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Effects of Random Vibration, Wick Structure and Body Forces
on the Capillary Limit of Heat Pipes for Electronics Cooling

By

Chad M. St. Louis

A thesis is submitted to
the Faculty of Graduate Studies and Research
in partial fulfillment of
the requirements of the degree of
Master of Applied Sciences

Ottawa Carleton Institute for Mechanical and Aerospace Engineering
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and Research acceptance of the thesis

Effects of Random Vibration, Wick Structure and Body Forces
on the Capillary Limit of Heat Pipes for Electronics Cooling

Submitted by Chad St.Louis, B.A.Sc.
in partial fulfillment of the requirements for
the degree of Master of Applied Sciences

Chair, Department of Mechanical Engineering

Thesis Advisor

Carleton University

October 7, 2002

To my family
and friends

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List of Abbreviations

CCA	Circuit Card Assembly
COTS	Commercial Off The Shelf
DMV	Dy 4 Military VME
INT	Internal
PWB	Printed Wiring Board
PSD	Power Spectral Density
PVC	Polyvinyl Chloride
THD	Threaded
VME	Versa Modules Eurocard
UUT	Unit Under Test

LIST OF SYMBOLS

A	cross-sectional area, m ²
A_l	liquid flow cross-sectional area, m ²
A_s	shear area, m ²
A_v	vapor core cross-sectional area, m ²
A_w	wick cross-sectional area, m ²
C	compressibility constant
C_l	wetted perimeter of liquid flow, m
C_w	wetted perimeter of wick, m
D_v	dynamic pressure term
d_i	inner diameter of heat pipe, m
d_o	outer diameter of heat pipe, m
d_v	vapor core diameter, m
d_w	wire diameter, m
F_l	frictional pressure term for liquid flow
F_s	shear force, N
F_t	surface force, N
F_v	frictional pressure term for vapor flow
f_l	friction coefficient for liquid flow
f_v	friction coefficient for vapor flow
g	gravitational acceleration, m/s ²
h_c	contact coefficient, W/m ² ·K
h_{conv}	convective heat transfer coefficient, W/m ² ·K
h_l	enthalpy of saturated liquid, J/kg

h_{sink}	heat transfer coefficient at sink, W/m ² ·K
h_{source}	heat transfer coefficient at source, W/m ² ·K
h_v	enthalpy of saturated vapor, J/kg
K	wick permeability, m ²
$K_{1,2}$	proportionality constants
k	thermal conductivity, W/m·K
k_{eff}	effective thermal conductivity of saturated wick, W/m·K
k_l	thermal conductivity of liquid, W/m·K
k_m	thermal conductivity of material, W/m·K
k_p	thermal conductivity of pipe, W/m·K
k_w	thermal conductivity of wick material, W/m·K
L_a	length of adiabatic section, m
L_c	length of condenser section, m
L_e	length of evaporator section, m
L_t	total working length of heat pipe, m
L_{eff}	effective length of heat pipe, m
M	molecular mass, kg/mol
M_v	vapor Mach number
$\dot{m}_l$	mass flow rate of liquid, kg/s
$\dot{m}_v$	mass flow rate of vapor, kg/s
N	mesh number, m ⁻¹
N_g	number of grooves
N_l	liquid figure of merit
P	pressure, Pa
P_c	capillary pressure, Pa
P_{pw}	saturation vapor pressure (at pipe-wick interface temperature), Pa

P_v	vapor pressure, Pa
Q	rate of heat transfer, W
Q_a	rate of heat transfer applied, W
Q_{in}	rate of heat transfer injected into heat pipe, W
Q_s	stray heat transfer rate (rate of heat transfer lost to surroundings), W
$Q_{h,max}$	maximum rate of heat transfer based on boiling limit, W
$Q_{c,max}$	maximum rate of heat transfer based on capillary limit, W
$Q_{e,max}$	maximum rate of heat transfer based on entrainment limit, W
$Q_{s,max}$	maximum rate of heat transfer based on sonic limit, W
$Q_{v,max}$	maximum rate of heat transfer based on viscous limit, W
R	radius, m
$\bar{R}$	universal gas constant, J/mol·K
R_s	effective thermal resistance of system, °C/W
R_v	ideal gas constant for vapor, J/kg·K
Re_l	Reynolds number of liquid flow based on liquid hydraulic diameter
Re_v	Reynolds number of vapor flow based on vapor hydraulic diameter
$R_{E1/A3}$	effective thermal resistance between points E1 and A3, °C/W
$R_{ext,c}$	external thermal resistance at condenser, °C/W
$R_{ext,e}$	external thermal resistance at evaporator, °C/W
R_{ic}	liquid/vapor thermal resistance at condenser, °C/W
R_{ie}	liquid/vapor thermal resistance at evaporator, °C/W
R_{pa}	axial thermal resistance through pipe, °C/W
R_{pc}	radial thermal resistance of pipe wall at condenser, °C/W
R_{pe}	radial thermal resistance of pipe wall at evaporator, °C/W
R_{tot}	total thermal resistance through heat pipe, °C/W
R_{va}	axial thermal resistance through vapor core, °C/W

R_{wu}	axial thermal resistance through wick, °C/W
R_{wc}	radial thermal resistance of wick at condenser, °C/W
R_{we}	radial thermal resistance of wick at evaporator, °C/W
R_1, R_2	principal radii, m
r	radial co-ordinate, m; radius of pore, m; radius of liquid flow passage, m
r_b	bubble radius, m
r_c	capillary radius inside condenser, m
r_e	capillary radius inside evaporator, m
r_i	inner pipe radius, mm
r_{eff}	effective capillary radius, m
$r_{eff,c}$	effective capillary radius in condenser, m
$r_{eff,e}$	effective capillary radius in evaporator, m
$r_{h,l}$	hydraulic radius of liquid flow, m
$r_{h,w}$	hydraulic radius of wick, m
$r_{h,v}$	hydraulic radius of vapor flow, m
r_n	nucleate radius, m
r_s	sphere radius, m
r_v	vapor core radius, m
r_1, r_2	outer and inner annular radius, m
S	crimping factor
s	groove pitch, m
T	temperature, K
T_{A3}	temperature at point A3, K; operating temperature, K
T_{base}	base plate temperature, K
T_{cu}	lower copper block temperature, K
T_v	vapor temperature, K

t_m	thickness of thermally conductive material, m
t_w	pipe wall thickness, m
T_{pw}	temperature at pipe-wick interface, K
$T_{p,c}$	surface temperature of heat pipe at condenser, K
$T_{p,e}$	surface temperature of heat pipe at evaporator, K
T_{wv}	temperature at wick-vapor interface, K
T_∞	bulk temperature, K
t	time, s; thickness, m
t_w	pipe wall thickness, m
V_l	liquid velocity, m/s
V_v	vapor velocity, m/s
V_∞	bulk velocity, m/s
We	Weber number
w	groove width, m; wire spacing, m
w_f	fin width, m
x	axial co-ordinate, m
x_{wp}	distance of wet point from end of condenser, m

Greek Symbols

α_b	body force parameter, N/m ²
α_c	capillary force parameter (capillary pressure), N/m ² (Pa)
α_l	liquid resistance parameter pertaining to fluid flow parameters, N·m/W
$\alpha_{g,l}$	liquid resistance parameter pertaining to geometry, m ⁻³
α_v	vapor resistance parameter pertaining to fluid flow parameters, N·m/W
$\alpha_{g,v}$	vapor resistance parameter pertaining to geometry, m ⁻³
β	half angle of groove, deg; variable coefficient

ΔP	pressure change, Pa
ΔP_b	body force pressure change, Pa
ΔP_c	capillary pressure difference, Pa
$\Delta P_{c,m}$	maximum capillary pressure difference, Pa
$\Delta P_{f,a}$	pressure change in adiabatic section due to friction, Pa
$\Delta P_{f,c}$	pressure change in condenser section due to friction, Pa
$\Delta P_{f,e}$	pressure change in evaporator section due to friction, Pa
$\Delta P_{i,c}$	inertial pressure change in condenser, Pa
$\Delta P_{i,e}$	inertial pressure change in evaporator, Pa
ΔP_l	liquid pressure drop, Pa
$\Delta P_{ph,c}$	pressure change across phase transition in condenser, Pa
$\Delta P_{ph,e}$	pressure change across phase transition in evaporator, Pa
ΔP_v	vapor pressure drop, Pa
$\Delta P_{//}$	axial hydrostatic pressure drop, Pa
$\Delta P_{\perp}$	radial hydrostatic pressure drop, Pa
ΔT	temperature change over a finite distance, K
δ	groove depth, m
$\partial P_v / \partial x$	sum of inertial and viscous pressure gradients in the vapor flow path, Pa
$\partial P_l / \partial x$	sum of inertial and viscous pressure gradients in the liquid flow path, Pa
ε	wick porosity
φ	angle of incline, deg
γ_v	vapor specific heat ratio
λ	latent heat of vaporization, J/kg
μ_l	liquid dynamic viscosity, kg/m·s
μ_v	vapor dynamic viscosity, kg/m·s

ρ_l	liquid density, kg/m ³
ρ_v	vapor density, kg/m ³
σ	surface tension of liquid, N/m
θ	wetting angle, deg
τ_l	frictional stress at liquid-solid interface, N/m ²

Other Symbols

%	percentage of convergence (NOVA software)
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Abstract

In this study, the effect of random vibration on the capillary limit of commercial heat pipes for rugged electronics cooling has been investigated. The heat pipes included in the study utilize sintered powder, axial groove, wrapped screen and fiber bundle wick structures. The four heat pipes have been tested with the gravitational force both normal and parallel to the axis of the heat pipe, with and without vibration. The configuration with axial gravitational force included both orientations, that is the body force both complementing and opposing the capillary force produced by the wick. A bench-top shaker was used to provide vibration in the same plane as the gravitational force at a level of $0.1 \text{ g}^2/\text{Hz}$ across a frequency range of 20 to 2000 Hz ($\sim 14\text{g}$ RMS). The capillary limit has been found to be sensitive to the wick structure, operating temperature (15°C to 75°C) and direction of gravity. These effects have been thoroughly documented in this thesis in the context of mathematical models used to predict heat pipe thermal performance. Negligible effects of vibration have been observed for all configurations tested.

1

Introduction

Based on the trends observed over the last forty years, the processing speed of electronic devices has increased continuously. In conjunction with this increase in performance, the size of chip packages has been decreasing at a greater rate. These improvements in performance and design have resulted in an exponential increase in the amount of heat to be dissipated from electronic devices. As such, their performance reliability and life expectancy cannot be maintained without concurrent improvements in cooling techniques. Thus, thermal management is becoming more important in the electronics industry.

1.1 Commercial Electronics

With the ever-increasing penetration of electronic devices into all aspects of our lives, the global electronics industry has grown at a very high rate within the last decade. Continuous development fueled by consumer demand has turned the microelectronics industry into a multi-billion dollar venture. As this industry is not too far from producing the world's first one-billion-transistor chip, the need for cooling has never been greater.

Today, commercial micro-processing units often use large heat sinks to provide the required amounts of cooling. In some instances, fans are mounted onto the heat sink to provide the desired forced-convection heat transfer from the surface of the heat sink. In instances where space is a constraint, however, the use of heat sink and fan-based systems to dissipate the heat close to the heat source is often not an option. It is these circumstances that have provided the stimulation for evolutionary advances in cooling technology. Unfortunately, evolutionary improvements will not be able to sustain tomorrow's high-heat-flux electronic systems. The

solution therefore lies in the discovery of revolutionary concepts that will not only cool the high-heat-flux devices anticipated in the near future, but will also accommodate leaps in cooling requirements that would likely be associated with revolutionary advances in the speed and packaging of electronic devices.

1.2 Military Electronics

In rugged military electronic devices, thermal management has already reached the epitome of cooling design. Consequently, engineers find themselves resorting to costly materials and processes in order to adequately cool electronic devices in these environments. Harsh conditions such as salt, humidity, and vibration typical of military applications may in many cases exclude the option of using conventional cooling technology. It therefore appears that electronic devices for military applications are at or near the top of the list of applications that are in immediate need of revolutionary advances in cooling technology.

1.2.1 Dy 4 Systems Inc.

Dy 4 Systems Inc., the current employer of the author, is a producer of complex Commercial Off the Shelf (COTS) VMEbus hardware for harsh environments of defense and aerospace. In response to the aforementioned shortcomings in cooling technology, two years ago Dy 4 Systems initiated a research program aimed at the evaluation of heat-pipe technology for military-grade circuit cards, and integration of this technology into existing product configurations. This thesis contains the results of the first phase of this research program.

Although numerous studies on the thermal performance of heat pipes exist in the published literature, the effects of vibration on this performance is relatively unknown. A number of studies have attempted to quantify the effects of vibration on heat pipes in recent years. However, these studies were typically limited to one type of heat pipe, oriented in one direction

relative to gravity, with vibration applied either perpendicular or parallel to the axis of the heat pipe. Furthermore, the conclusions drawn from the interpretation of the results contradict each other in several instances. In light of this incomplete understanding of the effects of vibration on heat pipe performance, a systematic study on the matter was deemed to be the right starting point for the heat pipe research project at Dy 4 Systems.

1.3 Objective

The first objective of this study is to determine the effects of random vibration on the thermal performance of heat pipes designed for electronics cooling. Secondly, the study is to establish the relative performances of several wick structures under vibration, and in the presence of body forces. Thirdly, the study aims to evaluate existing mathematical models for predicting the capillary limit of heat pipes with several variations in the wick structure, with and without body forces.

1.4 Approach

The approach to this study is to build a test setup that can be used to accurately evaluate commercial heat pipes under well-controlled conditions. The setup is to be designed to allow quick turn around in the testing of heat pipes with different wick structures. The setup is to also allow measurement of thermal performance under both vibrating and non-vibrating conditions. Furthermore, the setup is to accommodate different orientations of the heat pipe relative to gravity. On the basis of a systematic test matrix with the wick structure, vibration and orientation of gravity force as the primary variables, the optimum heat pipe configuration for operation in harsh environments is to be established, and its operating envelope is to be determined.

2

Background

The growing demand and ability to process more bits of data at even faster rates coupled with the development of smaller chip packages, pose tough thermal challenges for engineers. As previously stated in Chapter 1, today's techniques in cooling will soon be inadequate to cool the next-generation high-heat-flux electronic systems. In light of the shortcomings, this chapter examines heat pipes as a possible cooling technology suitable for COTS rugged electronic systems.

2.1 Heat Transfer in Electronics Cooling

Conduction is the mode of heat transfer within a solid. When a temperature gradient is present within a body, thermal energy is transferred from the region of higher-temperature to the region of lower-temperature. Jean Fourier stated that the total rate of heat flow is proportional to the area normal to the heat flow path, the temperature gradient along that path, and the thermal conductivity of the medium (Holman, 1997):

$$\frac{dQ}{dt} = -kA \frac{dT}{dx} \quad (2-1)$$

Thermal engineers in the electronics industry use Fourier's law, Equation (2-1), to model the heat flow within electronic components that dissipate heat. Heat is transferred by conduction from the circuitry of the chip through its molding and substrate towards the outer surfaces of the package. Upon reaching the outer surfaces of the package, the heat is transferred to the surroundings through conduction (e.g. through the package leads towards the printed circuit

board on which the package is mounted) and by convection if the surrounding medium is fluid. One must also consider the radiation heat transfer mode in some instances.

Heat pipes, thermoelectrics, and enhanced thermally conductive printed circuit boards are some of the potential technologies that can be used to improve heat transfer in electronic systems. If used properly, heat pipes can be several hundred to a thousand times more effective in transferring heat than any solid conductor known to date.

2.2 Heat Pipes

This passive two-phase heat transfer device was first invented by Gaugler in 1944. However, only when Grover first coined the term “heat pipe” and publicized the operating principles in 1964 did he demonstrate its effectiveness as a high-performance heat transfer device (Chi, 1976). At the beginning of their development, heat pipes were recognized as a costly alternative to extract heat from systems with large rates of heat dissipation, and were primarily used in applications involving extreme thermal conditions. Today heat pipes are used in many applications ranging from aerospace to laptop personal computers and have been standardized in both size and shape.

A heat pipe is essentially a two-phase heat transporter using latent heat of vaporization to facilitate transfer of heat from one of its ends to the other. It consists of an evaporator, a condenser and a section between these two that is normally considered adiabatic as shown in Figure 2.1.

Figure 2.1 : Example of a conventional heat pipe (Chi, 1976)

2.2.1 Operating Principle

A heat pipe operates by the process of evaporation and condensation of a working fluid. The working fluid is hermetically sealed within a material container paired with a wick structure. Within the evaporator, the region of heat input, the working fluid evaporates. Because of an increase in specific volume within the vapor core of the evaporator, vapor is accelerated along the center of the heat pipe to the opposite end where it then condenses releasing heat. During condensation, the flooding inside the condenser contributes to the capillary forces produced in the direction towards the evaporator. The amount of ‘pull’ in the evaporator is affected by the wick structure. As such, without the aid of body forces, it is the wick structure that enables the condenser to supply the evaporator with enough liquid for vaporization. Without enough liquid, the heat pipe cannot operate properly at steady state and “drying” of the wick occurs. This defines thermal failure of the heat pipe.

It should be noted that a heat pipe is a passive heat-transport device. When a temperature gradient is applied between the evaporator and condenser sections of the heat pipe, the vapor pressure begins to increase in the evaporator. This is accompanied by capillary forces developing in the wick. With the pressure gradient in the vapor core transferring vapor to the condenser and the capillary forces in the wick transferring the liquid to the evaporator, the heat pipe eventually reaches a steady state of operation.

2.2.2 Surface Tension

An important concept in understanding the functionality of a heat pipe is surface tension, for this phenomenon relates to the development of a capillary pressure difference between the condenser and evaporator ends of a heat pipe. The properties of fluid elements located at boundaries, where the fluid is in contact with dissimilar fluid or solid media, can be different from those of interior elements that are surrounded by contiguous fluid elements. For instance, in the interior of a water droplet, each water molecule is subject to forces imposed by neighboring molecules in such a manner to yield forces that are distributed equally in all directions. For molecules that are at the surface of the water droplet, however, only half of the space surrounding such molecules is occupied with water molecules. As a result, a different distribution of the intermolecular forces is needed to maintain the position of the molecules on the surface. While the liquid molecules in the interior of the droplet experience a repulsive force in average, the molecules on the surface experience attractive forces from the neighboring surface molecules (Figure 2.2).

This attractive (tensile) force per unit length is referred to as the surface tension, and is a property of the fluid in question, the properties of the adjacent dissimilar medium, and any impurities in the liquid congregate that may exist on the surface (e.g. surfactants). The tension forces prevailing amongst the surface molecules of the water droplet yield a higher pressure within the droplet compared to the pressure of the surrounding dissimilar fluid. This is analogous

to a balloon where the gas inside the balloon is at a higher pressure than the surrounding air, brought about by the tensile forces amongst the polymer molecules of the balloon.

If one considers the equilibrium force balance for one-half of a water droplet of radius r , the above-noted excess pressure inside the water droplet acting upon the cross sectional area, πr^2 , of the droplet is balanced by the surface tension acting on the circumference, $2\pi r$, of this cross section. Based on this argument of force balance, one notes that the pressure difference across the interface (i.e. the surface of the droplet in this instance) would tend towards zero as the radius of curvature of the surface approaches infinity, regardless of the magnitude of the surface tension. In other words, the magnitude of the pressure difference that is established across an interface is closely related to the radius of curvature of this interface.

It is common practice to consider the surface tension phenomenon from an energy perspective. The work that the aforementioned pressure difference across the surface of the water droplet would do in developing the droplet to its final radius (with the droplet being fed with water through a capillary tube, for example), is stored in the droplet as surface energy. It can be readily shown that this surface energy per unit area is equal to the surface tension.

Figure 2.2 : Schematic of tensile forces acting on a surface molecule

2.2.3 Wick Structures

Wick structures or capillary grooves serve two purposes in heat pipes: 1) to enhance capillary or pumping forces and 2) to evenly distribute liquid over the entire evaporator surface. Wicks are generally made of a porous or permeable material commonly identified by a wire screen, felt, or metallurgical powder. Grooves are sometimes used as a simple cost-effective alternative. However, these wick structures normally require the assistance of the gravitational force to transfer sufficient liquid from the condenser to the evaporator. Therefore, the orientation of the heat pipe should be such that the evaporator is below the condenser. Figure 2.3 shows the change in heat transfer capacity of a heat pipe depending on the direction of the gravitational force. A few wick structures, such as sintered powder, generate enough capillary forces to overcome gravity. These heat pipes are thus capable of operating at any orientation with respect to the gravitational force.

Figure 2.3 : Sensitivity of heat pipe performance to orientation relative to the gravitational force (Sergent and Krum, 1998)

Wick structures can be categorized into two classes; simple homogeneous and composite. Homogenous heat pipes contain wick structures that are made of a single material. Composite heat pipes, on the other hand, contain wick structures that are made of two or more materials. Examples of both types of wick structures are illustrated in Figure 2.4 together with several more advanced designs.

Figure 2.4 : Common wick structures (Bar-Cohen *et al.*, 1988)

One common wick structure is the wrapped screen wick. The porosity and permeability of this wick is controlled through the diameter of the wires in the screen, their spacing, and the number of layers wrapped into a circular shape inside the heat pipe.

Sintered metal (a.k.a. sintered powder) wick structures are made of metallic powders that are fused together. Smaller pores in this type of wick provide more homogenous heat-flow and fluid-flow paths, as well as larger capillary forces than a wrapped screen wick.

Lastly, axially grooved wicks are simple cost effective alternatives to wrapped screen and sintered powder wicks. Although the grooved design provides less resistance for radial heat flow into and out of the heat pipe, the pore size (i.e. groove spacing), is limited by the manufacturing techniques used today. As a result, the performance of the heat pipe is more sensitive to gravity or other body forces than the wrapped screen and sintered powder wicks.

Composite wick structures such as slab, pedestal artery, spiral artery, and tunnel artery use multiple channels and/or materials to enhance the capillary forces. One advantage of using artery wicks is the inert capability of reducing the resistance for liquid flow. Numerous variations exist within the “artery wick” category. A slab heat pipe operates with a slab wick inserted within an internally threaded container. A fine mesh at the surface of the slab generates the capillary forces necessary for proper operation. The threads along the circumference are used to evenly distribute the liquid and direct it to the slab of the wick structure. At the condenser, the threads provide efficient radial heat transfer (Chi, 1976).

Tunnel heat pipes operate using the “Clausius-Clapeyron effect” to fill the tunnel with liquid fluid. Instead of using threads within the container to direct liquid flow into the artery of the heat pipe, screen webs and wraps are used as capillary passages. Under uniform temperature, the tunnel structure contains saturated vapor that corresponds to the temperature of the artery wall. When heat is applied to the heat pipe, the vapor pressure outside the artery wall exceeds the tunnel vapor pressure. This difference in pressure causes liquid to flow into the tunnel (Chi, 1976).

2.2.4 Working Fluid

Selecting a suitable working fluid is an important factor in the design of heat pipes. The container and wick material must compliment the working fluid. Any corrosion deposits or other non-pertinent residue can hinder the heat pipe's overall performance. These particles can collect in the condenser and reduce the effective size of the condensation chamber. A list of materials and suitable working fluids is given in Table 2.1. Copper and water is a popular combination used in heat pipes for electronic systems cooling.

Table 2.1 : Working fluid, wick and container compatibility data (Dunn and Reay, 1982)

Material	Water	Acetone	Ammonia	Methanol
Copper	R	R	NU	R
Aluminum	GNC	R	R	NR
Stainless	GNT	PC	R	GNT
Nickel	PC	PC	R	R
Refrasil	R	R	R	R
R	Recommended by literature			
PC	Probably compatible			
NR	Not recommended			
NU	Not used			
UK	Unknown			
GNC	Generation of gas at all temperatures			
GNT	Generation of gas at elevated temperatures when oxide is present			

Another element to consider when selecting a working fluid is the operating temperature range. For example, applications well below 0°C can cause water to freeze inside the wick structure even at low charge pressures. Furthermore, water's unique property of expansion during solidification can damage the wick or the container altogether. The working fluid must therefore

be adequately selected for proper operation of the heat pipe throughout the application temperature range.

Working fluids can be categorized into one of three temperature ranges, Cryogenic, Moderate, and Liquid Metal. Cryogenic fluids are defined as those having boiling points below 120K when hermetically sealed. Examples of these fluids include Oxygen, Nitrogen, and Hydrogen.

Moderate-temperature fluids cover the range between 120K and 630K. These fluids include Water, Methanol, and Ammonia. For applications with higher temperatures, liquid metals with boiling points above 630K can be used. Silver, Lithium, and Sodium are some of the common fluids chosen for this purpose. Figure 2.5 shows a logarithmic temperature scale of typical working fluids.

Figure 2.5 : Logarithmic temperature scale and compatible working fluids (Chi, 1976)

Working fluids also differ in performance. The forces generated inside the wick are dependent on the surface tension of the working fluid, the liquid viscosity, and the liquid density. Latent heat of vaporization is directly related to the efficiency of conversion from liquid to vapor state, and the temperature drop across the wick structure is dependent on the thermal conductivity of the working fluid.

Chi (1976) combines these properties into a parameter referred to as the *Liquid Figure of Merit*. This parameter may be used to evaluate working fluids and compare them to each other. The liquid figure of merit, N_l , is expressed as,

$$N_l = \frac{\rho_l \sigma \lambda}{\mu_l} \quad (2-2)$$

where,

ρ_l liquid density of working fluid

σ surface tension of liquid

λ latent heat of vaporization

μ_l liquid dynamic viscosity

Thus increased performance requires a working fluid with a high latent heat of vaporization, high surface tension, high liquid density, and low liquid viscosity. Comparing working fluids to each other over a range of temperatures (i.e. cryogenic to liquid metal), water appears to be a robust fluid with a high liquid figure of merit in the moderate temperature range. Sodium (Na) and Ammonia (NH_3) appear to be the preferred fluids in the liquid metal and cryogenic temperature ranges, respectively. Methanol is a popular alternative to water for sub-zero temperatures. Figure 2.6 plots the liquid figure of merit for these working fluids.

Figure 2.6 : Figure of merit for some working fluids (Dunn and Reay, 1982)

2.2.5 Failure Mechanisms

Failure mechanisms of the heat pipe have been reduced to five distinct anomalies leading to failure or deficiency in the transfer of heat. The five mechanisms are associated with capillary, sonic, viscous, entrainment, and boiling limits. The value of these limits depend mainly on the properties of the working fluid, wick structure, geometry, and conditions of the heat pipe such as orientation with respect to the gravitational force and rate of cooling at the condenser. It is important to note that the equations presented in the following subsections for these limits are one-dimensional approximations applicable to conventional heat pipes.

2.2.5.1 Capillary Limitation

Capillary limitation is the condition wherein just enough capillary force is produced to overcome the “resisting” forces prevailing in the system. For a properly operating heat pipe, the capillary pumping forces must be equal to or greater than the sum of the pressure drops occurring

throughout the liquid and vapor flow paths (Peterson, 1994). This statement may be expressed mathematically through the following,

$$\Delta P_{c,m} \geq \int_{L_{eff}} \frac{\partial P_v}{\partial x} dx + \int_{L_{eff}} \frac{\partial P_l}{\partial x} dx + \Delta P_{ph,e} + \Delta P_{ph,c} + \Delta P_{\perp} + \Delta P_{\parallel} \quad (2-3)$$

where,

- $\Delta P_{c,m}$ maximum capillary pressure difference generated inside the wick structure between the evaporator and condenser
- $\partial P_v / \partial x$ sum of inertial and viscous pressure gradients in the vapor flow path
- $\partial P_l / \partial x$ sum of inertial and viscous pressure gradients in the liquid flow path
- $\Delta P_{ph,e}$ pressure drop across phase transition in evaporator
- $\Delta P_{ph,c}$ pressure drop across phase transition in condenser
- $\Delta P_{\perp}$ radial hydrostatic pressure drop (component of body forces acting along the radial length of the wick structure)
- $\Delta P_{\parallel}$ axial hydrostatic pressure drop (component of body forces acting along the length of the heat pipe)
- L_{eff} effective length of the heat pipe along which inertial and viscous pressure gradients exist

The capillary pressure is the difference in pressure between the two sides of a liquid-vapor interface. As stated previously, the operation of a heat pipe relies on capillary force to drive liquid from the condenser to the evaporator of the heat pipe. In a properly operating heat pipe, this force is able to overcome the resisting forces within the vapor and liquid flow. The capillary force is governed by the variation in the meniscus radius along the length of the heat pipe as shown in Figure 2.7.

(a)

(b)

Figure 2.7 : Variation of meniscus curvature (Sergent and Krum, 1998; Peterson, 1994)

Chi's work identifies the points based on which the difference in capillary pressure is to be evaluated inside a wick structure. The first point is named the "dry" point, the point at which the meniscus has a minimum radius of curvature due to vaporization. The point farthest from the condenser in the evaporator is the "dry" point of the heat pipe. Similarly, the "wet" point is the point at which flooding has occurred and can take place at any point within the condenser. These two points alone govern the effective length of the heat pipe, the length over which inertial and viscous forces develop to act against the capillary force.

In regions where flooding occurs, the wick is considered completely saturated with liquid. This point is dependent on the cooling rate applied in the condenser. For high cooling rates, the wet

point moves towards the evaporator, and the effective length of the heat pipe is shortened. For lower cooling rates, the wet point is generally near the end of the condenser, farthest from the evaporator.

Returning to Equation (2-3), the equation states that for proper operation of a heat pipe, the maximum capillary pressure must be equal to or greater than the sum of inertial and viscous pressure drops in the liquid and vapor flow paths, the pressure drop across the phase transition in the evaporator and condenser, and the sum of the components of body forces acting along the axial and radial lengths of the wick structure.

Figure 2.8 illustrates a typical pressure distribution encountered in a heat pipe. Three trends are sketched on the plot, one each for the vapor phase, the liquid phase with body forces, and the liquid phase without body forces. Factors affecting the vapor phase profile in the evaporator are the pressure drop across the liquid-vapor phase transition, $\Delta P''_{ph,e}$, the frictional pressure drop, $\Delta P''_{f,e}$, and the inertial pressure gradient, $\Delta P''_{i,e}$, which is a pressure drop related to the acceleration of the vapor flow. Through the adiabatic section of the heat pipe, the drop in pressure in the vapor core, $\Delta P''_{f,a}$, is due to friction. Pressure variation in the condenser is affected by the same mechanisms that are encountered in the evaporator. However, the inertial effects yield some pressure recovery in the condenser since the vapor flow decelerates in this instance (Peterson, 1994).

The pressure distributions experienced in the liquid phase are shown with two curves. The higher curve on the graph includes the effects of uniform and steady body forces such as gravitational acceleration exerted on the liquid. The lower curve ignores these effects. The primary mechanism affecting the pressure distribution in the liquid phase inside the condenser is the frictional pressure drop, $\Delta P^l_{f,c}$. The inertial pressure gradients in the liquid phase are typically ignored because of relatively low fluid velocity. In the adiabatic region of the liquid phase curve,

Figure 2.8 also illustrates the capillary pressure, ΔP_c , that is required to be equal to or greater than the sum of the aforementioned pressure drops experienced in the system. In order to predict the capillary operating limit of a heat pipe, each of the pressure gradients discussed above are to be estimated as described in the following subsections.

2.2.5.1.1 Capillary Pressure Difference - ΔP_c

Pressure drops associated with vapor and liquid flow as well as the capillary pressure increase with heat load. However, there is a maximum possible capillary pressure at any point along the heat pipe (Chi, 1976). This limit identifies the maximum heat transport capability of the heat pipe. The capillary pressure, P_c , can be found from the Laplace-Young equation:

$$P_c = \sigma \left(\frac{1}{R_1} + \frac{1}{R_2} \right) \quad (2-4)$$

where R_1 and R_2 are the principal radii of curvature shown in Figure 2.9a. For the maximum capillary pressure, Equation (2-4), is reduced to a single radius of curvature, r_{eff} (i.e. $R_1 = R_2 = r_{eff}$). The maximum capillary pressure drop, $\Delta P_{c,m}$, between the “dry” and “wet” points of a heat pipe is then expressed as,

$$\Delta P_{c,m} = 2\sigma \left(\frac{1}{r_{eff,e}} - \frac{1}{r_{eff,c}} \right) \quad (2-5)$$

where $r_{eff,e}$ and $r_{eff,c}$ denote the effective capillary radius in the evaporator and condenser, respectively.

At steady state, the flooding of the condenser causes the capillary radius, $r_{eff,c}$, to approach infinity. At this limit, the maximum capillary pressure drop can be reduced to a function of the capillary radius in the evaporator alone,

$$\Delta P_{c,m} = \frac{2\sigma}{r_{eff,e}} \quad (2-6)$$

The effective capillary radius in the evaporator represents the maximum possible value of $(1/R_1 + 1/R_2)$ in the Laplace-Young equation. The value of $r_{eff,e}$, hereafter r_{eff} , is dependent

upon the wick structure. Chi (1976) describes the effective capillary radius for several simple geometries including axially grooved heat pipes (rectangular and triangular cross-section), wrapped screen, and packed spheres. Although the r_{eff} value for simple geometries can be determined theoretically, for complex geometries, such as sintered powder that is fused to produce a wick, r_{eff} is obtained experimentally.

The following derivation for the effective capillary radius of a cylindrical pore is presented in Chi's work. In his first example, the principal radii R_1 and R_2 , as shown in Figure 2.9a, are equal to each other and redefined as R . In a cylindrical pore, the value of R can be determined by Equation (2-7).

$$R = \frac{r}{\cos \theta} \quad (2-7)$$

where,

r radius of the cylindrical pore

θ wetting angle

The meniscus geometry for a cylindrical pore is shown in Figure 2.9b.

Figure 2.9 : Meniscus geometry (Chi, 1976)

Substituting Equation (2-7) into the Laplace-Young equation gives the following expression for capillary pressure in a cylindrical pore,

$$P_c = \frac{2\sigma \cos \theta}{r} \quad (2-8)$$

According to this expression, the maximum capillary pressure occurs when the wetting angle is zero. Thus,

$$P_c = \frac{2\sigma}{r} \quad (2-9)$$

Comparing Equation (2-9) to the maximum capillary pressure as given by Equation (2-6) with $R_1 = R_2 = r_{eff}$, the effective capillary radius is noted to be equal to the radius of the pores,

$$r_{eff} = r \quad (2-10)$$

Similarly, the effective capillary radius for wicks with rectangular grooves, triangular grooves, and parallel wires can be derived theoretically. Table 2.2 summarizes Chi's findings of the effective capillary radius for simple geometries.

Table 2.2 : Effective capillary radii for several wick structures (Chi, 1976)

Wick Structures	Expressions for r_{eff}	
Cylindrical pores	$r_{eff} = r$	r = radius of pores
Rectangular groove	$r_{eff} = w$	w = groove width
Triangular groove	$r_{eff} = \frac{w}{\cos \beta}$	w = groove width β = half angle
Parallel wires	$r_{eff} = w$	w = wire spacing
Wire screens	$r_{eff} = \frac{w + d_w}{2}$	w = wire spacing d_w = wire diameter
Pack spheres	$r_{eff} = 0.41 r_s$	r_s = sphere radius

In the case of sintered powder, the effective capillary radius is dependent upon the size of the sintered particles, the mode of packing, and the degree of fusing between particles. Chi (1976) makes a conservative estimate for the effective radius based on his reference that uses cubic close-packed model of spheres,

$$r_{eff} = 0.41r_s \quad (2-11)$$

where r_s is the radius of the sintered particles.

Values and expressions of r_{eff} described so far assume that the working fluid is free of impurities. With impurities, a zero wetting angle may not be achieved. The effective capillary radius is then increased and the maximum capillary pressure is reduced.

2.2.5.1.2 Vapor Pressure Drop - ΔP_v

Calculating the vapor pressure distribution throughout a heat pipe is a challenging task when evaluating the capillary limit of a heat pipe. Velocity profiles and fluid properties vary depending on the extent of compressibility effects and whether the flow is laminar or turbulent. Although parabolic velocity profiles have typically been used for fully developed laminar flow and the one-seventh power law has been used for the turbulent regime, more elaborate models that are based on more complex approximations of the radial profiles of axial variation of velocity have also been considered (e.g. Busse, 1967). Acceleration of the vapor in the evaporator also contributes to the vapor pressure drop. For high vapor velocities, this may become the dominant factor in calculating the total pressure drop, accounting for as much as 80% of the total (Ivonavski *et al.*, 1982).

A one-dimensional approximation for ΔP_v was proposed by Chi (1976). The analysis begins with the following expressions.

$$\frac{dP_v}{dx} = -F_v Q - D_v \frac{dQ^2}{dx} \quad (2-12)$$

where,

Q rate of heat transfer

F_v, D_v frictional and dynamic-pressure terms given by,

$$F_v = \frac{(f_v \text{Re}_v) \mu_v}{2r_{h,v}^2 A_v \rho_v \lambda} \quad (2-13)$$

$$D_v = \frac{\beta}{A_v^2 \rho_v \lambda^2} \quad (2-14)$$

with,

f_v friction coefficient for vapor flow

μ_v vapor dynamic viscosity

$r_{h,v}$ hydraulic vapor core radius

A_v vapor core cross-sectional area

ρ_v vapor density

λ latent heat of vaporization

and

$$\beta = \frac{\rho_v^2 A_v}{\dot{m}_v^2} \int_{A_v} V_v^2 dA \quad (2-15)$$

For a conventional heat pipe, the dynamic pressure term, D_v , is relatively small, hence the 2nd term on the right hand side of Equation (2-12) can be ignored. By integrating the remainder of this equation over the previously defined effective length of the heat pipe, one obtains:

$$\Delta P_v = \left[\frac{(f_v \text{Re}_v) \mu_v}{2r_{h,v}^2 A_v \rho_v \lambda} \right] L_{eff} Q \quad (2-16)$$

The Reynolds number for vapor flow is defined as,

$$\text{Re}_v = \frac{2r_{h,v} Q}{A_v \mu_v \lambda} \quad (2-17)$$

which determines the fluid flow regime (laminar vs. turbulent).

The vapor Mach number, M_v , is used to characterize the compressibility effects in the vapor flow and is given by,

$$M_v = \frac{Q}{A_v \rho_v \lambda \sqrt{\gamma_v R_v T_v}} \quad (2-18)$$

where,

γ_v vapor specific heat ratio

R_v ideal gas constant of the vapor

The flow is considered laminar and incompressible when the values of Re_v and M_v are less than 2300 and 0.2, respectively. The value of $(f_v \text{Re}_v)$ can then be determined using Figure 2.10 for rectangular and annular cross-sectional geometries. For circular passages, the value of $(f_v \text{Re}_v)$ is equal to 16.

Figure 2.10 : Frictional coefficients for laminar flow (Chi, 1976)

For cases where Re_v is less than 2300 and M_v is greater than 0.2, the flow remains laminar, but the effect of compressibility on the value of $(f_v Re_v)$ must too be considered. The following equation relates the vapor friction coefficient for compressible laminar flow to that for incompressible laminar flow at the same Reynolds number.

$$(f_v Re_v) = 16 \left(1 + \frac{\gamma_v - 1}{2} M_v^2 \right)^{-1/2} \quad (2-19)$$

For flows where the Reynolds number exceeds 2300, Chi (1976) uses the one-seventh power law for the vapor velocity profile to estimate the value of f_v . The following expression can be used to estimate the vapor friction coefficient for incompressible turbulent flow,

$$f_v = \frac{0.038}{Re_v^{0.25}} \quad (2-20)$$

Concluding this discussion with the case where the flow is turbulent ($Re_v > 2300$) and compressible ($M_v > 0.2$), the vapor friction coefficient is estimated using,

$$f_v = \frac{0.038}{Re_v^{0.25}} \left(1 + \frac{\gamma_v - 1}{2} M_v^2 \right)^{-1/2} \quad (2-21)$$

Equation (2-16) assumes that the rate of phase change in the evaporator and condenser is uniform. Thus, the vapor mass flow rate varies linearly with axial distance in the evaporator and the condenser. As such, for typical heat pipes with uniform heat input and rejection over the surfaces of the evaporator and condenser, the effective length of the heat pipe may be expressed as,

$$L_{eff} = \frac{L_e}{2} + L_a + \frac{L_c}{2} \quad (2-22)$$

where,

- L_e length of evaporator (heat input)
- L_a length of adiabatic section
- L_c length of condenser (heat rejection)

In cases with higher cooling rates, flooding occurs prior to the end of the condenser. As previously discussed in Section 2.2.5.1, the wet point within the condenser is closer to the evaporator in these instances. In the absence of fluid motion within the flooded area of the condenser, the effective length is based on the location of the wet point. Equation (2-22) can thus be generalized as,

$$L_{eff} = \frac{L_e}{2} + L_a + \frac{L_c - x_{wp}}{2} \quad 0 \leq x_{wp} \leq L_c \quad (2-23)$$

where x_{wp} is the distance of the wet point from the end of the condenser.

2.2.5.1.3 Liquid Pressure Drop - ΔP_l

The primary factor responsible for the axial pressure gradient in the liquid flow is the frictional force,

$$\frac{dP_l}{dx} = -\frac{2\tau_l}{r_{h,l}} \quad (2-24)$$

where,

τ_l frictional stress along wetted perimeter of liquid flow

$r_{h,l}$ liquid hydraulic radius

The liquid hydraulic radius, $r_{h,l}$, is given by the following expression,

$$r_{h,l} = \frac{2A_l}{C_l} \quad (2-25)$$

where,

A_l liquid flow cross-sectional area

C_l wetted perimeter of liquid flow

The wall shear stress is related to the friction coefficient for liquid flow, f_l , through,

$$f_l = \frac{2\tau_l}{\rho_l V_l^2} \quad (2-26)$$

where ρ_l is the liquid density and V_l is the liquid velocity calculated from,

$$V_l = \frac{Q}{\varepsilon A_w \rho_l \lambda} \quad (2-27)$$

where,

A_w wick cross-sectional area

ε wick porosity

Substituting τ_l in terms of f_l into Equation (2-24), one obtains,

$$\frac{dP_l}{dx} = -\frac{2(f_l \text{Re}_l)\mu_l}{2\epsilon A_w r_{h,l}^2 \lambda \rho_l} Q \quad (2-28)$$

where,

$$\text{Re}_l = \frac{2r_{h,l}\rho_l V_l}{\mu_l} \quad (2-29)$$

Chi (1976) reduces Equation (2-28) to a form similar to the one used in calculating the vapor pressure drop,

$$\frac{dP_l}{dx} = -F_l Q \quad (2-30)$$

where F_l is a frictional term for liquid flow defined as,

$$F_l = \frac{\mu_l}{KA_w \lambda \rho_l} \quad (2-31)$$

in which K is the wick permeability that is related to porosity, ϵ , through,

$$K = \frac{2\epsilon r_{h,l}^2}{f_l \text{Re}_l} \quad (2-32)$$

The permeability represents a property of the wick structure that is independent of the flow path and is defined as “a measure of the wick resistance to axial liquid flow” (Faghri, 1995). The porosity of a wick, ϵ , is defined as the ratio of pore volume, V_p , to total volume of the wick structure, V_t . Expressions for the permeability and porosity of several common wick structures are listed in Table 2.3.

Table 2.3 : Wick permeability (Chi, 1976)

Wick Structures	<i>K</i> , Wick Permeability	
Circular cylinder (artery or tunnel wick)	$\frac{r^2}{8}$	r = radius of liquid flow passage
Open rectangular grooves	$\frac{2\varepsilon(r_{h,l})^2}{f_l \text{Re}_l}$	$\varepsilon = w/s$ w = groove width s = groove pitch $r_{h,l} = 2w\delta/(w + 2\delta)$ δ = groove depth
Circular annular wick	$\frac{2(r_{h,l})^2}{f_l \text{Re}_l}$	$r_{h,l} = r_1 - r_2$ $\varepsilon = 1$
Wrapped screen wick	$\frac{d_w^2 \varepsilon^3}{122(1 - \varepsilon)^2}$	d_w = wire diameter $\varepsilon = 1 - (1.05\lambda N d_w/4)$ N = mesh number
Pack spheres	$\frac{r_s^2 \varepsilon^3}{37.5(1 - \varepsilon)^2}$	r_s = sphere radius ε = porosity (dependent on packing mode)

For the wrapped screen wick listed in Table 2.3, Chi (1976) uses the crimping factor, 1.05, to take into account the complexity of the screen as opposed to simply crossed wires. This expression provides a conservative estimate for the maximum liquid pressure drop based on the assumption that the wire wraps are tight.

In a conventional axial groove heat pipe with trapezoidal grooves, the cross-section is represented by a trapezoid with the short edge acting along the liquid-vapor interface (Figure 2.11a). For the axial groove tested in the present study, close-up photographs of the cross-section of the heat pipe indicated that the grooves were trapezoidal shaped with the long edge acting along the liquid-vapor interface (Figure 2.11b). A photograph of the groove is included in Appendix A.

Figure 2.11 : Comparison of traditional trapezoidal groove (a) to actual groove observed (b)

Hereafter, the term *axial groove* will refer to the configuration shown in Figure 2.11b. The expressions for K and ϵ applicable to this configuration are,

$$K = \frac{r_{h,l}^2 \epsilon}{2(f_l \text{Re}_l)} \quad (2-33)$$

and,

$$\epsilon = \frac{w - \delta \tan \beta}{s} \quad (2-34)$$

where,

$$r_{h,l} = 2 \frac{\delta w - \delta^2 \tan \beta}{w - 2\delta \tan \beta + \frac{2\delta}{\cos \beta}} \quad (2-35)$$

The derivations of these expressions are given in Appendix B. Figure 2.11 shows the dimensions for the trapezoidal groove used in these expressions.

Figure 2.12 : Trapezoidal groove dimensions

Provided all parameters have been defined, the liquid pressure drop may be expressed as,

$$\Delta P_l = - \left(\frac{\mu_l}{KA_w \lambda \rho_l} \right) L_{eff} Q \quad (2-36)$$

2.2.5.1.4 Phase-Change Pressure Drop - $\Delta P_{ph,e}$ and $\Delta P_{ph,c}$

The phase-change pressure changes in the evaporator and condenser are of equal amounts but opposite sign. As such, these two terms in Equation (2-3) cancel each other out.

2.2.5.1.5 Hydrostatic Pressure Drop - $\Delta P_{\perp}$ and $\Delta P_{\parallel}$

The axial hydrostatic pressure drop in a heat pipe under the influence of gravitation is expressed as,

$$\Delta P_{\parallel} = \rho_l g L_t \sin \varphi \quad (2-37)$$

where,

g gravitational acceleration ($\pm$ depending on orientation)

L_t total working length of heat pipe

φ angle of inclination

The term $\rho_l g L_i \sin \varphi$ accounts for the gravitational force component along the axis of the heat pipe. The radial pressure gradient, ΔP_r , is ignored because of relatively small circumferential transmission of the liquid in the wick structure for typical heat pipes used in the electronic industry.

In summary, taking the various pressure-change terms into account and solving for the rate of heat transfer from Equation (2-3) based on the maximum capillary pressure, $2\sigma/r_{eff}$, the capillary limit may be expressed as,

$$Q_{c,max} = \frac{\frac{2\sigma}{r_{eff}} + \rho_l g L_i \sin \varphi}{(F_v + F_l) L_{eff}} \quad (2-38)$$

The vapor and liquid frictional terms, F_v and F_l , are defined by Equations (2-13) and (2-31).

2.2.5.2 Sonic Limitation

Sonic limitation is the limitation on the heat transport capability of a heat pipe through choking of the vapor flow. Chi (1976) points out the similarities between the vapor flow in a heat pipe and the flow of a compressible fluid in a converging-diverging nozzle.

Inside a heat pipe, the vapor stream accelerates and decelerates at the evaporator and condenser ends due to mass addition and removal across liquid/vapor phase planes. In a diverging converging nozzle, conversely, the axial velocity variations occur because of changes in cross sectional flow area.

In a converging-diverging nozzle, the increase in velocity with a decrease in cross-sectional area results in a pressure decrease as shown in Figure 2.13. Curve A describes the pressure distribution for a converging-diverging nozzle that does not reach sonic flow at the throat of the

nozzle. The diverging section reduces the flow velocity, thereby increasing the static pressure of the fluid. If the flow does reach sonic conditions at the throat, hence choking the nozzle, the velocity can increase even further in the diverging section and become supersonic. There is also the possibility of a shock wave once the flow reaches supersonic speeds. The specific flow conditions, and thus the amount of pressure recovery depends on the back-pressure at the outlet of the nozzle.

Figure 2.13 : Static pressure profiles in a diverging-converging nozzle (Chi, 1976)

The conditions in a heat pipe with compressible flow are very similar to a converging-diverging nozzle. A heat pipe also experiences high vapor velocities, possibly choking of the vapor flow, and pressure recovery at the condenser. Figure 2.14 illustrates the heat pipe wall temperature variation with axial length. The first curve illustrates subsonic flow through the heat pipe with near complete temperature recovery at the condenser. The temperature along the wall of the heat

pipe drops as the vapor flow is accelerated inside the evaporator. By lowering the condenser temperature, which is equivalent to lowering the back-pressure in a converging-diverging nozzle, the wall temperature is lowered throughout the heat pipe. Eventually, depending on the specific conditions and characteristics of the heat pipe and working fluid, the vapor reaches sonic speeds at the exit of the evaporator. Beyond this operating state, further reduction in condenser temperature has no effect on the evaporator wall temperature.

Figure 2.14 : Pipe wall temperature profiles in a heat pipe (Chi, 1976)

In calculating the sonic limit of a heat pipe, the vapor is commonly assumed to approximate ideal gas behavior and friction forces are neglected in comparison to inertia forces. These assumptions are valid since the sonic limit normally occurs in heat pipes operating at low vapor densities and high vapor velocities. Based on these assumptions, the heat input can be expressed as,

$$Q = \frac{A_v \rho_v \lambda (\gamma_v R_v T_v)^{1/2} M_v \left(1 + \frac{\gamma_v - 1}{2} M_v^2 \right)^{1/2}}{1 + \gamma_v M_v^2} \quad (2-39)$$

where M_v , T_v , and ρ_v correspond to the conditions at the throat (i.e. evaporator exit).

Equation (2-39) can be reduced further when the vapor velocity at the throat of the evaporator is sonic, that is the Mach number, M_v , reaches unity.

$$Q_{s, \max} = A_v \rho_v \lambda \left(\frac{\gamma_v R_v T_v}{2(\gamma_v + 1)} \right)^{1/2} \quad (2-40)$$

This expression was first derived by Levy and is commonly referred to as the Levy equation. An alternative solution to the sonic limit may be found from the Navier-Stokes equations for one-dimensional laminar flow. The derivation developed by Busse (1973) expresses the sonic limit as,

$$Q_{s, \max} = 0.474 \lambda A_v \sqrt{\rho_v P_v} \quad (2-41)$$

where the properties of the fluid are taken at the evaporator exit.

In summary, controlling the heat rejection rate can throttle the heat transfer rate up to the sonic limit of the heat pipe. However, beyond this limit, the sonic limitation does not result in thermal failure such as drying of the wick at the evaporator; it is merely a limit on the maximum heat transport rate.

2.2.5.3 Viscous Limitation

This limitation is particularly common at extremely low temperatures. As this limit is approached, the viscous forces in the vapor flow may become large enough to hinder circulation of the fluid in the heat pipe. Although this limit rarely occurs in the moderate temperature range (i.e. between 120K and 630K), it may be reached in heat pipes with extremely long condenser regions or in heat pipes undergoing cold start-up from a frozen state.

Literature on the derivation of a mathematical expression for this limiting state is sparse.

However, Peterson (1994) cites Busse's work (1973) and indicates that the following mathematical expression may be used as it agrees reasonably well with experimental results.

$$Q_{v,\max} = d_v^2 \lambda A_v \frac{P_v \rho_v}{4(f_v \text{Re}_v) \mu_v L_c} \quad (2-42)$$

Peterson (1994) also references experimental work presented in the Engineering Science Data Unit (ESDU, 1979), which indicated that the following relationship can be used to avoid the viscous limit.

$$\frac{\Delta P_v}{P_v} < 0.1 \quad (2-43)$$

2.2.5.4 Entrainment Limitation

Also related to viscous forces, high vapor velocities can retard liquid fluid returning to the evaporator. This is commonly referred to as the flooding limit. In severe cases, with high vapor velocities, waves can propagate through the vapor chamber. If these waves have shearing forces greater than the liquid surface tension of the working fluid, small droplets of liquid can be whisked away from the wick structure back to the condenser. Opposing liquid flow from the

condenser, the wick could eventually “dry out” in the evaporator. When this phenomenon occurs, the vapor mass flow rate is reduced due to a reduction in liquid supplied by the condenser. In turn, the vapor velocity and possibility of formation of waves is reduced. Thus, a heat pipe is expected to reach its entrainment limit in an intermittent manner. Consequently, the evaporator wall temperature would show an intermittent variation in temperature.

Cotter (1967) introduced a method to evaluate the entrainment limit of a heat pipe. The Weber number is used to compare the viscous shear force to the surface tension of the working fluid as a ratio (Peterson, 1994). For a Weber number less than 1, the entrainment limit is avoided as viscous forces are not strong enough to shear liquid fluid from the wick structure. The shear force, F_s , is expressed as,

$$F_s = K_1 \frac{\rho_v V_v^2 A_s}{2} \quad (2-44)$$

where K_1 is a proportionality constant and A_s is the shear area of individual surface pores of the wick.

The surface tension force, F_t , is proportional to the surface tension and wetted perimeter, C_w , of the surface pore given by the following expression, where K_2 is a proportionality constant.

$$F_t = K_2 C_w \sigma \quad (2-45)$$

Thus the Weber number, defined as the ratio of these two forces, can be expressed as,

$$We = \frac{2r_{h,w} \rho_v V_v^2}{\sigma} \quad (2-46)$$

where

$r_{h,w}$ hydraulic radius of wick

V_v vapor velocity

The hydraulic radius is found using the area of the individual surface pores and the wetted perimeter,

$$r_{h,w} = \frac{2A_w}{C_w} \quad (2-47)$$

Applying Equation (2-27), introduced earlier for liquid flow, to the vapor core yields,

$$Q = V_v A_v \rho_v \lambda \quad (2-48)$$

For $We = 1$, i.e. the entrainment limit, this equation reduces to,

$$Q_{c,max} = A_v \lambda \left(\frac{\sigma \rho_v}{2r_{h,w}} \right)^{1/2} \quad (2-49)$$

In this equation the ratio of K_1 and K_2 is assumed to be 8 based on correlation with experimental data (Chi, 1976).

2.2.5.5 Boiling Limitation

Boiling limitation defines the point at which bubbles, formed from nucleate boiling, overcome the surface tension of the working fluid and propagate. Formation occurs at nucleate sights and its extent is dependent on the number and size of nucleation sites on the solid surface. Growth is characterized by the vapor pressure and surface tension of the working fluid.

The formation of bubbles can not only cause hot spots inside the wick structure, but also impede the circulation of liquid through the wick at the evaporator end. Essentially, there would be a heat flux limit for proper evaporation in the heat pipe. It should be noted that this limitation

applies to the radial heat flux. The previously discussed four limitations all apply to axial heat flux. The radial heat flux is proportional to the axial heat flux provided that the evaporator geometry and surface distribution of heat flux at the evaporator are uniform.

The following mathematical derivation is cited from Chi (1976). As such, the mathematical analysis for the boiling limitation begins with the theory of nucleate boiling. Nucleate boiling can be reduced to two processes,

1. Nucleate formation
2. Propagation and motion

For a uniform spherical vapor bubble, the following relation can be written at equilibrium,

$$\pi r_b^2 (P_{pw} - P_l) = 2\pi r_b \sigma \quad (2-50)$$

where,

r_b bubble radius

P_{pw} saturation vapor pressure (at pipe-wick interface temperature)

The following expression can be obtained by rewriting equation (2-50) in terms of vapor and capillary pressure, P_v and P_c ,

$$\pi r_b^2 (P_{pw} - P_v + P_c) = 2\pi r_b \sigma \quad (2-51)$$

The Clausius-Clapeyron equation is given by,

$$\frac{dP}{dT} = \frac{\lambda \rho_v}{T_v} \quad (2-52)$$

Since the changes in saturation pressure and temperature through the wick are directly proportional, the Clausius-Clapeyron equation for a spherical vapor bubble is reduced to the form,

$$T_{pw} - T_{wv} = \frac{T_v}{\lambda \rho_v} \left(\frac{2\sigma}{r_b} - P_c \right) \quad (2-53)$$

Applying the conduction theory to a symmetrical cylinder of thickness (r_i - r_v), the total heat transfer can be calculated from the temperature drop across the wick structure at the evaporator section, (T_{pw} - T_{wv}). Thus,

$$T_{pw} - T_{wv} = \frac{Q \ln(r_i/r_v)}{2\pi L_e k_{eff}} \quad (2-54)$$

where,

- r_i inner radius of pipe container
- r_v radius of vapor flow
- k_{eff} effective thermal conductivity of liquid-saturated wick.

Table 2.4 summarizes Chi's (1976) notes related to calculating the effective conductivity of specific wick structures saturated with working fluid.

Table 2.4 : Wick-liquid effective conductivity, k_{eff} (Chi, 1976)

Wick Structures	k_{eff}
Wick and liquid in series	$\frac{k_l k_w}{\varepsilon k_w + k_l (1 - \varepsilon)}$
Wick and Liquid in Parallel	$\varepsilon k_l + k_w (1 - \varepsilon)$
Wrapped Screen	$\frac{k_l [(k_l + k_w) - (1 - \varepsilon)(k_l - k_w)]}{(k_l + k_w) + (1 - \varepsilon)(k_l - k_w)}$
Packed Spheres	$\frac{k_l [(2k_l + k_w) - 2(1 - \varepsilon)(k_l - k_w)]}{(2k_l + k_w) + (1 - \varepsilon)(k_l - k_w)}$
Rectangular Grooves, evaporator	$\frac{(w_f k_l k_w) + w k_l (0.185 w_f k_w + \delta k_l)}{(w + w_f)(0.185 w_f k_f + \delta k_l)}$
Rectangular Grooves, condenser	$(1 - \varepsilon)k_w + \varepsilon k_l$

where,

k_l thermal conductivity of liquid

k_w thermal conductivity of wick material

In addition to the common wick structures presented in Table 2.4, the effective thermal conductivity of the tested axial groove heat pipe within the evaporator is defined as,

$$k_{eff,e} = \frac{k_l (w - \delta \tan \beta)}{s} + \frac{\delta k_l k_w (w_f + \delta \tan \beta)}{s [\delta k_l + 0.185 k_w (w_f + \delta \tan \beta)]} \quad (2-55)$$

The derivation of this equation is provided in Appendix B.

Substituting $(T_{pw} - T_{wv})$ into the reformed Clausius-Clapeyron equation, Equation (2-52), gives,

$$Q = \frac{2\pi L_c k_{eff} T_v}{\lambda \rho_v \ln(r_i/r_v)} \left(\frac{2\sigma}{r_b} - P_c \right) \quad (2-56)$$

In summary, this expression provides the rate of heat transfer for a wick structure that contains vapor bubbles of radius r_b . Thus, if the actual heat transfer rate is lower than the value calculated through Equation (2-56), the vapor bubbles of radius r_b , if ever formed, will collapse. In cases where the heat transfer rate exceeds this limit, vapor bubbles will grow and propagate through the wick creating impedance to liquid flow. The boiling heat transport limit can therefore be expressed as,

$$Q_{b,max} = \left(\frac{2\pi L_{eff} k_{eff} T_v}{\lambda \rho_v \ln(r_i/r_v)} \right) \left(\frac{2\sigma}{r_n} - P_c \right) \quad (2-57)$$

where r_n is the nucleate radius.

The nucleate radius, r_n , of vapor bubbles is dependent on surface conditions and on the presence of dissolved non-condensable gases in the liquid. Experimental results indicate values for r_n in the order of 1×10^{-6} to 1×10^{-5} m for copper sheets immersed in water. For margin, Chi (1976) recommends values between 1×10^{-7} to 1×10^{-5} m. Dunn and Reay (1982) in their work have assumed a nucleation site radius of 2.54×10^{-7} m. In the absence of non-condensable gases, that is after the pipe has been properly cleaned prior to be charged with working fluid and hermetically sealed, the normally assumed nucleation radius values have been found to be substantially conservative.

2.2.6 Conclusion on Failure Mechanisms

The five failure mechanisms described in preceding sections are all dependent on the applied heat load and the operating temperature. Cryogenic applications can lead to viscous failure whereas the capillary limit is usually encountered first in the moderate temperature range. In the high temperature range where liquid metals are used as a working fluid the most likely limiting mechanism is the development of sonic conditions.

Using the definitions and calculations described in Section 2.2.5, each of the performance limits of a heat pipe can be computed over the desired temperature range thereby highlighting any potential shortcomings in either the design of the wick structure or choice of working fluid. Over the complete operational temperature range that can be envisaged for a heat pipe, Peterson (1994) illustrates the performance limits by mapping temperature vs. transport power as shown in Figure 2.15. If the transport power calculated lies outside the operational range, failure by that performance limit will occur.

Figure 2.15 : Map of failure mechanisms (Peterson, 1994)

2.2.7 Thermal Resistance of Conventional Heat Pipes

The thermal resistance of conventional heat pipes may be best described using a thermal resistance network diagram, as shown in Figure 2.15.

Figure 2.16 : Thermal resistance network diagram of a conventional heat pipe (Peterson, 1994)

In this figure,

$R_{ext,e}, R_{ext,c}$	external thermal resistance at the evaporator and condenser
R_{pe}, R_{pc}	radial thermal resistance of pipe wall at the evaporator and condenser
R_{we}, R_{wc}	radial thermal resistance of wick at the evaporator and condenser
R_{ie}, R_{ic}	liquid/vapor thermal resistance at the evaporator and condenser
R_{va}	axial thermal resistance through vapor core
R_{wa}	axial thermal resistance through wick
R_{pa}	axial thermal resistance through pipe

The typical relative magnitudes of these thermal resistances vary over a wide range, as listed in Table 2.5.

Table 2.5 : A comparison of thermal resistance by order of magnitude for a typical heat pipe (Peterson, 1994)

Thermal Resistance	Magnitude [°C/W]
R_{pe}, R_{pc}	10^{-1}
R_{we}, R_{wc}	10^{+1}
R_{ie}, R_{ic}	10^{-5}
R_{va}	10^{-8}
R_{pa}	10^{+2}
R_{wa}	10^{+4}

According to the orders of magnitudes presented in Table 2.5, the liquid/vapor thermal resistance and axial vapor thermal resistance may be ignored relative to the others. Furthermore, the large order of magnitude noted for the axial pipe and axial wick thermal resistances may be taken as infinite. As these are in parallel with R_{va} , they may be dropped from the network. From Fourier's law, the radial resistances are obtained using the following expressions,

$$R_{pe} = \frac{\ln(d_o/d_i)}{2\pi L_e k_p} \quad (2-58)$$

$$R_{we} = \frac{\ln(d_i/d_v)}{2\pi L_e k_{eff}} \quad (2-59)$$

$$R_{pc} = \frac{\ln(d_o/d_i)}{2\pi L_c k_p} \quad (2-60)$$

$$R_{wr} = \frac{\ln(d_i/d_v)}{2\pi L_c k_{eff}} \quad (2-61)$$

where,

d_o outside diameter of pipe

d_i inside diameter of pipe

d_v vapor core diameter

k_p pipe thermal conductivity

The effective thermal conductivity of the wick structure, k_{eff} , is calculated using the expressions given in Table 2.4.

2.2.8 Advantages and Disadvantages of Heat Pipes

The advantages of heat pipes begin with their superior performance in transporting heat. By depending on the latent heat of vaporization of working fluids, heat pipes have proven to be hundreds to thousands times more effective than any known solid conductor. By example, the temperature gradient in an aluminum rod of 10 cm length with 3 mm diameter transferring 20 Watts of heat is calculated to be 393°C. For pure copper, the temperature gradient is 182°C. For a conventional heat pipe, the temperature gradient is in the orders of a few degrees centigrade. This demonstrates the effectiveness of heat pipes when operating properly.

Another advantage of heat pipes is their simplicity and passive nature. Heat pipes do not have any moving parts. As a result, they produce no noise during operation and are virtually maintenance free. Heat pipes have an excellent history of reliability with life expectancies in excess of 180,000 hours without any signs of thermal degradation (Gernert, 1999).

In the context of applying heat pipes to electronics cooling, there are a few disadvantages. First is the performance of heat pipes being gravity dependent. Based on the characteristics of the wick structure and orientation of the heat pipe, some heat pipes vary significantly in performance. Another disadvantage to using heat pipes is the cost of custom-designed heat pipes. Although heat pipes are available at low cost in many standard sizes, bends and adjustments to the cross section may be required in order to embed heat pipes into existing electronic systems. These alterations may affect the cost and the thermal performance.

2.2.9 Experimental Testing of Heat Pipes under Vibration

One factor that may be critical for proper and reliable operation of heat pipes in electronic systems is the thermal response of the heat pipe to vibration. Although many papers have been published studying the effects of non-transient body forces on heat pipe performance (Ponnappan et al, 1992; Yerkes et al, 1992), little work has been done directly related to vibration. Deverall (1967) and Richardson (1970) are two of the few researchers to investigate the performance of heat pipes in simple harmonic motion. Both authors observed an increase in thermal resistance because of de-priming of the wick structure.

Charlton (1992), Huber (1993), and Carpenter (1994), in conjunction with the U.S. Air Force Institute of Technology, studied the effects of vibration on the capillary limit of heat pipes with vibration applied in the transverse and longitudinal directions. Charlton (1992) concluded that although a decrease in capillary limit was observed, the effect of transverse vibration on a wrapped screen copper/water heat pipe was not significant. It should be noted that Charlton's conclusions were limited because of the test setup. Charlton (1992) included some suggestions to help improve the quality of the data by making modifications to the test setup. Having applied Charlton's suggestions, Huber (1993) found that longitudinal vibration decreases the capillary limit of a wrapped screen wick copper/water heat pipe, with the amount of performance

degradation being dependent on the vibration amplitude. A decrease of approximately 2% was observed with a vibration amplitude of 0.2g. However, the uncertainty of the measurements exceeded this decrease in performance, thus no reliable conclusion could be drawn. Further testing by Huber showed a drop in capillary limit of approximately 10% and 6% with a vibration amplitude of 2.0 g at 30 Hz and 50 Hz, respectively. Carpenter (1994) considered vibration applied perpendicular to the axis (transverse) of an axial groove ammonia/aluminum heat pipe. A degradation of almost 13% was observed in the maximum heat transfer rate at a peak acceleration amplitude of 1.84g. The drop in performance increased to almost 15% with an increase in amplitude (2.50g). This supports Huber's observations with longitudinal vibration.

With regards to performance under random vibration, very little experimental results have been published (Deverall 1967, Richardson 1970). Given the limited proceedings, the performance of heat pipes under random vibration requires further investigation, and is the subject of the present study.

3

Heat Pipe Thermal Analysis

This chapter presents a software that was developed to calculate the thermal performance and five possible operating limits for sintered powder, wrapped screen, and axial groove heat pipes. The software was developed using the Visual Basic programming language and is based entirely on the theory presented in Chapter 2. The initial objective for developing the software was to assist in the experimental effort by providing estimates on anticipated heat-transfer rates. Once the theory used in the software was validated with the experimental results, the software became a tool that may be used at Dy 4 Systems for design and analysis of heat pipes efficiently and reliably. Thus, a professional user interface was added to the software which is also described in this chapter.

3.1 Software

In order to perform a thermal analysis of a sintered powder, wrapped screen, or axial groove heat pipe, the program applies the theory discussed previously in Chapter 2. The following subsections present a detailed explanation of the program, describing both the user interface and sequence of instructions.

3.1.1 Description of User Interface

Having explained the methods used to determine the five possible operating limits of a heat pipe in Chapter 2, the purpose of this section is to demonstrate the functionality of the computer program based on these methods. A set of step-by-step instructions with figures is used to direct the reader through the software. By the end of this section, the reader will be able to execute the software, create or modify a heat pipe model, view the working fluid properties, view a break down of the thermal resistance network throughout the heat pipe, and determine its five operating limits.

3.1.1.1 Start-up

On starting up the program, the user is welcomed with an introduction to the software. The introduction displays the name of the program, the developer, the advisor involved during development, and the date when the program was compiled. A tribute to Dy 4 Systems is also displayed. Figure 3.1 shows the *Title* form during start-up.

Figure 3.1 : Title form

After startup, the *Title* form is automatically closed and the *Main* form is activated. From this form the user may create, open, save and modify a heat pipe model. Figure 3.2 shows the form, which is composed of three frames containing information on the operating conditions, heat pipe material and dimensions, and wick structure, respectively. The definition of all parameters appearing on the *Main* form and all subsequent forms can be viewed by clicking on the parameter. This allows the user to quickly identify a parameter in the event that the symbol, character, or subscript associated with the parameter is not understood. An example of this feature is seen in Figure 3.2 for the heat pipe outside diameter, d_o .

NOVA - Heat Pipe Thermal Analysis

File Edit Run Results Wizard Help

Heat Pipe Operating Conditions

☒ Surface Temperature
☐ Rate of Heat Transfer and Heat Source/Sink Temperature
☐ Heat Source and Heat Sink Temperature

Surface Temperature

$T_{p,e}$ [°C]
 $T_{p,c}$ [°C]

Q and Heat Source/Sink Temperature

Q_s [W]
☐ Heat Source ☐ Heat Sink

Heat Pipe

Outside Diameter
 d_o [mm]
 d_i [mm]
 d_v [mm]
 L_e [mm]
 L_a [mm]
 L_c [mm]

Fluid [v]
 Material [v]
 k_p [W/m-K]

Wick

Type [v]
 Material [v]
 k_w [W/m-K]
 t_w [mm]

Properties

ϵ
 K [m²]
 r_{eff} [m]
☐ Calculate using geometry
☐ Use Values from Experiment

Figure 3.2 : Main form

The *Heat Pipe Operating Conditions* frame allows for input of evaporator/condenser temperatures and/or heat transfer rates. The heat pipe geometry, material and working fluid are specified in the *Heat Pipe* frame. The *Wick* frame is where the type of wick structure, wick material, and wick properties, specifically its porosity, ϵ , permeability, K , and effective capillary radius, r_{eff} , are specified. A detailed description of these frames is provided in Section 3.1.1.2.

In addition to the three frames, a series of menus can be seen along the top of the form beginning with features *File*, *Edit*, *Run*, *Results*, *Wizard*, and *Help*, respectively. The *File* menu contains traditional options such as *New*, *Open*, *Save*, *Save As*, and *Exit*. Figure 3.3 shows an example of

the menu options listed. As seen, short cut keys have been created for advanced users. This menu together with the remaining menus will be discussed in Section 3.1.1.2.

The screenshot shows the 'NOVA - Heat Pipe Thermal Analysis' window. The 'File' menu is open, displaying options: New (Ctrl+N), Open, Save (Ctrl+S), Save As, and Exit. Below the menu, there are several input fields and checkboxes for specifying the heat pipe model.

Heat Pipe Parameters:

- $T_{p,h}$ [°C]
- $T_{p,c}$ [°C]
- Q_b [W]
- ☐ Heat Source ☐ Heat Sink

Heat Pipe Geometry and Material Properties:

- d_o [mm]
- d_i [mm]
- d_v [mm]
- L_e [mm]
- L_a [mm]
- L_c [mm]
- Fluid: [Dropdown]
- Material: [Dropdown]
- k_p [W/m-K]

Wick Properties:

- Type: [Dropdown]
- Material: [Dropdown]
- k_w [W/m-K]
- t_w [mm]

Wick Properties (Advanced):

- ϵ [mm]
- K [m²]
- r_{eff} [m]
- ☐ Calculate using geometry
- ☐ Use Values from Experiment

Figure 3.3 : File menu

3.1.1.2 Specification of the Heat Pipe Model with the Wizard

The details of the contents of the *Main* form and its sub-forms are best described in the context of the built-in Wizard feature of the software. The Wizard is activated under the *Wizard* menu as shown in Figure 3.4.

Figure 3.4 : About Wizard

Once starting the Wizard, the user is prompted for the operating conditions of the heat pipe. In this step, the user is given three options as shown in Figure 3.5:

- 1) Surface Temperature
- 2) Rate of Heat Transfer and Source or Sink Temperature
- 3) Source Temperature and Sink Temperature.

The *Surface Temperature* option is selected in cases where the surface temperature at both the evaporator and condenser ends of the heat pipe is known. As discussed in the context of the flow chart presented in Section 3.1.2, the software uses the implied temperature gradient along the length of the heat pipe and the overall thermal resistance to determine the rate of heat transfer

through the heat pipe. This rate of heat transfer is then used to estimate the fluid temperature inside the evaporator based on the thermal resistance of the heat pipe wall and wick structure in the evaporator.

The second option is used when the rate of heat transfer is known together with the temperature of either the heat source or the heat sink. The thermal resistance of the heat source or heat sink is then included in the overall thermal resistance network the software uses for its calculations. This option may be popular in electronics cooling applications where the amount of heat to be dissipated is often known a priori.

The last option is similar to the surface temperature option except that the rate of heat transfer is now calculated based on a thermal resistance that includes the resistances of the heat source and heat sink. This may be a more practical option than the first described above since the temperatures of the heat source and heat sink would be more readily available than the temperatures at the outer surface of the heat pipe. Once the selection has been made, the user may proceed by using the command displayed in Figure 3.5.

Wizard - Step 1

The first step to building a heat pipe model is defining the operating conditions of the heat pipe. Three conditions are available. They are:

- 1) SURFACE TEMPERATURE: The heat pipe outer surface temperatures at both the evaporator and condenser are known.
- 2) RATE OF HEAT TRANSFER AND HEAT SOURCE/SINK TEMPERATURE: The heat transfer rate and either the heat sink or heat source temperature is known.
- 3) HEAT SOURCE AND HEAT SINK TEMPERATURE: Given the possibility of not knowing the heat transfer rate, the operating limit and thermal performance of the heat pipe may be evaluated using the heatsink and heatsource temperatures.

Heat Pipe Operating Conditions

Select one of the three conditions below:

- ☐ Heat Pipe Outer Surface Temperature
- ☐ Rate of Heat Transfer and Heat Source/Sink Temperature
- ☐ Heat Source and Heat Sink Temperature

Cancel **Next >>**

Figure 3.5 : Wizard form – Step 1: Operating conditions

Depending on the selection made in Step 1, Step 2 asks for input on the evaporator and condenser heat pipe surface-temperatures, rate of heat transfer and heat source (or sink) temperature, or heat source and heat sink temperatures. Figure 3.6 shows the three possible forms for Step 2. At this stage, the user is also provided with the option of scrolling back to previous steps by using the **Back <<** command. This allows the user to make adjustments to previous steps if necessary. The user is not allowed to proceed to the next step until all fields and/or options have been completed. This constraint has been put in place to ensure that sufficient information is made available to the software to complete the heat pipe model. The user may exit the Wizard at any time by using the **Cancel** command. If the *Cancel* command is used, all parameters entered up to the point of canceling the Wizard will have been copied to the *Main* form shown in Figure 3.2.

Wizard - Step 2

Surface Temperature

Enter the evaporator and condenser temperature of the heat pipe in the fields below

Evaporator Temperature [°C]

Condenser Temperature [°C]

Wizard - Step 2

Rate of Heat Transfer and Heat Source/Sink Temperature

Enter the rate of heat transfer in the field below and select either the heat sink or heat source definition.

Rate of Heat Transfer [W]

☐ Heat Source ☐ Heat Sink

Wizard - Step 2

The specification of both the Heat Source temperature and Heat Sink temperature has been selected. In the source and sink forms, define the thermal boundary conditions of the Heat Source and Heat Sink as either conduction or convection.

For conduction, the conductivity of a layer of solid medium (e.g. Thermal Grease) may be defined. Contact resistance between two surfaces may also be taken into account. The contact resistance value may be either specified directly or selected from typical surfaces.

For the convection condition, the convective heat transfer coefficient may either be specified by the user or calculated for one of the following conditions:

- 1) Cylinder in Cross Flow
- 2) Cylinder in Parallel Flow (i.e. aligned with its axis)

Figure 3.6 : Wizard form – Step 2: Surface temperature, rate of heat transfer, and heat source/heat sink forms

In Figure 3.6, the first form refers to the *Surface Temperature* condition. The required fields are the heat pipe outer surface temperatures at the evaporator and condenser ends. Once these values have been entered, the user may proceed to the next step. The second form provides fields for specification of *Rate of Heat Transfer and Heat Source/Sink Temperature*. The rate of heat transfer is to be entered first. The user is then given the option of either specifying the heat sink or heat source temperature. Once the choice is made, the *Heat Source* (or *Heat Sink*) form appears as shown in Figure 3.7. For the *Heat Source Temperature and Heat Sink Temperature* condition (third form shown in Figure 3.6), both the *Heat Source* and *Heat Sink* forms appear.

Figure 3.7 : Heat source and heat sink forms

In these forms, the user is given the option of defining either a conduction or convection condition. For conduction, the thickness and thermal conductivity of the conductive medium is to be specified. If a contact resistance is expected between this medium and the outer surface of the heat pipe, this is indicated on the form. In this case, the contact coefficient is either specified in the available field or the *Select from Typical Surfaces* option is chosen. For the latter, a surface is chosen on the *Typical Surfaces* form as shown in Figure 3.8.

Heat Sink

T_{sink} [°C]

☒ Conduction ☐ Forced Convection

Conduction

t_m [mm]

k_m [W/m·K]

☒ Include Contact Resistance

h_c

☒ Select from Typical Surfaces

Forced Convection

h_{conv}

☐ Calculate Heat

☐ Cylinder in

☐ Cylinder in

Typical Surfaces

	Roughness [μm]	Temperature [°C]	Pressure [atm]	h_c [W/m ² ·K]
☒ Aluminum, ground, air	2.54	150	12-25	11,300
☐ Aluminum, ground, air	0.25	150	12-25	55,500
☐ Copper, ground, air	1.27	20	12-200	142,800
☐ Copper, milled, air	3.81	20	10-50	55,500
☐ Copper, milled, vacuum	0.25	30	7-70	11,300

Cancel OK

OK

Figure 3.8 : Typical surfaces form

For convection, the heat transfer coefficient may be either specified by the user or calculated using one of two convection heat-transfer models built into the software. The two models are:

cylinder in cross flow, and cylinder in parallel flow with flow aligned to the heat pipe axis. In either case, the user is prompted for information on fluid temperature, velocity, and the outer diameter of the heat pipe as shown in Figure 3.9.

Heat Source

T_{source} [°C]

☐ Conduction ☒ Forced Convection

Conduction

t_m [mm]

k_m [W/m·K]

☐ Include Contact Resistance

h_c [W/m²·K]

☐ Select from Typical Surfaces

Forced Convection

h_{conv} [W/m²·K]

☒ Calculate Heat Transfer Coefficient

☒ Cylinder in Cross Flow
☐ Cylinder in Parallel Flow

OK

Cylinder in Cross Flow

Fluid

T_{∞} [°C]

V_{∞} [m/s]

d_o [mm]

Diagram: A cylinder of diameter d_o is shown with flow parameters T_{∞}, V_{∞} approaching it from the left.

Clear Cancel OK

Figure 3.9 : Steps to specifying a cross flow convection condition

Having defined the operating conditions of the heat pipe, the pipe material, section lengths, and diameters are then specified in Step 3. Figure 3.10 shows the input parameters required for this step.

Wizard - Step 3

Having defined the operating conditions of the heat pipe, the next step is to define its geometry. Specifically the length of the evaporator, adiabatic, and condenser sections; and the pipe outside and inside diameters.

The thermal resistance of the pipe and wick are dependent on these parameters. For example, an increase in evaporator surface area decreases the thermal resistance through the pipe and wick structure at the evaporator. As such, designs aiming to minimize the temperature gradient between the Heat Source and Heat Sink, should consider maximizing the surface area. Another option is to use more conductive materials or a wick structure known to have a high effective thermal conductivity.

Enter the required values below. The heat pipe material may be selected from a list of materials provided or entered directly. The thermal conductivity of the material is required only if the material is not chosen from the list.

Heat Pipe Length		Heat Pipe Diameter	
Evaporator	<input type="text"/> [mm]	Outside	<input type="text"/> [mm]
Adiabatic	<input type="text"/> [mm]	Inside	<input type="text"/> [mm]
Condenser	<input type="text"/> [mm]		

Heat Pipe Material

Material:

Thermal Conductivity: [W/m.K]

Figure 3.10 : Wizard form – Step 3: Heat pipe dimensions and material

The heat pipe material may either be entered by the user or selected from a list of materials in the drop down menu. When selecting from the list of materials, the thermal conductivity is updated automatically. Otherwise, the user must specify the thermal conductivity of the material after entering a description. The next step in the Wizard is defining the working fluid of the heat pipe. In this case the user is provided with five fluid choices suitable for moderate-temperature applications. The two most common are water and methanol. This fourth step is shown in Figure 3.11.

Figure 3.11 : Wizard form – Step 4: Working fluid

Once the working fluid has been specified, the last step required to complete the heat pipe model is information related to the wick structure. The wick type, material, and properties are specified in this step as shown in Figure 3.12. A choice of three wick types is made available; sintered powder, wrapped screen, and axial groove. The properties for each of these types may either be specified or calculated by the software.

Wizard - Step 5

The next step involves specification of the wick structure. Specifically the type, dimensions, and material of the wick. The properties of the wick may be either specified by the user or determined by the software.

Wick Structure

Enter the wick type and material using the combination boxes below. The properties of the wick structure may be entered directly, calculated by the software, or chosen from a list of published experimental results.

Wick Type: [Dropdown]

Wick Material: [Dropdown]

Thermal Conductivity: [Text] [W/m.K]

Thickness of Wick: [Text] [mm]

Properties

☒ Calculate ☐ Use Experimental Results

Porosity: [Text]

Permeability: [Text] [m²]

Eff. Capillary Radius: [Text] [m]

Cancel Back << Next >>

Figure 3.12 : Wizard form – Step 5: Wick structure

Once the wick type has been selected, the *Properties* frame becomes active. At this point, the user may specify or ask the software to calculate the properties of the wick. In specifying the wick properties, the user is also given the option of making a choice from a list compiled from published experimental results. If the *Calculate* option is chosen as shown in Figure 3.13, the *Property* form, shown in Figure 3.14, appears.

Wizard - Step 5

The next step involves specification of the wick structure. Specifically the type, dimensions, and material of the wick. The properties of the wick may be either specified by the user or determined by the software.

Wick Structure

Enter the wick type and material using the combination boxes below. The properties of the wick structure may be entered directly, calculated by the software, or chosen from a list of published experimental results.

Wick Type:

Wick Material:

Thermal Conductivity: [W/m-K]

Thickness of Wick: [mm]

Properties

☒ Calculate ☐ Use Experiment Results

Porosity:

Permeability: [m²]

Eff. Capillary Radius: [m]

Figure 3.13 : Example of calculating wick properties for sintered powder heat pipe

As seen in Figure 3.14, the *Property* form assumes a slightly different format depending on the chosen wick structure. The porosity, permeability and effective capillary radius of the wick are, for the most part, quantities common to all wick structures. For the sintered powder wick, the porosity of the wick is required in order to calculate the permeability and effective capillary radius. The remaining fields of data entry pertain to geometric information on the wick and as such, are unique for each wick structure.

Calculator - Wick Properties

r_s [m]
 ε

 ε
 K [m²]
 r_{eff} [m]

Calculate Cancel OK

Calculator - Wick Properties

No. Layers
 N
 d_w [m]
 S 1.05

 ε
 K [m²]
 r_{eff} [m]

Calculate Cancel OK

Calculator - Wick Properties

β [deg]
 w [m]
 δ [m]
 s [m]

 ε
 K [m²]
 r_{eff} [m]

Calculate Cancel OK

Number of Grooves

☐ Rectangular
 ☒ Trapezoidal

Figure 3.14 : Property form for sintered powder, wrapped screen, and axial groove heat pipes

If the user chooses to specify the wick properties with the help of the software (i.e. checking the box for *Use Experimental Results*), a form as shown in Figure 3.15, appears to inform the user of the porosity, permeability and effective capillary radius to be chosen. .

The figure shows a 'Wizard - Step 5' dialog box. The main text says: 'The next step involves specification of the wick structure. Specifically the type, dimensions, and material of the wick. The properties of the wick may be either specified by the user or determined by the software.'

Wick Structure

Enter the wick type and material using the combination boxes below. The properties of the wick structure may be entered directly, calculated by the software, or chosen from a list of published experimental results.

Wick Type: Sintered Powder (dropdown)
Wick Material: Beryllium Copper (dropdown)
Thermal Conductivity: 377.8 [W/m-K]
Thickness of Wick: [mm]
Properties:
☐ Calculate ☒ Use Experimental Results
Porosity: []
Permeability: [] [m²]
Eff. Capillary Radius: [] [m]

Buttons: Cancel, Back <<, Next >>

Experiment (sub-dialog box):

Properties based on Experimental Data
<Select Experiment> (dropdown)
ε: []
K: [] [m²]
r_{eff}: [] [m]
Buttons: Cancel, OK

Figure 3.15 : Example of using experimental results for the sintered powder heat pipe

Once the properties of the heat pipe have been defined, all the parameters required to predict the five operating limits of the heat pipe model have been made available to the software. However, as part of the *Advance* form (Figure 3.16), the user is given the opportunity to make adjustments to the angle of incline, acceleration, and the portion of the condenser length that contributes to L_{eff} (Section 2.2.5.1.2). On the *Advanced Options* form, detailed descriptions of these three parameters are provided to assist the user in quantifying them.

Wizard - Step 6

The last step gives the user the opportunity to make adjustments to advanced features such as angle of incline, presence of body forces, and location at which flooding in the condenser begins.

ANGLE OF INCLINE: Heat pipes in general are sensitive to the orientation of body forces relative to the axis of the heat pipe. In the case where the condenser is placed above the evaporator (with respect to the direction of acceleration), the body force assists the return of liquid to the evaporator. In the case where the condenser is placed below the evaporator, body forces act against capillary forces decreasing the capillary limit of the heat pipe. The default value for the angle of incline is set to 0. That is the axis of the heatpipe is assumed perpendicular to any body force.

ACCELERATION: The user is given the option of specifying a linear acceleration for the heat pipe (e.g. gravity).

FLOODING: The default calculation of the length over which viscous forces are considered is based on the sum of half the length of the evaporator section, adiabatic section, and half the length of the condenser section. If the user judges that part of the condenser may be flooded, thus containing working fluid in static state, this portion of the condenser should be excluded from the effective length of the heat pipe. The Wet Point is defined as the point at which flooding begins relative to the end of the heat pipe at the condenser.

Orientation			Flooding		
Angle of incline	0	[deg]	Wet Point	0	Back <<
Acceleration	9.81	[m/s ²]			Finish

Figure 3.16 : Wizard form – Step 6: Advanced options

Once the advanced features of the heat pipe are verified, the model is complete. At this point, the user may adjust any of the parameters specified on the *Main* form, Figure 3.2. Parameters associated with the heat sink, heat source, wick structure, and/or advance option may be modified using the *Edit* menu (Figure 3.17). Once the user is satisfied with all the specified parameters, the next step is to have the software perform a thermal analysis.

NOVA - Heat Pipe Thermal Analysis

File Edit Run Results Wizard Help

Heat Source
Heat Sink
Wick Properties
Advance

Surface Temperature
 $T_{p,e}$ [°C]
 $T_{p,c}$ [°C]

Q and Heat Source/Sink Temperature
 Q_a 10 [W]
☐ Heat Source ☒ Heat Sink

Heat Pipe
 d_o 4 [mm]
 d_i 2.7 [mm]
 d_v 0.3 [mm]
 L_e 63.5 [mm]
 L_a 71.5 [mm]
 L_c 60 [mm]
 Fluid Water
 Material Beryllium Copper
 k_p 377.8 [W/m-K]

Wick
 Type Sintered Powder
 Material Beryllium Copper
 k_w 377.8 [W/m-K]
 t_w 0.2 [mm]
 Properties
 ϵ 0.52
 K 9.00e-13 [m²]
 r_{eff} 9.39e-6 [m]
☐ Calculate using geometry
☒ Use Values from Experiment

Figure 3.17 : Edit menu

3.1.1.3 Thermal Analysis of the Heat Pipe

The execution of a thermal analysis is realized through the *Run Model* option in the *Run* menu of the *Main* form (Figure 3.18). Under the *Run* menu, there is also an *Error Check* option. This performs a check on the heat pipe model ensuring that all parameters have been defined. Having completed the model using the Wizard feature, this check is not necessary. For users not using the Wizard feature, this option is recommended. If a parameter is not defined, the software will display an error message.

NOVA - Heat Pipe Thermal Analysis

File Edit Run Results Wizard Help

Heat Pipe Run Model

☐ Surface
☒ Rate of heat transfer and Heat Source/Sink Temperature
☐ Heat Source and Heat Sink Temperature

Surface Temperature

$T_{p,e}$ [°C]
 $T_{p,c}$ [°C]

Q and Heat Source/Sink Temperature

Q_a 10 [W]
☐ Heat Source ☒ Heat Sink

Heat Pipe

d_o 4 [mm]
 d_i 2.7 [mm]
 d_v 2.3 [mm]
 L_e 63.5 [mm]
 L_a 71.5 [mm]
 L_c 60 [mm]

Fluid Water
 Material Beryllium Copper
 k_p 377.8 [W/m-K]

Wick

Type Sintered Powder
 Material Beryllium Copper
 k_w 377.8 [W/m-K]
 t_w 0.2 [mm]

Properties

ϵ 0.52
 K 9.00e-13 [m²]
 r_{eff} 9.39e-6 [m]

☐ Calculate using geometry
☒ Use Values from Experiment

Figure 3.18 : Run model

During the thermal analysis, the software informs the user of the vapor flow conditions inside the heat pipe (Figure 3.19). As discussed in Section 2.2.5.1.2, the vapor frictional loss is calculated based on the flow conditions and compressibility of the fluid.

Figure 3.19 : Example of vapor flow conditions displayed during a run

After the software has completed the thermal analysis, the *Results* menu heading on the *Main* form becomes active. Under this heading there are three options: *Fluid Properties*, *Operating Limits*, and *Thermal Resistance* (Figure 3.20). Shortcuts are again available for advance users.

The screenshot shows the 'NOVA - Heat Pipe Thermal Analysis' window. The 'Results' menu is active, displaying three options: 'Fluid Properties' (Ctrl+F), 'Operating Limits' (Ctrl+O), and 'Thermal Resistance' (Ctrl+R). Below the menu, there are several input fields and checkboxes for configuring the analysis.

Heat Pipe Operation:

- ☐ Surface Temperature
- ☒ Rate of Heat
- ☐ Heat Source and Heat Sink Temperature

Surface Temperature:

- $T_{p,e}$ [°C]
- $T_{p,c}$ [°C]

Q and Heat Source/Sink Temperature:

- Q_a [W] (Value: 10)
- ☐ Heat Source
- ☒ Heat Sink

Heat Pipe:

- d_o [mm] (Value: 4)
- d_i [mm] (Value: 2.7)
- d_v [mm] (Value: 2.3)
- L_e [mm] (Value: 63.5)
- L_a [mm] (Value: 71.5)
- L_c [mm] (Value: 60)
- Fluid: Water
- Material: Beryllium Copper
- k_p [W/m-K] (Value: 377.8)

Wick:

- Type: Sintered Powder
- Material: Beryllium Copper
- k_w [W/m-K] (Value: 377.8)
- t_w [mm] (Value: 0.2)

Properties:

- ϵ (Value: 0.52)
- K [m²] (Value: 9.00e-13)
- r_{eff} [m] (Value: 9.39e-6)
- ☐ Calculate using geometry
- ☒ Use Values from Experiment

Figure 3.20 : Results menu

The *Fluid Properties* form (Figure 3.21) displays the properties of the fluid based on the operating temperature of the heat pipe. These would be the properties that were used in predicting the operating limits of the heat pipe. The *Operating Limits* form (Figure 3.22) displays the maximum heat transfer rates based on Boiling, Capillary, Entrainment, Sonic, and Viscous limits of the heat pipe. The heat transfer rate applied at the evaporator, Q_{in} , is also shown. The software explicitly informs the user if the applied heat load exceeds any one of the operating limits.

Lastly, the *Thermal Resistance* form (Figure 3.23) itemizes the thermal resistances throughout the heat pipe including those associated with heat source and heat sink conditions, if applicable. The heat transfer coefficients and temperature gradients between the evaporator and condenser surface temperatures are also included.

Results - Fluid Properties

T_v [°C]

σ_l [N/m]

μ_l [N·s/m²]

μ_v [N·s/m²]

ρ_l [kg/m³]

ρ_v [kg/m³]

λ [kJ/kg]

OK

Figure 3.21 : Results form – Fluid properties

Results - Operating Limits

Operating Limits

$Q_{b,max}$ [W]

$Q_{c,max}$ [W]

$Q_{e,max}$ [W]

$Q_{s,max}$ [W]

$Q_{v,max}$ [W]

Q_a [W]

OK

Figure 3.22 : Results form – Operating limits

Results

Thermal Resistance

R_{source}	<input type="text"/>	[°C/W]
$R_{\text{p,e}}$	<input type="text"/>	[°C/W]
$R_{\text{w,e}}$	<input type="text"/>	[°C/W]
R_{v}	<input type="text"/>	[°C/W]
$R_{\text{w,c}}$	<input type="text"/>	[°C/W]
$R_{\text{p,c}}$	<input type="text"/>	[°C/W]
R_{sink}	<input type="text"/>	[°C/W]
R_{tot}	<input type="text"/>	[°C/W]

Heat Transfer Coefficients

h_{source}	<input type="text"/>	[W/m ² ·K]
h_{sink}	<input type="text"/>	[W/m ² ·K]

Temperature

$T_{\text{p,e}}$	<input type="text"/>	[°C]
$T_{\text{p,c}}$	<input type="text"/>	[°C]
ΔT	<input type="text"/>	[°C]

OK

Figure 3.23 : Results form – Thermal resistance

Having created, thermally analyzed, and viewed the results of a heat pipe model, the user may then decide whether or not the heat pipe satisfies the design requirements. If not, adjustments to a wide number of parameters can be made in order to find a heat pipe that is suitable. Alternatively, a sensitivity analysis may be conducted using the software by monitoring the operating limit of the heat pipe while systematically varying any of the parameters of interest.

3.1.2 Flow Chart

A flow chart for the core portion of the software, which excludes the user interface routines, is presented in Figures 3.24 to 3.32. These sets of instructions are executed after the user has selected a particular heat pipe model with one of the available sets of boundary conditions. While summarizing the functionality of the software, the flow chart also demonstrates the application of the theory described in Chapter 2.

Figure 3.24 : Flow chart of thermal analysis module of NOVA – part 1

Figure 3.25 : Flow chart of thermal analysis module of NOVA – part 2

Figure 3.26 : Flow chart of thermal analysis module of NOVA – part 3

Figure 3.27 : Flow chart of thermal analysis module of NOVA – part 4

Figure 3.28 : Flow chart of thermal analysis module of NOVA – part 5

Figure 3.29 : Flow chart of thermal analysis module of NOVA – part 6

Figure 3.30 : Flow chart of thermal analysis module of NOVA – part 7

Figure 3.31 : Flow chart of thermal analysis module of N

Temperature is known. Determine
Calculate R_{src} .

Flow chart of thermal analysis module of NOVA – part 8

Heat Source Temperature is known. Determine Parameters to Calculate R_{src} .

Heat Sink Temperature is known. Determine Parameters to Calculate R_{snk} .

Figure 3.32 : Flow chart of thermal analysis module of

32 : Flow chart of thermal analysis module of NOVA – part 9

4 Experimental Setup

4.1 Introduction

This chapter describes the experimental arrangements and procedure that were used for testing of heat pipes under vibrating conditions. Several arrangements are described since the experiments of this study are divided by the type of wick structure and orientation of the gravitational force relative to the heat pipe axis: *Transverse* and *Longitudinal*.

The setup is designed to investigate the effects of vibration on the thermal operating limit of heat pipes. The operating limit under such conditions is anticipated to be a function of vibration frequency, amplitude, and direction. As such, considering a typical operating environment, the frequency range and amplitude are chosen based on Dy 4 guidelines, and the orientation is varied systematically. The setup has been designed to allow variations in heat pipe wick structure to establish the sensitivity of thermal performance to this feature under vibrating conditions. The accuracy of the measurements is such that the results can be used to evaluate existing mathematical models for predicting heat pipe thermal performance under static conditions, and to develop models for predicting heat pipe thermal performance while subjected to dynamic conditions.

4.2 Selection of Heat Pipes

As noted in the preceding section, variations are considered for the heat pipe design with respect to the wick structure and orientation relative to the direction of vibration and gravitational force.

The extent of variations given in Table 4.1 is driven by the intent of the application, i.e. cooling of printed circuit boards.

Table 4.1 : Heat pipe configurations

Heat Pipe	1	2	3	4
Length [mm]	200	200	200	200
- Working	190 ± 1	192 ± 1	192 ± 1	194 ± 1
- Evap/Cond	$63\frac{1}{2}/50 \pm 1$	$63\frac{1}{2}/52 \pm 1$	$63\frac{1}{2}/52 \pm 1$	$63\frac{1}{2}/54 \pm 1$
Cross Section	Circular	Circular	Circular	Circular
Working Fluid	Water	Water	Water	Water
Material	Copper	Copper	Copper	Copper
Wick Structure	Sintered Powder	Wrapped Screen	Axial Groove	Fiber Bundle
Wick Material	Copper	Copper	Copper	Copper

The four wick structures tested were sintered powder, wrapped screen, axial groove, and fiber bundle. These four heat pipes were chosen because of their availability commercially. The performance of each heat pipe depends on the capillary force generated by the wick. As described in Chapter 2, the extent of the capillary force in a heat pipe largely depends on the discrete corners or abrasions associated to the wick structure. The four wick structures and their known properties are shown in Table 4.1. All four heat pipes are Commercial Off The Shelf (COTS) components strategically sized by the intent of the application.

Water is chosen as the working fluid because of its potential for producing a high capillary force compared to other working fluids as was presented in Section 2.2.4. The liquid figure of merit for water exceeds those of other working fluids in the moderate temperature range because of its high latent heat of vaporization, high surface tension, high liquid density, and low liquid viscosity relative to alternative fluids. Other advantages of water are low costs, availability, non-toxicity, and compatibility with the selected temperature range for cooling of integrated circuits.

The overall length of each heat pipe is 200 mm. The working length of the heat pipe, which excludes the crimped section, is measured using a straight edge with a resolution of 1 mm. Based on the experimental setup, the lengths of the evaporator and condenser sections are 63.5 mm and approximately 52.0 mm, respectively. Outer diameter measurements are performed using a dial caliper with a resolution of 0.025mm. The wall thickness of each pipe is determined using a low-speed cutter to first section the heat pipe and an Ersascope, a high definition digital microscope equipped with real-time video capture, for magnified viewing of its cross-section. A micrometer with a resolution of 0.001 mm is used to calibrate the displacement projected by the camera. Measurements are made using point-to-point displacements provided by the application software and a calibration factor specified by the user. Sections of each heat pipe are shown in Appendix A.

4.3 Vibration

All the variations in heat pipe configuration described in the preceding section are tested under vibration. The experimental setup for vibrating heat pipes is comprised of an electromagnetic shaker system and a vibration test fixture located at Dy 4 System facilities. The electromagnetic shaker system is composed of several components: a shaker, blower, field generator, power amplifier, signal conditioner, and a control console. A schematic of the vibration system is shown in Figure 4.1.

Figure 4.1 : Schematic diagram of the vibration test system

This vibration test system can be used to perform sine sweeping, random vibration, and shock testing. Accelerometers are used to collect displacement, velocity, and/or acceleration information from the shaker outputs. One of these accelerometers is used to provide the driver controls with feedback. Up to three others are response accelerometers that are at the disposal of the user. Outputs from these accelerometers are in units of Pico-coulombs/g sensitivity. The signal conditioner processes signals from the accelerometers to provide clean amplified signals from all control and response accelerometers and delivers these signals to the Spectral Dynamics Controller in units of mV/g. The shaker consists of a moving shaker table (armature) driven by a surrounding driver coil. The shaker table provides a planar mounting surface to install fixtures. A remote blower is used to duct cooling air across the driver coil to control the coil temperature. A compressed air line is used to pump up pneumatic pads to isolate the shaker vibration from the floor and allows the operator to adjust the shaker table height to the center of its stroke in order to compensate for the fixture and UUT weight. The power amplifier consists of a large cabinet

housing the amplifier and a wall-mounted field generator. The power amplifier receives signals from the vibration controller and allows the operator to activate the power amplifier circuits, field generator, and cooling blower. The vibration controller consists of a Spectral Dynamics Controller and a SUN workstation connected through the 'S' bus. The vibration controller provides digital real-time closed-loop shaker control and contains multiple internal microprocessors running in parallel. It can carry up to 12 simultaneous input channels for control or analysis. A single channel contains a 16-bit Digital-to-Analog converter that has a sampling rate of 51.2 kHz and an output voltage range of $\pm 10V$ full scale.

The shaker is the principle part of the system. A schematic diagram of it is shown in Figure 4.2. It provides support and motion to the test items attached to its upper surface and is designed such that the shaker table vibrates perpendicular to this surface.

Figure 4.2 : Schematic diagram of the shaker

The air-cooled field coils, connected to a DC power source, generate magnetic fields in the annular air gap. The table and driver coil are suspended from the shaker body by a table suspension. When an alternating current, provided by the power amplifier, is passed through the

driver coil, an electrodynamic force is generated causing the table to move. The frequency of the alternating current determines the frequency of vibration. The table has a resonant frequency of 2100 Hz nominal. The maximum frequency applied to the table is therefore kept at or below 2000Hz. The acceleration of the table is dependent on the magnitude of the force generated, the mass of the table and attached test item, and the stiffness of the table suspension. The maximum free table acceleration is 100g. The extent of this motion is limited by the maximum stroke of the shaker and available voltage from the power amplifier. The maximum stroke of the shaker is 51 mm (2 in.) peak-to-peak. The power amplifier maximum output currents are 100 amps RMS for Sine operation and 90 amps RMS for Random vibration. Table 4.2 lists the components of the shaker system and describes the manufacturer, model, and if applicable, calibration dates.

Table 4.2 : Description of the shaker system

Description	Manufacturer	Model	Last Calibration	Next Calibration
Shaker	Unholtz-Dickie	S202-12	N/A	N/A
Power Amplifier	Unholtz-Dickie	SA-15	N/A	N/A
Signal Conditioner	Unholtz-Dickie	CVA-4	04/06/01	04/06/02
Controller	GenRad	2252B	04/01/01	04/01/02

The part to be tested under vibration is mounted onto the shaker table through a vibration test fixture that is designed for the current experiment to accommodate two 76 × 228 mm (3 × 9 in.) extruded polystyrene blocks. Figure 4.3 shows an isometric view of the test fixture model constructed using Pro/ENGINEER. A photograph of the actual fixture used is shown in Figure 4.4.

Figure 4.3 : Isometric view of the vibration test fixture (model)

Figure 4.4 : Photograph of the vibration test fixture

Vibration input levels are controlled using an accelerometer (Endevco Model 2226C) mounted onto the base of the vibration test fixture. A second and third accelerometer (Endevco Model 2222C) is mounted onto the base in the two planes, X and Y, perpendicular to the motion of the shaker table. The responses of these two accelerometers are monitored to verify that the motion of the test fixture was predominantly perpendicular to the planar surface of the shaker table.

To facilitate testing with vibration along the axis of the heat pipe, an L-shaped test fixture is mounted onto the shaker armature instead. The test fixture shown Figure 4.4 is then mounted onto the L-shaped test fixture for testing in the longitudinal direction. Figure 4.5 shows a photograph of this L-shaped fixture mounted onto the shaker armature.

Figure 4.5 : Photograph of the L-shaped test fixture

For testing, the heat pipes are embedded within extruded polystyrene insulation. Inserted between two 25 mm (1 in.) thick blocks of this material, the structural elastic deformation of the heat pipe is minimized under vibration. The blocks are encapsulated inside an aluminum cast casing that compresses the two blocks together. Since the overall length of the heat pipe exceeds the inner dimensions of one casing, two such cast casings are used together. Figure 4.6 shows an exploded view of the two casings with a 76 mm wide channel cutout interconnecting the two casings. The casings are mounted onto the test fixture as shown previously in Figure 4.4.

Figure 4.6 : Exploded view of the cast aluminum casings

4.4 Vibration Testing

As previously noted, the results of this study will be used as a precursor to an application-specific testing phase that focuses on cooling of printed circuit boards under harsh conditions. In these applications, the type of vibration is mostly described by a broad band of frequencies rather than a simple harmonic oscillation. As such, the type of vibration considered herein is based on Dy 4's standard guidelines for harsh environments. The vibration profile is restricted to a flat PSD spectrum of $0.1 \text{ g}^2/\text{Hz}$ between 20 and 2000Hz. The input tolerances are $\pm 3\text{db}$ below 500 Hz and $\pm 6\text{db}$ above 500 Hz. The vibration tests are performed with the heat pipe axis either parallel or perpendicular to the direction of vibration.

4.5 Thermal Setup

The thermal setup is composed of three groups of components: components that are used to create and regulate heat flow through the heat pipe; sensors that determine the temperature distribution along the heat pipe; and a data acquisition system used to record measurements.

4.5.1 Heat Load System

4.5.1.1 Heater Elements

Two resistance heater elements, Minco Model No.5275, are used to supply heat to the evaporator section of the heat pipe. Kapton polyamide film with aluminum backing and Teflon adhesive is used to bond the elements to a copper block encapsulating the evaporator of the heat pipe. The aluminum backing on each element assists in spreading the heat evenly. The total heat supplied is either limited by the adhesive or the electric power supply. For the current setup, the power per heater element is limited to 34 Watts for a plate temperature of +80°C. Exceeding this temperature level may break down the adhesive, which in turn may reduce the contact area between the heater element and the hot plate.

The power supply used for testing can produce up to 15V, 2A, and has a resolution of 0.1V, 0.01A. Two such power supplies were used to yield a combined maximum power input of 110 Watts. Taking the losses in the wires attached to the heater elements into account, the corresponding maximum heat load is 53 watts per heater element, which is greater than the limit imposed by temperature restrictions on the adhesive. For the operating temperatures considered in the present experiments, the capillary limit of the heat pipe should be reached below either of the two aforementioned limits. Thus, the capacity of the heat-load system meets the requirements of the present test matrix. Figure 4.7 shows a schematic of the heat-load system.

Figure 4.7 : Schematic of the heat-load system

4.5.1.2 Block Heater

Because of the relatively small diameter of the heat pipe, the heater elements cannot be directly applied to the heat pipe. A highly conductive block heater provides a planar surface to mount the elements, and a uniform distribution of heat along the surface of the heat pipe. The block is designed to contain a 63½ mm section of the heat pipe, and is divided into two halves. Two pieces are used to safely adjust the thermocouple wires on the evaporator. A thermal grease (OmegaTherm 201) is used between the heat pipe and block heater to minimize the thermal resistance due to diminutive air gaps. This paste does not cure or harden with temperature, hence the block heater can be removed, cleaned, and reused on other units. The block heater is sandwiched between two composite plates made of ECP+ material as shown in Figure 4.8. This ECP+ composite serves to insulate the heater elements attached to the bottom and upper halves of the block heater. A sudden rise in temperature, as experienced during dry-out, could damage the polystyrene blocks encapsulating the block heater. The ECP+ composite material is designed to withstand high temperatures and heat flux. The remainder of the space inside the aluminum casings is filled with pink fiber insulation to further reduce stray heat transfer. The overall

assembly (Figure 4.9) is fastened to the test fixture to eliminate displacements during vibration. The assembly of the block heater has been carefully designed such that the block heater can be raised or lowered as desired to prevent bending between the evaporator and the adiabatic section of the heat pipe.

Figure 4.8 : Exploded view of the block heater assembly

Figure 4.9 : Exploded view of the overall evaporator assembly

Figure 4.10 shows the fiber insulation used inside the aluminum casing at the evaporator end.

Figure 4.10 : Photograph of the evaporator setup with fiber insulation

4.5.1.3 Cooling-Jacket

A cooling-jacket is used to extract the heat from the condenser of the heat pipe, as shown in Figure 4.11. The heat is transferred to water that flows steadily through the jacket. The cooling-jacket is a 12.5 mm cavity bored into polystyrene. Two PVC $\frac{1}{2}$ " internal thread adapters at the inlet and outlet of the cooling jacket are fixed to the polystyrene using Pres-tite Green contact cement. PL300 Foam Board Adhesive is used as a seal along the edges of the two adapters and the polystyrene. A second $\frac{1}{2}$ " external thread to $\frac{1}{2}$ " Barb PVC adapter is used to connect nylon tubing for water supply and drain as shown in Figure 4.11.

Figure 4.11 : Photograph of the condenser section

For support of the heat pipe in the cooling-jacket cavity, two circular stand-offs with triangular spokes are used. The supports are embedded within the polystyrene as shown in Figure 4.12. The heat pipe is not shown in this schematic for clarity.

Figure 4.12 : Exploded view of the cooling-jacket and its components

4.5.1.4 Control of Coolant Temperature

Mixing of cold building water with water heated locally in a 36 Liter tank provides temperature control of the coolant. The hot-water tank has a single heater element with a maximum heat transfer rate of 1500 Watts and is capable of supplying water at 70°C water at a rate of up to 4 liters/min. Needle valves are used for precise adjustment of hot and cold water in the mixture that is fed to the condenser cooling-jacket of the heat pipe enclosure. The coolant water temperature is measured with thermocouples at the inlet and outlet of the cooling-jacket. The flow rate is measured with a Sho-rate 1355 series rotameter installed upstream of the cooling-jacket. The accuracy of the flow meter is $\pm 5\%$ ¹. Figure 4.13 shows a photograph of each of the aforementioned items.

Figure 4.13 : Photograph of the hot-water tank, needle valves, and flow meter

¹ Brooks Instruments; Design Specifications, DS-1350E; April 1997.

4.5.1.5 Bleed Line

During initial measurements, vapor pockets were observed in the water line feeding from the hot-water tank, causing fluctuations in the condenser temperature of the heat pipe. Such fluctuations would adversely affect the time to reach steady-state operation of the heat pipe. A bleed line was installed before the inlet of the cooling-jacket to eliminate this problem. Figure 4.14 shows this bleed line used to capture vapor pockets.

Figure 4.14 : Photograph of the bleed line

4.5.1.6 Brass Sleeve

The condenser and adiabatic portions of the heat pipe (i.e. $\approx 70\%$ of the heat pipe length) are placed in a brass tube of 4.5 mm and 5.5 mm inner and outer diameters, respectively. Thermal grease is used to fill the gap between this sleeve and the heat pipe to minimize the thermal resistance, hence temperature gradient between the heat pipe and sleeve. Thermocouples are mounted along the length of the sleeve rather than directly onto the heat pipe thereby facilitating quick replacement of heat pipes for testing.

4.5.2 Temperature Sensors

K-type Nickel-Chromium, Nickel-Aluminum beaded thermocouples are used for all temperature measurements. The thermocouples are Teflon wrapped wires each 0.25 mm (0.010 in.) in diameter and 1.8 m (72 in.) long. The tips of the thermocouples are cement mounted to the outside surface of the brass sleeve using Omegabond 101, an epoxy adhesive. They are spaced 15 mm apart along the length of the heat pipe beginning 5 mm from the throat of the adiabatic section (i.e. evaporator end). At every second position along the length of the heat pipe, two thermocouples are installed such that, relative to the direction of vibration, temperature can be measured at both the top and side of the heat pipe. An example of five thermocouples mounted onto a portion of the brass sleeve is shown in Figure 4.15. These pairs will confirm whether or not the wall temperatures are uniform about the circumference of the heat pipe.

Figure 4.15 : Thermocouple placements on sleeve

To reduce the possibility of leaks in the cooling-jacket, thermocouple wires on the condenser are directed through the outlet of the cooling-jacket to a tee on the discharge line. The foam adhesive used to bond the two polystyrene blocks in each of the two aluminum cast casings seals the thermocouple wires in the adiabatic section.

Three thermocouples are mounted onto the evaporator section of the heat pipe using a conservative amount of Kapton tape. These thermocouples are positioned at distances 15 mm, 35 mm, and 55 mm from the end of the heat pipe. The exact placements are measured and recorded at the time of installation.

All thermocouples are labeled numerically with an alphanumeric prefix. Thermocouples in the condenser are prefixed with a 'C', thermocouples in the adiabatic section are prefixed with the letter 'A', and the three thermocouples in the evaporator section are labeled E1, E2, and E3, where E1 is near the end of the heat pipe the farthest away from the adiabatic section.

4.5.3 Temperature Data Acquisition System

A Keithley Model 2700 Multimeter/Data Acquisition system manufactured by Keithley Instruments is used to acquire and digitize the thermocouple signals. The system can measure up to 40 differential analog signals of which only 23 of these channels are needed to measure the thermocouple outputs of the present setup. The analog input voltages are all preconditioned (amplified) as appropriate and then digitized with a 22-bit type Dual Slope Integrated A/D converter and the digitized voltages are converted to a temperature reading. Manufacturer's specifications for the uncertainty introduced by the digitization process and the unit conversion of the signal is $\pm 1.0^{\circ}\text{C}$. The system has a single A/D converter for all 40 channels. As such, the analog signals on these channels are digitized sequentially. The system can acquire samples at a rate of up to 600 samples/sec. However, to reduce noise, the A/D converter was set to acquire 50 samples/sec as recommended in the user's manual. The samples from each thermocouple collected over a period of approximately 0.2 seconds are then averaged to determine the time-mean value of the temperature measurements. The duration of sampling is chosen to be greater than the period of fluctuations in the measured temperature signal, and to yield a sufficiently large

sample count to produce reliable time-mean values. For the range of vibration frequencies that were investigated, the thermal inertia of the thermocouple sensor would be too large to follow the temperature fluctuations caused by the vibrating environment. Thus temporal analysis of the measured temperature data was not attempted.

4.5.3.1 Data Logging and File Storage

The data collected by the data acquisition system is first stored in on-board memory and then transmitted to a personal computer through an RS-232 Serial port at a baud rate of 9600. The software used to control the off-the-shelf data acquisition system is Keithley XILINX. This start-up software provides an environment in which the user can set the number of channels being measured, the sampling rate for each channel, and the number of samples to be collected on each channel. The software also provides the user a choice of thermocouples including B, E, J, K, N, R, S, and T.

After storing the collected data on the hard drive of the computer, data interpretation and analysis are performed using a spreadsheet, Microsoft Excel.

4.6 Quantifying Q_{in}

In order to determine the capillary limit of a heat pipe, the rate at which heat is applied at the evaporator, Q_{in} , is required. In steady-state operation, the heat influx in the evaporator is equal to the heat efflux in the condenser, provided the remainder of the length of the heat pipe is perfectly insulated. Thus, one method of determining Q_{in} is to measure the flow rate and temperature rise of the cooling water in the cooling jacket where heat is extracted from the condenser. Based on the current setup and properties of water, the resulting temperature rise is typically less than 0.5°C depending on the rate of heat transfer at the evaporator and capillary limit of the heat pipe tested.

Because of the uncertainty associated with temperature measurements, an alternative method to determine Q_{in} is thus required.

To determine Q_{in} on the basis of the heat input provided by the heater elements, one must account for losses. These losses, or stray heat transfer, occur because of the temperature difference between the evaporator and ambient conditions of the setup. Large amounts of insulation would be required in order to reduce these losses to negligible amounts. The alternative is to quantify the stray heat transfer, Q_s , and subtract it from the applied heat load, Q_u , with the heater elements.

4.6.1 Stray Heat Transfer Q_s

The stray heat transfer in the system can be quantified one of two ways: through empirical testing or numerically, e.g. by Finite Element Analysis (FEA). As described in the following subsections, both approaches aim to determine an effective thermal resistance.

4.6.1.1 Empirical Testing

A series of tests without the heat pipe embedded inside the housing were conducted to characterize the unit and estimate the heat lost to the surroundings. Using the power applied at the heater element and the temperature rise between the lower copper block and the base plate (vibration test fixture, Figure 4.3), an effective thermal resistance of the system can be written as,

$$R_s = \frac{T_{Cu} - T_{Base}}{Q_u} \quad (4-1)$$

where,

R_s	effective thermal resistance of the system
T_{Cu}	lower copper block temperature
T_{Base}	base plate temperature directly below the casing
Q_u	applied heat load

Having a large mass, the base plate temperature tends to be insensitive to the temperature field in the housing of the heat pipe setup. Thus, using the temperature of the base plate as the lower ambient temperature that drives the stray heat transfer from the unit is a suitable choice. Brief fluctuations in the laboratory temperature also have little effect on the base plate temperature.

The total power applied at the copper blocks is estimated using the voltage applied and current drawn by the heater elements. The uncertainty of this applied heat load is largely dependent on the resolution of the current given by the power supply (0.01 A). The voltage applied is measured using a multi-meter that displays up to 0.1 mV precision. The readings of one multi-meter are then compared to two others as a precaution. The variations were noted to be less than 10 mV. The resultant uncertainty in Q_a is less than 0.1 Watts, which is small as percentage of heat loads applied to the heat pipes during the experiments.

After the heat load is applied at each copper block and the voltage and current are measured, temperatures are recorded until the system reaches equilibrium. The temperature difference between the copper block and the base plate is then used to construct the plot given in Figure 4.16. The plot illustrates the effective thermal resistance of the system as a function of block temperature. The error bars presented in the figure are determined on the basis of uncertainty in Q_a , T_{cu} , and T_{base} .

The non-zero slope indicates that the effective thermal resistance of the system varies with temperature. An increase in copper block temperature yields a reduction in resistance. This slight variation is accredited to the fact that the thermal conductivity of the materials used in the setup is dependent on temperature. The material in the evaporator section of the unit is largely composed of fiberglass. The thermal resistance of this insulation varies with temperature since the thermal conductivity of the air trapped between the glass fibers is sensitive to temperature. Based on the

four data points collected, the trend in Figure 4.16 is consistent with the expectation of increased thermal conductivity of air with increasing temperature.

Figure 4.16 : Plot of R_s as a function of copper block temperature

For a given copper block temperature, the effective thermal resistance of the system can be calculated using the equation of the curve presented in Figure 4.16. With this estimate of R_s and the temperature difference ($T_{Cu} - T_{Base}$), the stray heat loss can then be estimated using,

$$Q_s = \frac{T_{Cu} - T_{Base}}{R_s} \quad (4-2)$$

4.6.1.2 Finite Element Analysis

Another method for determining the stray heat transfer of the system is through numerical analysis. By creating a computer model of the setup and using the method of finite elements to

predict the temperature difference between the base plate and copper block, the effective thermal resistance of the system can be estimated using Equation (4-1).

A model representing the setup was created, constrained, and solved using Pro/MECHANICA.

Figure 4.17 shows an exploded view of the assembly used. Table 4.3 following the figure gives a description of the numbered items.

Figure 4.17 : Exploded view of the FEA model at the evaporator end of the setup

Table 4.3 : Description of parts

No.	Description
1	Aluminum Casing (T-6061)
2a	Nylon Nut (#6)
2b	Stainless Steel Nut (#6)
3	Fiber Insulation (b/w ECP+ Composite and Casing)
4	ECP+ Composite
5	Copper Block
6	Fiber Insulation (Top)
7	Stainless Steel Rod (6-32)
8	Fiber Insulation (b/w Casing and Base)
9	ECP+ Standoff
10	Polystyrene (b/w Cu Blocks and Housing)
11	Fiber Insulation (b/w Cu Blocks and End)
12	Fiber Insulation (End)
13	Fiber Insulation (b/w End and Casing)
14	Fiber Insulation (surrounding rod)
15	Fiber Insulation (b/w ⑭ and Casing)
16	Polystyrene Housing (Lower)
17	Polystyrene Housing (Upper)

The material properties of each component were specified based on the temperature range of the copper block. Table 4.4 lists the thermal conductivity ranges of the materials used in the setup. It should be noted that available datasheets did not specify a thermal conductivity value for some components within the temperature range of the setup. The thermal conductivity of these components was therefore estimated on the basis of conductivity values available at temperatures closest to the temperature range in question. Furthermore, for the extruded polystyrene, the thermal conductivity of the material was calculated from the R-value specified by the manufacturer. In order to calculate the thermal conductivity, the inverse of the R-value and material thickness representing the R-value was used. Because the R-value of a material is

expressed as an integer and has been rounded either up or down by the manufacturer, the lowest and highest possible value for the thermal conductivity of the material was used in the FEA model. Lastly, it was found that a number of sources for thermal conductivity properties conflicted with each other. The value chosen was based on the value tabulated most often.

Table 4.4 : Material thermal conductivity

Material	k [W/m·K]
ECP+	0.36 – 0.40
Fiber Insulation	0.035 – 0.068
Extruded Polystyrene	0.029 – 0.036
Aluminum (Die cast 360)	146
Stainless Steel	14.0 – 16.2
Nylon	0.25 – 0.27

A temperature constraint was applied uniformly on the outer surfaces of the assembly. For simplicity, this temperature constraint was set to 0°C. Heat flux was specified over the area of the copper blocks interfaced to the heater elements. The model was then meshed and solved. The guidelines for mesh and node distributions and convergence criteria are described in Appendix C. The resulting temperature rise ($T_{Cu} - T_{Base}$) was then used to determine the effective thermal resistance of the system. Because of variations in thermal properties with temperature, two simulations (one with the lower k values of Table 4.4 and one with the higher values) were performed to capture the effect of these variations on the effective thermal resistance of the system. Figure 4.18 compares the results obtained from empirical testing to the results obtained through numerical analysis.

Figure 4.18 : Empirical vs. numerical results for R_s

The plot shows a consistent difference between the two methods. This difference is judged to be due to the uncertainty in the thermal conductivity values used in the FEA model. The fiber insulation used in the current setup is essentially designed to create air pockets. The extent of packing of the fiber insulation affects the size and number of air pockets, thereby effectively changing the convective currents, hence, the effective thermal conductivity of the material. The fact that the difference between the measured and predicted values of R_s is independent of temperature as per results in Figure 4.18 lends support to this argument.

In light of these results, the effective thermal resistance for the system was estimated using the results of empirical testing.

4.7 Procedure

The procedure developed for testing is presented in three parts. The first is the procedure used to build the housing, the second describes the installation of the heat pipe and setup prior to testing, and the third discusses the measurement procedure followed for each test.

4.7.1 Construction of the Housing

1. The surface of the brass sleeve to be embedded inside the housing is first cleaned using either a lather or alcohol.
2. Thermocouples are then mounted to the wall of the sleeve using Omega Bond 101 Epoxy adhesive. The thermocouples are spaced equally 15 mm apart and situated such that they are installed at the same circumferential position. Every 2nd thermocouple is mounted in pairs as shown in Figure 4.15. The epoxy adhesive is allowed to cure for 24 hours before continuing with the assembly of the setup.
3. For support of the condenser portion of the heat pipe in the cooling jacket (Figure 4.12), circular stand-offs are installed at the end and midpoint of the condenser section. Thermocouple wires on the condenser portion of the sleeve weaved through these supports towards the outlet of the housing.
4. The polystyrene blocks are prepared as shown in Figure 4.12. Contact cement is used to bond the ½" INT THD fitting at the inlet of the cooling jacket. The second ½" INT THD fitting is not installed at this time. This second fitting is installed *after* the heat pipe and components are sandwiched inside the polystyrene blocks.

5. The sleeve is then placed inside a groove etched out in the polystyrene housing. The two supports are positioned inside etched grooves in the cooling jacket of the housing as seen in Figure 4.12. To avoid adjustments to the bonded thermocouple ends, the sleeve is rotated such that the majority of thermocouple wires along the adiabatic portion of the heat pipe rest on the polystyrene block. The few thermocouple wires originating on top of the sleeve are carefully adjusted along side the other thermocouple wires to prevent damage to the thermocouple bond.
6. PL300 foam adhesive is then applied to the mating surfaces of the two polystyrene halves except in close vicinity of the cooling jacket to prevent adhesive from spreading into the cavity.
7. The sleeve, thermocouple wires, and circular supports are then sandwiched between the two polystyrene blocks.
8. The polystyrene housing is placed inside the two aluminum casings that would already be mounted to the vibration base plate. The lids of the casings are then torqued with 6-32 socket cap screws to compress the unit and seal the brass sleeve and its components inside.
9. The second ½" INT THD fitting is bonded to the outlet of the cooling jacket using contact cement. For seal, foam adhesive is applied to the edges of both INT THD fittings in contact with the housing. The contact cement and foam adhesive are allowed to cure for 24 hours before continuing.

4.7.2 Installation of the Heat Pipe and Setup

1. Removing the unit from inside the aluminum casings, the heat pipe is inserted into the brass sleeve. Before doing so, Omegatherm 201 thermal grease is injected into the sleeve until the sleeve is completely filled.
2. The heat pipe is then carefully inserted into the sleeve, crimped end first, through the evaporator end of the sleeve. Inserting the heat pipe slowly prevents voids that could occur between the heat pipe and sleeve. Frictional forces, if forced quickly, could shear grease away from the inside wall of the sleeve. Excess amounts of grease protruding from the opposite end in the cooling jacket can be recaptured for later use. Any grease remaining inside the cooling jacket is removed using a q-tip.
3. Once the heat pipe has been inserted, the evaporator length of the heat pipe is measured using either a dial caliper or a straight edge. The evaporator length is fixed at $63\frac{1}{2}$ mm for the current setup.
4. Three thermocouples, E1, E2, and E3 are then installed onto the evaporator section of the heat pipe using Kapton tape. Conservative amounts of tape are used to prevent inadequate contact between the heat pipe and copper blocks.
5. Prior to installing the block heater, heater elements would have been placed on the top and bottom halves. The first, on the bottom half at the bottom surface, and the second on the top half at the top surface. The +/- leads of the heater elements are situated such that the wires point directly away from the heat pipe.
6. Each copper block is then snapped into the ECP+ insulating material specifically designed to sandwich the copper blocks.

7. The lower half of the evaporator section is then assembled using eight stainless steel rods, and 24 nylon nuts (Figure 4.9). 8 of the nuts act as standoffs separating the copper block from the base of the casing. The 8 nuts acting as standoffs are adjusted along the stainless steel rod until the bore in the lower copper block is aligned with the brass sleeve inside the housing. The heat pipe when secured must be centered inside the brass sleeve and block heater.
8. An angle locator is used to check the alignment of the copper block with respect to the base plate.
9. Once the placement of the copper block is verified, the block is removed and lightly packed insulation is placed between the lower copper block and the aluminum casing.
10. The unit containing the heat pipe is then reinserted into the casing. The heat pipe is positioned within the groove of the lower copper block. Thermal grease may be applied between the heat pipe and copper block for increased thermal efficiency.
11. The leads of the evaporator thermocouples are placed into the grooves on the copper block. The upper copper block heater is then inserted, encapsulating the heat pipe inside. 8 additional nylon nuts are used to fasten the ECP+ and copper blocks to the casing. Additional amounts of insulation are used to fill any remaining air gaps between the copper blocks and the housing.
12. The lids of the aluminum casing are installed compressing the unit inside.
13. The base plate is attached to the shaker table.
14. Thermocouple +/- leads are attached to the thermocouple input card. Channel numbers are recorded against the placement of the thermocouple on the heat pipe.

15. The acquisition system is triggered to verify that each thermocouple is functional.
16. The coolant water lines are attached to the inlet and outlet fittings of the cooling jacket.
 $\frac{1}{2}$ " barbed fittings are used to join the nylon coolant lines to the inlet and outlet of the cooling jacket. The inlet and outlet water lines are each monitored with a thermocouple. When attaching the inlet and outlet water lines, the placement of the thermocouple in the stream is verified.
17. The +/- leads of the heater elements are attached to the DC power supply.

4.7.3 Test Procedure

1. The data acquisition system is triggered to begin recording temperatures.
2. The coolant water flow is turned on to a maximum of 450 cc/min. Higher flow rates should be avoided as they may cause leaks in the coolant line at the barbed fittings.
3. Using the acquisition system, the temperature of the coolant is monitored. In order to achieve a desired coolant temperature, adjustments to the mixture of hot and cold water are made using the pin-needle valves on the hot and cold water lines.
4. Once the desired coolant temperature is reached, the DC power supply is powered on and the voltage applied is measured using a multi-meter. The current drawn is measured using the display on the power supply.
5. The heat pipe is then allowed to reach steady state and the evaporator is monitored for sharp increases in temperature.

6. If the heat pipe reaches steady state without any indication of dry-out, the voltage is increased and measurements for both voltage and current are re-recorded. Thermocouple readings in the evaporator are continuously monitored for failure.
7. Steps 5 and 6 are repeated until the operating limit of the heat pipe is reached. Sharp increases in temperature at the evaporator are used as signs of dry-out. The effective thermal resistance, $R_{E/A3}$, is plotted to demonstrate the increase in evaporator temperature, $E1$, with respect to the operating temperature of the heat pipe, $A3$.
8. If a sharp increase in temperature is seen, the power supply is powered down and the heat pipe is allowed to return to near-coolant temperatures. The heat load is then reapplied at a lower level and the test is repeated.
9. Once the capillary limit is verified, the two closest data points that respectfully qualify as PASS and FAIL is used. The separation between the two data points contributes to the uncertainty in the capillary limit.
10. Steps 4 through 8 are repeated under vibration. The duration of a test can vary from 4 to 6 hours depending on the number of iterations needed to verify the operating limit of the heat pipe and the tentative effects of vibration on the operating limit.

5 Experimental Results

This chapter discusses the results obtained in the experiments. The presentation of the data begins with the results found under non-vibrating conditions. For each heat pipe, these results are subsequently used as a reference to identify the effects of vibration on the capillary limit. For both vibrating and non-vibrating conditions, the capillary limit is presented for transverse and longitudinal orientations of gravitational force, with the latter being in both favorable and unfavorable directions. Based on measurements at multiple operating temperatures with a range of wick structures, sensitivity of the capillary limit to these factors is established.

5.1 Results

5.1.1 Sample Data Set

In order to determine the capillary limit of a given heat pipe, a series of transient measurements are required. By determining the effective thermal resistance between points E_1 and A_3 , as described in Section 4.7.3, failure is defined as an abnormal increase in this quantity. As an example, Figure 5.1 displays the temporal variation of the effective thermal resistance of the axial groove heat pipe under a fixed heat load. Following start-up at ambient conditions, the heat pipe is noted to reach thermal equilibrium after approximately 800 seconds. At this point, the effective thermal resistance of the heat pipe is seen to converge at 0.54°C/W . In this example of operation under 6.5 Watts of heat load, the heat pipe is said to be functioning properly. In the experiment, once the heat pipe reached steady state, the operating temperature, T_{A3} , was recorded.

Figure 5.1 : Example of a properly operating heat pipe

Figure 5.2 shows an example of a failed heat pipe. Continuing from the previous test of Figure 5.1, the power supply was adjusted such that the heat load at the evaporator was increased by 0.1 Watts. A sudden increase in thermal resistance is noted approximately 250 seconds after the increase in heat load. The capillary limit of the axial groove heat pipe at an operating temperature of 37.0°C is therefore said to be between 6.5 and 6.6 Watts. It should be noted that the duration of each such test varied depending on the time the heat pipe reached thermal equilibrium and the number of increments in heat load that was considered in search of the point of failure.

Figure 5.2 : Example of a heat pipe experiencing thermal failure

The tests summarized in Figures 5.1 and 5.2 together allow identification of the operating limit, and form one data set. Depending on the initial estimate of the capillary limit, often more than two data points are needed in a data set to deduce the capillary limit to within 0.1 Watts. The following figure provides a typical example of two data sets of different operating temperatures, containing four and nine data points, respectively.

Figure 5.3 : Capillary limit of the axial groove heat pipe

Each data point is labeled as PASS or FAIL based on whether or not the temporal variation of $R_{EI/A3}$ followed the trend in Figure 5.2. The dashed line, identified as a trend line in Figure 5.3, illustrates the capillary limit of the heat pipe over a temperature range described by the two data sets. The chosen operating temperature range corresponds to Dy 4 qualification requirements. Because of the narrow temperature range in question ($\sim 50^{\circ}\text{C}$), only two data sets are used in forming the trend line. The assumed linear variation between the data sets is supported by analytical results, as discussed in Section 5.1.2.1.2. When establishing the trend line in each data set, the midpoint between the two closest data points that respectively qualify as PASS and FAIL is used. The separation between the two data points contributes to the uncertainty in the capillary limit.

For the sake of clarity, heat pipe performance results presented in the following sections are based on the trend lines only. Detailed plots that present the trend lines together with the data sets are included in Appendix D.

5.1.2 Non-vibrating Test Results

5.1.2.1 Results with Transverse Gravitational Force

Figure 5.4 presents the capillary limit trend line for heat pipes with different wick structures. In these cases, the gravitational force was oriented 90° (i.e. transverse) to the axis of the heat pipe.

Figure 5.4 : Capillary-limit trend lines for the four heat pipes under transverse gravitational force

It is noted that the capillary limit for the sintered powder heat pipe increases from 4.5 to 10.5 Watts between operating temperatures of 18°C and 74°C . This illustrates the substantial influence

operating temperature has on the capillary limit. In this context, the capillary limit of the fiber bundle heat pipe increases from 5.6 to 15.2 Watts between 29°C and 73°C, and is thus the most sensitive to the operating temperature amongst the four heat pipes tested. In contrast, the wrapped screen heat pipe displays the lowest sensitivity, with its capillary limit rising from 4.1 to 7.6 Watts between 28°C and 73°C.

The variations in sensitivity of the capillary limit to operating temperature, indicated by differences in the slopes of the trend lines, is directly related to the geometry of the wick. The type and design of the wick structure influences the capillary limit through the amount of pressure loss experienced along the length of the heat pipe and the total capillary force produced.

The capillary force generated in a heat pipe is controlled by the surface tension of the working fluid and the resulting meniscus radius in the evaporator. The discrete corners or abrasions in the wick structure dictate the size of the meniscus radius. As described in Chapter 2, the smaller the radius of meniscus in the evaporator, relative to the condenser, the more 'pull' is created throughout the saturated wick. Thus, one way to improve the maximum allowable rate of heat transfer in a heat pipe is to design a wick structure that creates very small meniscus radii in the evaporator. This is the principle behind wrapped screen and sintered powder wick designs.

However, the capillary limit is based on a force balance. An increase in capillary pumping force through a fine wick structure also leads to a reduction in permeability, hence an increase in resistance to liquid flow through the wick along the length of the heat pipe. For sintered powder or wrapped screen heat pipes, the liquid pressure drop through the wick is greater than for an axial groove or fiber bundle heat pipe having the same overall dimensions. Depending on the operating temperature, this higher flow resistance may more than offset the benefits brought about by a smaller meniscus radii.

5.1.2.1.1 Theoretical Analysis

The above noted assertions are best illustrated by the mathematical expression for the capillary limit of a heat pipe, restated here in Equation (5-1). This equation is applicable to a conventional heat pipe in the absence of body forces along the axis of the pipe.

$$Q_{c,max} = \frac{\frac{2\sigma}{r_{eff}}}{L_{eff} \cdot (F_l + F_v)} \quad (5-1)$$

In the equation, σ is the liquid surface tension of the fluid, r_{eff} is the effective capillary radius, L_{eff} is the effective length of the heat pipe, and the liquid and vapor frictional terms, F_l and F_v , are defined as:

$$F_l = \frac{\mu_l}{KA_w \rho_l \lambda} \quad (5-2)$$

$$F_v = \frac{(f_v Re_v) \mu_v}{2r_{h,v}^2 A_v \rho_v \lambda} \quad (5-3)$$

where,

μ_l, μ_v	liquid and vapor dynamic viscosities
K	wick permeability
A_w, A_v	wick and vapor cross sectional areas
ρ_l, ρ_v	liquid and vapor densities
λ	latent heat of vaporization of the working fluid
$r_{h,v}$	hydraulic vapor radius
f_v	vapor friction coefficient
Re_v	Reynolds number based on density, viscosity, hydraulic diameter, and mean velocity of vapor flow

Substituting the expressions of F_l and F_v into Equation (5-1) and regrouping the terms in the denominator based on association with geometry and fluid properties, one obtains:

$$Q_{c,\max} = \frac{\frac{2\sigma}{r_{eff}}}{\frac{\mu_l}{\rho_l \lambda} \cdot \frac{L_{eff}}{KA_w} + \frac{(f_v Re_v) \mu_v}{\rho_v \lambda} \cdot \frac{L_{eff}}{2r_{h,v}^2 A_v}} \quad (5-4)$$

$$Q_{c,\max} = \frac{\alpha_c}{\alpha_l \cdot \alpha_{g,l} + \alpha_v \cdot \alpha_{g,v}} \quad (5-5)$$

where,

- α_c capillary force parameter
- α_l liquid resistance parameter pertaining to fluid flow parameters
- $\alpha_{g,l}$ liquid resistance parameter pertaining to geometry
- α_v vapor resistance parameter pertaining to fluid flow parameters
- $\alpha_{g,v}$ vapor resistance parameter pertaining to geometry

To demonstrate the relative significance of these α parameters with respect to the performance of a heat pipe, Equation (5-5) is applied to the axial groove and wrapped screen heat pipes next.

All four heat pipes tested in this study are commercial type heat pipes designed for large volume production. Because of the nature of fabrication, information pertaining to the design and geometry is not readily available. Geometry measurements were thus performed by sectioning the axial groove and wrapped screen heat pipes that were equal in size and design to those tested.

Images of the cross-sections were captured using an Ersa-scope as previously discussed in Section 4.2. Appendix A contains example photographs of cross-sections obtained for the axial groove and wrapped screen heat pipes. Tables 5.1 and 5.2 summarize the dimensions acquired through measurements from these photographs. The uncertainties tabulated for each quantity are based on the repeatability of the measurements.

Table 5.1 : Axial groove dimensions

Property	Measurement	Uncertainty
Number of Grooves, N_g	40	-
Groove Shape	Trapezoidal	-
Groove Width, w	0.157 mm	± 0.025
Groove Depth, δ	0.127 mm	± 0.013
Outside Diameter of Pipe, d_o	4.00 mm	± 0.03
Pipe Wall Thickness, t_w	0.49 mm	± 0.03
Half Angle of Groove, β	16°	$\pm 4^\circ$

Figure 5.5 : Cross-section of axial groove**Table 5.2 : Wrapped screen dimensions**

Property	Measurement	Uncertainty
Number of Layers	1.10	± 0.05
Wire Diameter, d_w	0.044 mm	± 0.006
Wire Spacing, w	0.075 mm	± 0.006
Outside Diameter of Pipe, d_o	3.99 mm	± 0.03
Pipe Wall Thickness, t_w	0.38 mm	± 0.02

Figure 5.6 : Schematic view of screen

The mesh number for the wrapped screen heat pipe, defined as the number of openings per unit length (m^{-1}), was estimated by measuring the wire diameter and spacing one of three ways. First by measuring the wire diameter and spacing independently, secondly by measuring the wire diameter and spacing together and using the total to calculate the mesh number, and lastly, using the hypotenuse of a square to calculate the sum of the wire diameter and spacing. A mesh number of 7874 m^{-1} (200 in^{-1}) was obtained using the third method. As this is a round number in imperial units, and is reasonably close to those obtained with the other measurement techniques (8403 and 8000 m^{-1}) the mesh number of the wrapped screen heat pipe was taken to be 7874 m^{-1} .

The number of layers was estimated based on examining the cross-section after the cut. A slight overlap was noticed placing the effective number of layers somewhere between 1 and $1\frac{1}{2}$. The exact value of 1.1 was determined by extracting the mesh from the heat pipe, unwrapping it, and measuring its width. This width was then compared to the inner circumference of the heat pipe, yielding the fractional overlap of the mesh.

Using the quantities tabulated in Tables 5.1 and 5.2, the permeability, effective capillary radius, vapor core diameter, and wick cross-sectional area were evaluated for both heat pipes. For this,

the correlations listed in Table 5.3 were used. It is important to note that the correlations listed for the wrapped screen wick structure do not take the presence of scattered sintered particles, as noted in Appendix A, into account. The resultant values of $\alpha_{g,v}$ and $\alpha_{g,l}$ for the two heat pipes are listed in Table 5.4. The $\alpha_{g,v}$ parameter is noted to be very similar in the two cases, whereas $\alpha_{g,l}$ is seen to be roughly five times greater for the wrapped screen heat pipe.

Table 5.3 : Analytical expressions for r_{eff} , L_{eff} , K , and ε

Parameter	Axial Groove	Wrapped Screen
Effective Capillary Radius, r_{eff}^*	$\frac{w}{\cos \beta}$	$\frac{w + d_w}{2} = \frac{1}{2N}$
Effective Length, L_{eff}	$\frac{1}{2}L_e + L_a + \frac{1}{2}L_c$	
Permeability, K	$\frac{2\varepsilon(r_{h,l})^2}{f_l Re_l}$	$\frac{d_w^2 \varepsilon^3}{122(1-\varepsilon)^2}$
Porosity, ε	$\frac{w - \delta \tan \beta}{s}$	$1 - \frac{1.05\pi N d_w}{4}$

Table 5.4 : Liquid and vapor geometry parameters

Wick	$\alpha_{g,l} \times 10^{15}$	$\alpha_{g,v} \times 10^9$
Axial Groove	0.39 ± 0.05	4.11 ± 0.09
Wrapped Screen	1.99 ± 0.23	4.03 ± 0.08

* The effective capillary radius expression for the axial groove applies to both triangular and trapezoidal grooved heat pipes, provided the depth of the groove is no less than half the groove width.

Calculation of the liquid fluid flow parameter, α_l , is straight forward as it is evaluated using the thermodynamic properties of water alone. Calculation of the vapor fluid flow parameter, α_v , is more involved since the frictional term, $f_v Re_v$, is required. For a conventional heat pipe with circular passage, this term is assumed to be equal to 16 (Chi, 1976). To verify this value, a simple steady-state, steady-flow energy balance was performed for a control volume enclosing the evaporator as shown in Figure 5.7.

Figure 5.7 : Control volume enclosing the evaporator

Neglecting kinetic and potential energy terms, the energy balance for this control volume reduces to:

$$Q_{in} = -\dot{m}_l h_l + \dot{m}_v h_v \quad (5-6)$$

Taking the vapor and liquid to be at their respective saturated states, and recognizing that for steady flow $\dot{m}_l = \dot{m}_v$, we obtain:

$$\begin{aligned} Q_{in} &= \dot{m}_v (h_v - h_l) \\ &= \dot{m}_v \lambda \end{aligned} \quad (5-7)$$

From this expression, $\dot{m}_v$ can be determined using the measured value of Q_{in} and the λ value obtained on the basis of the evaporator vapor temperature. Then the Reynolds number for the vapor flow is determined from,

$$Re_v = \frac{2\dot{m}_v r_{h,v}}{A_v \mu_v} \quad (5-8)$$

with the dynamic viscosity, μ_v , evaluated at the evaporator temperature.

The axial thermal resistance through the vapor core is orders of magnitudes smaller than the resistance through the pipe and wick structure (Peterson, 1994). As such, the vapor temperature through the heat pipe is nearly isothermal. Therefore the latent heat of vaporization, λ , and dynamic viscosity, μ_v , may also be evaluated using the operating temperature, T_{A3} , of the heat pipe measured at position A3 (Figure 4.15, Section 4.5.2) with little loss of accuracy.

Using this procedure, the value of Re_v for the axial groove and wrapped screen heat pipes was determined to be 92 and 64, respectively. As these Reynolds numbers are well below the critical values for transition in fully-developed pipe flows, the vapor flow is laminar. Thus, the vapor friction coefficient, f_v , can be calculated using the analytical expression for laminar, fully developed pipe flows,

$$f_v = \frac{16}{Re_v} \quad (5-9)$$

Hence, the vapor friction term, $f_v Re_v$, appearing in Equation (5-5) is 16 for both the axial groove and wrapped screen heat pipes. With the evaluation of $f_v Re_v$, all information required to complete

the calculation of all α parameters appearing in Equation (5-5) is now available. The resultant α values are summarized in Table 5.5 for a series of operating temperatures.

Table 5.5 : Geometry and flow parameters for use in Equation (5-5)

	Axial Groove Wick Structure			Wrapped Screen Wick Structure		
T_v [°C]	$\alpha_l \alpha_{g,l}$ [N/m²·W]	$\alpha_v \alpha_{g,v}$ [N/m²·W]	α_c [N/m²]	$\alpha_l \alpha_{g,l}$ [N/m²·W]	$\alpha_v \alpha_{g,v}$ [N/m²·W]	α_c [N/m²]
20	162	15.0	890	821	14.7	2290
30	131	9.0	872	664	8.8	2243
40	108	5.2	852	549	5.1	2192
50	92	3.5	833	464	3.5	2142
60	79	2.4	812	400	2.3	2088
70	69	1.6	790	351	1.6	2032
80	62	1.1	768	313	1.1	1975

Over the operating temperature range in question, the frictional resistance of the vapor flow, represented by $\alpha_v \alpha_{g,v}$, is noted to be very small in comparison to that of the liquid flow for the wrapped screen heat pipe. For the axial groove heat pipe, the liquid frictional resistance, identified by the $\alpha_l \alpha_{g,l}$ term, is significantly lower. As such, the vapor frictional resistance has a relatively larger influence on the capillary limit of the axial groove heat pipe, particularly at the lower end of the operating temperature range. Nonetheless, the ratio of $\alpha_l \alpha_{g,l}$ to $\alpha_v \alpha_{g,v}$ is still greater than one order of magnitude for this heat pipe.

Although the frictional forces affecting liquid flow are significantly larger for the wrapped screen wick, this adverse feature is countered by the larger capillary forces evident in the relative values of α_c .

Between the two sets of parameters, $\alpha_l \alpha_{g,l}$ and α_c , which dominantly affect the value of $Q_{c,max}$, $\alpha_l \alpha_{g,l}$ is noted to decrease more rapidly with operating temperature than α_c . Consequently, the capillary limit is expected to increase as the operating temperature increases, as was noted in the experimental results.

5.1.2.1.2 Theoretical vs. Experimental Results for Axial Groove and Wrapped Screen Heat Pipes

Having completed the calculation and analysis of all parameters upon which $Q_{c,max}$ depends as per Equation (5-5), the predicted $Q_{c,max}$ values are compared to experimental results in Figures 5.8 and 5.9.

Figure 5.8 : Capillary limit of the axial groove heat pipe under transverse gravitational force – theoretical vs. experimental results

Figure 5.9 : Capillary limit of the wrapped screen heat pipe under transverse gravitational force – theoretical vs. experimental results

As illustrated for both heat pipes, the predicted capillary limit over the tested temperature range varies linearly. This justifies the use of two data sets to establish the experimental capillary limit trend line between 15°C and 75°C. The uncertainties in the experimental results were determined as per the procedure outlined in Section 4.7.3. The uncertainty of the theoretical $Q_{c,max}$ values on the other hand, were determined by varying the various geometric and flow parameters (Equation 5-5) within their respective uncertainty ranges (e.g. Table 5.1 and Table 5.2). Details on the latter are provided in Appendix E.

As for the agreement between the predicted and measured capillary limits, the observations in the two figures are encouraging. The rate of change of $Q_{c,max}$ with T_{A3} is captured fairly accurately for both heat pipes, particularly for the wrapped screen wick structure. The $Q_{c,max}$ values for the axial groove heat pipe are predicted within the uncertainty range of the analytical results. However, the wrapped screen heat pipe $Q_{c,max}$ values are consistently under predicted. This is to be expected

given that the mathematical model does not take the sintered particles of the tested heat pipe into account. In the presence of such particles, the effective porous radius should be lower. As such, the capillary force produced would be higher, thereby increasing the capillary limit of the heat pipe. Arguably, the porosity of the wick is affected adversely by the presence of these particles. Thus some of the improvement in performance brought about by the increased capillary force would be offset by an increase in liquid flow resistance.

Having established a good agreement between the experimental and theoretical $Q_{c,max}$ values in the absence of gravitational forces, the differences in $dQ_{c,max}/dT_{A3}$ (i.e. the slopes of the trend lines for the respective heat pipes) can be readily explained through Equation (5-5) and Table 5.5. As demonstrated in Table 5.5, $\alpha_v\alpha_{g,v}$ has a relatively small effect on the $Q_{c,max}$ value. The capillary limit is thus dictated primarily by the capillary and liquid viscous forces, specifically $\alpha_c/\alpha_l\alpha_{g,l}$. This ratio is noted to increase more rapidly with T_{A3} for the axial groove heat pipe than for the wrapped screen heat pipe. α_c is roughly three times smaller for the axial groove heat pipe and does not vary significantly over the temperature range in question. Therefore, one concludes that the greater value of $dQ_{c,max}/dT_{A3}$ of this heat pipe is brought about by $\alpha_l\alpha_{g,l}$ being significantly lower.

Based on the trend lines presented in Figure 5.4, it is straightforward to identify the heat pipes with favorable thermal performance. Since all four heat pipes deliver fairly similar $Q_{c,max}$ values at the low end of the operating temperature range, those pipes with greater $dQ_{c,max}/dT_{A3}$, i.e. the axial-groove and fiber bundle heat pipes, would be the preferred choices.

5.1.2.2 Results with Longitudinal Unfavorable Gravitational Force

When the gravitation force or any other body forces act along the longitudinal axis of a heat pipe, they must be included in the balance of forces of the working fluid that dictates the maximum heat the pipe can transfer. Thus, in these instances the steady-state operating point of the heat pipe

at a given temperature is determined by the balance between capillary, frictional, and body forces acting on the working fluid.

When body forces act along the longitudinal axis of the heat pipe, two possibilities exist: the body force is either in the direction of liquid flow or acts in the opposite direction. The latter is considered unfavorable since the body force, which is much larger in the liquid state than vapor, opposes the capillary driving force.

The experiments in the unfavorable longitudinal orientation of the heat pipe involved once again data sets at two different operating temperatures for each heat pipe. The resultant trend lines are shown in Figure 5.10. The individual points in each data set leading to the trend lines given in this figure are provided in Appendix D. The figure does not include trend lines for the axial groove and fiber bundle heat pipe. For both of these heat pipes, proper steady-state operation was not observed when oriented in the unfavorable direction.

Figure 5.10 : Capillary-limit trend lines with longitudinal gravitation forces in the unfavorable direction

When the trends in Figure 5.10 are compared to those with transverse gravitational force in Figure 5.4, a substantial decrease in the capillary limit is noted due to the gravitational force acting longitudinally in the unfavorable direction. Comparison of the maximum heat transfer rates at 30°C and 60°C for the two conditions is given in Table 5.6.

Table 5.6 : Comparison of maximum heat transfer rate with gravitational force in transverse and longitudinal unfavorable directions

Wick Structure	$Q_{c,max}$ at $T_{A3} = 30^{\circ}\text{C}$		$Q_{c,max}$ at $T_{A3} = 60^{\circ}\text{C}$	
	Transverse g	Longitudinal g	Transverse g	Longitudinal g
Sintered Powder	5.9 W	4.5 W	9.1 W	6.4 W
Wrapped Screen	4.4 W	1.5 W	6.7 W	2.4 W
Axial Groove	5.3 W	N/A	10.7 W	N/A
Fiber Bundle	5.6 W	N/A	12.4 W	N/A

The temperature variation of the effective thermal resistance is shown Figure 5.11 for the axial groove heat pipe at close to ambient temperature.

Figure 5.11 : Transient behavior of axial groove heat pipe with longitudinal gravitational force in the unfavorable direction

Test results for heat loads of 0.6 and 1.3 Watts clearly indicate failure by the magnitude and rate of increase in thermal resistance. This state of failure also holds true at 0.2 Watts. The high level of scatter and continued increase in thermal resistance between 1000 and 2000 seconds for this heat load suggests that the heat pipe is unlikely to settle into a steady state operation, eventually resulting in evaporator dry-out.

At the higher end of the operating temperature range tested, the axial groove heat pipe showed scatter and a slight increase in thermal resistance with a heat load of 0.4 Watts, similar to the 0.2 Watt case of Figure 5.11. The capillary limit of the axial groove heat pipe at or below 60°C operating temperature is thus concluded to be less than 0.4 Watts. This amount of heat transfer is negligible compared to the amount recorded under a transverse gravitational force. The trends of the fiber bundle heat pipe were observed to be similar to those of the axial groove heat pipe.

5.1.2.2.1 Theoretical Analysis

As evident in Figure 5.10, the sintered powder and wrapped screen wick structures provide notable amounts of heat transfer despite unfavorable longitudinal gravitational force. As was conducted in the cases of transverse gravitational force, it is most effective to explain the observed trends with the aid of a simple mathematical analysis. The analysis is merely an extension of the one presented in Section 5.1.2.1.1. In the present case, the heat load at the capillary limit is determined by a balance in capillary, viscous, and body forces acting on the working fluid. The resultant Equation for $Q_{c,max}$ replacing Equation (5-5) is:

$$Q_{c,max} = \frac{\frac{2\sigma}{r_{eff}} + \rho_l g L_l \sin \varphi}{\frac{\mu_l}{\rho_l \lambda} \cdot \frac{L_{eff}}{KA_w} + \frac{(f_v Re_v) \mu_v}{\rho_v \lambda} \cdot \frac{L_{eff}}{2r_{h,v}^2 A_v}} \quad (5-10)$$

$$Q_{c,\max} = \frac{\alpha_c + \alpha_b}{\alpha_l \cdot \alpha_{g,l} + \alpha_v \cdot \alpha_{g,v}} \quad (5-11)$$

where,

- g gravitational acceleration
- L_l total working length of heat pipe
- ϕ angle of incline relative to horizontal orientation
- α_b body force parameter

The body force parameter, α_b , accounts for the effects of gravitational force. For $\phi = 0$ or 180° , the gravitational force is transverse to the axis of the heat pipe, and Equation (5-11) reduces to Equation (5-5). For $\phi = -90^\circ$ the condenser is situated directly below the evaporator and the body force is said to be in the longitudinal unfavorable direction.

As it was not possible to measure the effective capillary radius, r_{eff} and effective permeability, K , of the sintered powder heat pipe, an alternative means of acquiring this information is needed in order to apply Equation (5-10) to this heat pipe. One possibility is to make use of Equation (5-10) itself to quantify r_{eff} and K , by utilizing the measured $Q_{c,\max}$ values at two different operating temperatures. However, since $Q_{c,\max}$ data is available for only two operating temperatures, using the two $Q_{c,\max}$ values for the unfavorable orientation of the gravitational force to establish r_{eff} and K would negate the possibility of assessing the accuracy of Equation (5-10) in predicting $Q_{c,\max}$ for this configuration. To circumvent this problem, the alternative of using one data point ($Q_{c,\max}$ vs T_{A3}) with transverse gravitational force in Equation (5-4) and a second data point with unfavorable longitudinal gravitational force in Equation (5-10) is opted for. The resultant values of r_{eff} and K are listed in Table 5.7.

Table 5.7 : r_{eff} and K values for sintered powder heat pipe

r_{eff}	$2.17 \times 10^{-5} \text{ m}$
K	$2.42 \times 10^{-11} \text{ m}^2$

It is worth pointing out that it is not possible to implement this approach solely on the basis of Equation (5-4), i.e. with both data points belonging to the transverse gravitational force configuration. This is due to the fact that the vapor frictional term appearing in the denominator of Equation (5-4) is typically much smaller than the liquid frictional term (as per Table 5.5). With the vapor frictional term neglected, Equation (5-4) effectively becomes an equation for K/r_{eff} , which does not provide for the solution of independent values of K and r_{eff} .

The theoretical performance analysis of the wrapped screen heat pipe with transverse gravitational force, presented earlier, did not account for the presence of scattered particles embedded in the screen. It is feasible to determine the K and r_{eff} values which account for the presence of these particles by applying the same procedure described for the sintered powder heat pipe. The resultant values of K and r_{eff} together with those obtained on the basis of the screen geometry alone are listed in Table 5.8.

Table 5.8 : Effective capillary radius and permeability of the wrapped screen heat pipe with and without sintered particles taken into account

	Wrapped Screen (as per Table 5.3)	Wrapped Screen with Sintered Particles
r_{eff}	$6.35 \times 10^{-5} \text{ m}$	$4.62 \times 10^{-5} \text{ m}$
K	$7.08 \times 10^{-11} \text{ m}^2$	$5.67 \times 10^{-11} \text{ m}^2$

As expected, both the effective capillary radius and the permeability of the wick is reduced due to the presence of particles embedded in the screen.

The permeability and effective capillary radius pertaining to the fiber bundle heat pipe could not be determined through the use of Equation (5-10) with experimental $Q_{c,max}$ values because of difficulties in measuring the wick cross-sectional area and thus the vapor core cross-section.

Photographs of the fiber bundle cross-section, such as the one given in Appendix A, indicated the

absence of a well-defined wick along the perimeter of the heat pipe. As a result, a theoretical analysis of this heat pipe could not be performed.

The numerous α parameters in Equation (5-11) are listed in Table 5.9 for the wrapped screen wick and in Table 5.10 for the sintered powder wick. The relative magnitudes of the α parameters, specifically the value of α_b compared to α_c and the sum of $\alpha_l, \alpha_{g,l}$ and $\alpha_v, \alpha_{g,v}$, readily demonstrates why the amount of reduction in $Q_{c,max}$ that occurs when the angle of incline, ϕ , is reduced from 0 to -90° (as per Table 5.6) is greater for the wrapped screen wick than for the sintered powder wick structure. It is important to note that the α parameters presented in Table 5.9 correspond to the wrapped screen wick structure with the scattered sintered powder particles being taken into account. Consequently, the liquid flow resistance and capillary forces are substantially higher than those for the basic wrapped screen wick tabulated in Table 5.5.

Table 5.9 : Capillary, viscous, and body force parameters for the wrapped screen heat pipe with longitudinal gravitational force in the unfavorable direction

Wrapped Screen Wick Structure				
T_v [°C]	$\alpha_l \alpha_{g,l}$ [N/m ² ·W]	$\alpha_v \alpha_{g,v}$ [N/m ² ·W]	α_c [N/m ²]	α_b [N/m ²]
20	874	13.3	3147	-1881
30	707	7.9	3082	-1874
40	584	4.6	3013	-1865
50	494	3.1	2944	-1857
60	426	2.1	2870	-1847
70	374	1.4	2792	-1837
80	333	1.0	2714	-1826

Table 5.10 : Capillary, viscous, and body force parameters for the sintered powder heat pipe with longitudinal gravitational force in the unfavorable direction

Sintered Powder Wick Structure				
T_v [°C]	$\alpha_l \alpha_{g,l}$ [N/m ² ·W]	$\alpha_v \alpha_{g,v}$ [N/m ² ·W]	α_c [N/m ²]	α_b [N/m ²]
20	1899	44.1	9321	-1861
30	1536	26.3	9128	-1854
40	1270	15.4	8923	-1846
50	1074	10.4	8718	-1837
60	926	6.9	8500	-1828
70	813	4.7	8269	-1818
80	724	3.3	8039	-1807

Based on the tabulated values for the sintered powder heat pipe, arguments similar to those for the axial groove and wrapped screen heat pipes under transverse gravitational force can be made regarding the relative magnitudes of the capillary force, liquid frictional forces, vapor frictional forces, and their sensitivity to the operating temperature for this heat pipe. The equilibrium state between the three forces and changes in this state with operating temperature ultimately translates into the $Q_{c,max}$ vs T_{A3} trends for the sintered powder heat pipe shown in Figure 5.4.

Lastly, it is interesting to note that when Equation (5-10) is applied to the case of the axial groove heat pipe with the gravitational force oriented unfavorably along the pipe's axis at 20°C operating temperature, the result is -5.6 Watts. This is consistent with the experimental result of the pipe not performing properly at this temperature.

5.1.2.3 Results with Longitudinal Favorable Gravitational Force

With gravitational force oriented longitudinally to promote the flow of liquid from the condenser to the evaporator, a new balance is established between the capillary, viscous, and body forces. Since in this instance the body and capillary forces act against the viscous forces, the heat transfer rate at the capillary limit is expected to improve compared to the cases of unfavorable longitudinal and transverse orientations of the gravitational force. This is evident in Figure 5.12 where the trend lines for the three configurations are shown for the wrapped screen and sintered powder heat pipes. The increases in $Q_{c,max}$ with favorable orientation of the gravitational force were such that the experimental setup was incapable[†] of bringing the fiber bundle and axial groove heat pipes to failure. Hence, they are excluded from the figure. Appendix D contains the data points collected during these measurements.

[†] As Q_{in} is increased in search of $Q_{c,max}$, the control over T_{A3} is limited because the minimum coolant temperature of the setup was approximately 15°C. Thus in instances of high $Q_{c,max}$, it was not always possible to remain within the operating temperature range of interest. Another limitation of the setup is the upper limit imposed on Q_{in} by the electric power supplies. In the tests with axial groove and fiber bundle heat pipes, the lower limit of the coolant temperature was the restricting factor.

Figure 5.12 : Capillary-limit trend lines for the sintered powder and the wrapped screen heat pipe with longitudinal gravitational force in the transverse, favorable, and unfavorable directions

As noted in Figure 5.12, the percentage increase in $Q_{c,max}$ with the longitudinal, favorable orientation of the gravitational force is notable higher for the wrapped screen heat pipe than for the sintered powder heat pipe. The slope of the trend line is noted to have increased in both instances compared to transverse orientation of the gravitational force. To explain these trends, Equation (5-11) presented in Section 5.1.2.2.1 was applied to these two cases by setting ϕ to $+90^\circ$. The predictions are compared to the measured trend lines in Figures 5.13 and 5.14, and the α_b parameter to replace those listed in Tables 5.9 and 5.10 is given in Table 5.11.

Figure 5.13 : Capillary-limit trend lines for the wrapped screen heat pipe with longitudinal gravitational force in the favorable direction – theoretical vs. experimental results

Figure 5.14 : Capillary-limit trend lines for the sintered powder heat pipe with longitudinal gravitational force in the favorable direction – theoretical vs. experimental results

Table 5.11 : Body force parameter for the wrapped screen and sintered powder heat pipe with gravitational force in the longitudinal favorable direction

	Wrapped Screen	Sintered Powder
T_v [°C]	α_b [N/m ²]	α_b [N/m ²]
20	1881	1861
30	1874	1854
40	1865	1846
50	1857	1837
60	1847	1828
70	1837	1818
80	1826	1807

The agreement of $dQ_{c,max}/dT_{A3}$ between the predicted and measured results in Figure 5.13 is good and is comparable to what was observed for transverse and longitudinal unfavorable gravitational force orientations. However, for both heat pipes, the experimental results are seen to be consistently greater than the predicted $Q_{c,max}$ values. The slight difference can be explained through a bias error imposed by the start-up conditions of the heat pipe. When the heat pipe is situated with the gravitational force in the longitudinal favorable direction, working liquid tends to accumulate inside the evaporator section of the heat pipe. At the start of each test session, with what was perceived to be a properly operating heat pipe after heat was initially applied to the evaporator, the load was increased in search of the capillary limit. It is plausible that the capillary limit of the heat pipe may have been already exceeded early on in these sessions as the liquid that had collected in the evaporator turned into vapor. This physics is not accounted for by the mathematical model, thus is argued to be a potential cause of the slight under prediction of the capillary limit in these instances.

The relative magnitudes of the α_b parameters in Tables 5.9 and 5.10 vs. 5.11 demonstrate that the increase in the slope of the trend line, $dQ_{c,max}/dT_{A3}$ with longitudinal, favorable gravitational force is brought about by notably larger values of $\alpha_c + \alpha_b$. The axial groove heat pipe is particularly sensitive to orientation as observed in Section 5.1.2.2. Although experimental results could not locate the capillary limit of the heat pipe when oriented with the gravitational force in the longitudinal favorable direction, the mathematical model predicts a significant increase in performance. Figure 5.15 shows the effect of g-force orientation on the theoretical trend line for the axial groove heat pipe.

Figure 5.15 : Capillary-limit trend lines for the axial groove heat pipe with transverse and longitudinal gravitational force in the favorable direction – theoretical

Table 5.12 lists the geometry- and flow- α parameters for the axial groove heat pipe including the body force parameter, α_b , over the temperature range of interest. The relative magnitude of α_b

explains the significant effect the orientation of the g-force has on the capillary limit of the axial groove heat pipe.

Table 5.12 : Geometry and flow parameters for the axial groove heat pipe with gravitational force in the longitudinal favorable direction.

Axial Groove Wick Structure				
T_v [°C]	$\alpha_l \alpha_{g,l}$ [N/m ² ·W]	$\alpha_v \alpha_{g,v}$ [N/m ² ·W]	α_c [N/m ²]	α_b [N/m ²]
20	162	15.0	890	1881
30	131	9.0	872	1874
40	108	5.2	852	1865
50	92	3.5	833	1857
60	79	2.4	812	1847
70	69	1.6	790	1837
80	62	1.1	768	1826

Results on the fiber bundle heat pipe are not available because of the previously-noted limits of the test setup and inability to quantify the wick structure of this heat pipe. Nonetheless, the fiber bundle heat pipe is expected to perform similar to the axial groove heat pipe based on observations presented in Sections 5.1.2.1 and 5.1.2.2.

Although the axial groove heat pipe demonstrates the highest $Q_{c,max}$ when the gravitational force is in the longitudinal favorable direction, it fails to perform when oriented in the opposite direction. Given the trade off, the sintered powder heat pipe is favored over the other three heat pipes tested at least for the static (non-vibrating) conditions presented thus far.

Having established the performance of each of the four heat pipes under the influence of transverse and longitudinal gravitation forces, in the next section these results are used as a baseline to determine the effects of vibration on thermal performance.

5.1.3 Test Results under Vibration

5.1.3.1 Vibration Frequency and Input Levels

The vibration frequency range and power-spectral density (PSD) level as given in Table 5.13 were chosen in accordance with Dy 4 requirements for conduction-cooled CCAs. The typical vibration input used during the experiments is shown in Figure 5.16.

Table 5.13 : Input vibration level and frequency range

PSD Level	Frequency Range
0.1 g ² /Hz	15 - 2000 Hz

Figure 5.16 : Example of measured vibration level applied by the shaker table

5.1.3.2 Temperature Measurements under Vibration

Preliminary test results in a vibrating environment indicated notably increased scatter in temperature recordings. In a separate test with the thermocouple leads directly attached to the shaker, Figure 5.17 plots the measurements recorded while the thermocouple was intermittently subjected to vibration. The increased scatter during vibration periods is obvious. The scatter was traced to movements of the thermocouple leads, producing variations in the output signals of the order of a few milli-Volts prior to signal conditioning. The wires were therefore constrained to the aluminum casing, base plate, and the shaker body. As shown in Figure 5.18, temperature measurements continued to display somewhat higher levels of scatter under vibration. However, the extent of scatter was sufficiently lowered to not interfere with the observation of actual trends in the temperature signals.

Figure 5.17 : Effects of motion on thermocouple readings

Figure 5.18 : Example of thermocouple reading under vibration with the leads constrained

5.1.3.3 Results with Transverse Gravitational Force and Transverse Vibration

Figure 5.19 presents the capillary limit trend line for heat pipes with different wick structures, with the gravitational force and direction of vibration oriented 90° (i.e. transverse) to the axis of the heat pipe. The sintered powder, axial groove, and fiber bundle heat pipes were noted to yield similar thermal performance under vibrating conditions.

Figure 5.19 : Capillary-limit trend lines with transverse vibration under transverse gravitational force

To quantify the effects of vibration on $Q_{C,MAX}$ for each heat pipe, plots comparing results with vibration to non-vibrating conditions are required. Such a plot for the sintered powder heat pipe is shown in Figure 5.20. An improvement in capillary limit is noted when the heat pipe is subjected to vibration. To explain this trend, the temperature distribution along the heat pipe under vibrating and non-vibrating conditions are compared in Figure 5.21. The temperature measurements on the top and side of the brass sleeve containing the heat pipe were noted to be essentially identical. This suggests absence of any significant circumferential asymmetry developing in the heat pipe under vibration. Distance X on this plot is measured from the evaporator end of the heat pipe.

Figure 5.20 : Capillary-limit trend lines for the sintered powder heat pipe with and without transverse vibration under transverse gravitational force

Figure 5.21 : Typical temperature distributions for a heat pipe in vibrating and non-vibrating environments

It is evident that the temperature distribution along the length of the heat pipe is notably less uniform when subjected to vibrating conditions. This is attributed to an increase in cooling within the cooling jacket. Under vibration, the convective heat transfer coefficient inside the cooling jacket is expected to be higher bringing the surface temperature of the condenser section closer to the temperature of the coolant. As a result, the effective length of the heat pipe is shortened under vibrating conditions.

In summary, the observed change in the capillary limit of the heat pipe with the introduction of vibration appears to be the result of an inadvertent change in the test conditions of the heat pipe, rather than a fundamental change in the thermo/fluid dynamics prevailing inside the pipe.

5.1.3.3.1 Comparison with $Q_{c,max}$ under Non-vibrating Conditions

Based on the temperature distribution shown in Figure 5.21, it is judged that under vibrating conditions the contribution of the condenser to L_{eff} is reduced from the original value of $1/2 L_c \approx 26$ mm, excluding the crimped section, to roughly 10 mm. The comparison of the $Q_{c,max}$ trend lines for vibrating and non-vibrating conditions based on this L_{eff} is shown in Figure 5.22 for the sintered powder heat pipe. Since the mathematical model for $Q_{c,max}$ has already been validated for non-vibrating results, this model has been used to estimate the $Q_{c,max}$ value for non-vibrating conditions with an effective length, L_{eff} , that matches the value observed under vibration.

Figure 5.22 : Capillary-limit trend lines for the sintered powder heat pipe with and without transverse vibration under transverse gravitational force and the same value of L_{eff}

The two curves agree with each other within the combined uncertainty ranges of the theoretical and experimental results. Based on this result, transverse vibration does not appear to have an effect on $Q_{c,max}$ within the temperature range of interest.

Figure 5.23 : Capillary-limit trend lines for the axial groove heat pipe with and without transverse vibration under transverse gravitational force and the same value of L_{eff}

Figure 5.24 : Capillary-limit trend lines for the wrapped screen heat pipe with and without transverse vibration under transverse gravitational force and the same value of L_{eff}

Although the trends for the axial groove and wrapped screen heat pipes shown in Figures 5.24 and 5.25 are consistent with the conclusion drawn from the results of the sintered powder heat pipe, the support for this conclusion is not as strong in these instances due to the larger uncertainty of the theoretical results.

5.1.3.3.2 Fiber Bundle Heat Pipe with Transverse Vibration

Sensitivity of the performance of the fiber bundle heat pipe to vibration is shown in Figure 5.25. A slight difference in $Q_{c,max}$ between vibrating and non-vibrating conditions is evident at higher operating temperatures. However, the difference is too close to the combined uncertainty ranges of the data. Thus, further discussion on it is not justified. It is important to note that the non-vibrating trend line in this figure is experimental hence the two trend lines correspond to difference values of L_{eff} . The use of the empirical trend line for the non-vibrating case was necessary for this heat pipe because sufficient information on the wick structure to predict $Q_{c,max}$ could not be obtained.

Figure 5.25 : Capillary-limit trend lines for the fiber bundle heat pipe with transverse vibration under transverse gravitational force – correction to L_{eff} not taken into account

5.1.3.4 Results with Longitudinal, Unfavorable Gravitational force and Longitudinal Vibration

All four heat pipes were tested under longitudinal vibration with unfavorable gravitational force. Figure 5.26 plots the trend lines observed for the sintered powder and wrapped screen heat pipe and compares the results to those under non-vibrating conditions. As per the discussions in Section 5.1.3.3.1, the results without vibration in Figure 5.26 were determined theoretically to enable comparison for the same values of L_{eff} . The shift in the trend lines with the introduction of vibration is noted to be quite small, and is within the uncertainty range of the data. Thus, one may conclude that the effect of longitudinal vibration on both heat pipes, with the gravitational force acting in the unfavorable direction, is negligible. The axial groove and fiber bundle heat pipes failed at negligible levels of heat loads, both with and without vibration.

Figure 5.26 : Capillary-limit trend lines with longitudinal vibration under longitudinal unfavorable gravitational force and the same value of L_{eff}

5.1.3.5 Results with Longitudinal, Favorable Gravitational Force and Longitudinal Vibration

All four heat pipes were tested under longitudinal vibration with favorable gravitational force.

Figure 5.27 plots the trend lines observed for the sintered powder and wrapped screen heat pipe.

$Q_{c,max}$ is not available for the other two heat pipes since the limitation of the setup did not permit this heat transfer limit to be reached (see Section 5.1.2.3). As such, the axial groove and fiber bundle heat pipes are excluded.

Figure 5.27 : Capillary-limit trend lines with longitudinal vibration under longitudinal favorable gravitational force and the same value of L_{eff}

The sintered powder heat pipe shows an increase in $Q_{c,max}$ under vibrating conditions. At the high end of the operating temperature range, the wrapped screen heat pipe also shows an increase in $Q_{c,max}$ under vibration. These differences between the empirical trend lines for the vibrating condition and the theoretical trend lines for the non-vibrating state are believed to be due to the effect of evaporator flooding on the start-up of the experiments, as discussed in Section 5.1.2.3.

6 Conclusions and Recommendations

6.1 Conclusion

This thesis presents thermal performance test results for several conventional heat pipes. Each of the four tested heat pipes contained a different wick structure, namely sintered powder, axial groove, wrapped screen, and fiber bundle. The heat pipe dimensions were chosen to be compatible with typical electronics cooling applications. The thermal performance of each heat pipe was evaluated for several orientations of the gravitational force under vibrating and non-vibrating conditions.

By virtue of the extent of the dataset, the relative influence of various physical mechanisms that affect heat pipe thermal performance were thoroughly evaluated. This was done by a systematic analysis of the numerous parameters that exist in current mathematical models for the prediction of the capillary heat transfer limit. This approach provided a basis for thorough interpretation of the operating envelopes of the tested heat pipes with respect to the influence of operating temperature, relative direction of gravitational force, wick structure, and presence of vibration.

These parameters have been found to affect the capillary limit in an interactive manner. In the absence of gravitational force and vibration, the capillary limit was found to be the most sensitive to the operating temperature for the fiber bundle wick structure ($dQ_{c,max}/dT = 0.22 \text{ W/}^\circ\text{C}$), and the least sensitive for the wrapped screen wick structure ($dQ_{c,max}/dT = 0.08 \text{ W/}^\circ\text{C}$).

The capillary limits of all four heat pipes were very similar at the lower end of the measured operating temperature range ($Q_{c,max} \approx 5$ Watts at 30°C). At the high end of the measured operating temperature range, the fiber bundle wick structure displayed the highest capillary limit ($Q_{c,max} \approx 15$ Watts at 70°C).

Under the influence of favorable gravitational force (gravitational force complementing capillary force), the aforementioned trends in terms of peak heat transfer performance and sensitivity to operating temperature were observed to change as follows. As expected, the capillary heat transfer limit increased for all four heat pipes under these conditions. For the fiber bundle and axial groove heat pipes, this increase was beyond the maximum heat load of 60 Watts limited by the experimental setup. The improvement in the capillary limit at the low end of the operating temperature range was noted to be similar for the wrapped screen and sintered powder heat pipe ($\Delta Q_{c,max} \approx 3$ Watts). The increase in the temperature sensitivity of the capillary limit, however, was greater for the wrapped screen heat pipe ($dQ_{c,max}/dT$ increase of 120% vs. 50%).

Under the influence of unfavorable gravitational force, the following changes were observed in the capillary limit and its sensitivity to operating temperature, with respect to the case without gravitational force. Since in this instance the gravitational force opposes the capillary force, the fiber bundle and axial groove heat pipes failed to perform properly in the temperature range tested. For the same reason, the capillary limit of the other two heat pipes were noted to be lower. At the lower end of the temperature range, this reduction was greater for the wrapped screen heat pipe ($Q_{c,max}$ reduction of 65% vs. 25% at 30°C). The reduction in the temperature sensitivity of the capillary limit was also greater for the wrapped screen heat pipe ($dQ_{c,max}/dT$ decrease of 70% vs. 40%).

In the presence of random vibration, with all three orientations of the gravitational force noted above, the capillary limits of all four heat pipes were observed to be within the uncertainty range ($\approx 10\%$) of the corresponding non-vibrating capillary limits.

6.2 Recommendations

Although the results acquired in this study have achieved all the objectives that were set at the start, the experimental setup that has been developed and the knowledge gained provide a good platform for further investigations. The following list of recommendations is based on the interpretation of the present results and experience with the experimental setup.

1. Flooding inside the condenser: Attempts should be made to locate the wet point more precisely in order to determine L_{eff} more accurately.
2. Control of cooling rate: An alternative design for the condenser heat sink that facilitates heat rejection that is insensitive to vibration should be considered. One possibility is to rely on conduction rather than convection in extracting heat from the condenser.
3. Maximum allowable heat transfer rate: An increase in the maximum allowable heat transfer rate in the experimental setup may be worthwhile to establish the capillary limit of those heat pipes that are more sensitive to orientation (e.g. fiber bundle and axial groove heat pipes with gravitational force in the longitudinal favorable direction).
4. Fluid motion under vibration: A more fundamental study of the simple harmonic motion of the heat pipe on the liquid dynamics inside the wick would be beneficial. This would assist in interpreting the different conclusions that have been drawn on the effects of vibration on heat pipe thermal performance in several independent studies thus far.

5. Test matrix: Testing under vibration with a power spectral density greater than those of the present study may identify a point at which the capillary limit of the heat pipe is affected by vibration. Non-circular cross-sections and bends in the heat pipe may also be introduced to determine if either of these variables causes the capillary limit to be susceptible to vibration.

APPENDIX A : Photographs of Heat Pipe Cross-Sections

For each of the cross-sections shown, the cut was made using a low-speed cutter designed specifically for micro-sectioning. The quality of the cut is determined mainly by the speed of the cutter. It was observed that higher speeds resulted in a decrease in the quality of the cut. As such, when sectioning the heat pipes, a lower speed of 60 revolutions per minute was used. The following subsections describe the photographs of each of the cross-sections observed. In some instances, measurements were made from these photographs to quantify the geometry of the wick structure.

Sintered Powder Heat Pipe

A photograph of the sintered powder heat pipe cross-section is shown in Figure A.1. Fine sintered particles are readily visible along the perimeter of the vapor core. It should be noted that the particles are of elongated shape, which was established by viewing a particle sample under 40 $\times$ magnification.

Figure A.1 : Cross-section of sintered powder heat pipe

In the second photograph of the sintered powder heat pipe (Figure A.2), the sintered particles were removed using a fine point to scrap away the powder. Based on this cross-section, the thickness of the heat pipe wall was measured. The difference in the mean vapor core diameter (Figure A.1) and inner diameter was taken to be the thickness of the wick.

Figure A.2 : Cross-section of sintered powder heat pipe after particles have been removed

Wrapped Screen Heat Pipe

A photograph of the wrapped screen cross-section is shown in Figure A.3. It should be noted that the wrap that appears in the figure was damaged during the cut. However, other portions of the wick structure confirmed that the screen was tightly wrapped and properly seated along the inside surface of the pipe. As previously discussed in Chapter 2, the tightness and placement of the wrap are expected to have a significant effect on the thermal performance of the heat pipe.

Figure A.3 : Cross-section of wrapped screen heat pipe

The number of layers of the screen in its installed configuration was determined based on the width of the screen and the inner diameter of the heat pipe. This calculation indicated a slight overlap, which was subsequently taken into account. The properties of the screen were measured by magnifying the screen extracted from the cut cross-section. Figure A.4 shows a close-up view of the screen. The wire diameter and spacing were measured on this view using the Ersa-scope software. From these measurements, the mesh number of the screen was calculated to be 7874 m^{-1} (200 in^{-1}).

Figure A.4 : Close-up view of screen

In addition to the observations made on the wrap tightness, placement of the screen, and mesh geometry, fine sintered particles were seen after examining the screen under 40× magnification. As such, it was recognized that the presence of these particles has an effect on the capillary limit of the heat pipe, specifically the capillary force produced by the wick.

Axial Groove Heat Pipe

The cross-section of the axial groove heat pipe is shown in Figure A.5. Measurements of the pitch, groove width, groove depth, and half angle were made on this and other views using the Ersascope software. The uncertainty associated with these parameters was based on the repeatability of the measurements.

Figure A.5 : Upper right quadrant of axial groove cross-section

Because of the nature of the cut, the quality of the grooves varied depending on where the cut was initiated. This made it difficult to count all of the grooves along the perimeter. An alternative method was discovered by using a droplet of water to magnify the cross-section at some arbitrary distance away from the cut. Figure A.6 illustrates the result of this approach, which yielded a groove count of 40.

Figure A.6 : Magnification of the total number of grooves ($N_g = 40$)

Fiber Bundle Heat Pipe

For the fiber bundle heat pipe, the wick structure showed a spaghetti-like formation of fibers along one side of the heat pipe. On the other side, presence of fibers was not evident, as shown in Figure A.7. Because of this arbitrariness, the wick properties could not be quantified reliably. As such, the study of this heat pipe relied on empirical testing alone.

Figure A.7 : Cross-section of fiber bundle heat pipe

APPENDIX B : Derivation of Wick Properties for the Axial Groove Heat Pipe

Since the groove configuration of the tested axial groove heat pipe differs from that presented in most published literature, the derivation of the wick properties are presented here for completeness.

Wick Cross-Sectional Area, A_w

The wick cross-sectional area for the axial groove heat pipe is expressed as,

$$A_w = N_g \cdot (w\delta - \delta^2 \tan \beta) \quad (\text{B-1})$$

where,

N_g total number of grooves

w groove width

δ groove depth

β groove half-angle

The groove parameters used in the Equation (B-1) are illustrated in Chapter 2, Figure 2.11.

Permeability, K

For non-circular channels, the permeability is expressed in terms of the hydraulic radius for liquid flow as follows,

$$K = \frac{2r_{h,l}^2 \varepsilon}{2(f_l \text{Re}_l)} \quad (\text{B-2})$$

where the porosity of the wick, ε , is defined as the ratio of pore volume, V_p , to the total volume of the wick structure, V_t ,

$$\begin{aligned} \varepsilon &= \frac{V_p}{V_t} \\ &= \frac{w - \delta \tan \beta}{s} \end{aligned} \quad (\text{B-3})$$

The hydraulic radius for liquid flow, $r_{h,l}$, is defined as twice the groove cross-sectional area, A_w , divided by the wetted perimeter, C_l ,

$$r_{h,l} = 2 \frac{w\delta - \delta^2 \tan \beta}{w - 2\delta \tan \beta + \frac{2\delta}{\cos \beta}} \quad (\text{B-4})$$

Given the unique cross-section, a correlation for the liquid friction term, $f_l \text{Re}_l$, is not readily available. However, it may be argued that the groove closely resembles a rectangular cross-section given that the half-angle of the groove is very small. As such, the liquid friction term is approximated using the expression for a rectangular groove,

$$f_l \text{Re}_l = 24(1 - 1.3553\vartheta + 1.9467\vartheta^2 - 1.7012\vartheta^3 + 0.9564\vartheta^4 - 0.2537\vartheta^5) \quad (\text{B-5})$$

where,

$$\vartheta = w/\delta \quad (\text{B-6})$$

Effective Thermal Conductivity, k_{eff}

The effective thermal conductivity of the wick in the evaporator and condenser sections differ from each other in an axial groove heat pipe. Similar to a rectangular groove cross-section, the effective thermal conductivity within the condenser portion of the heat pipe for a trapezoidal groove may be calculated from parallel heat conduction in the liquid and solid portions of the wick approximated by the following expression,

$$k_{eff,c} = (1 - \varepsilon)k_w + \varepsilon k_l \quad (B-7)$$

Unlike the condenser portion of the wick, the effective thermal conductivity inside the evaporator must take into account the additional interface resistance acting between the land area of the fin and vapor core. In addition, the expression for a rectangular groove does not account for the variation in area through the fin and groove. As such, the effective area is derived from the volume of liquid within the trapezoidal groove. Thus at the top of the groove, the cross-sectional area of heat flow is equal to (wL_e) , where L_e is defined as the evaporator length of the heat pipe. The cross-sectional area at the base of the groove is reduced to $(w - 2\delta \tan \beta)L_e$. Thus, the volume of liquid may be calculated from the following integral,

$$\begin{aligned} V_l &= L_e \int_0^\delta (w - 2t \tan \beta) \cdot dt \\ &= L_e \delta (w - \delta \tan \beta) \end{aligned} \quad (B-8)$$

and the effective area through the liquid, A_l , is simply,

$$A_l = L_e (w - \delta \tan \beta) \quad (B-9)$$

Using the effective area of the groove, the resistance through the liquid, R_l , may be defined as,

$$R_1 = \frac{\delta}{k_l L_e (w - \delta \tan \beta)} \quad (\text{B-10})$$

Similarly, the thermal resistance through the solid portion of the wick may be defined as,

$$R_2 = \frac{\delta}{k_w L_e (w_f + \delta \tan \beta)} \quad (\text{B-11})$$

where w_f is the fin width of the groove.

Lastly, taking the liquid-vapor interface thermal resistance in series with the thermal resistance of the solid portion of the wick into account, Chi (1976) defines the heat transfer coefficient through the land area of the fin as,

$$h = \frac{k_l}{0.185 w_f} \quad (\text{B-12})$$

Thus,

$$\begin{aligned} R_3 &= \frac{1}{h A_f} \\ &= \frac{0.185}{k_l L_e} \end{aligned} \quad (\text{B-13})$$

Solving the two conductive paths in parallel, the effective thermal conductivity of a trapezoidal groove within the evaporator section may be expressed as,

$$k_{eff,e} = \frac{k_l (w - \delta \tan \beta)}{s} + \frac{\delta k_l k_w (w_f + \delta \tan \beta)}{s [\delta k_l + 0.185 k_w (w_f + \delta \tan \beta)]} \quad (\text{B-14})$$

Knowing the effective conductivity of the trapezoidal groove, the overall thermal resistance through the wick may be calculated.

APPENDIX C : Element/Node Distribution for FEA Model

The spatial mesh used as the basis of the FEA analysis to determine the effective thermal resistance of the system, R_s , was developed using PTC's Pro/MECHANICA. Pro/MECHANICA provides users with the flexibility of constructing assemblies using default settings for meshing, solving, and post processing the results. Consequently, a majority of the time is spent on developing the geometry of the model and setting the proper boundary conditions. Once the user has completed these tasks, AUTOGEM may be used to quickly generate the spatial mesh. Furthermore, since Pro/MECHANICA makes use of higher-order P-version elements instead of smaller rigid H-version elements, the total node count remains at reasonable levels. As such a standard desktop computer may be used to solve complex geometries such as the evaporator assembly presented in this study.

In order to validate the suitability of the mesh generated by AUTOGEM, the mesh was fine-tuned by introducing surface regions into the evaporator assembly. This forced AUTOGEM to generate more elements. In the cases presented, the total number of elements was increased from approximately 22,000 to 28,000. As displayed in Figures C.1 to C.3, this adjustment in node count had a negligible impact on the temperature distribution in each part. A second iteration that produced 31,000 elements confirmed these results.

From Figure C.1, the small temperature differential is due to the zero degree boundary condition applied to the outside surfaces of the casing and the high thermal conductivity of aluminum. It should be noted that the temperatures shown are in degrees centigrade.

Figure C.1 : Effect of mesh node-count change on the temperature distribution in the aluminum casing

Figure C.2 : Effect of mesh node-count change on the temperature distribution in the copper block

Figure C.3 : Effect of mesh node-count change on the temperature distribution
 in the polystyrene housing

Based on these findings, AUTOGEM was used to mesh the final assembly. By minimizing the total number of elements, the computing time was kept within approximately 2 hours with a Pentium 866 MHz Processor and 1 Gb memory, without affecting the accuracy of the results.

APPENDIX D : Capillary Limit Trend lines with Data Points

Figure D.1 : Sintered powder heat pipe in transverse gravitational force without vibration

Figure D.2 : Sintered powder heat pipe in transverse gravitational force with vibration

Figure D.3 : Axial groove heat pipe in transverse gravitational force without vibration

Figure D.4 : Axial groove heat pipe in transverse gravitational force with vibration

Figure D.5 : Wrapped screen heat pipe in transverse gravitational force without vibration

Figure D.6 : Wrapped screen heat pipe in transverse gravitational force with vibration

Figure D.7 : Fiber bundle heat pipe in transverse gravitational force without vibration

Figure D.8 : Fiber bundle heat pipe in transverse gravitational force with vibration

Figure D.9 : Sintered powder heat pipe with longitudinal gravitational force in unfavorable direction without vibration

Figure D.10 : Sintered powder heat pipe with longitudinal gravitational force in unfavorable direction with vibration

Figure D.11 : Axial groove heat pipe with longitudinal gravitational force in unfavorable direction without vibration

Figure D.12 : Axial groove heat pipe with longitudinal gravitational force in unfavorable direction with vibration

Figure D.13 : Wrapped screen heat pipe with longitudinal gravitational force in unfavorable direction without vibration

Figure D.14 : Wrapped screen heat pipe with longitudinal gravitational force in unfavorable direction with vibration

Figure D.15 : Fiber bundle heat pipe with longitudinal gravitational force in unfavorable direction without vibration

Figure D.16 : Fiber bundle heat pipe with longitudinal gravitational force in unfavorable direction with vibration

Figure D.17 : Sintered powder heat pipe with longitudinal gravitational force in favorable direction without vibration

Figure D.18 : Sintered powder heat pipe with longitudinal gravitational force in favorable direction with vibration

Figure D.19 : Axial groove heat pipe with longitudinal gravitational force in favorable direction without vibration

Figure D.20 : Axial groove heat pipe with longitudinal gravitational force in favorable direction with vibration

Figure D.21 : Wrapped screen heat pipe with longitudinal gravitational force in favorable direction without vibration

Figure D.22 : Wrapped screen heat pipe with longitudinal gravitational force in favorable direction with vibration

Figure D.23 : Fiber bundle heat pipe with longitudinal gravitational force in favorable direction without vibration

Figure D.24 : Fiber bundle heat pipe with longitudinal gravitational force in favorable direction with vibration

APPENDIX E : Uncertainty of Theoretical Model and Sensitivity Analysis

For the axial groove and wrapped screen heat pipes, the uncertainty of the theoretical $Q_{c,max}$ was calculated based on the uncertainty of the measurements shown in Table 5.1 and 5.2. For each heat pipe, a random function was used to calculate a string of $Q_{c,max}$ values at a fixed operating temperature. The standard deviation of a large sample size (~ 500) was then calculated to estimate the overall uncertainty of the theoretical model. A sensitivity analysis was conducted a prior to determine which parameters had the greatest and least effect on $Q_{c,max}$. Those parameters showing a negligible effect on $Q_{c,max}$ were excluded from the uncertainty of the theoretical model.

Figures E.1, E.2, E.3, and E.4 illustrate the effects of the groove depth, groove width, groove angle, and angle of incline on $Q_{c,max}$ for the axial groove heat pipe. Based on the uncertainty of the measurements, the outside diameter and wall thickness was found to have a negligible effect on the capillary limit. As shown in the aforementioned figures, the axial groove heat pipe is extremely sensitive to geometry of the groove and angle of incline. The resulting overall uncertainty was estimated to be $\pm 25\%$ for the theoretical model.

For the wrapped screen heat pipe, $Q_{c,max}$ was found to be sensitive to the number of wraps, mesh number, and angle of incline as shown in Figures E.5, E.6 and E.7. The wire diameter, outside diameter, and wall thickness showed a negligible effect on the capillary limit. The resulting overall uncertainty was estimated to be $\pm 11\%$ for the theoretical model.

Figure E.1 : Sensitivity of $Q_{c,\text{max}}$ to groove depth, δ , for the axial groove heat pipe

Figure E.2 : Sensitivity of $Q_{c,\text{max}}$ to groove width, w , for the axial groove heat pipe

Figure E.3 : Sensitivity of $Q_{c,\max}$ to half angle, β , for the axial groove heat pipe

Figure E.4 : Sensitivity of $Q_{c,\max}$ to angle of incline, ϕ , for the axial groove heat pipe

Figure E.5 : Sensitivity of $Q_{c,\text{max}}$ to mesh number, N , for the wrapped screen heat pipe

Figure E.6 : Sensitivity of $Q_{c,\text{max}}$ to number of layers for the wrapped screen heat pipe

Figure E.7 : Sensitivity of $Q_{c,\text{max}}$ to angle of incline, ϕ , for the wrapped screen heat pipe

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