (bottom) for the initial active controlled gas loaded heat pipe test with an RGA.

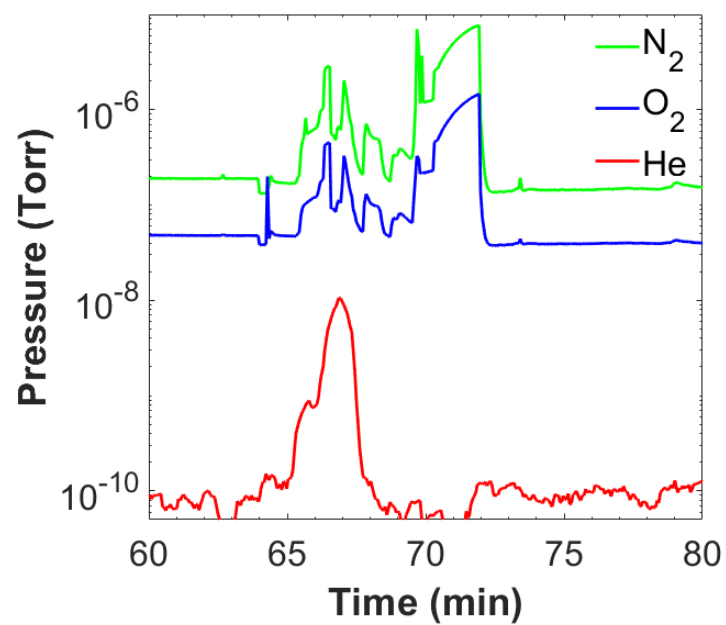
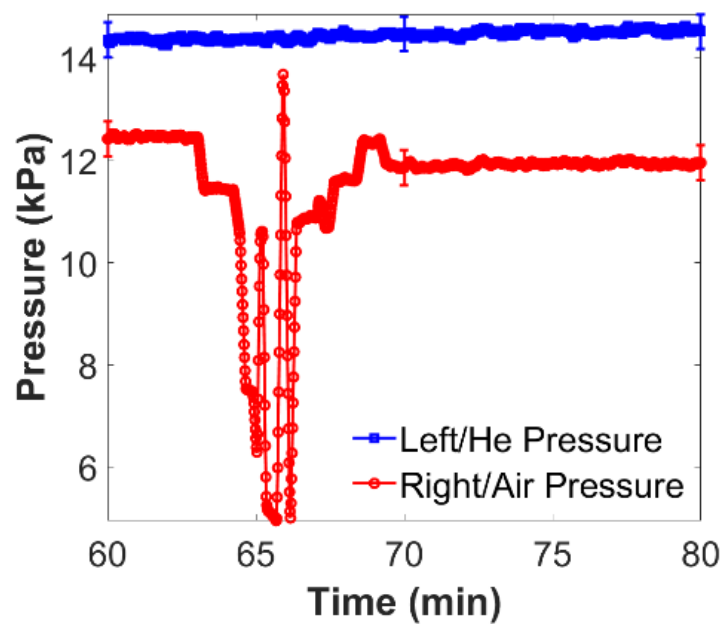


Figure 5.42: Pressure controller data (left) and RGA data (right) taken during the successive dilution process.

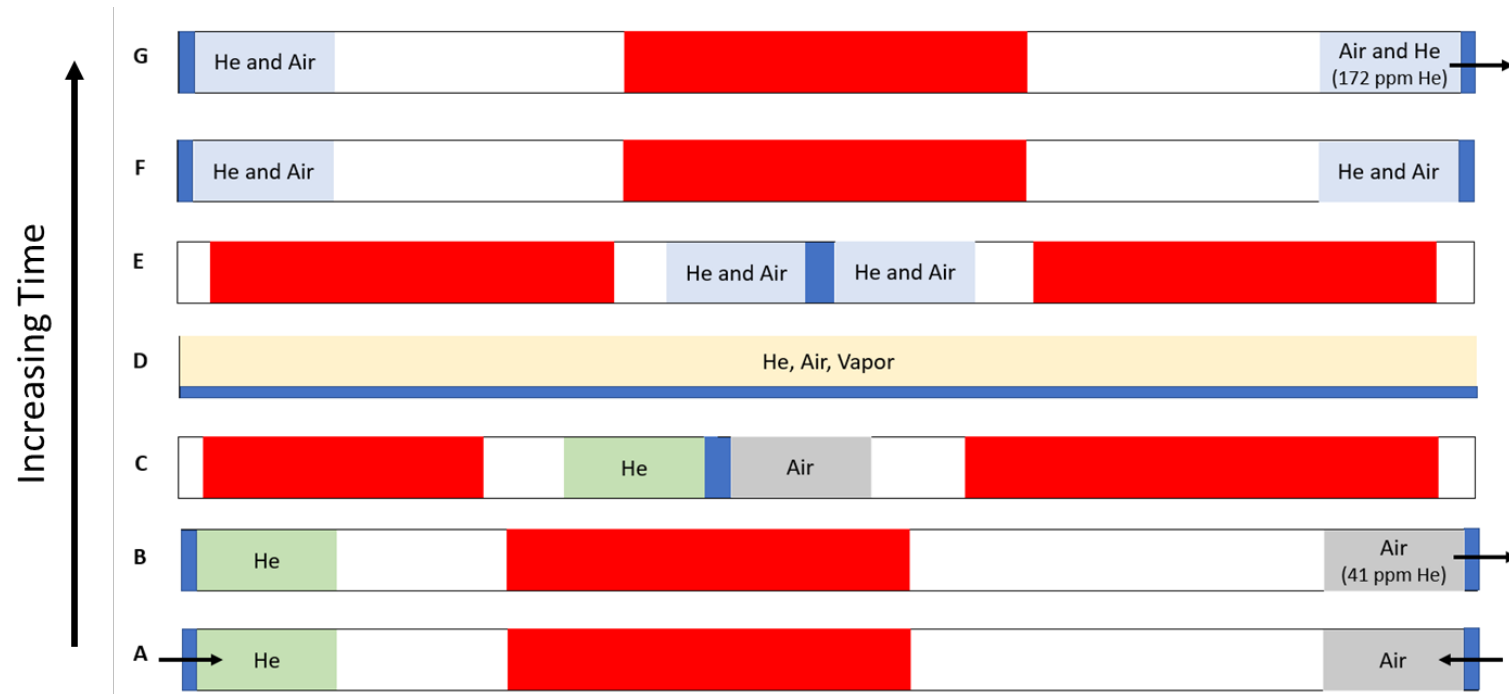


Figure 5.43: Concentration of helium in the helium-air mixture on the right side of the heat pipe measure during the first RGA test.

At approximately 30 minutes in condition B the sample volume was successively diluted with air. Following this successive dilution, a sample was taken from the right side of the heat pipe. This sample showed only the baseline helium measurement. The left side of the heat pipe was sampled to ensure helium was present in this side of the heat pipe. The RGA found that this sample contained a partial pressure of helium $4 \times 10^{-8} \pm 2 \times 10^{-11}$ Torr. The sample volume was cleared by opening all the bypass valves. A secondary non-condensable gas sample was taken from the right side of the heat pipe at a time of 86 minutes to ensure that no helium had reached this side of the heat pipe during 30 minutes of operation. This sample also showed only the baseline level of helium. A closer view of these measurements is shown in Figure 5.45. Like the previous test, the helium data presented uses a moving average filter was applied using 10-point subsets to reduce noise.

The heat pipe configuration changed to a dual evaporator centered condenser configuration at 93 minutes for condition C. In this condition, heater zones 1 and 2 supplied 17.3 ± 0.90 W each for a total thermal power input of 34.6 ± 1.25 W. The heat pipe remained in this condition until a time of 174 minutes. For condition D the heat pipe transitioned back to a single centered evaporator and dual condenser configuration with a thermal power of 35.4 ± 1.34 W. At 240 minutes the non-condensable gas on the right side of the heat pipe was sampled. The RGA measured a small $5 \times 10^{-10} \pm 2 \times 10^{-11}$ Torr helium partial pressure indicating a slight amount of non-condensable gas mixing occurred in the dual evaporator centered condenser configuration. The heat pipe transitioned from a single centered evaporator dual condenser configuration to a left single evaporator single condenser configuration for Condition E. After operating in this configuration for over an hour, a non-condensable gas sample was taken from the right side of the heat pipe. The RGA measured a helium partial pressure of $6 \times 10^{-9} \pm 2 \times 10^{-11}$ Torr indicating a greater amount of helium present on the right side of the heat pipe. The concentration of helium was calculated using the partial pressures of helium, nitrogen, and oxygen. Figure 5.44 shows that after the injection of the air and helium, most of the helium remains on the left side of the heat pipe as measured in Condition B. At Condition D, a slight increase in helium concentration was measured on the right side of the heat pipe. This indicates a small amount of mixing occurred when transitioning to and from Condition C. Lastly, in

Condition E the concentration of helium in the helium-air mixture on the right side of the heat pipe is significantly higher. It should be noted that the concentration of helium measured on the right side of the heat pipe in Condition E is greater than the initial concentration of helium on the left side of the heat pipe in Condition B. This may be caused by axial concentration gradients in the non-condensable gas which have been observed in the literature. Further testing with an axially traversing RGA probe would provide insight into this effect.

Figure 5.46 through Figure 5.47 show graphical data taken during this test. Figure 5.46 shows the wall temperature of the heat pipe during the test. The horizontal white lines in Figure 5.46 divide the results into several conditions. The non-condensable gas region is larger and cooler in this test compared to the results shown in Figure 5.41. The evaporator is also warmer in this test which indicates the volume of non-condensable gas is larger in this test. This may be caused by the considerably cooler room temperature during the test. In this test the room temperature was $24 \pm 1.0^{\circ}\text{C}$ instead of $29 \pm 1.0^{\circ}\text{C}$ leading to a shorter active condenser. Figure 5.47 shows the heater power data, pressure controller data, and RGA data measured throughout the test. To reduce the baseline noise of 1.8×10^{-10} Torr present in the helium data, a moving average filter was applied using 10-point subsets.

The second test showed less mixing in the dual evaporator single centered evaporator test. This could be from either the difference in liquid inventory available for pool formation or the fact that the second test ran continuously without a shut down. The second tests suggests that new cold and hot stagnation points begin to form rapidly after a significant change in external thermal boundary conditions. This formation limits the amount of mixing of two different non-condensable gases and suggests it may be possible for the heat pipe to return to its previous state with minor differences if the number of stagnation points remains the same. If the number of stagnation points changes such as the heat pipe temporarily shutting down or the heat pipe changing from a three stagnation point configuration to a two stagnation point configuration, the heat pipe may not be able to experience greater changes when attempting to return to its previous state. Further testing with varied heat pipe fill amounts is suggested to confirm the required conditions to prevent mixing.

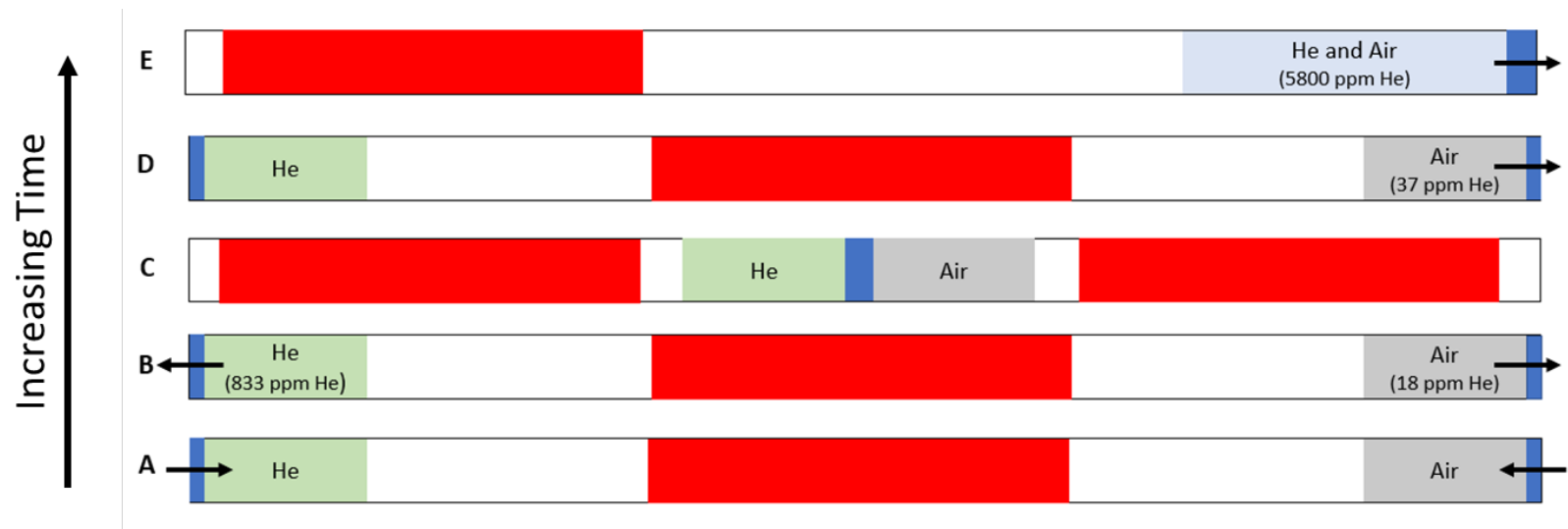


Figure 5.44: Concentration of helium in the helium-air mixture on the right side of the heat pipe measure during the second RGA test.

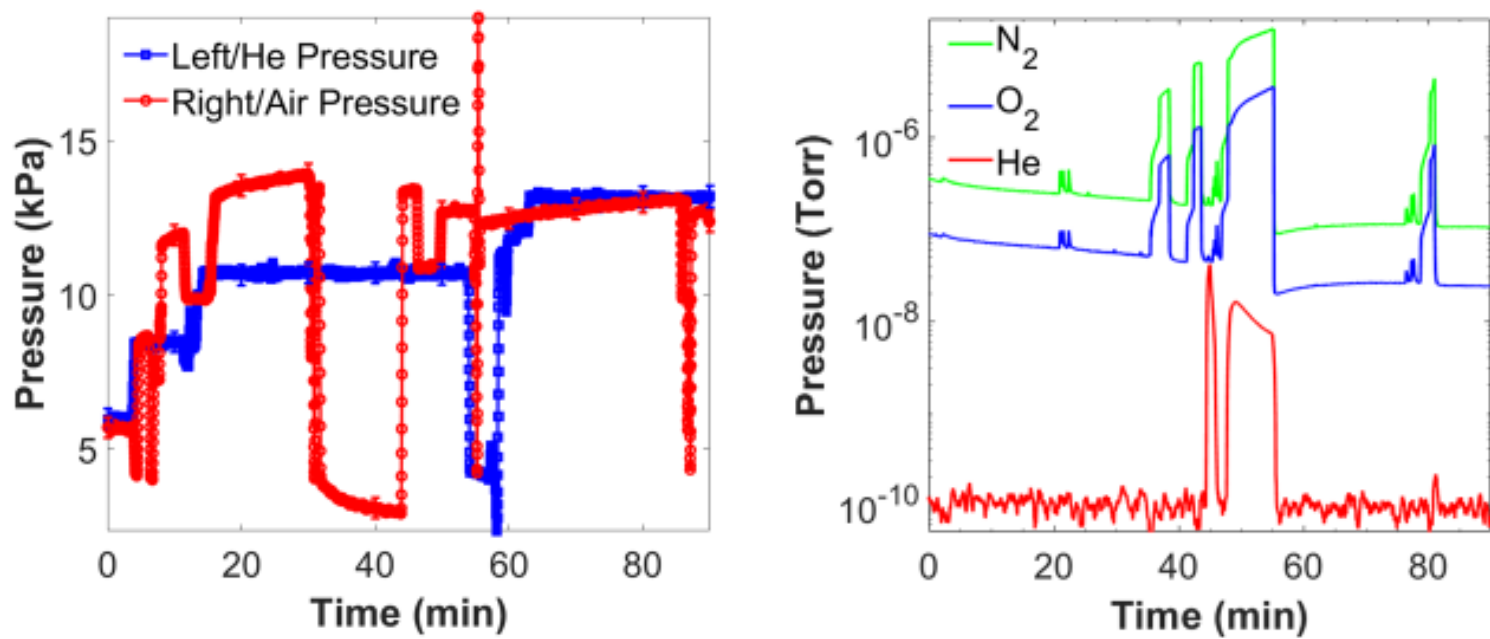


Figure 5.45: Pressure controller data (left) and RGA data (right) taken after helium injection.

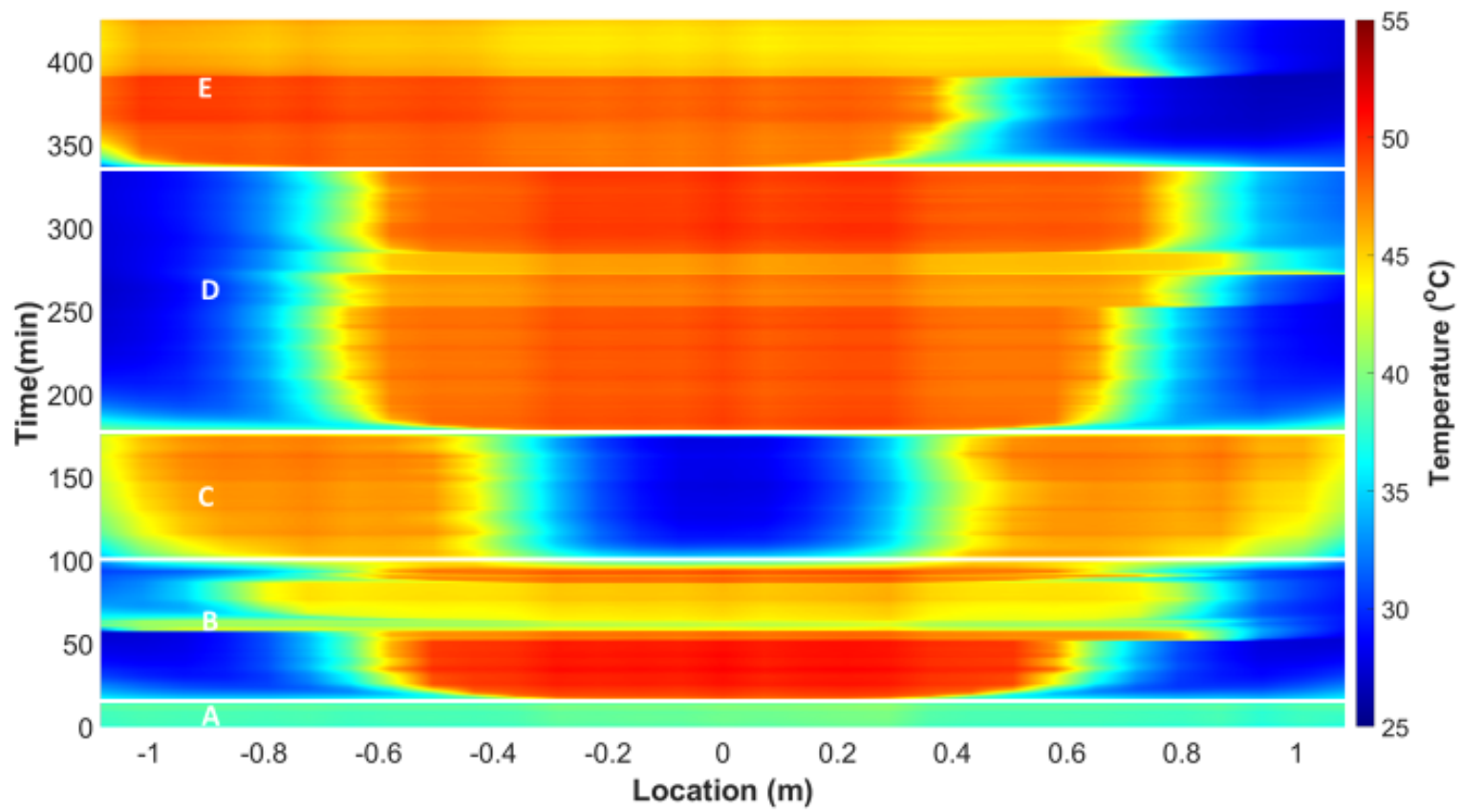


Figure 5.46: Two-dimensional contour plot of heat pipe wall temperature with time and axial location of the second active controlled gas loaded heat pipe test with an RGA.

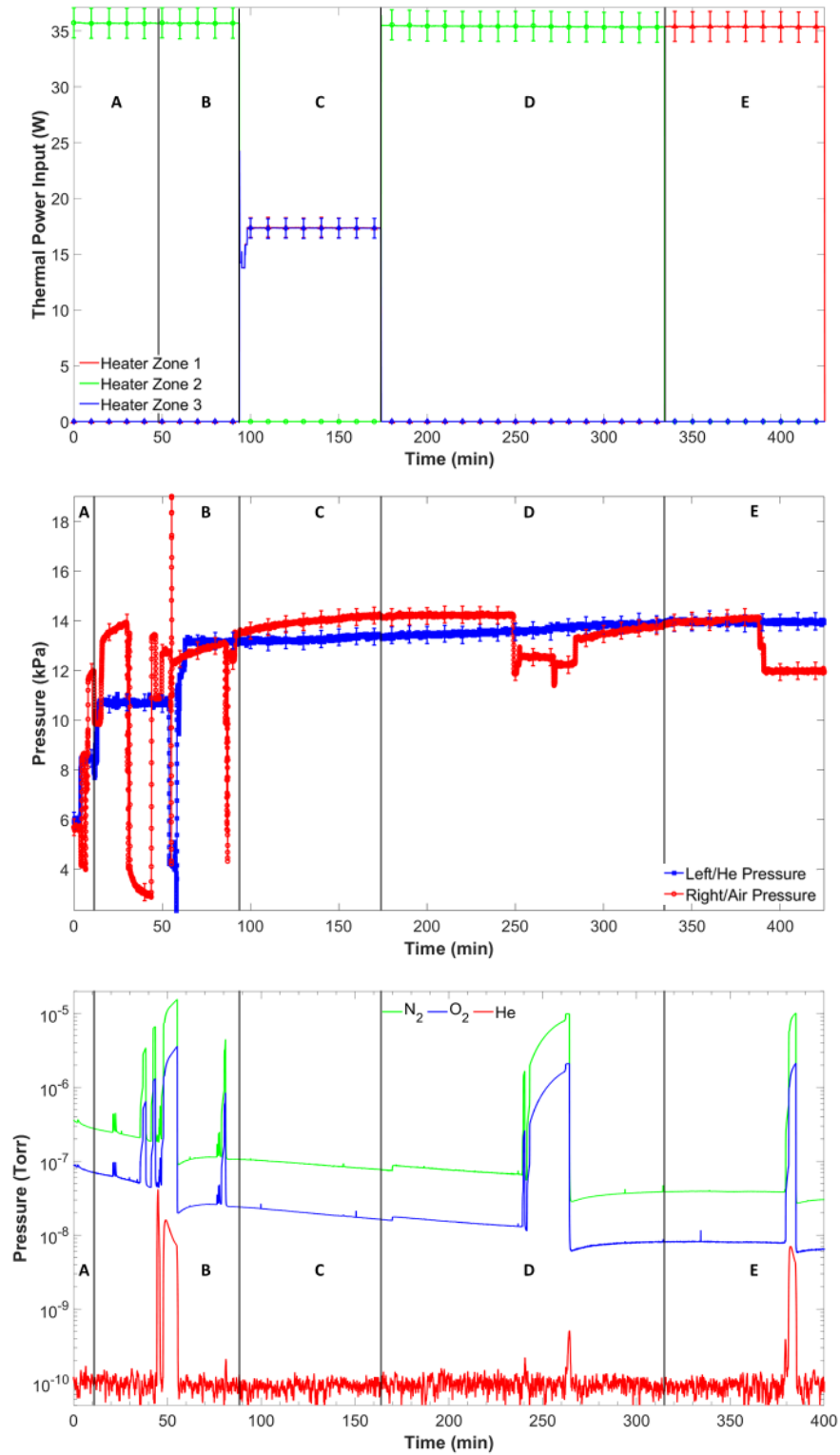


Figure 5.47: Heat power data (top), pressure controller data (middle), and RGA data (bottom) for the second active controlled gas loaded heat pipe test with an RGA.

CHAPTER SIX

CONCLUSIONS AND RECOMMENDATIONS

This work focused on heat pipes with multiple evaporators or multiple condensers. Feasibility tests conducted on a fixed conductance heat pipe show that heat pipes can operate stably in various evaporator and condenser configurations. Non-condensable gas was introduced into the heat pipe for two purposes. The first purpose was to confirm symmetric gas distribution in passive gas loaded heat pipes with symmetrical thermal boundary conditions. The second purpose was to use the non-condensable gas to identify the location of cold stagnation points.

For this heat pipe the non-condensable gas and excess liquid regions (inactive) were modeled using the adiabatic fin tip approximation that incorporates natural convection, forced convection, and linearized radiation heat transfer coefficients. The adiabatic fin tip approximation under predicts the temperature at the end of inactive regions. Under prediction is attributed to combined heat transfer coefficient uncertainty and residual heat in the heat pipe end fitting.

When symmetric thermal boundary conditions are applied to the heat pipe, non-condensable gas distributes symmetrically. When thermal boundary conditions are asymmetric, the length of the inactive condenser region caused by non-condensable gas is also asymmetric. This is supported by the passive gas loaded heat pipe tests and is shown when the single centered evaporator and dual condenser test is compared to the single off centered evaporator and dual condenser test.

The active controlled non-condensable gas tests showed that it may be possible to effectively shut down one of the condensers of a multiple condenser heat pipe. An asymmetric gas distribution may be created with symmetric thermal boundary conditions if active control is used. Active control may also create a symmetric gas distribution with asymmetric thermal boundary conditions. These features may be of use in applications where the heat pipe is thermally coupled to more than one heat exchanger.

Consideration must be made for the rate of non-condensable gas addition. Rapid introduction of non-condensable gas can cause the non-condensable gas to pass hot stagnation points and reside on the opposite side of the heat pipe. The state of the heat pipe must be considered before increasing the number of stagnation points. In passive gas loaded heat pipes, the gas may remain on one side of a newly formed hot stagnation point despite symmetric boundary conditions creating hysteresis.

The results of the calorimeter cooled tests indicate that passive variable conductance heat pipes with multiple separate condensers distribute heat to each condenser evenly, but additional testing is required to confirm this. The results also suggest that for a passive controlled gas loaded heat pipe, it can operate if heat exchanger conditions change rapidly. The clamps used on the calorimeter in the single evaporator and single condenser configuration were prone to develop leaks across the vacuum fittings when installing insulation. Leaks change the thermal resistance of the gas gap creating a condition that is unknown and not repeatable. The thin gas gap with bends in the heat pipe created inconsistent thermal resistance across the gas gap in the single evaporator and single condenser configuration. However, these effects were not noticeable in the dual condenser configuration.

The RGA tests suggest that cold and hot stagnation points begin to form rapidly after a significant change in thermal boundary conditions. The formation of the cold stagnation points limits the amount of mixing of two different non-condensable gases. This demonstrates repeatable operation: the heat pipe returns to its previous state with minor differences so long as the number of stagnation points remains the same. In these transitions the heat pipe behavior is repeatable. However, if the number of stagnation points decreases such as the heat pipe temporarily shutting down or the heat pipe changing from a 3-stagnation point configuration to a 2-stagnation point configuration, the heat pipe behavior is not repeatable. This insight may be used in the design and application of multiple evaporator or multiple condenser heat pipes. If returning to a similar state after the heat pipe is shut down or the number of stagnation points decreases, is desired, active gas control must be used to reset the heat pipe.

Overall, the results of these tests lead to several observations about non-condensable gas loaded heat pipes with multiple evaporators or multiple condensers. The observations below apply to fixed conductance, passive variable conductance, and active variable conductance heat pipes:

1. A stagnation point is a point of zero liquid and vapor velocity that bounds flow cells within a heat pipe when there is no excess liquid or non-condensable gas in the heat pipe.
2. The number and type (hot or cold) of stagnation points in a heat pipe are determined by the thermal boundary conditions applied to the heat pipe.
3. Non-condensable gas and excess condensed liquid may alter the axial heat flux and thus the stagnation point location.
4. Inactive condenser regions composed of non-condensable gas and/or liquid pools surround cold stagnation points.
5. Active cold stagnation points are formed at the interface of the vapor and inactive condenser regions.
6. An active cold stagnation point is a point of zero liquid and vapor velocity that bounds flow cells within a heat pipe that has an inactive condenser region.
7. Cold and hot stagnation points form rapidly in response to changes in thermal boundary conditions.

Although this work provides general insight into the behavior of gas loaded heat pipes with multiple evaporators or condensers, further work is recommended to better characterize their behavior. Further testing may include, improved calorimeter testing, more advanced RGA testing, the incorporation of thermocouples to measure azimuthal temperature gradients, testing different heat pipe outer tubes, and testing in a more controlled environment.

Improved calorimeter testing may provide more insight on the thermal power distribution of multiple evaporator or multiple condenser heat pipes. Improved testing includes improving the insulation, testing different calorimeter designs, and increasing the run time of the test. Improved insulation could be a combination of using multi-layer insulation and testing in a vacuum chamber. Different calorimeter designs may be

calorimeters that are direct contact, have a larger gas gap, or are shorter to limit changes in thermal coupling caused by the heat pipe contacting the calorimeter along the bends. Additional upgrades would limit the number of fittings or fitting types used to secure and seal the calorimeter around the heat pipe. Increased test run time would allow for the heat pipe and calorimeters to approach a steady state.

These heat pipe configurations may benefit from more advanced RGA tests. This includes improving the spatial resolution by increasing the length of a traversing sample tube so samples may be taken when the gas is centered in the heat pipe. Another advancement could be to include a method to take radial measurements. This would allow investigation of the parabolic gas front observed in the infrared images. Another method to monitor the gas front would be including thermocouples around the circumference of the heat pipe.

As suggested by the literature, a diffusion layer of working fluid vapor and gas may develop and change the temperature profile of the non-condensable gas region. A heat pipe made from a material that has a lower thermal conductivity should be tested to determine if the other methods of capturing the inactive gas region need to be used.

Lastly, testing this type of heat pipe in a more controlled environment would be beneficial. This may include both the ability to monitor any working fluid removed from the heat pipe during active tests as well as the ability to have better control over the heat transfer coefficient. Greater control of both the working fluid inventory and the heat transfer coefficient could provide insight into temperature profiles that are caused by the liquid pool in comparison to those caused by the non-condensable gas.

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APPENDIX

Appendix A: Unpublished Dual Condenser Heat Pipe Characterization

Below is an unpublished fragment that characterizes the operation of heat pipes in contact with more than one hot or cold reservoirs that was generated by Dr. Robert Reid in the 2013-to-2015-time frame. An alternative theory found elsewhere in this dissertation was adapted and extended from this derivation.

Physical System Considered

Consider the heat pipe depicted in Figure A.1 comprising a hermetically sealed cylindrical tube with a working fluid of ample quantity to saturate and wet a capillary wick lining the tube perimeter. The wick enables a capillary driving pressure to form across a liquid-vapor interface of the working fluid. This driving pressure allows working fluid circulation through the heat pipe by its liquid surface tension forces.

A single hot reservoir contacts the heat pipe midway down its length. Cold reservoirs contact the heat pipe near each end. Heat transfers from the hot reservoir through the heat pipe wall. This heat vaporizes working fluid near the wick surface beneath the hot reservoir. Mass transfer by evaporation at the liquid-vapor interface creates capillary forces via a low-pressure zone in the liquid and a high-pressure zone in the vapor.

Vapor from the zone in contact with the hot reservoir (also called an evaporator) flows to the zones in contact with the cold reservoirs (also called condensers). At each cold zone, the vapor condenses to liquid. The latent heat of vaporization from the condensed working fluid conducts across the cold zone walls to the cold reservoirs. The wick that lines the inside surface of the heat pipe tube then draws liquid condensate from the cold zones back to the hot zone by capillary forces completing the circuit.

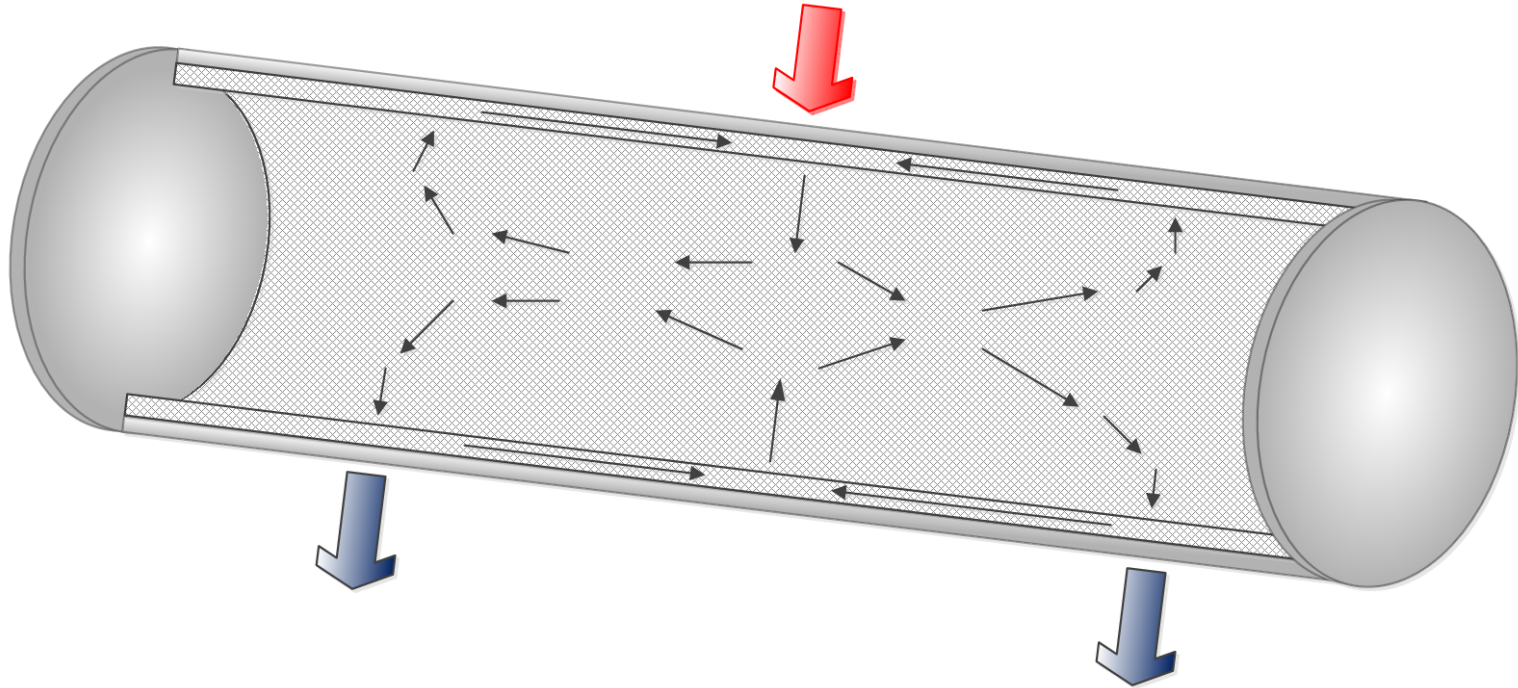


Figure A.1: Heat pipe in contact with heat from a hot thermal reservoir (red arrow), working fluid movement (black arrows), and heat transfer from heat pipe to cold thermal reservoirs (blue arrows).

Thermal Boundary Conditions

Figure A.1 represents one possible case for a larger class of systems. This larger class may include placement of any number of thermal reservoirs at arbitrary locations along the heat pipe length. Loading a heat pipe with a constant or varying quantity of non-condensable gas offers more combinations for this larger class. Here we examine solely simple examples of reservoir placement.

Consider a heat pipe with applied power per unit length $\dot{Q}'(z)$ over domain, $0 \leq z \leq L$. For regions along the heat pipe where $\dot{Q}'(z) > 0$, a hot reservoir supplies thermal power to the heat pipe, for regions where $\dot{Q}'(z) = 0$, the heat pipe is adiabatic with respect to its surroundings, and for regions where $\dot{Q}'(z) < 0$, the heat pipe supplies thermal power to a cold reservoir. Regardless of the applied power location, the heat transfer through the heat pipe over domain, $0 \leq z \leq L$, is:

$$\dot{Q}_T = \frac{1}{2} \int_0^L |\dot{Q}'(z)| dz \quad (\text{A.1})$$

Equation (A.1) generally applies to heat pipes in contact with numerous hot and cold reservoirs over, $0 \leq z \leq L$. The following treatment considers two restrictive cases for heat pipes lacking excess liquid or non-condensable gas loading. The first case considers heat pipes in contact with a single hot reservoir and two cold reservoirs. The second case considers heat pipes in contact with a single cold reservoir and two hot reservoirs. Within a heat pipe, the area average vapor velocity (m/s) in the axial direction, $u(z)$, relates to the applied power per unit length, (W/m), $\dot{Q}'(z)$, through:

$$\int_0^z \dot{Q}'(z) dz = \rho_v u(z) A_v h_{fg} \quad (\text{A.2})$$

Where z is axial position (m), and A_v is the axial cross-sectional area (m^2) of the vapor region, and ρ_v and h_{fg} are the vapor density (kg/m^3) and latent heat of vaporization (J/kg), respectively, at the saturation temperature.

Figure 2 shows examples of linear heat rate compared to area average axial vapor velocity as a function of axial position from Equation (A.2). The left panel shows an example of a heat pipe in contact with a single hot reservoir and a single cold reservoir (a classic heat pipe). Here, the hot reservoir applies power uniformly with length, vaporizing the working fluid. Uniform vaporization per unit length linearly accelerates the vapor from zero velocity at the left end of the heat pipe (the evaporator entrance). For incompressible flow, this vapor reaches its maximum velocity beneath the right end of hot reservoir (the evaporator exit). Uniform power per unit length extracted from the cold reservoir results in a linear decrease in area average axial vapor velocity with position. The flow decelerates from a maximum velocity at the entrance of the cold reservoir (the condenser entrance) to zero velocity at the end of the heat pipe (the condenser end). Note that power applied to the heat pipe assumes a positive value and power extracted from the heat pipe assumes a negative value. At steady condition, the integrated linear heat rate above the horizontal axis equals the integrated linear heat rate beneath the horizontal axis.

The right panel of Figure A.2 shows an example of a heat pipe in contact with a single hot reservoir and cold reservoirs on either side of the hot reservoir (also known as a dual condenser heat pipe). Interpretation of the right panel is like the left panel. Here, vapor flow from left to right has a positive velocity sign convention and vapor flow from right to left has a negative velocity sign convention. There exists an axial location, ξ_A , over the heat pipe length that partitions the working fluid flow into successive counter-rotating flow circuits. This location satisfies:

$$\int_0^{\xi_A} \dot{Q}'(z) dz = 0 \quad (\text{A.3})$$

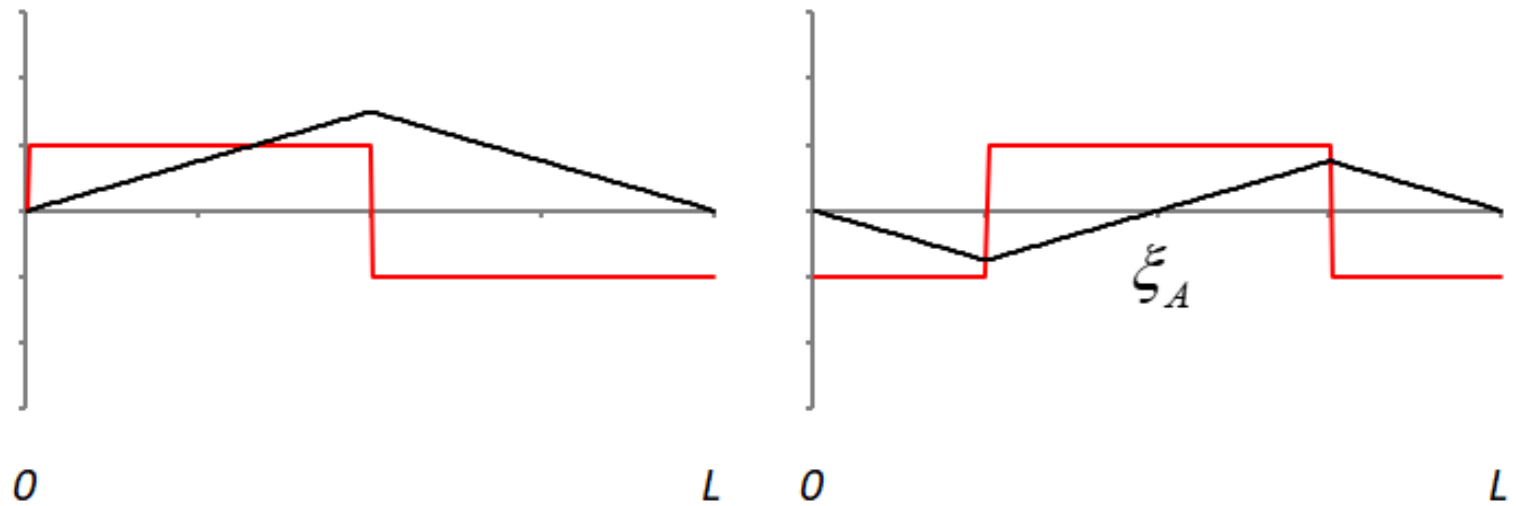


Figure A.2: Linear heat rate profile (red curves) and area average axial vapor velocity (black curves) versus axial position for a heat pipe in contact with a single hot and cold reservoir (left) and a heat pipe in contact with a single hot reservoir and two cold reservoirs (right).

It follows if $\xi_A = L$, then the heat pipe is in contact with a single hot reservoir and a single cold reservoir (a classic heat pipe). The net total heat transfer rate along the length of a heat pipe having two flow circuits is:

$$\dot{Q}_1 = \frac{1}{2} \int_0^{\xi_A} |\dot{Q}'(z)| dz \quad (\text{A.4})$$

And

$$\dot{Q}_2 = \frac{1}{2} \int_{\xi_A}^L |\dot{Q}'(z)| dz \quad (\text{A.5})$$

So that,

$$\dot{Q}_T = \sum_{i=1}^2 \dot{Q}_i \quad (\text{A.6})$$

On Figure A.2 the partition point is located where the axial vapor velocity curve crosses the horizontal axis. The integrated area under the applied power per unit length curve on either side of the partition point sums to zero.

Figure A.3 shows qualitative examples of applied power per unit heat pipe length (red curves) and area average axial vapor velocity (black curves) versus position along the heat pipe axis. In all cases shown in Figure A.3, the area in contact with the hot reservoir is the same as the area in contact with the cold reservoir.

Panel [1,1] on Figure A.3 shows uniform power applied to and rejected from a heat pipe in contact with a single hot reservoir and a single cold reservoir. This plot is marked “Tr” as uniform power is applied from the hot reservoir and uniform power is rejected to the cold reservoir, allowing solution by analytically tractable pressure profile relations such as those found in Busse (1967) in the viscous and incompressible flow regimes. Examples

of configurations shown in subsequent panels that are not analytically tractable are marked “n-Tr.”

Panel [1,2] shows non-uniform power with exponential shape applied to and rejected from a heat pipe in contact with a single hot reservoir and a single cold reservoir. Since power is non-uniformly applied to either the hot or cold regions of the heat pipe (in this case both), this configuration does not permit solution by analytically tractable pressure profile relations. The sharp transitions in slope of the area average velocity with position cause sharp discontinuity in velocity profile with position further complicating evaluation of pressure profile even by numerical means.

Panel [1,3] shows another example of an analytically non-tractable heat pipe configuration with a cosine heating profile on one end and an exponential cooling profile on the other end. The cosine heating profile in Panel [1,3] more closely approximates the uniform heating profile in Panel [1,1] than the exponential heating profile in Panel [1,2]. An analytical pressure profile solution for the heating profile in Panel [1,3] may be conceivably approximated. However, downstream of the heating zone the transition to an exponential cooling profile leads to significant adjustments in vapor velocity profile rendering futile all hope for an analytical pressure profile solution over the heat pipe length in contact with the cold reservoir.

The second row, Panels [2,1], [2,2] and [2,3] show the transition from a heat pipe in contact with a single cold reservoir to a heat pipe in contact with cold reservoirs on either side of a hot reservoir. Since all three of these configurations are heated uniformly, they are analytically tractable. Flow partition points, ξ_A , move from left to right in Panels [2,2] and [2,3]. Note that less heat is transferred to the left end of Panel [2,2] than to the right end. For uniform heating and cooling this results in the magnitude of the leftward area average vapor velocity to be less than that of the velocity in rightward direction.

The third row shows dual condenser with various heating and cooling profiles. These profiles are very similar to those depicted in the first row of Figure 3. In all three panels the flow partition points, ξ_A , reside at the axial mid-point of the heat pipe. The profile in Panel [3,1] is clearly analytically tractable for a pressure profile solution, while Panel [3,2] and Panel [3,3] are not.

The fourth row shows configurations with the heat pipe mid-point in contact with a high temperature reservoir. The reservoir transfers heat uniformly to the heat pipe in Panel [4,2]. The heat transfer rate to cold reservoir on the left side of the heat pipe is lower than the heat transfer rate to the cold reservoir on the right side of the heat pipe. Since heat is applied and rejected uniformly past the flow stagnation point, Panel [4,2] is analytically tractable. In a similar vein, the configuration in Panel [4,3] is analytically tractable since the heat transfer rates are uniform with position on either side of the flow partition point.

The fifth row shows further examples of different heating cases. The configuration in Panel [5,1] is analytically tractable since the heat transfer rates are uniform with position on either side of the flow partition point. The configurations in Panels [5,2] and [5,3] are tractable for the flows toward the right but is non-tractable for the flows toward the left since the flow stagnation point is non-coincident with the step changes in linear heat rates versus axial position at the point marked γ in Panel [5,2].

The sixth row shows the transition from a heat pipe in contact with two hot reservoirs to a heat pipe in contact with a single hot and a single cold reservoir. All three of these configurations are tractable. In summary, Panel [3,1], Row 4, Panel [5,1], and Row 6 are analytically tractable in the laminar, incompressible limit by piece-wise application of relations developed by Busse (1967). The remaining panels are not analytically tractable since the applied power per unit length is non-uniform or otherwise makes changes downstream of flow partition locations.

Condensate returns from the region in contact with the cold reservoir to the region in contact with the hot reservoir. The pressure drop of the vapor flowing between the regions in contact with the hot and cold reservoir, $\Delta p_{v,o}$, is assumed to be much greater than the pressure drop of liquid condensate flowing through a compound artery wick, $\Delta p_{l,o} \ll \Delta p_{v,o}$. For viscous vapor flow through a heat pipe in contact with a single hot reservoir and a single cold reservoir, the pressure drop is given by:

$$\Delta p_{l,o} \ll \Delta p_{v,o} \sim \frac{4\mu_v \dot{Q}_T}{\pi \rho_v R_v^4 h_{fg}} (L_e + 2L_a + L_c) \quad (\text{A.7})$$

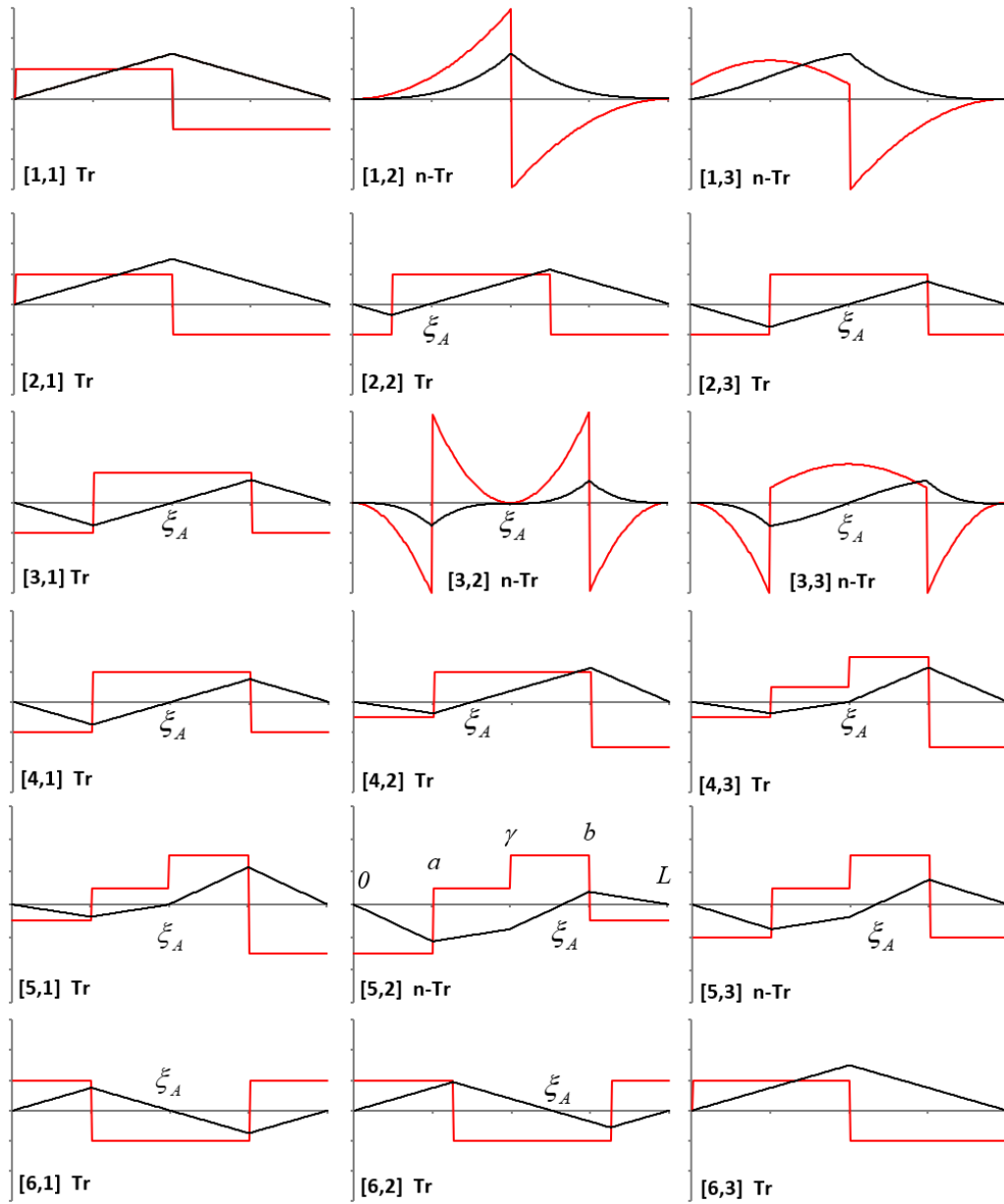


Figure A.3: Qualitative examples of applied power per unit heat pipe length (red curves) compared to area average axial vapor velocity (black curves) versus position along the heat pipe axis for various cases.

Where μ_v is vapor dynamic viscosity, ρ_v , is vapor density, h_{fg} , is latent heat of vaporization of the working fluid all evaluated at the saturation temperature, R_v , is the radius of the vapor space, L_e , is the length of the heat pipe under the hot reservoir (the evaporator), L_a , is the unheated length of the heat pipe, and L_c , is the length of the heat pipe under the cold reservoir (the condenser). Here $L = L_e + L_a + L_c$ and for a uniform linear heat rate $L_{eff} = L_e/2 + L_a + L_c/2$. In the laminar, incompressible limit, the ratio of pressure drops for a heat pipe in contact with two cold reservoirs to the pressure drop for a heat pipe in contact with a single cold reservoir can be approximated as:

$$\frac{\Delta p_{v,i}}{\Delta p_{v,o}} \sim \frac{L_{eff,i}}{L_{eff,o}} \frac{\dot{Q}_i}{\dot{Q}_T} \quad (\text{A.8})$$

Generalizing the coordinate system to allow placement of the flow partition at any position along the heat pipe length yields a non-dimensional variable describing flow partition symmetry:

$$\Xi_A = \frac{1}{2} - \left| \frac{\xi_A}{L} - \frac{1}{2} \right| \quad (\text{A.9})$$

Generalizing the power transfer to allow application of power to either end of a heat pipe having laminar vapor flow, yields a non-dimensional variable describing power distribution symmetry:

$$\Xi_B = 1 - \left| \frac{\dot{Q}_i}{\dot{Q}_T} - \frac{1}{2} \right| \quad (\text{A.10})$$

Consider two analytically tractable heat pipes in contact with hot and cold reservoirs. If both heat pipes are in the laminar and incompressible flow regimes, then the ratio of pressure drop for a heat pipe in contact with two cold reservoirs to a heat pipe in contact with a single cold reservoir is:

$$\frac{\Delta p_v}{\Delta p_{v,o}} \sim [1 - 2(\Xi_A \Xi_B) + (\Xi_A \Xi_B)^2] \quad (\text{A.11})$$

Table 1 summarizes values for this relation as a function of ξ_A and $\dot{Q}_i/\dot{Q}_T$ from Equation 11. The vapor pressure drop with the wet point at the condenser end of the heat pipe with $\max(\dot{Q}_1, \dot{Q}_2)$ may be estimated using:

$$\Delta p_v \sim \frac{4\mu_v \dot{Q}_T}{\pi \rho_v R_v^4 h_{fg}} (L_e + 2L_a + L_c) [1 - 2(\Xi_A \Xi_B) + (\Xi_A \Xi_B)^2] \quad (\text{A.12})$$

Inspecting Table 1, if heat transfers in equal measure to each side of a flow partition located at the axial mid-point of a heat pipe, $\dot{Q}_i/\dot{Q}_T = 0.5$ and $\xi_A/L = 0.5$, then $\Delta p_v/\Delta p_{v,o} \sim 0.25$. This limiting case comports with reason as a centrally located heating zone distributes heat through two vapor cross sections instead of one vapor cross section for a heat pipe heated from one end. Double the vapor cross sections cuts the vapor velocity in half. For laminar vapor flow this action also reduces pressure drop by a factor of two. Central location for heat pipes having equal heated and cooled length reduces the heat pipe effective length in half. These factors together reduce the vapor pressure drop by a factor of four. A flow stagnation point located at the axial mid-point of the heat pipe with equal heating to either end of the heat pipe minimizes pressure drop compared to other configurations shown in Table A.1.

To validate the estimates found in Table 1, vapor and liquid pressure profile solutions were computed in a piece-wise fashion using relations adapted from Busse (1967) as found in the HTPIPE code. These results are shown in Figures 4, 5, and 6.

Figure 4 shows vapor and liquid pressure drops versus position for uniformly applied and rejected linear heat rate. The four panels depict the transition of a heat pipe in contact with a single hot reservoir at its left end to contact with a hot reservoir at its midpoint. Shown on the top row: heat pipe vapor pressure (red solid curves) and heat pipe liquid pressure (black solid curves) versus position along the heat pipe axis. Shown on the bottom row: applied power per unit heat pipe length (red dashed curves) compared to area average axial vapor velocity (black dashed curves) versus position along the heat pipe axis.

Figure 5 shows vapor and liquid pressure drops versus position with asymmetry in rejected power per unit length. The four panels depict the transition of a heat pipe in contact with a single hot reservoir at its left end to contact with hot reservoir at its midpoint. Shown on the top row: heat pipe vapor pressure (red solid curves) and heat pipe liquid pressure (black solid curves) versus position along the heat pipe axis. Shown on the bottom row: applied power per unit heat pipe length (red dashed curves) compared to area average axial vapor velocity (black dashed curves) versus position along the heat pipe axis.

Table A.1: Vapor pressure drop ratios, $\Delta p_{v,o}/\Delta p_v$, from Equation A.11.

	ξ_A/L	0.0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
$\dot{Q}_i/\dot{Q}_T$	$\Xi_B \setminus \Xi_A$	0.0	0.1	0.2	0.3	0.4	0.5	0.4	0.3	0.2	0.1	0.0
0.0	0.5	1.000	1.108	1.235	1.384	1.563	1.778	1.563	1.384	1.235	1.108	1.000
0.1	0.6	1.000	1.132	1.291	1.487	1.731	2.041	1.731	1.487	1.291	1.132	1.000
0.2	0.7	1.000	1.156	1.352	1.602	1.929	2.367	1.929	1.602	1.352	1.156	1.000
0.3	0.8	1.000	1.181	1.417	1.731	2.163	2.778	2.163	1.731	1.417	1.181	1.000
0.4	0.9	1.000	1.208	1.487	1.877	2.441	3.306	2.441	1.877	1.487	1.208	1.000
0.5	1.0	1.000	1.235	1.563	2.041	2.778	4.000	2.778	2.041	1.563	1.235	1.000
0.6	0.9	1.000	1.208	1.487	1.877	2.441	3.306	2.441	1.877	1.487	1.208	1.000
0.7	0.8	1.000	1.181	1.417	1.731	2.163	2.778	2.163	1.731	1.417	1.181	1.000
0.8	0.7	1.000	1.156	1.352	1.602	1.929	2.367	1.929	1.602	1.352	1.156	1.000
0.9	0.6	1.000	1.132	1.291	1.487	1.731	2.041	1.731	1.487	1.291	1.132	1.000
1.0	0.5	1.000	1.108	1.235	1.384	1.563	1.778	1.563	1.384	1.235	1.108	1.000

Figure 6 shows vapor and liquid pressure drops versus position with asymmetries in applied and rejected power per unit length. The four panels depict the transition of a heat pipe in contact with a single hot reservoir at its left end to contact with hot reservoir at its midpoint. Shown on the top row: heat pipe vapor pressure (red solid curves) and heat pipe liquid pressure (black solid curves) versus position along the heat pipe axis. Shown on the bottom row: applied power per unit heat pipe length (red dashed curves) compared to area average axial vapor velocity (black dashed curves) versus position along the heat pipe axis.

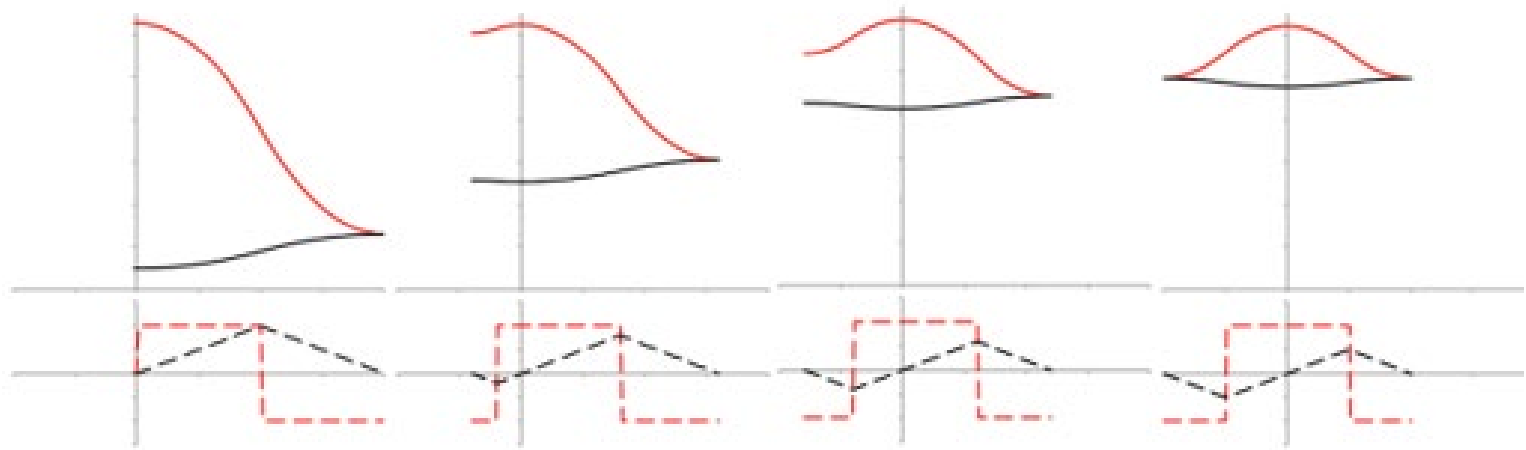


Figure A.4: Transition of heat pipe in contact with a single hot reservoir at its left end to contact with hot reservoir at its midpoint.

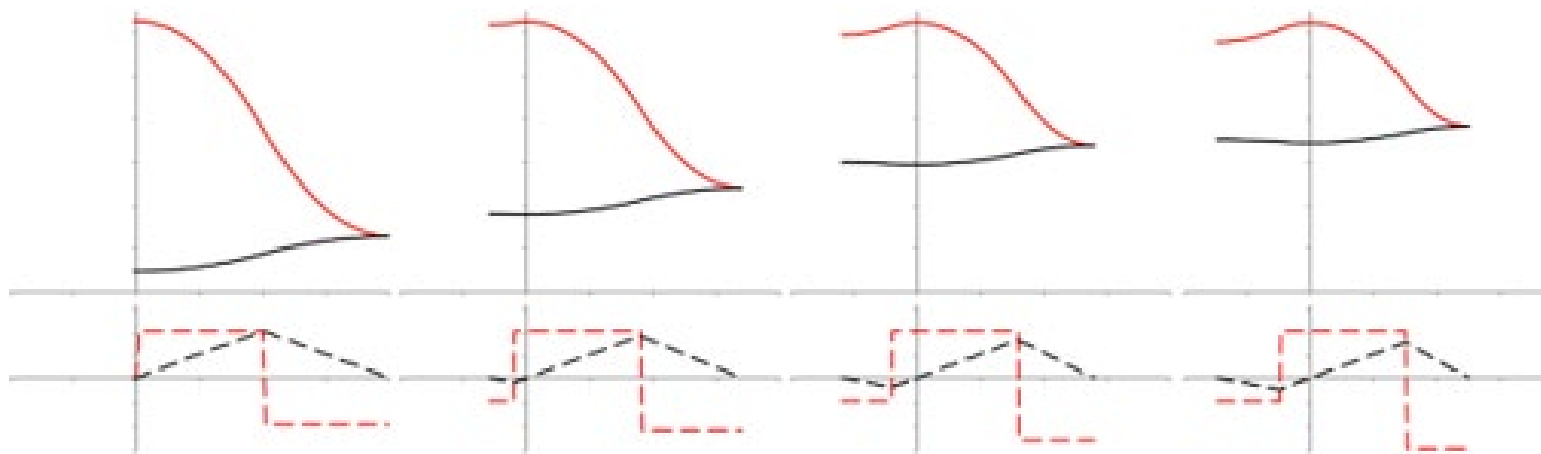


Figure A.5: Transition of heat pipe in contact with a single hot reservoir at its left end to contact with hot reservoir at its midpoint, with asymmetry in rejected power per unit length.

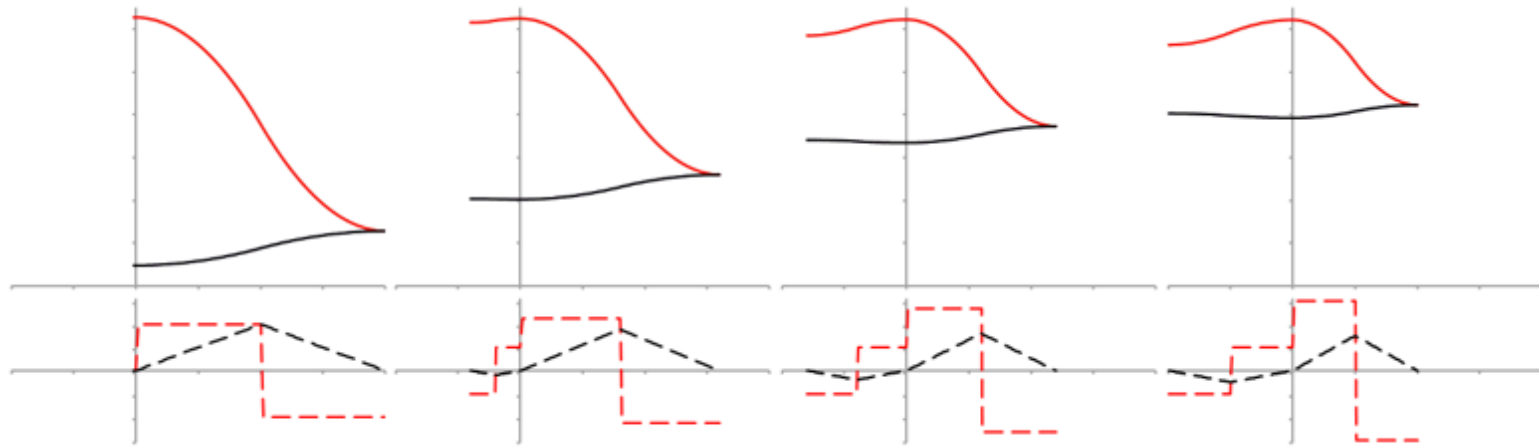


Figure A.6: Transition of heat pipe in contact with a single hot reservoir at its left end to contact with hot reservoir at its midpoint, with asymmetries in applied and rejected power per unit length.

Appendix B: Rotameter Calibration Data

Table B.1: Rotameter# 1 Calibration Data.

Scale (mm)	Time(s)	Mass (g)
0±1	603.4±2	0±0.1
5±1	607.0±2	4.6±0.1
10±1	608.7±2	9.4±0.1
15±1	611.4±2	14.7±0.1
20±1	608.8±2	20.2±0.1
25±1	619.7±2	27.0±0.1
30±1	610.0±2	31.5±0.1
35±1	611.6±2	37.2±0.1
40±1	604.9±2	42.0±0.1
45±1	618.7±2	48.1±0.1
50±1	622.2±2	52.8±0.1
55±1	606.9±2	56.3±0.1
60±1	590.6±2	59.8±0.1

Table B.2: Rotameter# 2 Calibration Data.

Scale (mm)	Time(s)	Mass (g)
0±1	632.9±2	0±0.1
5±1	606.5±2	57.2±0.1
10±1	606.4±2	104.8±0.1
15±1	604.9±2	161.4±0.1
20±1	604.6±2	209.7±0.1
25±1	609.3±2	274.1±0.1
30±1	606.5±2	327.8±0.1
35±1	606.7±2	375.9±0.1
40±1	604.0±2	419.8±0.1
45±1	605.3±2	467±0.1
50±1	600.5±2	514.7±0.1
55±1	634.6±2	590.8±0.1
60±1	637.0±2	618.7±0.1

VITA

Katrina Sweetland was born in Los Alamos and graduated from Los Alamos High School in May 2010. She started her undergraduate education New Mexico Institute of Mining and Technology in August of 2010. Starting in May 2011 she worked as a summer intern at Los Alamos National Laboratory. She graduated with a BS in Mechanical Engineering in May 2015. She then enrolled in the PhD program University of Tennessee in August of 2015 working with Dr. Trevor Moeller on hot reacting flow diagnostics until December 2018 when she returned to Los Alamos National Laboratory as a full-time graduate student.

Since her return to Los Alamos National Laboratory, she has worked under Dr. Robert Reid on heat pipes. In the summer of 2020, she became a full-time technical staff member at Los Alamos National Laboratory. From this point forward she worked on her dissertation while also completing related research and development work.