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Effective thermal conductivity of an aluminum foam+water two phase system

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EFFECTIVE THERMAL CONDUCTIVITY
OF AN
ALUMINUM FOAM+WATER TWO PHASE SYSTEM

A Thesis
Presented To
The Faculty Of The Department Of Materials Engineering
San Jose State University

In Partial Fulfillment
Of The Requirements For The Degree
Master Of Science

by
John Moskito
Summer 1999

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ABSTRACT

EFFECTIVE THERMAL CONDUCTIVITY OF AN ALUMINUM FOAM + WATER SYSTEM

by John Moskito

The effect of volume fraction and pore size on the effective thermal conductivity of an aluminum foam and water system was investigated. Nine specimens of aluminum foam representing a matrix of three volume fractions (4-8% by vol.) and three pore sizes (2-4 mm) were tested with water to determine relationships to the effective thermal conductivity. Increases in volume fraction of the aluminum phase correlated to increases in the effective thermal conductivity. It was not possible, statistically, to determine if changes in pore size of the aluminum foam correlated to changes in the effective thermal conductivity. However, interaction effects between the volume fraction and pore size of the foam were statistically significant. Ten theoretical models were selected from the published literature to compare against the experimental data. Models by Asaad, Hadley [1986] and de Vries provided effective thermal conductivity predictions within a 95% confidence interval.

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List of Abbreviations

A	Amperes	f	Empirical factor in Hadley equation
bulk	Apparent volume	$f(\tau)$	Function of the dimensionless term τ
BTU	British thermal units	ft	Foot
c	Empirical factor in Asaad equation	F	Empirical factor in de Vries equation
cm	Centimeters	$^{\circ}\text{F}$	Degrees Farenheit
C	Slope of R versus $f(\tau)$	$g(^{*})$	Geometrical shape factor (de Vries equation)
C_p	Heat capacity	gm	Gram
$^{\circ}\text{C}$	Degrees Celsius	Gr. Pr	Grashof-Prandtl number
d	Diameter of heater wire	h	Thickness of conducting matrix in direction of heat flow
D	Cross sectional area of flow	$\bar{h}$	Average heat transfer coefficient
D	Thickness of conducting matrix in direction of heat flow	h	Thickness of hot strip
D_{circle}	Diameter of circle	H_o	Null hypothesis
DIFHS	Direct Interface Fusible Heat Sink	hr	Hour
erfc	Complimentary error function	I	Current (amperes)
Ei(-u)	Exponential integral of -u	ISS	International space station
EMU	Extravehicular mobility unit	k	Thermal conductivity
EVA	Extravehicular activity	kg	Kilogram

K	Kelvin	P	Pore size
L	Length	Pr	Prandlt number
L_{fusion}	Heat of fusion	PLSS	Personal life support system
L_{pore}	Length of pore in direction of heat flow	q	Heat flow
lb	Pound	r	Radius
lbm	Pound mass	R	Resistance
LCG	Liquid cooling garment	°R	Degrees Rankine
LEO	Low earth orbit	Re	Reynolds number
m	Mass	t	Time
m	Meter	T	Temperature
mm	Millimeter	t	Student's t-test statistic
m_{Asaad}	Empirical term in Asaad equation	t_c	Time constant for heat flow through an insulative layer
mag	Magnification	<i>vol. frac</i>	Volume fraction
mass	Mass	V	Volts
$\bar{N}$	Average number of intersections	Vol	Volume
NASA	National Aeronautic and Space Administration	V_∞	Velocity
Nu_D	Nusselt number	W	Watt
pore	Average pore size	w	Half-width of heater strip
psia	Pounds per square inch absolute	Z	Thickness

Greek symbols

α_H	Empirical factor in Hadley equation	ν_f	Kinematic viscosity of the fluid
α	Temperature coefficient of resistivity (TCR)	π	Pi, 3.14159265
β	Multiple regression coefficients	θ	Characteristic time
δ	Infinitesimal change in a property	ρ	Density
ε_f	Volume fraction of fluid phase	σ	Stephan-Boltmann constant 5.6696×10^{-8} W / m ² K
ε_s	Volume fraction of solid phase	τ	Dimensionless function of time
ε	Emissivity	ξ	Penetration depth of heat wave of hot strip
φ	Characteristic pore ratio	Δ	Finite change in a property
κ	Diffusivity	Ω	Ohms

Subscripts/Superscripts

0 Initial value

a Container

avg Average

c Constant

eff Effective

end End

f Fluid

freezer Freezer

H₂O Water (liquid)

ice Ice

max Maximum

s Solid

w Water

1. Introduction

In the United States of America the exploration of space has been divided into five programs: Mercury and Gemini, which laid the groundwork for future exploration; Apollo which visited the Moon; Skylab which established a long-term presence in space; and the current Shuttle program which focuses on short-term, Low Earth Orbit-based missions. The next program for the United States is the International Space Station, which will provide a base for a continuous multi-national presence in space.

A common thread among each of these space programs is the use of astronauts, outside of their spacecraft, in a self-sufficient, self-contained capacity, who conduct experiments, perform maintenance and repairs, and explore new worlds. The technology that allows these astronauts to perform in space is directly related to the environment each program encounters.

1.1 Extravehicular Activity

When the astronaut leaves the spacecraft to perform a task he/she performs an Extravehicular Activity (EVA)*. Alexei A. Leonov and Edward H. White were the first Soviet and American astronauts, respectively, to perform EVAs. White ventured from his Gemini capsule on June, 3 1965.

---

* See List of Abbreviations located at the beginning of the paper for a complete listing of terms and acronyms used.

His spacewalk lasted about one-half hour.⁽¹⁾

Over the course of the Gemini program, astronauts logged 12 more hours of EVA time. In later programs even more EVA hours were logged. One hundred sixty-five hours were spent outside the spacecraft during the Apollo program. Of those hours, 161 were spent walking on the surface of the moon. The Skylab program added 82 more EVA hours.⁽²⁾ Space Shuttle astronauts have so far logged hundreds hours of EVA time.⁽³⁾

There is a continuing need for astronauts to “Go EVA”. Mobility outside and away from the ship is required for repair and maintenance of the craft itself. Sixty-five hours of unscheduled EVA were conducted during the Skylab program to repair damage the space station suffered during deployment. Conducting experiments and the repair and service of satellites and other space hardware will require astronauts to venture outside the spacecraft. The building of the International Space Station is anticipated to require hundreds of EVA-hours.⁽⁴⁾

1.2 Components of a space suit

When an astronaut ventures outside the protective environs of the spaceship he/she must take along an environment that will provide the life support systems needed to survive. During an EVA the astronaut wears a number of life support items that are collectively termed the Extravehicular

Mobility Unit (EMU).

There are 19 separate systems, or modules, that make up the current Shuttle spacesuit or EMU.⁽¹⁾ These systems are shown in Figure 1. The Shuttle EMU systems provide the astronaut with pressure, thermal insulation and micrometeorite protection. Oxygen, climate control, drinking water, food and waste collection are all provided within the suit. Electrical power and communications equipment are also supplied. Table 1 contains a breakdown of the EMU equipment and lists their primary uses.

1.3 Spacesuit improvements

The Shuttle EMU provides an excellent basis from which to perform Low-Earth Orbit (LEO) EVAs. However, suit requirements for the International Space Station (ISS) may necessitate changes to some of the current Shuttle EMU systems due to the long mission duration and multiple EVA scenarios envisioned during the construction and occupancy phases of the ISS.

Changes envisioned for the Shuttle EMU to enable it to be used as an International Space Station EMU include increasing suit pressure to 8.3 psia, allowing the capability to resize and service the EMU in orbit and conserving

Shuttle Extravehicular Mobility Unit

Figure 1. Components of the Shuttle EMU

Table i. Shuttle EMU Item List (From Ref. 1)

Item	Description	Item	Description
Portable Life Support System (PLSS)	Provides oxygen, CO ₂ removal, warning system, electrical power, water cooling system, and radio	Display And Control Module (DCM)	Contains all controls, digital, and external interfaces
EMU Electrical Harness (EEH)	Provides bio-instrumentation and communications connections to PLSS	Secondary Oxygen Pack	Supplies 30-minute emergency supply
Service & Cooling Umbilical (SCU)	Provides interface between EMU and orbiter	Battery	Supplies electrical power
Contaminant Control Cartridge (CCC)	Cleanses suit atmosphere	Hard Upper Torso (HUT)	Upper torso of suit
Lower Torso	Suit pants	Arms (left And right)	
EVA Gloves	Tool tethers and wristwatch built-in	Helmet	Plastic pressure bubble
Liquid Cooling Garment (LCG)	Provides cooling water	Urine Collection Device (UCD)	Urine collection for male crew members
Disposable Absorption And Containment Trunk (DACT)	Urine collection for female crew members	Extravehicular Visor Assembly (EVA)	Provides sun-filter in visor, headlamps, and TV camera-transmitter
In-Suit Drink Bag (IDB)	Provides drinking water for crew members	Communications Carrier Assembly (CCA)	Provides earphones and microphone for crew member
Airlock Adapter Plate (AAP)	Provides mounting and storage fixture for suit		

cooling water that is vaporized in the suit's thermal regulation sub-system during EVA. Current suit pressure is 4.3 psia but the shuttle cabin pressure is 14.7 psia. The difference in pressures requires a lengthy pre-breathe period prior to EVA to acclimate the astronaut to the reduced suit pressure. By constructing a suit designed for an 8.3 psia pressure, the pre-breathe time can be saved.

The ability to resize the suit in orbit would allow greater flexibility in crew selection and EVA work schedules. In-orbit servicing of the Space Station EMU will be required for crew members and their EMU's staying in orbit for weeks or months at a time. Current EMU design restricts a suit to 21 hours of use before return to Earth for complete cleaning and re-certification,^(1, 4) which is not amenable to Space Station service.

In the current Shuttle EMU design, cooling water used for thermal regulation of the suit is not conserved. To remove metabolic heat generated by the astronaut water is sublimated into the vacuum of space. Up to 5.44 kg (12 lb.) of water can be vaporized into space with each 8-hour EVA.^(5, 6) Eliminating this loss of water during EVA will conserve a resource that is precious in a space environment.

1.4 An alternative technology for ISS EMU thermal regulation

Thermal regulation for the spacesuit consists primarily of removing metabolic heat generated by the astronaut from the body area of the suit. The thermal regulating system collects, transports, then either stores or removes this heat.

An alternative technology for the ISS EMU, proposed by the NASA-Ames Advanced Life Support Branch, is the Direct Interface Fusible Heat Sink (DIFHS).^{7, 8, 9)} The system offers a means to store the metabolic heat generated by the astronaut. Storage is possible due to the absorption of the metabolic heat as a rise in the temperature of and a phase change in the DIFHS material. The DIFHS medium is water, which changes from ice (solid) to water (liquid) during use. By storing the heat for expulsion after the conclusion of the EVA, no resources, such as water, are expended during the EVA. Expulsion of the heat is done after the completion of the EVA, by freezing, or regenerating, the DIFHS water for the next EVA. This offers the advantage of eliminating the sublimation system used in the Shuttle EMU, thereby conserving the water that would have been sublimated into space during each EVA. The use of the Direct Interface Fusible Heat Sink for thermal regulation of the EMU in multiple-EVA sorties conserves resources compared to the Shuttle spacesuit sublimator unit.

A full description of the Direct Interface Fusible Heat Sink is presented in Chapter 2. In general terms the DIFHS absorbs metabolic heat by passing cooling water across the astronaut's body, transports the absorbed heat by pumping the water to the backpack area of the EMU, then allows the now warm water to directly contact and transfer the absorbed heat to the DIFHS material, which is ice. The DIFHS material increases in temperature, then melts, in order to accept the metabolic heat. At start-up, the transporting water must melt a passage through the DIFHS ice to form a channel through which the water may flow continuously during the remainder of the EVA. After the EVA, the DIFHS is removed from the EMU backpack, placed in a freezer and refrozen for use in an up-coming EVA.

The two disadvantages of the DIFHS has had hitherto were inconsistent heat transfer rates in the melted channel during use and long refreezing times during regeneration.^(7, 8)

Heat transfer within the DIFHS is primarily a function of the mixing of the inlet water and the ice block inside the DIFHS.⁽⁷⁾ Based on experiments, Lomax and Webbon⁽⁷⁾ found that at flow initiation heat transfer rates were high. After a time, the heat transfer rate would drop due to coring through the block. This occurred after the water had melted a passage through the ice. Laminar flow through the passage insulated the inner core of water which contributed to reduced heat transfer rates.

Regeneration of the DIFHS consists of removal of the tank from the PLSS system to re-freeze the water in a freezer. In a 1-g environment, freezing is enhanced by natural convection caused by temperature-induced density gradients within the container. Convection allows the physical transport of cold water away from the container walls and its replacement with warmer water from the center. In a micro-gravity situation density changes are not accompanied by convection, and the transport of heat from the center must be accomplished by thermal conduction alone.^(8, 10) As is shown in Chapter 2, the reliance on thermal conduction alone may exceed the allowable time period.

A possible solution to the disadvantages of uneven heat transfer during use and a long regeneration time afterwards is the introduction of an interlaced material into the DIFHS container. This material would be in intimate contact with the fusible medium and the container walls to provide thermal pathways for improved heat conduction. Heat transfer during use would be increased due to the heat exchange between the fusible media, the interlaced material and the flow stream. Long regeneration times would be reduced by increasing the effective thermal conductivity of the material, both in the liquid and solid (ice) phases.

A possible candidate for use as an interlaced material in the DIFHS is an aluminum foam, which possesses a continuous open-pore matrix

structure and a high thermal conductivity.⁽⁸⁾ Open pores allow the efficient flow of the fluid stream through the DIFHS. A continuous matrix aids in structural stability. These attributes can be seen in the foam shown in Figure 2. The high thermal conductivity relative to water, which is the fusible medium, provides the increased heat transfer properties needed to reduce the disadvantages of coring and long regeneration times in the DIFHS.

The improvement to the heat transfer properties to the DIFHS system, with the addition of the aluminum foam, can be estimated from empirical correlation equations. The heat transfer coefficient for the addition of the aluminum foam into the fluid stream can be estimated to a first approximation as a bank of staggered aluminum rods. The estimation of the heat transfer by this method provides a reliable means to predict the benefit attained in using an interlaced material.

The effective thermal conductivity is the sum of the combined thermal conductivities of the water and aluminum foam. Estimation of the effective thermal conductivity, compared to the heat transfer, is not well established or consistent at this time.^(11, 12) As explained in Chapter 2, the empirical prediction of the effective thermal conductivity can vary by as much as an order of magnitude, based on the orientation of the materials and their relative thermal conductivities. Therefore, the effective thermal conductivity is typically measured experimentally.

Figure 2. Aluminum foam as a candidate interlaced material (Photo courtesy of NASA-Ames Advanced Life Support Branch)

In this study the effective thermal conductivities of nine sets of aluminum foam and water were measured. It was postulated that the addition of the aluminum foam would increase the effective thermal conductivity of the DIFHS system, thereby decreasing the time required for regeneration.

Chapter 2 of this report contains a review of the cooling requirements for the Shuttle EMU and the proposed use of the Direct Interface Fusible Heat Sink as a replacement technology for the current sublimator for thermal

regulation. The use of aluminum foam as a medium for increasing the effective thermal conductivity of the system, with emphasis on a selection of models that predict the effective thermal conductivity is also described.

2. Background

This chapter is divided into four sections. First, EMU thermal regulation subsystem requirements and the use of the Direct Interface Fusible Heat Sink as a potential technology for the ISS EMU are discussed. Second, the effect of using an interlacing material, such as the aluminum foam in this study, to improve heat transfer rates are examined. Equations are developed to predict the benefit of an interlacing material in the DIFHS. Third, the relevance of using effective thermal conductivity models from the published literature is discussed, and, fourth, an overview of the issues concerning the experimental measurement of the effective thermal conductivity of a solid/liquid system is presented.

2.1 Thermal regulation subsystem performance requirements

Metabolic heat generated by the astronaut during an EVA is the primary source of heat the cooling system must remove from the body area. About 100 Watts (400 BTU/hr) of heat is produced by an astronaut at rest. This heat output can easily increase by an order of magnitude during periods of heavy work.^(6, 7, 8) Current total capacity requirements are 3420 Watts (11680 BTU/hr) for an 8-hour EVA with a maximum cooling capacity of 585.5 Watts (2000 BTU/hr) for any 15 minute period.⁽⁸⁾ Table 2 is a listing of the cooling requirements for a Low Earth Orbit EVA regulation system.

Table 2. Cooling Requirements for a Low Earth Orbit EVA (From Ref. 8)

Parameter	Requirement
EVA Duration	8 hours
EVA Environment	LEO (any Orientation)
Minimum Heat Load	117.1 Watt (400 BTU/hr)
Maximum Heat Load	585.6 Watt (2000 BTU/hr)
Total Heat Capacity	3420 Watt-hr (11680 BTU)
Outlet Temperature	<15.6°C (60°F)
Regeneration Time	<16 hours
EVA Frequency	8 hours/day , 6 days/ week
Mission Duration	6 months
Operational Life	1228 hours

Additional heat sources and sinks exist which include heat generated by the life support equipment and external sources. The average heat load generated by the life support equipment in the PLSS is 186 Watts (635 BTU/hr).^(7, 8, 10)

The astronaut's overall heat balance is affected by the ambient conditions encountered during the EVA. External heat transfer to and from the suit are mainly through radiation. Infrared energy can be absorbed from the sun, earth and nearby hot structures such as the payload bay of the shuttle.

The astronaut simultaneously radiates energy into deep space.

Additional heat transfer to/from the suit can occur via conduction through contact with objects such as structures, tools and equipment. Contact with the lunar or Martian surface during planetary missions will also impose a heat load (positive or negative) on the astronaut.

Convective heat transfer is also possible in certain situations. Mars has an atmosphere in which convective heat transfer will play a significant part of the total heat load equation.

2.1.1 Components of the thermal regulation system

The thermal regulation system is composed of three processes: 1) acquisition of the thermal energy from its sources (metabolic, instrumentation, ambient, etc.); 2) transport of the energy through the system; and 3) storage and/or removal of the heat from the suit environment.

The acquisition of metabolic heat is accomplished by use of the Liquid Cooling Garment (LCG). The LCG is worn against the body under the spacesuit. The LCG is a Spandex™ garment laced with tubing within which the cooling water is circulated, and is shown in Figure 3.^(1, 2) Heat transportation to and from the LCG and within the EMU is accomplished by pumping the cooling water through the system.

Storage/removal of the energy delivered by the water transportation system can be accomplished by a number of technologies. The current process of sublimating the heat into the vacuum of space has been proven over years of service to be an economical and reliable system for use in the Shuttle EMU. Possible heat storage and/or removal technologies for use in an ISS EMU include heat pipe/radiators, evaporative membranes and the aforementioned Direct Interface Fusible Heat Sink.^(6, 9)

Figure 3. Liquid Cooling Garment, (LCG)
(From Ref. 1)

Current NASA Shuttle EMU's utilize a flat plate sublimator as the heat

removal unit. A schematic of the flat plate sublimator is shown in Figure 4. The sublimator unit is comprised of three main sections: a porous stainless steel plate, a feed-water channel located below the plate and the LCG water loop. One side of the porous plate is vented to the vacuum of space. The other side is connected to the feed-water channel. Water pumped from the feed-water channel is allowed to percolate into the porous metal plate. A phase transformation occurs as the water "sees" the vacuum on the far side of the plate. The feed-water may freeze in the pores of the stainless steel plate and then evaporate/sublimate. The phase transformation cools the remaining water in the feed-water channel which in turn absorbs heat from the LCG coolant -water loop.^(5, 6, 13)

2.1.1.1 Advantages and disadvantages of the sublimator

The flat plate sublimator currently used in the NASA Shuttle suit has the advantage of low unit weight and small size compared to alternative cooling systems. These two advantages weigh heavily in the decision to continue using this technology. Conversely, mass depletion, fouling of the porous plate and a required maximum pressure for sublimation are the flat plate sublimator's three main deficiencies, which encourage alternative technologies to be studied.

Figure 4. Flat Plate Sublimator (From Ref. 5)

The flat plate sublimator possesses the advantage of in-use replenishment of its fusible material (feed-water) and a compact size within the PLSS. The sublimator requires a 5.12 kg (12 lbm) supply of feed water for sublimation cooling for an 8-hour EVA.⁽⁵⁾ This feedwater supply is augmented during the EVA by water vapor condensed from the perspiration and exhaled air of the astronaut.⁽¹⁾ This allows the sublimator system to extend its usable supply and/or maintain reserves for emergency use. The

largest component of the sublimator is the flat porous stainless steel plate which presents an area of approximately 930 cm^2 (1 ft^2) towards the vacuum of space.

The disadvantages of the flat plate sublimator are the loss of feed-water, fouling of the porous metal plate and the inability to operate above a maximum ambient pressure. During use the feed-water is constantly sublimated into space. This mass cannot be recaptured. Multiple-EVA orbital missions, such as during the construction phase of the International Space Station and long-duration missions which include lunar missions and Space Station assignments, will be required to carry a large supply and/or source of feed-water for sublimator use.

Fouling or clogging of the porous metal plate with prolonged use is also a possible problem. Scheduled servicing of the sublimator plate will be required for long-duration missions. The sublimator is restricted to 21 hours of use before return to Earth for complete cleaning and re-certification.^(4, 5)

The sublimation of solid water (ice) to vapor requires that the low-pressure side of the sublimator be below the triple point of water, which is (0.088 psia at 32°F). The thermal environment of Mars consists of an atmosphere whose total pressure is approximately 0.08 psia and a surface temperature which can range from -130 to +40°F depending upon the location

and season.⁽⁶⁾ Therefore, because of the higher pressures that may be encountered, the flat plate sublimator would not be suitable for a Mars expedition.

2.1.2 Alternatives to the sublimator system

There are several alternative cooling systems to the flat plate sublimator, of which the Heat Pipe/Radiator, Evaporative Membrane Heat Exchanger and Direct Interface Fusible Heat Sink are primary technologies. These systems offer both advantages and compromises over the current flat plate sublimator system.

The heat pipe/radiator technology uses a vaporizing fluid such as liquid nitrogen or ammonia to transport metabolic heat to radiator plates placed on the outside of the suit backpack. The heat pipe fluid absorbs heat from the LCG water loop and vaporizes. A thermal gradient along the heat pipe causes the vapor to travel to the radiator end where it condenses. The radiators eject the transported heat into deep space. The condensed fluid circulates back to the vaporizing end of the heat pipe by capillary action along re-entrant grooves.^(14, 15) The cross-section of a heat pipe tube in Figure 5 shows the re-entrant grooves around the inner diameter.

Figure 5. Heat Pipe Cross-Section with Re-Entrant Grooves (From Ref. 14)

The heat pipe/radiator system takes advantage of the 7.2K (4°R) deep space radiation sink temperature by exposing a large area to space. However, radiators sized to handle the maximum heat loads anticipated during EVA would be prohibitively large.⁽⁶⁾ Use of a radiator-based system for EVA use would probably be sized for average EVA heat loads and coupled with a “top-off” cooling system, such as a sublimator or evaporative membrane heat exchanger, to handle the higher heat loads during strenuous astronaut activity.

2.1.2.1 Evaporative Membrane Heat Exchanger

The Evaporative Membrane Heat Exchanger ejects metabolic heat into space by boiling or vaporizing a small portion of the LCG water through a polymeric membrane. The Evaporative Membrane uses a membrane that is impervious to liquid water but allows water vapor to pass through. The opposite side of the membrane is exposed to the vacuum of space. LCG water is pumped past the membrane. The low pressure on the opposite side causes the water next to the membrane to evaporate or boil. The energy required to vaporize this water is taken from the remaining LCG water behind it. The now vaporized water can escape through the membrane into space. The remaining LCG water is pumped back to the Liquid Cooling Garment. The vaporization of water is controlled by controlling the partial pressure of water on the vacuum side of the membrane. In preliminary form the membrane is formed as a tube with the liquid water on the outside and the inner side vented to space. The partial pressure is controlled by sizing the inner tube diameter to match the diameter for choked flow through an orifice.^(16, 17, 18) A prototype version of the Evaporative Membrane Heat Exchanger is shown in Figure 6.

Figure 6. Evaporative Membrane Heat Exchanger During Testing in a Vacuum Chamber

The Evaporative Membrane promises to substantially reduce the overall size and mass of the thermal regulating unit compared to the current flat plate sublimator. The Evaporative Membrane uses a membrane surface area approximately 85% smaller and 90% lighter than the flat plate sublimator. The feed-water loop is discarded, resulting in additional weight and mass savings due to direct use of the LCG water as the phase transformation medium. Since the Evaporative Membrane uses the latent heat of vaporization as the cooling source, higher atmospheric (vacuum)

pressures can be endured. Atmospheric pressures up to the equilibrium vapor pressure at the LCG water temperature, which is typically 20°C (68°F), are possible. This higher pressure ceiling allows the Evaporative Membrane to be a candidate for a Mars expedition suit.

The disadvantages of the Evaporative Membrane result from its development as a new technology. The reliability of the polymer membrane in a space environment is not known at this time and the practical problem of controlling the partial pressure on the vacuum side requires further study.

2.1.3 Direct Interface Fusible Heat Sink

The Direct Interface Fusible Heat Sink (DIFHS) stores the metabolic heat generated by an astronaut. The heat is stored by using the heat of fusion to transform a suitable material from one phase to another. Such a suitable phase change material is water from solid (ice) to liquid. The liquid water is refrozen during a system regeneration after the EVA is completed. The DIFHS is unique in that the fusible material, water, is also the medium used to transport the heat from the garment to the PLSS. Through direct interface, or contact, of the LCG water and the DIFHS ice, cooling by forced convection as well as phase transformation can be obtained.^(7, 8)

The DIFHS has advantages that make it a viable alternative to the flat plate sublimator. Water is conserved so multiple-EVA operations become

more economical. The DIFHS can be regenerated for additional EVA sorties on-board. Additionally, the large ice mass required for an 8-hour EVA is not a hindrance in micro-gravity (orbital) EVA scenarios.⁽⁷⁾

The relatively low heat of fusion of water requires that a large mass of ice/water be carried by the astronaut. While not an important issue for micro-gravity EVA's, the DIFHS may not be an acceptable alternative for a Mars expedition due to the relatively large gravitational force there.⁽⁷⁾

A selection of alternative methods for cooling an astronaut during EVA have been presented. For use in the International Space Station EMU, the DIFHS may provide the best choice among alternative methods, due to its capability for no expendables and in-flight regeneration. In Section 2.2 the capacity requirements required for the DIFHS and the use of an interlaced material to assist in the heat transfer to and from the system are developed.

2.2 Heat transfer

The mass of water required for a full sized DIFHS can be calculated. With a total cooling capacity of 3420 W-hr (11680 Btu) required for an 8-hour EVA, a DIFHS starts with ice initially cooled to -17.8°C (0°F) which melts, then warms to 4.4°C (40°F). With heat capacities, C_p , of ice and water of 2.052 Joule/ $^{\circ}\text{C gm}$ and 4.216 Joule/ $^{\circ}\text{C gm}$, respectively and a heat of fusion, L_{fusion} , of 333.7 Joule/gm, the mass of water can be calculated from Equation [1]:

$$q = mCp_{ice} \Delta T_{ice} + mL_{fusion} + mCp_{water} \Delta T_{water} \quad [1]$$

Therefore, a full sized DIFHS would hold approximately 31.7 kg (70 lb.) of water . The volume equivalent to this is 0.032 m³ (1.12 ft³) of water or 0.035 m³ (1.25 ft³) of ice.^(7, 8)

Prototype DIFHS units have been developed and tested at NASA-Ames Research Center.^(7, 8) Design issues addressed in the DIFHS prototypes have included measurements of the heat transfer rate during use and the time required for regeneration.

Inconsistent heat transfer during use and long regeneration times to freeze the melt water are the major development issues for the DIFHS. In experiments by Lomax and Webbon,⁽⁷⁾ after the initialization of flow through the DIFHS, heat transfer between the fluid stream and the surrounding ice decreased due to inadequate fluid mixing. Attempts to increase the turbulence of the fluid stream resulted in increased heat transfer rates. In later work, Lomax and Kader⁽⁸⁾ measured the regeneration time of a 1/2 scale DIFHS unit at 17 hours. Possible solutions proposed in these studies include baffling the interior of the tank with fins or using an interlaced material.

2.2.1 Baffles

Baffling the tank with cooling fins could increase the conduction

surface area for heat transfer and allow passage of the LCG water flow without large penalties in pressure.⁽⁷⁾ However, baffles require mass and volume increases to the overall DIFHS system. Baffling could also slow or prevent flow initiation due to long passages between inlet and outlet couplings. Additionally, channeling may still occur in a baffled tank.⁽⁷⁾

The increased heat transfer that could be obtained using baffles can be estimated to a first approximation by using a simple laminar forced convection flow model. Assuming the LCG water melts first, then flows through a passage of constant circular cross section, the heat transfer within the DIFHS can be modeled as a simple flow through a tube. Figure 7 is an illustration of both the non-baffled tank and one with a baffle which increases the flow stream's length. In the baffled model the total length of the melt passage was assumed to be double that of the unbaffled model.

A maximum mass flow rate of 240 lb/hr for the LCG water system was used to determine the laminar flow regime.⁽⁸⁾ The average heat transfer coefficient, $\bar{h}$, was calculated from the product of the average Nusselt number and the thermal conductivity, k , divided by the cross-sectional diameter, D , in Equation [2]. The Hausen equation⁽¹⁹⁾ for a constant wall temperature, presented below as Equation [3], provides the average Nusselt number, $\overline{Nu_D}$.

Figure 7. Fluid Streams Through Non-Baffled And Baffled Tanks

$$\bar{h} = \overline{Nu_D} \frac{k}{D} \quad [2]$$

$$\overline{Nu_D} = 3.656 + \frac{K_1 \left[\frac{D}{L} Re_D Pr \right]}{1 + K_2 \left[\frac{D}{L} Re_D Pr \right]^n} \quad [3]$$

where L is the length, Re_D and Pr are the Reynolds and Prandtl numbers, respectively, and K_1 , K_2 and n are constants based on inlet conditions. In Figure 8, the heat transfer coefficients for laminar flow forced convection are plotted for a non-baffled and baffled DIFHS. The heat transfer lengths were normalized to equivalent volumes of ice and melt water. Although the effects of adding a baffle added only about 9% to the overall heat transfer coefficient in this example, there is a positive result to manipulating the fluid stream.

Figure 8. Heat Transfer Coefficient For Laminar Flow Forced Convection

2.2.2 Interlaced materials

The use of an interlaced material to aid in heat flow in the DIFHS was first proposed in the work by Lomax and Kader.⁽⁸⁾ They suggested the use of an expanded aluminum foam to reduce the time required for regeneration of the melt water. It was hypothesized that the relatively high thermal conductivity of the interlacing material would improve the transport of heat from the center of the melt water to the outside of the container where it would be convected away by the freezer system. The use of an interlaced

material would also be used to increase the heat transfer properties of the LCG water in the DIFHS during use.

In the same manner as for the addition of baffles, it can be shown that an interlaced material may also improve the heat transfer coefficient of the fluid stream in the DIFHS. As a first approximation, the interlaced material can be modeled as a bank of staggered tubes lying perpendicular to the fluid flow.

The average heat transfer coefficient is estimated by Grimson⁽¹⁹⁾ and is presented in Equation [4] as

$$\bar{h} = \frac{k_f}{D} C_1 (\text{Re}_{\max})^n \quad [4]$$

where k_f is the thermal conductivity of the fluid, D is the diameter of the tendrils, C_1 and n are constants. The maximum Reynolds number, $\text{Re}_{\max}$, is defined as

$$\text{Re}_{\max} = \frac{V_{\infty} L_{\text{pore}}}{(L_{\text{pore}} - D)} \frac{D}{\nu_f} \quad [5]$$

where V_{∞} , L_{pore} and ν_f are the fluid velocity, length of the pore in the direction of heat flow and the kinematic viscosity, respectively.

Applying Equation [4] to a DIFHS with a maximum LCG mass flow of 240 lb/hr, and with an interlaced material with tendril diameter of $D = 1$ mm and pore length of 4 mm results in predicted heat transfer rates over 12 times greater than without an interlaced material. It is also significantly better than for the case with baffles. The graph of predicted heat transfer coefficients versus a normalized volume of melt water is shown in Figure 9.

The addition of an interlaced material into the DIFHS fluid stream would greatly increase the heat transfer over using no additional material or baffles during use. An interlaced material may provide a practical solution to the inconsistent cooling rates encountered in previous DIFHS prototype testing.

2.2.3 Time to freeze

The use of an interlaced material may decrease the regeneration time for the DIFHS. In a microgravity environment, heat transfer through the melt water is by conduction or forced convection only, due to the lack of natural convection which is the physical movement of fluid caused by buoyancy forces arising from density changes. The effects of buoyancy forces, that is, forces due to gravity, are negligible in a microgravity environment.⁽⁶⁾

Figure 9. Heat Transfer Coefficients Using An Interlaced Material Of 135W/mK.

The time required for regeneration of the DIFHS can be separated into three distinct processes. First the warm melt water is cooled to the phase transformation temperature. Second, the phase transformation from water to ice occurs; third, the ice is cooled further to the DIFHS start temperature. As a first approximation for the regeneration time, these three processes can be considered to be independent of each other which leads to Equation [6]

$$t_{regeneration} = t_{coolwater} + t_{phase\ transformation} + t_{coolice} \quad [6]$$

Using a lumped thermal capacity approach to determine the time for each process, the resulting Equations [7], [8] and [9] are

$$t_{cool\ water} = \frac{T_i - 32}{T_i - T_{freezer}} c_p^{H_2O} \rho_{H_2O} \left[\frac{r^2}{2} \left(\frac{1}{2k_w} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{k_a} \right) \right] \quad [7]$$

$$t_{phase\ transformation} = \frac{-h_f \rho_{H_2O}}{32 - T_{freezer}} \left[\frac{r^2}{2} \left(\frac{1}{2k_i} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{k_a} \right) \right] \quad [8]$$

$$t_{cool\ ice} = \frac{32 - T_{end}}{32 - T_{freezer}} c_p^{ice} \rho_{ice} \left[\frac{r^2}{2} \left(\frac{1}{2k_i} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{k_a} \right) \right] \quad [9]$$

where T_i is the initial melt water temperature, T_{end} is the final temperature of the ice, $T_{freezer}$ is the temperature of the freezer, and c_p , ρ , r and r_o are the heat capacity, density and inner and outer radii of the container, respectively. In order to alleviate confusion, the heat capacity and density of each phase is denoted with superscripts and subscripts, respectively.

Assuming an initial melt water temperature of 4.4 °C (40 °F), a final temperature of -17.8 °C (0°F), and a freezer temperature of -23.8 °C (-6 °F) this equation predicts a regeneration time of 17.8 hours for a 0.32 m (12.5 inch) diameter cylinder. This compares favorably to the regeneration time of 17 hours reported by Lomax and Kader for their 1/2 capacity DIFHS unit.⁽⁸⁾

From the above equations, economies in the regeneration time can be realized from the use of lower freezer temperatures, decreased radii for the DIFHS container and increased thermal conductivities for the fluid and/or the solid phases.

Lower freezer temperatures can reduce the time to freeze for the DIFHS unit. However, reduced temperatures can be directly related to higher heat loads. For radiator freezers, reduced freezer temperatures create higher heat rejection loads which in turn require increased radiator areas to expel the heat into space. Refrigeration systems operating at a greater ΔT require increased power to reject these higher heat loads.⁽⁶⁾

Decreasing the total conduction distance in the DIFHS has the cost of increasing the mass of the container. Given a constant total heat load that the DIFHS must absorb, the fusible material volume of the DIFHS remains constant. Deviations from a spherical container, which has the minimum surface area to volume ratio, increases the mass of the container. Changing the container shape from spherical to cuboid will probably be necessary to fit the unit into the EMU backpack while simultaneously improving the regeneration time. Increasing the thermal conductivity of the liquid and solid phases will reduce the regeneration time by increasing the heat transfer capabilities within the system.

The effect of increased effective thermal conductivities for both the liquid and solid phases on the regeneration time is shown in the graph in Figure 10. The time required for regeneration is plotted against the increased effective thermal conductivity. In this model, it was assumed that the increases in the conductivity values of the liquid and solid phases are proportional. This assumption will be shown to be a valid approximation in Chapter 3.

The addition of an interlacing material inside the Fusible Heat Sink could be used to increase the heat transfer properties of the warm fluid stream to the ice block during use and to increase the relative, or effective, thermal conductivity of both the melt water and the forming ice mass during regeneration. The use of an interlaced material may be advantageous over baffles because the volume of ice that the fluid must melt through during the start-up condition would be shorter. Additionally, the stresses induced on the tendrils of the interlaced material as the fluid freezes may be less severe than on fixed baffles.

Figure 10. Plot Of The Time To Freeze As A Function Of Effective Thermal Conductivity

2.2.4 Duocel Aluminum Foam Metal

Prior to this study, a search for potential interlaced materials was performed.⁽⁸⁾ Interlaced materials made of a variety of metals including aluminum, nickel, silver and copper, are available. Foams made of other materials, such as silicon carbide,⁽²⁰⁾ vitreous carbon⁽²¹⁾ and cordierite⁽²²⁾ are also available, but are typically considered thermal insulators. These interlaced materials have material properties typical of their parent materials.⁽²⁰⁾

A candidate interlaced material is the Duocel Aluminum Foam Metal. It is a proprietary product of the ERG Materials and Aerospace Corporation, of Oakland, CA. A photograph of this material is shown in Figure 11. Formed of 6061-T0 aluminum alloy, it has an open-celled structure. The material can be characterized as a skeletal structure of continuous solid metal ligaments forming a matrix of cell-like voids of nominally uniform shape. The void space is a tetrakaidecahedral structure which approximates the equiaxed grain structure in a solid material. The density, strength and thermal conductivity of the tendrils are all typical of the parent 6061 alloy material.⁽²³⁾

Duocel is manufactured in a range of pore sizes and volume fractions, from 5 to 40 pores per inch and 3 to 12 percent by volume, respectively. A selection of different pore sizes, each with a relative volume fraction of 8% is shown in Figure 12. Pore size and relative volume fraction are independently variable within a range of values.⁽²³⁾

Duocel represents a candidate interlaced material that can overcome some of the performance issues of the Direct Interface Fusible Heat Sink. Inefficiencies in heat transfer due to coring, channeling, inefficient mixing and long regeneration times may be reduced or even eliminated. It fulfills the high strength, low density, high void fraction and open pore configuration required for an ideal interlacing material. Its thermal conductivity value is that of the parent material, approximately

Figure 11. Duocel Aluminum Foam Metal (Photo courtesy of NASA-Ames Advanced Life Support Branch)

135 Watt/mK. The ability to specify both the pore size and volume fraction to optimize the material for use in the DIFHS makes it an attractive candidate.

2.2.4.1 Thermal conductivity of the aluminum foam

The effect of the interlaced material on the time-to-freeze was a primary factor in the selection process. The thermal conductivities of 6061 aluminum alloy and water are 135 Watt/mK and 0.607 Watt/mK, respectively.⁽²⁴⁾ Therefore, the heat transfer properties of the aluminum in concert with the DIFHS water may be utilized to enhance the thermal conductivity of the system.

Figure 12. Aluminum Foam Specimens With Equal Volume Fraction (8%) But Different Pore Sizes (Photo courtesy of NASA-Ames Advanced Life Support Branch)

2.3 Overview of effective thermal conductivity models

The effective thermal conductivity of a multi-phase system is a function of the total heat transfer of the system. In theory, the effective thermal conductivity is the summation of all the various paths of heat transfer across and between each of the phases. In practice, models have been developed that use a set of measurable system parameters to predict the overall heat transfer. In this section a selection of models are examined, to

determine whether a model may be useful in predicting the effective thermal conductivity of the aluminum foam and water system.

A search of the published literature resulted in ten models being identified. Each predicts the effective thermal conductivity of a two phase system based either exclusively or primarily on volume fraction of the phases. The models presented here are divided into four groups. First, the bounding equations for effective heat flow are reviewed. Second, models which are based on a system of volume averaging are examined. In the third group, structure-based models are presented and fourth, models which include additional terms for heat transfer via convection and radiation are reviewed. Excellent reviews of the existing literature are presented by Woodside and Messmer⁽¹¹⁾ and Collinshaw and Evans.⁽¹²⁾

2.3.1 Boundary equations

The absolute upper and lower bounding equations for a two-phase system can be set by using the volumetric law of mixtures. Weighted averages of the conductivities and resistivities of the constituent phases produce the absolute upper and lower bound equations, respectively. These bounding equations are presented as the arithmetic and harmonic models.^(11, 12)

The arithmetic mean equation mimics the volumetric law of mixtures

for parallel streams. This represents an upper bounding equation for heat flow in two phases. As shown in Equation [10], it equates the effective thermal conductivity as the sum of the weighted averages of the conductivities of the individual phases.

$$k_{eff} = \varepsilon_f k_f + (1 - \varepsilon_f) k_s \quad [10]$$

The arithmetic mean model is continuous; as ε_f , the volume fraction of the fluid phase, approaches 0, k_{eff} approaches k_s . The model implicitly assumes one or more structures for each phase parallel to the direction of heat flow with no interaction between the phases.

The harmonic mean model represents the lower bound of effective conductivity. The harmonic mean assumes heat flow through independent phases located in series. It is mathematically represented as the weighted average of the resistivities of the phases and is presented here as Equation [11].

$$k_{eff} = \frac{k_s k_f}{\varepsilon_f k_f + (1 - \varepsilon_f) k_s} \quad [11]$$

In the same fashion as the arithmetic model, the harmonic mean model is continuous and assumes no interaction between phases. In practice, the effective conductivity is biased towards the lesser conductive material.

Maxwell, in his treatise on electrical conductivity in 1861, developed bounding equations for two special cases of combined phase materials. Although these equations were originally developed for electrical conductivity, they are equally valid for thermal flows.^(11, 12, 25, 26)

Maxwell's upper bound equation examined a solid matrix containing a suspension of spherical fluid-filled voids or closed pores. It was assumed that the pore volume was small and at the limit, as $(1 - \varepsilon_f) \rightarrow 1$, Maxwell's upper equation approaches Equation [12].

$$k_{eff} = k_f \left[\frac{2 \frac{k_s}{k_f} (1 - \varepsilon_f) + (1 - 2\varepsilon_f) \frac{k_s}{k_f}}{(1 - 2\varepsilon_f) \frac{k_s}{k_f} + (1 - \varepsilon_f)} \right] \quad [12]$$

Maxwell's lower bound equation is presented as Equation [13]. Here Maxwell assumes a random distribution of solid spheres in a continuous medium.

$$k_{eff} = k_f \left[\frac{2\varepsilon_f k_f + (3 - 2\varepsilon_f) k_s}{(3 - \varepsilon_f) k_f + \varepsilon_f k_s} \right] \quad [13]$$

Because of Maxwell's assumption that the spheres do not interact with each other, Equation [12] is limited to when ε_f is large, that is when $\varepsilon_f \rightarrow 1$.

Maxwell's bounding equations are based on the assumption that the secondary phase of pores or particles are discrete and therefore non-interacting with other pores/ particles. This assumption limits the porosity range although no parameter for limiting the relative size of the pores is given in Maxwell's equation.^(11, 25) Comparing the Maxwell equations to the arithmetic and harmonic models it is observed that due to the assumption of non-interaction of the dispersed voids and particles these equations are more limiting.

These equations for maximum and minimum thermal conductivity are important to the current study in that they explicitly relate volume fraction of the phases and implicitly relate the interaction of the phases to the effective thermal conductivity.

2.3.2 Volume averaging models

For multi-phase systems, the simplest expressions for the effective thermal conductivity are those that rely only on the volume fractions of the constituent phases. In their drive towards simplicity, the effect of structure on the effective thermal conductivity is ignored. Three models, namely the geometric mean, Asaad's empirical model, and one by Hadley for packed metal powders are presented.

The Geometric Mean model is a volume fraction averaging method,

which is an intermediate of the arithmetic and harmonic models presented earlier. It is based on the weighted logarithms of the individual conductivities. Two variations, by Woodside and Messmer and by Langton and Matthews, have appeared in the literature and are presented here as Equations [14] and [15].

Woodside and Messmer's version of the geometric mean model was developed as the product of the separate thermal conductivities of each phase, raised to the power of its respective volume fraction.⁽¹¹⁾

$$k_{eff} = k_f^{\varepsilon_f} k_s^{1-\varepsilon_f} \quad [14]$$

Langton and Matthews presented the weighted geometric mean in the form of logarithms.

$$\log k_{eff} = \varepsilon_f \log k_f + (1 - \varepsilon_f) \log k_s \quad [15]$$

The advantage of the geometric mean model is its intermediate value between the arithmetic and harmonic mean models. The geometric mean assumes a linear relationship between the logarithms of the thermal conductivities of the components. Like the arithmetic and harmonic models, it is continuous across the full range of porosities. The geometric mean model does not consider the geometry of the heterogeneous system, nor does it account for any interaction effects between the individual components.

Asaad,⁽²⁷⁾ in his doctoral thesis at U.C. Berkeley submitted in 1955, noted that measured effective thermal conductivities tended to follow the power law of the geometric mean model. He allowed for an empirical factor, c (where c is approximately equal to 1), to be added into the power term of the geometric model to compensate for deviations from unity. Asaad's empirical equation is presented in Equation [16]

$$\frac{k_{eff}}{k_s} = \left(\frac{k_f}{k_s} \right)^{m_{Asaad}} \quad [16]$$

where m_{Asaad} is equal to ce_f and c is approximately unity. The value of c is adjusted to fit the experimental results. If $c=1$, then the equation reverts to the Geometric Mean equation.

Hadley⁽²⁶⁾ applied the technique of volume averaging to produce a series of equations from which the effective thermal conductivity could be calculated. Equation [17] is based on a weighted relationship between two competing routes of heat transfer. One route is conduction through a series of particles in contact with each other (parallel conduction) and the other is series conduction through a suspension of separated particles and a low conducting matrix. This second term is equivalent to Maxwell's upper bounding equation presented in Equation [12].

$$k_{eff} = k_f \left[(1 - \alpha_H) \frac{\varepsilon_f f + \frac{k_t}{k_f} (1 - \varepsilon_f f)}{1 - \varepsilon_f (1 - \varepsilon_f f) + \frac{k_t}{k_f} \varepsilon_f (1 - \varepsilon_f f)} + \alpha_H \frac{2 \left(\frac{k_t}{k_f} \right)^2 (1 - \varepsilon_f) + (1 + 2\varepsilon_f) \frac{k_t}{k_f}}{(2 - \varepsilon_f) \frac{k_t}{k_f} + 1 - \varepsilon_f} \right] \quad [17]$$

The resulting equation uses two empirical terms to weight the effects of these two modes of heat transfer. Alpha is an empirical factor that balances the first and second terms and can vary within the range of $0 < \alpha_H < 1$. The term f represents the propensity of the suspension to operate as if the heat flow was in series, from phase to phase, or in parallel, where the heat flows through each individual material independent of the other material.⁽²⁶⁾ Both of the empirical terms are determined from experimental data.

The three models presented in this section attempt to trade simplicity in the equation for exactness in the results. The equations were formulated using only the volume fractions of the phases, just as the bounding equations presented previously, but without the implicit restriction of structure, and therefore interaction, between the phases. The models developed by Asaad and Hadley use empirical factors to compensate for their generality.

2.3.3 Structure-based parameter terms

This class of models adds a parameter to describe the structure of a multi-phase system so that structure is explicitly considered in predicting the effective thermal conductivity. Mathematical schemes are used to describe

the secondary phase structure more accurately than the spherical shape typically assumed in most of the previous models.

De Vries,⁽¹¹⁾ in 1952, amended Maxwell's lower bound equation, presented previously as Equation [13], to accept particles of various shapes. This modification allowed changes to the assumed spherical nature of the particle component. De Vries applied this structure component to the calculation of the effective thermal conductivity of unconsolidated soils. The form of de Vries' equation presented as Equation [18] is for a two-phase system with a continuous fluid phase and dispersed solid phase.

$$k_{eff} = \frac{\epsilon_f k_f + (1 - \epsilon_f) F k_s}{\epsilon_f + (1 - \epsilon_f) F} \quad [18]$$

$$\text{where } F = \frac{1}{3} \sum_i^3 \left[1 + \left(\frac{k_s}{k_f} - 1 \right) g(*) \right]^{-1} \quad [19]$$

The $g(*)$ terms in Equation [19] represent geometric shape factors where $g(x)+g(y)+g(z)=1$. The set of $g(*)$ terms: $g(x)=g(y)=g(z)=1/3$ represents a sphere. In his original work, de Vries chose $g(x)=g(y)=1/8$ and $g(z)=6/8$ which represented an ellipsoidal solid whose major axis was 6 times that of the minor axes. Of course, the implicit assumption of non-interaction between the dispersed phase stated in Maxwell's development continues to hold in de Vries' equations.

The Unit Cell Model of Dulnev^(28, 29, 30), was adapted for use in porous solid materials by Luikov. In this model the heterogeneous material was modeled as two intertwined three-dimensional networks (solid-fluid) composed of repeated units. The model assumes that the overall thermal conductivity can be developed by determining the conductivity of an elementary volume, or unit cell, whose edges create a skeletal framework that approximates the continuous phase. The non-continuous phase is the resulting volume remaining.

Luikov allowed for four independent modes of heat transfer: conduction through the solid phase, conduction through the fluid medium, radiative heat transfer between the surfaces and gas convection within the pores. Additionally, contact resistance between particles may also be accounted for. The Luikov equation for a two component system with interconnected pores⁽²⁸⁾ is presented in Equation [20].

$$k_{eff} = k_s \left[\left(\frac{h}{\varphi L_{pore}} \right)^2 + \frac{k'_g}{k_s} \left(1 - \left(\frac{h}{\varphi L_{pore}} \right) \right)^2 + \frac{2 \frac{k'_g}{k_s} \frac{h}{\varphi L_{pore}} \left(1 - \frac{h}{\varphi L_{pore}} \right)}{1 - \frac{h}{\varphi L_{pore}} + \frac{k'_g}{k_s} \frac{h}{\varphi L_{pore}}} \right] \quad [20]$$

where $k'_g = k_{g:conduction} + k_{g:radiation}$ and h is the tendril diameter.

The Dulnev/Luikov model has the capability of explicitly modeling

dual continuous systems such as the aluminum foam and water system considered in this study. The pore geometry, thermal conductivities and volume fractions of the individual components are explicitly input. The addition of a characteristic pore length, ϕL_{pore} however, allows the model to compensate for deviations from strict one-dimensional flow that this modeling scheme relies on. That is, the characteristic length is calculated from empirical data so that the model agrees with experimentally determined results.

The structure-based models explicitly consider the interaction between the phases in predicting the effective thermal conductivity. These models require more information about the system than the volume averaging equations presented earlier, which may or may not increase the accuracy of the prediction. The use of empirical coefficients in some of the volume averaging models and the use of a characteristic pore length in the Luikov/Dulnev model allow the models to correlate to the experimental results which may be influenced by other factors.

2.3.4 Contact resistance and other additional terms

Additional modes of heat transfer such as contact resistance, convection and radiation may influence the thermal conductivity. Contact resistance can impede, or decrease, the heat transfer between particles or

phases. As observed in the previous model by Dulnev, the effect of contact resistance between particles can be explicitly accounted for. Heat flow via convection and/or radiation within the pores increases the thermal conductivity.^(19, 22) In practice the effects of convection and radiation decrease as the pore size and mean temperature decrease, respectively. Convection is considered negligible at pore diameters less than 6-10 mm or when the Grashof-Prandtl number (Gr. Pr) is less than 10^3 .^(12, 19, 31) Similarly, the effects of radiation are considered negligible at temperatures less than 600K.^(12, 19)

The inclusion of additional heat transfer terms such as radiation, convection and contact resistance are usually simplified as being additive. That is, the effective conductivity is the sum of the separate terms as shown in Equation [21].

$$k_{eff} = k_{conduction} + k_{radiation} + k_{convection} + k_{contact} \quad [21]$$

While the assumption of additive paths allows convenient simplicity it can also detract from exactness.⁽¹¹⁾

A model for ceramic foams with interconnected pores was developed by Zumbrunnen, Viskanta and Incropera,⁽²²⁾ and is presented as Equation [22]. This model considers many of the same parameters as the Dulnev/Luikov model. It assumes an elementary set of pathways for heat transfer within the material. For example, a set of pathways could be as follows: heat flow across

a pore, through the matrix phase, and through the matrix and then across a pore. The total thermal conductivity then is the sum of the heat flows across these pathways, weighted according to the probability of their occurrence.

$$k_{eff} = \left[\zeta + \Lambda \left(\frac{\beta + 1}{\beta} \right) + 1 \right] \left[\frac{1}{\lambda} \left(\frac{\Lambda \psi}{\beta} + \zeta \right) + (1 + \Lambda) \left(v h_{rv} + \frac{k_f}{\varphi} \right)^{-1} \right]^{-1} \quad [22]$$

Where $\varphi = \text{characteristic pore ratio} = \frac{\text{effective length for conduction in pore}}{\text{characteristic size of pores in solid}}$ [23]

$$\beta = \frac{\varphi^3}{1 - \varphi^3} \quad [24]$$

$$\begin{aligned} \psi &= 0 \text{ for open pore systems} \\ &= 1 \text{ for closed pore systems} \end{aligned}$$

$$v h_{rv} = \text{radiation contribution term} \quad [25]$$

$$\Lambda = 1 + \ln \left(\frac{\beta + 1}{\beta} \right)^{-1} \quad [26]$$

$$\text{and} \quad \zeta = \frac{(1 + \beta) \left(\frac{2 + \beta}{1 + \beta} \right) \ln(1 + \beta) + 1}{\beta \ln(1 + \beta)} \quad [27]$$

The Zumbrunnen model assumes that convection within a pore is negligible but defines conduction across a pore as a ratio proportional to the effective length for conduction across a pore to the characteristic pore length.

Therefore, in practice the effects of convection may be considered. The model allows for conduction through two continuous phases (parallel conduction).

2.3.4.1 Model comparison

In order to visualize the range of predictions for the effective thermal conductivity that these ten models generate, a hypothetical two phase system was analyzed. The system was assumed to be equivalent in structure to the aluminum foam and water systems considered in this study. Both phases were continuous through the bulk with the low conductivity phase having the greater volume fraction. The phases were assigned conductivities of 0.6 and 135 Watt/mK, respectively. Structure parameters required for some models are given with the tabulated data in Tables 3 and 11 .

For the range of volume fractions from 0 to 10% for the high conductivity material the effective thermal conductivities predicted by the ten models are presented in tabular and graphical form in Table 3 and Figure 13, respectively.

Table 3 . Effective Thermal Conductivities Predicted By Ten Models

% Volume Fraction, Foam	3.7%	3.8%	5.2%	5.6%	6.0%	6.1%	7.7%
Arithmetic	5.62	5.79	7.67	8.23	8.85	8.92	11.04
Harmonic	0.63	0.63	0.64	0.65	0.65	0.65	0.66
Upper Maxwell	4.00	4.12	5.41	5.80	6.23	6.27	7.76
Lower Maxwell	0.68	0.68	0.71	0.72	0.73	0.73	0.76
Geometric	0.74	0.75	0.81	0.82	0.85	0.85	0.92
de Vries	1.13	1.15	1.35	1.41	1.48	1.49	1.73
Asaad	1.32	1.33	1.42	1.45	1.48	1.48	1.60
Dulnev	1.24	3.29	2.09	1.69	2.13	1.42	0.78
Zumbrennen, et al	1.47	1.47	1.48	1.49	1.49	1.49	1.50
Hadley	1.16	1.18	1.36	1.42	1.48	1.49	1.69
Correlation coefficients							
Assad		$c = 0.89$					
Hadley		$\alpha = 0.04$					
		$f = 0.082$					
de Vries		$F = 0.100$					
Zumbrennen et al		$\psi = 0$					
		$h(\text{rad}) = 0$					
		$\varphi = 0.63$					

From Table 3 and Figure 13 it can be deduced that each of the models lie between the arithmetic (upper bounding) and harmonic mean (lower bounding) models, as was expected. The models generally predict a conservative effective thermal conductivity.

Figure 13. Plot Of Model Predictions Of Effective Thermal Conductivity As A Function Of Interlaced Volume Fraction

2.4 Experimental method development

Due to the wide disparity in effective conductivities that the empirical models predicted, the next step was to experimentally determine the effective thermal conductivity. A literature review was conducted to obtain an overview of the methods used to measure thermal conductivity. Additionally, a number of independent testing labs, testing equipment distributors and equipment producers were interviewed to determine whether any testing method had a practical advantage over the other methods.

From the literature review, a number of different methods to measure the effective thermal conductivity were found. In general, they can be separated into two classes; steady state and transient. Steady state methods rely on an equilibrium state of heat flow. The thermal conductivity is then solved directly utilizing Fourier's law. Steady state methods are computationally simpler for extraction of the thermal conductivity, but heat leaks out of the system make practical measurements difficult.

Transient methods, in general, relate the temperature change over time of a known sensor/heat source in contact with the test material to the thermal properties of the material itself. Transient methods have the advantage of not having to attain and maintain an equilibrium state of heat flow. The disadvantages of transient systems include the inability to test across a large volume; most transient methods minimize the test surface to a line or point. This deficiency increases possible error where the measured heat conductance over the test distance may not be representative of the thermal conductivity.

Interviews with engineers associated with testing labs, test equipment distributors and producers revealed no consistent method for determining thermal conductivity. Of the nearly one dozen companies that were contacted, only one method, the Guarded Heat Stack (ASTM

E1225) was offered by more than one respondent.

Table 4 is a listing of a selection of methods, taken from the literature and from recommendations by thermal engineers, that can be used to determine thermal conductivity. Some methods are applicable only to solids or to insulative materials, while others are very general in theory. Oftentimes, the experimental apparatus needs to be optimized to the range of conductivities expected from the test material.

2.5 Requirements for the experimental method

The measurement of the effective thermal conductivity of the foam + water system is difficult due to the complex nature of the system. In general, any applicable method must consider the conditions imposed by the material to be tested. The appropriate experimental method must have the ability to measure a liquid and solid composite material. The presence of a liquid phase raises the possibility of convective heat transfer during the testing must be at least that of the pore diameter. More appropriately, the distance should be much greater than the pore diameter because of inhomogeneous pore size distribution throughout the foam.

For steady state methods, where a temperature gradient is used, additional test constraints apply. The absolute temperature range of the

Table 4. Summary Of Methods To Measure The Effective Thermal Conductivity

Method		Description
ASTM 2717-90 ⁽³²⁾	Steady state	The 4-lead platinum heat source in a thin glass tube. Known energy input, temperature gradient is measured. Small conduction distance (gap) required to prevent convection. Non-metallic liquids only. Precision reported to 10%
ASTM E-1225-87 Guarded Stack (Comparator) ^(33, 34, 35)	Steady state	Specimen is sandwiched between two reference materials. ΔT between reference blocks is measured. Heated guards and insulation are utilized to minimize heat losses through sides. Relative method. Solids. Precision reported to 6.8% ($300K < T < 600K$)
Unguarded Stack ⁽³⁶⁾	Steady State	A variation of Guarded Stack Method (ASTM 1225-87) without the use of guards. Larger error possible. Insulative materials only.
Concentric Spheres ⁽³⁷⁾	Steady state	Heater in central sphere, sample in gap between spheres. Realizes theoretical conditions of no heat loss without the use of guard heaters. Liquids, solid-liquid composites (microbeads) and powdered solids.
Transient Hot Strip (THS) ^(38, 39, 40)	Transient	Constant power planar heat source. Change in resistance across heater related to q . Resistance across heater measured. Low temperature increase (1 degree K). Powdered metals, liquids and solids of low thermal conductivity. Precision reported to within 3%.
Transient Hot Wire ^(41, 42, 43)	Transient	Constant power radial heat source. Assumes the relation of temperature vs. $\ln(\text{time})$ is linear and the slope contains the TC of the material the wire is pressed against. Insensitive to plastic flow. Good contact is required to attain true readings. Precision to within 2% of reported standards. Solids and liquids under pressure.
Longitudinal Test with Profiled Guard ^(43, 44)	Steady state	This method uses a close fitting guard around a cylindrical sample plus a second outer guard at the heat sink temperature. Solids and powdered samples
Horizontal Flat plate ⁽⁴⁵⁾	Steady State	Thin fluid film between two metal plates. Heated guard plate required. Convection can be suppressed by decreasing the thickness of the fluid film. Flat plate method is only optimal for fluids at their critical point with gap widths of a few hundredths of a millimeter. Fluids.
Kohlrausch Heat Conductivity ⁽⁴⁶⁾	Steady State	Constant electrical current is passed through material. Joule heating creates thermal gradient along bar. Thermal conductivity is determined from electrical conductivity measured.

system is between 0°C and 100°C. The largest temperature gradient possible across the system will be less than 100° C. It is possible to maintain a larger gradient across the test fixture where reference materials are stacked above and/or below the specimen but the gradient through the specimen must not exceed 100°C.

For transient testing, typically a heat front is applied to one side of the system and allowed to penetrate through a given distance within the material. The penetration of that heat front must at least exceed the maximum pore size of the aluminum foam test specimens. The maximum pore size of the aluminum foam specimens to be tested is approximately 5 mm.

Of the various measurement techniques listed in Table 4, the three most likely candidate methods to determine the effective thermal conductivity of the foam + water system in this study are the Guarded Heat Stack (ASTM E1225), Hot Wire (ASTM C1113) and the Transient Hot Strip techniques.

2.5.0.1 Guarded Heat Stack ASTM E-1225-87

The Guarded Heat Stack method^(33, 34, 35) is the primary ASTM standard method for measuring the thermal conductivity of solid materials. In theory, a one-dimensional heat flow is established in the test material. The thermal

conductivity is then measured directly using Fourier's equation for steady state heat flow

$$k = -q \frac{Z}{\Delta T} \quad [28]$$

where Z is the thickness of the material in the heat flow direction. The heat flow, q, is determined by the use a reference material in-line with the test sample.

In practice, the test specimen is placed in a vertical stack between two reference materials. Heat is supplied from above by a heater; heat flow, q, is determined by temperature readings above and below each reference block. The temperature gradient across the test specimen determines the thermal conductivity. Heated guards and insulation along the sides are utilized to minimize heat losses through the sides of the reference blocks and test specimen.

The difficulty encountered with the Guarded Heat Stack is the establishment of one-dimensional heat flow.⁽³³⁾ Heater guards and insulation are required to maintain this condition. To compensate for heat losses through the sides the thermal conductivity of the insulation between stacks and guards must be known. A heat shunting equation is provided in the ASTM write-up to calculate these heat losses. Reference materials of equal or

slightly greater thermal conductivity as the test specimen aid in establishing linear heat flows. Precision is reported by ASTM to be within 6.8% in the temperature range of 300K to 600K (80°F to 620°F).⁽³³⁾

With respect to the aluminum foam and water system, the Guarded Heat Stack offers an easy test equipment set-up; however, the temperature differential across the test material limits its applicability. The absolute temperatures of the foam + water system are 0° and 100°C (32° and 212°F) due to phase transitions. This allows a maximum differential of 100°C. This range also is just outside the recommended temperature, which may reduce accuracy. Additionally, convection may occur during testing due to the temperature differential which would introduce inaccuracies into the results.

2.5.0.2 Hot Wire, ASTM C 1113-90 (Platinum Resistance Thermometer Technique)

The Hot Wire method^(41, 42, 43) was developed to measure the thermal conductivity of non-carbonaceous, dielectric refractories. In this method a platinum wire is used as both a heat source and a resistance thermometer. A constant current electrical load through the wire causes Joule heating to occur, which in turn heats the surrounding sample material. By measuring the temperature increase and the power input to the wire the thermal conductivity can be calculated by using the time dependent Fourier equation

for heat flow from a line source

$$k = \frac{q \ln(t)}{4\pi dT} \quad [29]$$

The method is specified for dielectric materials with a thermal conductivity under 15 watt/mK and within a temperature range of room temperature to 1500°C (2700°F). Only small changes in the bulk temperature occur during testing, reported by the authors to be approximately 1°C (2°F). The time required for testing is approximately 10 minutes.^(41, 42)

The Hot Wire method is limited to dielectric materials due to the use of a bare metal lead (platinum) as the heat source. However, this constraint may be circumvented by the use of an electrically insulating paint over the wire for use with the electrically conductive foam + water system.⁽⁴²⁾

The Hot Wire method is limited by the use of a line source as the heating element. The effective test volume may be less than the pore size of the foam material. This method is not recommended for measurement of anisotropic materials, especially those containing fibers.⁽⁴³⁾

2.5.0.3 Transient Hot Strip (THS)

The Transient Hot Strip method^(38, 39, 40) is similar in theory to the Hot Wire method but uses an extended metal strip (at least a 20:1 length to width

ratio) as a planar source of heat and temperature sensor. As in the previous method, a constant voltage is passed through the strip but here the change in resistance of the strip over time is recorded. The relationship between the resistance and time is given as Equation [30]

$$R = R_0 + Cf(\tau_c) \quad [30]$$

where time is represented by a dimensionless function, τ , for the material.

The slope, C , of this function is proportional to the thermal conductivity, k , and can be determined from Equation [31]

$$k = \frac{\alpha R_0 I_0^2}{2\sqrt{\pi} h C} \quad [31]$$

This method for measuring thermal conductivity has been cited in numerous recent papers for a wide selection of materials and conductivities. For example, the thermal conductivity of powdered metals,⁽⁴⁷⁾ liquids,⁽³⁹⁾ and solids of low thermal conductivity^(38, 39, 48) have been measured using this method. Reasons cited in these studies for using the Transient Hot Strip method include low increase in the bulk temperature (1° K) and a test time interval of less than one minute. Electrically conductive materials have been tested with this method, using an insulating layer between the strip and material.⁽⁴⁰⁾ Accuracy was reported by the original authors to be within $\pm 3\%$.⁽³⁹⁾

The Transient Hot Strip method has specific advantages, over the other two candidate methods, for determining the effective thermal conductivity of a foam+water system. The low bulk temperature increase, which will minimize convection, is a clear advantage over the Guarded Heat Stack method. In comparison to the Hot Wire method, the volume of analysis using the THS method is greater due to greater power inputs and the increased surface area of the heat source. This may increase the repeatability of the analysis for an anisotropic material. The analysis time period is shorter, on the order of one minute as compared to ten minutes for the Hot Wire method. However, a shorter analysis time may increase error due to the heat capacity of the insulating cover required for electrically conductive samples.

Alternatives to the sublimator system for cooling astronauts during EVA have been presented. The Direct Interface Fusible Heat Sink (DIFHS) is a viable option for use in an ISS EMU provided the regeneration time between sorties can be reduced. The use of an interlaced aluminum foam in the DIFHS water to increase the effective thermal conductivity may provide that reduction in time. A review of the literature revealed that models to predict the effective thermal conductivity may yield results that vary by over an order of magnitude or require experimentally derived coefficients. Experimental methods to determine the effective conductivity were

reviewed. The Transient Hot Strip (THS) method was determined to overcome most of the difficulties in measuring this two-phase system.

In Chapter 3 the research for this investigation is presented. Following the objective statement, the procedures and results of the experiments conducted are reported in Chapters 4 and 5, respectively. In Chapter 6 the results are discussed and in Chapter 7 conclusions based upon those results are presented.

3. Research Objective

The primary objective of this investigation was to measure the effective thermal conductivity of an aluminum foam + water system. It was hypothesized that the effective thermal conductivity is both a function of the volume fractions of the individual components in the system and their structure. The contribution of both the volume fraction and pore size to the effective thermal conductivity was to be determined.

4. Experimental procedure

To measure the effective thermal conductivity of the aluminum foam and water system, the Transient Hot Strip method was used. In this section the unique considerations of the foam+water system are presented and the design of the test system are described. Procedures are outlined for establishing system coefficients, obtaining calibration checks and acquiring and reducing data to determine the effective thermal conductivity of the foam+water samples.

In addition to measuring the effective thermal conductivity, the average pore size and volume fraction of the aluminum foam samples were measured. These procedures are also outlined in this section.

4.0.1 Predicted range of measured thermal conductivity

In these experiments the range of expected thermal conductivities was from 0.6 to 12.0 Watt/mK. This range was based on the conductivity of pure water and the conductivity predicted by the arithmetic, or parallel flow model, for a 92% water, 8% aluminum (Al 6061) system.

4.1 Equipment design

The equipment used to determine the effective thermal conductivity of the aluminum foam and water specimens consisted of four separate

subsystems: the computer / data acquisition system, controlling circuitry, power supply and the test fixture which included the heater strip. A schematic of these sub-systems is shown in Figure 14.

4.1.1 Computer and data acquisition equipment

Data collection was controlled by a Macintosh IIfx personal computer using Strawberry Tree data acquisition boards and Workbench 4.0 data acquisition software. Timers and switch relays were triggered directly by the software. The voltages across the hot strip and the current sensing resistor were logged at a rate of 10 readings per second (10 Hz) at a resolution of 0.000001 V. Temperature readings from the T-type thermocouple were calculated and logged directly by the software.

4.1.2 Control circuit design

The electrical circuit for the experiment consisted of a constant voltage power source which fed two parallel circuits. A schematic of the circuit is shown in Figure 15. The primary circuit was made up of the heater strip in series with a time independent resistor, R_s , from which the current for the circuit could be measured. In the secondary circuit, a parallel resistor, R_p , was placed. The resistor, R_p , was used to offset the internal resistance of the power supply.

Figure 14. Schematic Of Testing Equipment

In order to avoid even a small time dependence in R_s , a balancing circuit with a relay and balance resistor, R_b , was included. The balancing circuit allowed the system to attain steady state before the test was actually run. The balancing circuit was wired parallel to the heater strip. The circuit was controlled by a relay which allowed current to pass through the balancing circuit and resistor R_b prior to the test, and through the balancing circuit and the heater strip during the test. The relay was triggered by a 5 volt signal provided by a port in the data acquisition board. The value of R_b was chosen to be very nearly the same as the resistance of the heater strip so that changes in the voltage across R_s at the start of testing, due to the difference in

Figure 15. Circuit Design For Hot Strip

resistance between R_b and the heater strip, could be minimized. The resistors and relay were soldered directly to a circuit board.

4.1.3 Power source

A Hewlett-Packard E3617A variable power supply was used as a power source for the hot strip circuit. The unit had a maximum power output of 150 Watts, providing 3A of current at 50V.

4.1.4 Test Fixture

A fixture to house the foam and water specimen and heater strip during testing was constructed and schematic is shown in Figure 16. Made of an acrylic block, a test cell was machined into the block to accept the test specimens and heater strip in a horizontal position. The test cell had inner dimensions of approximately 8.5 cm x 4.0 cm x 4.0 cm. The outer dimensions

Figure 16. Test Fixture Schematic

were 15.0 cm x 10.0 cm x 6.0 cm.

4.1.5 Heater strip(s)

Typically, the heater/ temperature sensor in a Transient Hot Strip apparatus is a metal strip which is an electrically-conductive heat source.⁽³⁹⁾ The heat source used in this investigation was a heater element developed by Southwall Technologies. The heater strip was formed from vapor-deposited copper on a Kapton substrate. Electrical leads were soldered to the ends of the formed strip. A second layer of Kapton was then fused to the first to obtain a permanent seal around the metal strip.

The heater leads were silver soldered to longer lead wires which were in turn terminated into a Molex-style plug. This plug allowed heater strips of different widths to be interchanged, depending upon the expected thermal conductivity. A typical heater strip is shown in Figure 17.

The width of the heater strip used for the tests was based on the expected diffusivities that would be encountered and the practical time period required to acquire a sufficient number of measurement points. The practical time span was 20-25 seconds. This time span allowed a sufficient number of measurement points to be collected but was shorter than the minimum time before convection was observed in the water-only sample. The relationship between the length of the test (characteristic time, θ) and the half-width, w , of the strip is given by Equation [32]

$$\theta = \frac{w^2}{k/\rho C_p} = \frac{w^2 \rho C_p}{k} \quad [32]$$

where k is the thermal conductivity, ρ is the density and C_p is the specific heat at constant pressure for the test material.

Given a thermal conductivity, density and specific heat for pure water of 0.6 Watt/mK, 0.998 g/cm³ and 4.18 J/gK, respectively, the strip width for pure water was determined to be 3.0 mm. Similar calculations for a 92%

Figure 17. Heater strip

water, 8% aluminum (Al 6061) system yielded a strip width of 3.5 mm.

The requirement for the length of the heater strip is that the ratio of length to width be approximately 20.⁽⁴⁰⁾ This is to minimize the effects of heat flow along the length (z-direction). Therefore, for these experiments, the strip length was approximately 70 mm.

The actual dimensions of the strip used for these experiments for foam + water combinations was 79 mm x 30 mm x 0.025 mm.

4.1.5.1 Determining the TCR of strip(s)

The temperature coefficient of resistivity (TCR) for each strip was determined by measuring the resistance across the strip at a series of temperatures between 0° and 100°C.

Each strip was tested by connecting the strip to the test apparatus. The strip was then placed in a water bath above a hot plate/magnetic stirrer. The

magnetic stirrer allowed the water in the bath to circulate to attain a uniform temperature. The hot plate was used to maintain the temperature of the bath above room temperatures for additional readings.

A low current, 0.1 to 0.2A, was applied to the strip and its resistance was measured to the nearest 0.01 Ω . The temperature of the water bath was recorded to within 0.1°C, with an NIST-traceable thermometer. The resistance and temperature were recorded a minimum of three times at each temperature.

The strip was placed in a 0°C ice bath and the resistance and temperature recorded. The ice bath was replaced with a water bath at room temperature and additional temperature and resistance measurements were recorded. The strip and thermometer were then immersed in a boiling water bath where resistance and temperature readings were recorded.

The data were plotted graphically as resistance as a function of temperature. A second order equation was fitted to the data. The temperature coefficient of resistivity, α , was then calculated according to Equation [33].

$$\alpha = \frac{1}{R} \frac{\partial R}{\partial T} \quad [33]$$

The temperature coefficient of resistivity for the strip was determined to be $4.14 \times 10^{-3} \text{ }^{\circ}\text{C}^{-1}$ at 22.0°C , which compares favorably with a listed value of $0.0068 \text{ }^{\circ}\text{C}^{-1}$ for pure copper at 20°C .⁽⁴⁹⁾

4.1.6 Calibration of equipment

Preliminary tests were made on a standard material in order to determine the accuracy of the test equipment. The reference material chosen, pure water, was within the range of expected thermal conductivities. Additionally, testing of a liquid (water) allowed the effects of convection on the test results to be determined.

4.2 Procedure for data collection

For this study a matrix of aluminum foam + water specimens were used to determine the effect of pore size on the effective thermal conductivity. The specimens represented three volume fractions crossed with three pore sizes, resulting in an array of nine specimens. The pore size, volume fraction and effective thermal conductivity were determined for each specimen.

Bias was minimized during testing of the foam and water specimens by testing in a round robin sequence. Each specimen was mounted in the fixture, allowed to equilibrate, tested, then removed. Replications for each

specimen were done independently.

4.2.1 Mounting of specimens to reduce contact resistance

Specimens were mounted in the chamber so that the strip would lie in a horizontal position with the separate halves of the foam above and below the heater strip. A wedge was used to reduce contact resistance. The bottom half of the aluminum foam specimen inserted in the test cell with the heater strip in position is shown in Figure 18.

To mount the specimen for testing, the test chamber was filled with de-ionized water. The first, or bottom, half of the specimen was placed in the fixture and the entire fixture was tamped against the table top to dislodge any air bubbles that may have been trapped inside the pores of the specimen. The heater strip was laid on top of the specimen. The second half of the foam specimen was then laid on top of the heater strip. Additional water was added to completely fill the test chamber. Additional tamping removed any trapped air bubbles in the second piece of foam. Contact resistance was minimized by compressing the two halves of the foam and the heater strip against the bottom of the chamber while inserting a wooden wedge between the sides of the foam and the chamber wall. A T-type thermocouple was placed into the foam to determine when thermal equilibrium of the test system was attained and to record the initial temperature.

Figure 18. Aluminum Foam Sample In Test Fixture

4.2.2 Equilibrium temperature

Prior to each test the specimen and test fixture was allowed to equilibrate in temperature. A T-type thermocouple was inserted into the specimen prior to testing. Temperature equilibrium was assumed to be attained when the temperature indicated by the thermocouple fluctuated less than 0.5°C over a 30-second time period.

4.2.3 Recording of data

The collection of data was initiated, controlled and logged by a data acquisition system run through a desktop computer. Data logged consisted of bulk temperature, elapsed time, voltage generated by the power source, voltage across the heater strip and voltage across the current sensing resistor.

Separate data files were created for each test and saved electronically for later data analysis. A sample of the logged data is shown in graphical form in Figure 19.

4.3 Calculations to determine the thermal conductivity

In order to determine the effective thermal conductivity from the data acquired from the Transient Hot Strip method three intermediate steps are required. The first step was to determine the time delay, t_c , caused by the Kapton layer between the hot strip and the sample material. Second, a non-dimensional function was applied to the time delay, and third, a characteristic value for time, θ , was determined. The value for the characteristic time was assumed to be correct when a linear relationship existed between the resistance across the strip and the time function. The thermal conductivity of the sample was determined from the slope of that line.

4.3.1 Determination of the time delay, t_c , of strip(s)

A time delay, t_c is encountered in the heat flow from the hot strip to the test material due to the thermally-insulating layer surrounding the strip. This strip was required due to the conductive nature of the test samples. According to Gustafsson, Karawacki and Chohan⁽⁴⁰⁾ the time delay due to a thin insulating strip can be expressed by the relationship presented as Equation [34]

Figure 19. Typical Data Of Resistance Versus Time.

$$R = R_{\star} + A(t - t_c)^{1/2} + B(t - t_c) \quad [34]$$

where t_c is the time delay, $R_{\star}$ is the apparent resistance, and A and B are constants.

By plotting resistance, R, versus $(t - t_c)^{1/2}$, a t_c was chosen through an iterative procedure such that the deviations of the experimental measurements from the theoretical values were minimized. Additionally, the apparent resistance, $R_{\star}$, was obtained from the y-intercept, by extrapolation.

4.3.2 Determination of $f(\tau)$

By plotting the resistance versus the function $f(\tau)$, a linear regression of the data yielded an equation of the form

$$R = R_0 + Cf(\tau) \quad [35]$$

where the parameters of the line, C , the slope, and R_0 , the y-intercept, are used to determine both the diffusivity and the thermal conductivity.

The function $f(\tau)$ is defined as

$$f(\tau) = -\operatorname{erfc}\left(\frac{1}{\tau}\right) - \frac{\tau^2}{\sqrt{4\pi}} \left[1 - \exp\left(\frac{-1}{\tau^2}\right) \right] + \frac{1}{\sqrt{4\pi}} \left[-Ei\left(\frac{-1}{\tau^2}\right) \right] \quad [36]$$

where

$$\tau = \sqrt{\frac{t - t_c}{\theta}} \quad [37]$$

$$\operatorname{erfc}(u) = 2(\pi)^{-1/2} \int_u^{\infty} \exp(-v^2) dv \quad [38]$$

$$-Ei(-u) = \int_u^{\infty} v^{-1} \exp(-v) dv \quad [39]$$

4.3.2.1 Polynomial approximation of Ei(x)

The mathematical function - Ei(-x), the exponential integral, is approximated in this report by a polynomial function and a set of constants derived by Abramowitz and Stegun⁽⁵⁰⁾ and presented here as Equation [40]

$$-Ei(-x) = E_1(x) \approx \left[\frac{x^4 + a_1x^3 + a_2x^2 + a_3x + a_4}{x^4 + b_1x^3 + b_2x^2 + b_3x + b_4} + \varepsilon(x) \right] (xe^x)^{-1} \quad [40]$$
$$|\varepsilon(x)| < 2 \times 10^{-8}$$

where

$$\begin{aligned} a_1 &= 8.57332 \ 87401 & b_1 &= 9.57332 \ 23454 \\ a_2 &= 18.05901 \ 69730 & b_2 &= 25.63295 \ 61486 \\ a_3 &= 8.63476 \ 08925 & b_3 &= 21.09965 \ 30827 \\ a_4 &= 0.26777 \ 37343 & b_4 &= 3.95849 \ 69228 \end{aligned}$$

4.3.3 Determination of the diffusivity and thermal conductivity

According to prior work by Gustafsson, Karawacki and Khan,⁽³⁹⁾ the deviations of the experimental measurements from a linear equation will be minimized when θ , the characteristic time value, is correct. By using an iteration procedure it is possible to determine a θ such that the correlation coefficient attains a maximum value. The diffusivity, κ , can then be determined directly from the relationship

$$\kappa = \frac{w^2}{\theta} \quad [41]$$

where w is the half-width of the strip. From the slope of the line, C , of the function resistance versus $f(\tau)$, the effective thermal conductivity, k_{eff} , was calculated from the relationship

$$k_{eff} = \frac{\alpha R_0 R_0 I_0^2}{2\sqrt{\pi} h C} \quad [42]$$

4.4 Procedure for measuring pore size

The pore sizes of the aluminum foam specimens were determined by using the Circular Intercept Method of the ASTM standard E112-88 *Standard Test Methods for Determining Average Grain Size*.⁽⁵¹⁾

To analyze the foam specimens a two-dimensional representation of each specimen was required. The two large faces of each specimen were pressed into a thin layer, approximately 0.5 cm thick, of modeling clay. The impression in the clay layer was photocopied on a Xerox photocopy machine to enhance the contrast of the marks left on the clay by the foam tendrils and to provide a permanent record.

Circles of known diameter were drawn directly on the photocopied images such that the circles fell completely within the field of the image. The diameter of the circle was chosen to provide approximately 35 counts or

intersections with the pore boundaries.^(51, 52)

Intersections of the circle and each pore boundary were marked directly on the image. An intersection of the circle with a pore boundary was given a count of one. When the circle coincided with a junction of three pores that intersection was given a count of two. After all intersections were marked along the circumference of the circle a total count was taken.

The average number of counts per circle was determined for each specimen. The average pore diameter, p_{avg} , was then calculated by

$$p_{avg} = \frac{\pi D_{circle} / Mag}{\bar{N}} \quad [43]$$

where $\bar{N}$ is the average number of counts, D is the diameter of the circle and Mag is the magnification of the image used.

4.5 Procedure for measuring volume fraction

The volume fraction of each foam specimen was determined by the ratio of mass to bulk volume. Each specimen was weighed on an analytical balance to a resolution of 0.00001gm. The bulk volume was determined by measuring the volume of the test cell with water. A Plexiglas plate with a single 1/4" hole was screwed to the top of the chamber to provide a constant volume. Water from a graduated burette was used to fill the test cell volume.

The test chamber was considered full when water reached the bottom of the filling hole and no air bubbles remained. The volume ratio of the foam was determined as the ratio of mass volume to bulk volume. The ratio, presented here as Equation [44], is given as

$$\% \text{ vol. fraction} = \frac{vol_{cell}}{Vol_{bulk} \rho_{Al_{6061}}} \quad [44]$$

where $\rho_{Al_{6061}}$ is the density of 6061 aluminum, vol_{cell} is the test cell volume measured and Vol_{bulk} is the bulk volume in milliliters.

5. Results

Using the experimental procedures outlined in the previous section, the effective thermal conductivity for nine aluminum foam+water specimens, plus water alone, were measured using the Transient Hot Strip method (THS) presented in the Chapter 4. In addition to this measurement, the average pore size and volume fractions of the same nine aluminum foam samples were measured. The values for volume fraction and pore size were then used to generate predictions for the effective thermal conductivity from each of the ten models presented in Chapter 2. For models with empirical factors, the experimentally measured thermal conductivities were used to determine these factors.

In this chapter the results are presented. This is followed by Chapter 6, Discussion, where the experimental results are reviewed and placed under statistical scrutiny. The experimentally determined effective thermal conductivities and the predictions based on selected models are then used to predict reduction in the time to freeze a volume of water using an aluminum foam + water system over that of water alone.

The test specimens consisted of three volume fractions of foam, nominally, 4, 6 and 8% of total volume, crossed with three different pore sizes of 10, 20 and 40 pores per inch (2.5, 1.3 and 0.6 mm). This resulted in a total of

nine specimens. In order to differentiate the samples, each specimen was labeled with its nominal volume fraction and pore size. For example, the 4%, 10 pores per inch specimen was labeled as specimen 410.

5.1 Results of effective thermal conductivity measurements

The effective thermal conductivities for the nine aluminum foam and water specimens were measured by using the Transient Hot Strip (THS) Method. Ten replications were acquired for each test specimen and the values averaged to produce the conductivity value presented. As might be expected, the effective thermal conductivity varied, depending upon the volume fraction and pore size. The 4% volume, 10 pores per inch specimen had the lowest thermal conductivity at 1.27 Watt/mK while the 8% volume, 40 pore per inch specimen had the highest value at 2.38 Watt/mK. Additionally, water alone was tested and its thermal conductivity was measured as 0.60 Watt/mK. The results for the nine test specimens plus water, together with their standard deviations, are presented in Table 5.

5.2 Results of volume fraction measurement

The volume fraction of each of the nine aluminum foam specimens were measured using the water displacement method. Three replications were performed on each specimen. The volume fractions of the solid (foam) that were measured ranged from a low of 3.7% to 8.4%. The experimental

Table 5. Effective Thermal Conductivity For The Nine Foam+Water Specimens As Measured Using The Transient Hot Strip (THS) Method

Specimen	Effective Thermal Conductivity (W/mK)	Std. dev.
410	1.27	0.05
420	1.52	0.07
440	1.42	0.15
610	1.70	0.06
620	1.44	0.09
640	1.80	0.07
810	1.41	0.13
820	1.85	0.32
840	2.38	0.09
Water	0.60	0.05

values for the volume fraction are presented in Table 6.

5.3 Results of pore size measurement

The average pore size for the nine specimens were measured using ASTM method E112-88. Ten replications were performed in order to determine an average pore size for each of the nine test specimens. The pore sizes ranged from 2.08 mm to 3.94 mm. The average pore size measured for each foam specimen is presented in Table 7.

Table 6 . Results Of Volume Fraction Testing Of Nine Al Foam Specimens By Water Displacement Method

Specimen	Nominal Volume Fraction of Al (%)	Measured Volume Fraction of Al (%)
410	4%	3.7%
420	4%	5.2%
440	4%	3.8%
610	6%	6.1%
620	6%	5.6%
640	6%	6.0%
810	8%	7.7%
820	8%	7.8%
840	8%	8.4%

5.4 Predictions of the effective thermal conductivity based on theoretical models

Using the experimental results presented in the previous section, theoretical predictions for the effective thermal conductivity of an aluminum foam + water system were calculated, using the ten models described earlier in Chapter 2. Five models used the experimentally measured volume

Table 7. Results of Average Pore Size Measurement Using ASTM
Method E112-88

Specimen	Nominal Pore Size (mm)	Measured Average Pore Size (mm)
410	2.50	3.94
420	1.30	2.51
440	0.60	2.08
610	2.50	2.91
620	1.30	2.93
640	0.60	2.15
810	2.50	3.89
820	1.30	2.73
840	0.60	2.06

fractions to predict the effective thermal conductivity. The models include the four bounding models, namely, the Arithmetic Mean, Harmonic Mean, and Maxwell's Upper and Lower Bounding models, plus the Geometric Mean model.

Four of the remaining five models: Hadley, Asaad, de Vries, and Zumbrennen, Viskanta and Incropera, used the volume fractions and the thermal conductivities of the foam + water specimens to generate effective thermal conductivity equations. The experimentally determined thermal

conductivities were used to back-calculate empirical coefficients for an Al + water system. These results in turn could lead to predictions of the effective thermal conductivity of similar systems; i.e. each model could then be used to predict the effective thermal conductivity for any ratio of aluminum to water.

The remaining model, namely the unit cell model by Dulnev, used the results of the average pore size plus an additional measurement of the average tendril diameters of the aluminum foam specimens in order to predict an effective thermal conductivity for each specimen.

5.4.1 Predicted effective thermal conductivity by five models using the measured volume fraction

The volume fraction of the interlacing aluminum foam material was used to predict an effective thermal conductivity based on five models. The predicted thermal conductivities ranged from 0.63 to 0.66 Watt/mK for the Harmonic Mean, 0.68 to 0.77 Watt/mK for Maxwell's lower limit equation, 0.74 to 0.96 Watt/mK for the Geometric Mean, 4.00 to 8.43 Watt/mK for Maxwell's upper boundary equation, and 5.62 to 12.00 Watt/mK for the Arithmetic Mean model. The effective thermal conductivities predicted by the Arithmetic, Harmonic, Upper and Lower Maxwell's equations, and the Geometric Mean model are presented in Table 8. The same data for the five models are presented in a graphical form in Figure 20.

Table 8. Measured Effective Thermal Conductivities Of Nine Al+Water Specimens Compared Against the Predictions of Five Models

Exp'l Results			-----Theoretical Models-----				
Specimen ID	Volume Fraction Foam (%)	Effective Thermal Conductivity (W/mK)	Arithmetic	Harmonic	Upper Maxwell	Lower Maxwell	Geometric
410	3.7%	1.27	5.62	0.63	4.00	0.68	0.74
420	5.2%	1.52	7.67	0.64	5.41	0.71	0.81
440	3.8%	1.42	5.79	0.63	4.12	0.68	0.75
610	6.1%	1.70	8.92	0.65	6.27	0.73	0.85
620	5.6%	1.49	8.23	0.65	5.80	0.72	0.82
640	6.0%	1.80	8.85	0.65	6.23	0.73	0.85
810	7.7%	1.41	11.0	0.66	7.76	0.76	0.92
820	7.8%	1.85	11.2	0.66	7.89	0.76	0.93
840	8.4%	2.38	12.0	0.66	8.43	0.77	0.96

5.4.2 Predicted effective thermal conductivity by five additional models

Four of the five remaining models, namely those by Hadley, Asaad, de Vries, and Zumbrennen, Viskanta and Incropera, used both the measured volume fractions and the experimentally determined thermal conductivities to first determine empirical coefficients in order to predict the

Figure 20. Predicted Effective Thermal Conductivity Using Four Different Boundary Models

effective thermal conductivity of an aluminum + water system. The correlation factors in each of these models, namely, c , for Asaad, α and f for Hadley, F for de Vries, and ϕ , $h(\text{rad})$ and ψ , for the model by Zumbrennen, Viskanta and Incropera, respectively, were optimized by using the experimentally measured thermal conductivities to more closely align the predictions with the experimentally measured results. From these optimized correlation coefficients, values for an idealized system could then be

predicted.

The tenth model, namely the unit cell model by Dulnev, used the results of the average pore size plus an additional measurement of the average tendril diameter of the aluminum foam specimens in order to predict an effective thermal conductivity for each specimen.

The effective thermal conductivities predicted by each of these five models ranged in value from a low set of 1.13 to 1.84 Watt/mK for the model by de Vries to 1.16 to 1.78 Watt/mK for the model by Hadley. The models by Asaad, Dulnev, and Zumbrennen, Viskanta and Incropera predicted effective thermal conductivities that ranged from 1.32 to 1.65 Watt/mK, 0.78 to 3.29 Watt/mK, and 1.47 to 1.50 Watt/mK, respectively. The predicted values for the effective thermal conductivity are presented in Table 9. The empirical coefficients used for these models are listed at the end of the table. The same data are presented in a graphical form in Figure 21.

Table 9. Measured Effective Thermal Conductivities of Nine Al+Water Test Specimens Compared to The Predictions Of Five Models

Specimen ID	Exp'l Results			-----Theoretical Models-----				
	Effective Thermal Conductivity (W/mK)	Volume Fraction Foam (%)	Pore Size (mm)	De Vries	Asaad	Dulnev	Zumbrennen et al	Hadley
410	1.27	3.7%	3.94	1.13	1.32	1.24	1.47	1.16
420	1.52	5.2%	2.51	1.35	1.42	2.09	1.48	1.36
440	1.42	3.8%	2.08	1.15	1.33	3.29	1.47	1.18
610	1.70	6.1%	2.91	1.49	1.48	1.42	1.49	1.49
620	1.49	5.6%	2.93	1.41	1.45	1.69	1.49	1.42
640	1.80	6.0%	2.15	1.48	1.48	2.13	1.49	1.48
810	1.41	7.7%	3.89	1.73	1.60	0.78	1.50	1.69
820	1.85	7.8%	2.73	1.75	1.61	1.22	1.50	1.71
840	2.38	8.4%	2.06	1.84	1.65	1.85	1.50	1.78

Correlation coefficients derived from the measured effective thermal conductivity

Asaad $c = 0.89$

Hadley $\alpha = 0.04$

$f = 0.082$

De Vries $F = 0.100$

Zumbrennen et al $\psi = 0$

$h(\text{rad}) = 0$

$\varphi = 0.63$

Figure 21. Predicted Effective Thermal Conductivity Values For Six Models

6. Discussion

The experimental results were analyzed to determine the validity of the data and are reported in this section. First, an overview of the estimated errors in measurement in the experiment are presented. Second, analyses in the form of linear, higher order and multiple variable regression analysis were performed on the data to determine the correlation of the volume fraction and pore size to the effective thermal conductivity. These regression analyses provided the statistical ability to compare the theoretical predictions by the models to the experimental measurements. Third, the predictions of the effective thermal conductivity by the ten models presented in Chapter 5 are compared to the experimental results and three models are shown to provide acceptable results. Fourth, these accepted models were then used to predict the time to freeze a volume of water.

6.1 Estimated errors in measurement

Measurement errors that might be expected to occur in this experiment come from a variety of sources. The major sources of error include contact resistance, penetration depth of the heat flux, and additional modes of heat transfer including convection and radiation.

6.1.1 Standard material testing

In order to determine the accuracy of the test fixture, a standard material was tested and the results compared to its reference thermal conductivity. Water was chosen as the reference material due to its well-measured thermal conductivity, availability, and use as baseline measurement against which the test specimens of foam+water could be compared.

6.1.1.1 Calibration with water as a test material

Water represented the lower thermal conductivity limit required for the test equipment for this experiment. Water at room temperature (25°C) has a thermal conductivity of 0.607 Watt/mK.⁽⁴⁹⁾

The results of measurements on water yielded an average thermal conductivity of 0.597 W att/ mK at an initial temperature of 22°C. This compares with an interpolated value for water at 22°C of 0.602 Watt/mK calculated from values listed in the CRC Handbook of Chemistry and Physics.⁽⁴⁹⁾ The difference between the measured value and the interpolated value is 1% which is within the scope of experimental error for measurements of this nature.

6.1.1.2 Effects of convection in measurements

The testing of water allowed the effects of convection to be studied. It was observed in the results of the tests that the orientation of the strip affected the variance in the data. It was determined that a horizontal orientation for the heater strip, i.e. with the strip faces parallel to the floor, resulted in the most consistent results. Therefore all testing was done with the heater strips in the horizontal configuration.

Analysis of the data indicated that near the end of the test period the relationship between the measured resistance and the time function $f(\tau)$ ceased to be linear. It was determined that this was due to the onset of convection. This effect was not observed when the material being tested was a solid or solid/liquid system (foam+water). Water tests were run with an abbreviated test period of approximately 18 seconds compared to the standard test period of 25 seconds. The results of the abbreviated tests indicated that prior to the onset of convection the method is accurate and valid for the testing of fluids at this thermal conductivity range. A data set for water alone, where the effect of convection on the results is apparent, is shown in Figure 22.

Figure 22. Data Showing Onset Of Convection In Water-Only Sample

6.1.2 Error due to specimen thickness

Implicit in the Transient Hot Strip method is the assumption of infinite specimen thickness in the direction of the heat flux. This is a direct requirement of one dimensional heat flow in the specimen. It has been shown in previous work by Gustafsson, Karwacki and Khan⁽³⁹⁾ that the probe depth, ξ , can be defined as

$$\xi = 1.42\sqrt{\kappa t} \quad [30]$$

where κ is the diffusivity and t is the measurement time.

Additionally, according to Log and Metallinou,⁽⁵³⁾ in actual experiments the probe depth was determined to be approximately equal to the width of the heater strip, $2w$. Therefore, the thickness of the specimens in this study needed to be at least 12 mm since the heat flux flowed from the center of the specimens equally in the positive and negative z -directions. The thickness of the specimens in the direction of heat flow (z -direction) was 18 mm.

6.1.3 Error due to contact resistance

Contact resistance was minimized by using a wooden (non-conductive) wedge to firmly press the two specimen slabs against the heater strip. The error due to non-uniform contact resistance can be determined by comparing the variance in results from tests using the same experimental set-up (i.e. repeated testing without re-installing the specimen in the test fixture) to results of independent replications (complete removal and installation for each test). The variance between repeated tests was 0.0110. This compares to a variance of 0.0197 for replicated tests on the same specimens. The error due to changes in contact resistance was assumed to be the difference between the replication and repeated test variances. The difference in the variances was 0.0087, which represents the error due to changes in the contact resistance.

Compared to a variance for all samples of 0.322, the variance due to contact resistance represents only 2.7% of this total. Therefore the variance due to contact resistance was determined to be minimal. All test data reported were from replicated testing.

6.1.4 Error due to radiative heat transfer

The error due to neglecting heat flow due to radiation can be calculated by determining the ratio of radiative versus conductive heat flow.⁽⁵⁴⁾ Using the general equations for radiative and conductive heat transfer yields the ratio

$$\frac{\text{radiation}}{\text{conduction}} = \frac{4\sigma\epsilon T_{avg}^3 P}{k_{eff}} \quad [36]$$

where ϵ represents the emissivity, σ is the Stephan-Boltzmann constant, T is the test temperature in Kelvin, P is the average pore diameter, and k_{eff} is the effective thermal conductivity. The emissivity for the system was on the order of 1.0, the temperature was 325 K, the largest average pore size was 3.9 mm, and the effective thermal conductivity was estimated as 2.0 Watt/mK. The ratio of radiative to conductive transfer was calculated as

$$\frac{\text{radiation}}{\text{conduction}} = 0.4\% \quad [37]$$

Therefore the effect of radiation can be neglected without a significant effect

on the accuracy of the measurements.

6.2 Correlating experimental results

To determine whether a correlation existed between the effective thermal conductivity and the volume fraction or average pore size the following steps were taken:

- Plot graphs of effective conductivity versus volume fraction and effective conductivity versus pore size.
- Observe general relationships.
- Determine independent linear relationships.
- Perform tests for linear correlation.
- Perform tests for higher order terms.
- Determine confidence intervals.

Additionally, a multiple variable regression analysis was performed to determine whether interaction between the volume fraction and the pore size had an effect on the effective thermal conductivity. This analysis showed that statistically there was no direct relationship between the pore size and the effective thermal conductivity. However, there was a statistically significant interaction effect of pore size and volume fraction on the effective thermal conductivity.

6.3 Effect of volume fraction on the effective thermal conductivity

The relationship between the volume fraction and the effective thermal conductivity is plotted in Figure 23. As expected, increases in the volume fraction of the aluminum foam resulted in increases in the thermal conductivity of the system.

A linear relationship between the volume fraction and conductivity was first assumed and a single factor regression analysis was performed. The straight line equation of conductivity as a function of volume fraction is presented here as Equation [38]

$$Th\ Conductivity = 16.255(vol.\ frac) + 0.66309 \quad [38]$$

A test for the utility of the regression equation was performed on the slope. A 95% confidence interval test resulted in a value of 30.51. This result, compared to a rejection value of 2.00 for the slope, indicated that it was statistically relevant that the effective thermal conductivity increased with increasing volume fraction.

The r^2 coefficient of determination for the conductivity to volume fraction data was determined to be 0.77. Hence, 77% of the variation in the effective thermal conductivity measured can be attributed to changes in the volume fraction.⁽⁵⁵⁾

Figure 23. Effective Thermal Conductivity As A Function Of
Volume Fraction

6.4.1 Second order regression

A second order regression of the independent variable, volume fraction, was performed on the data to determine the effects of higher order terms. The second order regression equation, presented here as Equation [39], for the volume fraction was determined to be

$$Th\ Cond. = -28.476(vol.\ frac)^2 + 18.731(vol.\ frac) + 0.6282 \quad [39]$$

Student's t-test was used to determine whether the higher order term was statistically beneficial in predicting the effective thermal conductivity.

The null hypothesis for the test was

$$H_o : \text{2nd Order Coefficient} = 0$$

$$H_a : \text{2nd Order Coefficient} \neq 0$$

$$\text{test statistic } t = \frac{\text{2nd OC}}{s_{\text{2nd OC}}} = \frac{\text{2nd OC}}{\frac{s}{\sqrt{SS_{xx}}}} \quad [40]$$

$$t = 0.243$$

$$\text{rejection region : } |t| > t_{crit} = 2.365$$

The calculated t-value of 0.243 determined from Equation [40] was less than the critical value of 2.365. Therefore the test was not able to reject the hypothesis. This indicates that the addition of higher order terms do not provide a better estimate of the correlation of volume fraction and effective conductivity.

6.4.2 Prediction interval for the effective thermal conductivity

The 95% prediction interval was determined for the volume fraction of foam as a predictor of the effective thermal conductivity. The upper and lower boundaries of this interval are shown in Figure 24. The symbols in the chart represent the measured thermal conductivities of this study, the solid line being the best-fit linear equation, and the dashed lines the high and low (95%) prediction interval boundaries.

Figure 24. Effective Thermal Conductivity Results With 95% Confidence Intervals Added

The prediction interval maps the range of values within which an unknown sample's effective thermal conductivity would lie with a confidence interval of 95%. The prediction interval is quite large: ± 0.843 Watt/mK at 5% volume fraction. This is probably due to the small size of the sample studied. Larger samplings would be expected to decrease the width of the prediction interval.⁽⁵⁵⁾

The volume fraction was determined to have a strong linear correlation to the effective thermal conductivity. This agrees with the results predicted by the theoretical models in the earlier section. However, the r^2 correlation is not strong enough to draw the conclusion that volume fraction

alone is sufficient to predict the effective thermal conductivity. The effect of pore size may also influence the effective thermal conductivity. In the following sections the significance of this parameter is examined.

6.5 Effect of pore size on the effective thermal conductivity

The relationship between the average pore size and the effective thermal conductivity was determined in the same fashion as that for the volume fraction was in the previous section. The graph of the effective conductivity as a function of the average pore size is presented here as Figure 25.

A linear relationship was first assumed between the thermal conductivity and the pore size. A linear regression analysis was performed which yielded the following straight line equation:

$$Th\ Conductivity = -0.2825(pore\ size) + 2.440 \quad [41]$$

The slope of -0.2825 represents the trend of decreasing effective thermal conductivity with increasing pore size.

The r^2 coefficient of determination for this model was determined to be 36.4%. This is a low measure of correlation between the effect of increasing pore size and decreasing thermal conductivity.

Figure 25. Effective thermal conductivity as a function of average pore size

A test of the utility of the straight line equation was performed. The null hypothesis was that the slope = 0, *i.e.*, that the pore size had no linear effect on the effective thermal conductivity.

$$\begin{aligned}
 H_o: & \text{ slope} = 0 \\
 H_a: & \text{ slope} \neq 0 \\
 \text{test statistic } t = & \frac{\text{slope}}{s_{\text{slope}}} = \frac{\text{slope}}{s / \sqrt{SS_{xx}}} \quad [42] \\
 t = & -1.247 \\
 \text{rejection region: } & |t| > t_{\text{crit}} = 2.365
 \end{aligned}$$

The value of the 95% confidence t-statistic was determined to be $t = -1.247$. Compared to the critical t-value of $t_{\text{crit}} = 2.365$, the absolute value of

the t-statistic was not greater and the null hypothesis failed to be rejected. Therefore the data was not able to substantiate the hypothesis that changes in the pore size correlated to changes in the effective thermal conductivity.

Failure to reject the null hypothesis can be attributed to three possible reasons: effects due to the volume fraction or interaction between the pore size and volume fraction, insufficient sampling size in the study, or no actual correlation of the conductivity with respect to the pore size.

The regression analysis performed above was for the independent relationship between the pore size and the effective thermal conductivity. It has already been shown that changes in the volume fraction have a strong relationship to changes in the effective thermal conductivity. Interaction, or secondary, effects between the volume fraction and pore size may have contributed to the failure to show an independent correlation. A multiple variable regression analysis including both volume fraction and pore size would allow these effects to be determined. Multiple variable regression is discussed in Section 6.6.

The number of measurements in this experiment was nine. Small sampling sets, such as with this experiment, decrease the magnitude of the t-statistic. Larger sample sizes provide a more accurate t-statistic. Given resources to include a larger number of measurements, a comprehensive

study may indicate that pore size significantly affects the effective thermal conductivity.

No correlation between pore size and effective thermal conductivity may exist. In a strict one-dimensional heat flow model only the tendrils oriented in the direction of the heat flow would transport heat via axial conduction. All the other tendrils would participate in serial heat conduction in-line with the fluid. In areas much larger than the average pore size, the total cross-sectional area for parallel conduction, either through the tendrils or through the fluid, is constant with respect to pore size, assuming constant volume fraction. Likewise, for serial conduction, the cross-sectional area available remains constant with changing pore size. Assuming first, strictly one-dimensional heat flow through the system, and second, that the contributions of each heat pathway are additive, then there would be no change in the heat conduction with changes in pore size. Therefore, the result of no significance of conductivity to pore size changes in these experiments would agree with theoretical expectations.

Because of the strong correlation between the volume fraction and effective thermal conductivity the influence of pore size alone is not significant. Of course, this may not entirely discount any effect due to pore size. The next step to developing relationships between the variables and the conductivity was to analyze the two variable system in a multiple variable

regression analysis.

6.6 Multiple variable regression analysis

It must be considered that the unrejected test statistic for the pore size was for the *independent* correlation of the pore size to the effective thermal conductivity. It was shown previously that the volume fraction does have a linear correlation to the effective conductivity. The effect of the volume fraction and of possible interaction between the volume fraction and pore size may influence the effect of pore size on the effective thermal conductivity.

A multiple variable regression analysis was performed using a complete second order equation to determine the main and secondary effects of volume and pore size on the effective thermal conductivity. The form of the equation is

$$\begin{aligned} Th \text{ Cond.} = & \beta_0 + \beta_1(vol. \text{ frac}) + \beta_4(pore) + \\ & \beta_2(vol. \text{ frac})^2 + \beta_3(pore)^2 + \beta_5(pore)(vol. \text{ frac}) \end{aligned} \quad [43]$$

Tests for confidence in the β coefficients led to the removal of some of the terms. It was found that the volume fraction and the interaction term between the volume fraction and the average pore size were the only significant terms in the correlation to the effective thermal conductivity. The final regression equation for volume fraction and pore size is presented here

as Equation [44]

$$Th\ Cond. = \beta_0 + \beta_1(vol.\frac{frac}{}) + \beta_2(pore)(vol.\frac{frac}{}) \quad [44]$$

where the remaining β coefficients, determined through an iteration process of curve fitting the data, were found to be

$$\beta_0 = 0.6833$$

$$\beta_1 = 29.33$$

$$\beta_2 = -4.836$$

The average pore size was not a significant term and was removed to simplify the equation. A comparison of the predicted thermal conductivity, using this regression model, versus the measured thermal conductivity is shown graphically in Figure 26.

The r^2 coefficient of determination for this regression model was determined to be 0.96, or 96%. The r^2 values for correlating thermal conductivity to volume fraction and pore size separately were 76% and 36%, respectively. Compared to the r^2 for volume fraction alone, it was found that by using the additional information provided by the pore size the predicted effective thermal conductivity can be substantially improved.

Figure 26. Predicted Thermal Conductivity Versus Experimental Results

The multiple variable regression equation suggests that volume fraction is the primary parameter in determining the effective thermal conductivity. However, this relationship is not necessarily linear. The interaction term of the volume fraction and pore size modulates this effect which may suggest that conductivity is not a linear function of volume fraction but possibly logarithmic or some other function. It may also suggest that structure may be weakly influential, or that another, unspecified

parameter may be influencing the results.

6.7 Theoretical models

The experimental results can now be compared against the models presented in Chapter 2. The experimental values of the volume fraction and effective thermal conductivity tabulated against the predicted values of each of the models are listed in Table 10. The sum of the squares of the error, SSE, for each model is included.

The range of predictions for the effective thermal conductivity was from a low of 0.63 to a maximum of 11.98 Watt/mK. The majority of the models predicted conductivity values in the range of 1.0-2.0 Watt/mK. The sums of the squares for error, or SSE's, ranged from less than one to almost 500. An SSE value of zero signifies perfect correlation.

Table 10. Experimental Effective Thermal Conductivity Results And Theoretical Predictions Based On The Volume Fraction Measurements

Exp'l Results		Theoretical Models									
% Volume Fraction Foam	Eff. Thermal Conductivity	Arithmetic	Harmonic	Upper Maxwell	Lower Maxwell	Geometric	de Vries	Asaad	Dulnev	Zumbrennen	Haley
3.7%	1.27	5.62	0.63	4.00	0.68	0.74	1.13	1.32	1.24	1.47	1.16
5.2%	1.52	7.67	0.64	5.41	0.71	0.81	1.35	1.42	2.09	1.48	1.36
3.8%	1.42	5.79	0.63	4.12	0.68	0.75	1.15	1.33	3.29	1.47	1.18
6.1%	1.70	8.92	0.65	6.27	0.73	0.85	1.49	1.48	1.42	1.49	1.49
5.6%	1.49	8.23	0.65	5.80	0.72	0.82	1.41	1.45	1.69	1.49	1.42
6.0%	1.80	8.85	0.65	6.23	0.73	0.85	1.48	1.48	2.13	1.49	1.48
7.7%	1.41	11.04	0.66	7.76	0.76	0.92	1.73	1.60	0.78	1.50	1.69
7.8%	1.85	11.22	0.66	7.89	0.76	0.93	1.75	1.61	1.22	1.50	1.71
8.4%	2.38	11.98	0.66	8.43	0.77	0.96	1.84	1.65	1.85	1.50	1.78
SSE		496	9.87	202	8.46	6.44	0.68	0.8	5.11	1.09	0.71

6.7.1 Bounding equations

The experimental results for the effective thermal conductivity and the four bounding equations are plotted in Figure 27. As expected, plotting the experimental results and the bounding equations show that the experimental values lie within the limits of both the Arithmetic and Harmonic Mean models as well as the more stringent Upper and Lower Maxwell's bounding

Figure 27. Experimental Results Plotted With Bounding Models

equations.

6.7.2 Comparison of theoretical predictions to experimental results

To compare the predictions to the experimental data a 95% confidence interval was used. The confidence interval defines a range of effective conductivities that lie within a 95% confidence interval for the true mean value. The confidence interval for the mean defines a more narrow range than the prediction interval presented earlier.⁽⁵⁵⁾ The 95% confidence interval for the experimental data is presented in Figure 28. The confidence interval can be seen to widen at either end of the range of conductivities tested. This

Figure 28. Experimental Results Versus Volume Fraction With Confidence Intervals

is due to the increasing distance from the average value of volume fractions tested.

The predictions generated by the models were compared against the experimental data and confidence intervals in order to determine the utility of the models. The comparison consisted of two steps. First, if a model's prediction fell outside the confidence limits of the experimental data the model was judged to be not acceptable. Second, models whose predictions fell within the confidence limits were further tested to determine the statistical relevance of their slopes. A comparison of the slope of the regression line for

the model was compared to the 95% confidence interval for the slope for the experimental values. Slopes that fell outside the confidence range were judged to be not acceptable models.

Models judged unacceptable because their predictions of the effective thermal conductivity fell outside of the 95% confidence limits were the Arithmetic, Harmonic, and Geometric Mean Models and the Upper and Lower Maxwell Equations. The Arithmetic Mean and the Upper Maxwell Equation fell above the upper limit while the other models had predictions that were below the lower limit. The positions of these models in relation to the confidence interval are illustrated in Figures 29 and 30.

6.7.1.1 Dulnev Unit Cell Model

The Unit Cell model by Dulnev presented an atypical result. From the graph of Dulnev's predictions, in Figure 31, the predicted effective thermal conductivity strongly relates increases in volume fraction of the aluminum foam to decreases in the effective conductivity. This inference to the relationship between volume fraction and conductivity is related to the use of the average pore size in determining the predicted values. As was demonstrated previously, pore size was mildly inversely proportional to the conductivity. Dulnev's predictions mirror these results.

Figure 29. Predicted Effective Thermal Conductivity Of Three Models

The assumption made earlier in developing the Dulnev predictions was that the average pore size was equal in length to the body diagonal of the Dulnev cubic system. Tortuosity within the true foam cell, or the amount of deviation in the actual cell from the simplified model, is probably high.

If the assumption between measured pore size and the characteristic unit cell is relaxed the unit cell length, L , can be determined empirically from the data. Plotting the sum of the squares for error as a function of unit cell length resulted in the plot shown in Figure 32.

From the plot in Figure 32 the optimum value of the length of the characteristic unit cell was determined to be $2.20 \times \text{Pore Size}$. This optimum

Figure 30 . Predicted Effective Thermal Conductivity Of Two Models

value suggests the tendrils of aluminum foam are less conductive than the parent material of which they are made.

Plotting the predictions of this optimized length value, which is presented as Figure 33, still resulted in an inversely proportional relationship between the volume fraction and the effective thermal conductivity. Since the slope of this optimized line was still outside the 95% confidence interval for the experimental data the Unit Cell Model by Dulnev was deemed not acceptable.

Figure 31. Dulnev Predictions Versus Volume Fraction Compared To Experimental Results

6.7.1.2 Determination of slope

The remaining four models proposed by Asaad, Hadley, de Vries, and Zumbrennen, Viskanta, and Incropera each predicted conductivity values that were within the 95% confidence interval. Each of these models were then tested for the utility of their slope value. The equations of the line for the predictions by each of the four models were determined to be:

Figure 32. Sum Of Squares For Error Versus Average Pore Size For Dulnev Model

<i>Asaad</i>	$7.0701x + 1.0544$
<i>Hadley</i>	$13.131x + 0.6815$
<i>de Vries</i>	$15.124x + 0.5678$
<i>Zumbrennen, Viskanta, and Incropera</i>	$0.6673x + 1.4479$

In Chapter 5, the 95% confidence interval for the slope of the experimental data was found to range from 2.00 to 30.51. The graph in Figure 34 shows the agreement of the slopes of the models to this confidence interval. The models of Asaad, Hadley, and de Vries each fall within the

Figure 33. Predictions By Dulnev Of Effective Thermal Conductivity
Compared To Experimental Results

range of the slope interval. The slope of Zumbrennen, Viskanta, and Incropera lies outside the interval with a slope of 0.6673 compared to the minimum interval value of 2.00.

The models by Asaad, Hadley, and de Vries, were determined to satisfy the conditions of acceptance. Each model predicted values for the effective thermal conductivity within the 95% confidence interval and possessed values for their slopes that were within the limits of confidence of the slope for the experimental data. The graphs of Asaad, Hadley, and de Vries are

Figure 34. Slope Values For Four Models Compared To Experimental Confidence Interval

presented in Figures 35, 36, and 37, respectively.

If the experimental data are accepted to be an accurate measurement of the effective thermal conductivity of aluminum foam and water system and that the Asaad, Hadley, and de Vries models can predict values for the effective thermal conductivity that lie within a specified confidence interval about these experimental data, then any of these models can be used to predict the effective thermal conductivity for a given foam volume fraction and pore size. Following these predictions, the time to freeze, or regenerate, a system

Figure 35. Asaad Model Predictions Compared To Experimental Results

of foam + water can be estimated from the equations presented in Chapter 2.

Using the models of Asaad, Hadley, and de Vries, the estimated effective thermal conductivity can be found for a set of foam+water systems. The predicted values are presented in Table 11. The foam parameters of volume fraction and pore size were arbitrarily selected but represent the range of conditions that may be possible using this particular foam product.

Figure 36. Hadley Model Predictions Compared To Experimental Results

6.7.1.3 Time to freeze

The effective thermal conductivities predicted by models which yielded results within the confidence limits of the experimental data can now be used to predict the time to freeze for an interlaced foam + water system.

This study focused on the effective thermal conductivity of a two phase system; the effective thermal conductivity of the solid-solid (foam+ice - single phase) system was not determined. However, if the assumption of heat transfer via conduction is valid, then the effective conductivity of any two-component system, whether two phase or single phase, can be predicted

Figure 37. De Vries Model Predictions Compared To Experimental Results

by the above empirical models given that the structure of the system is equivalent and the relative ratio of thermal conductivities are comparable. The predicted effective thermal conductivities of the water (liquid-solid) and ice (solid-solid) conditions for an arbitrary set of volume fractions are listed in Table 12. Since each of the three models Asaad, Hadley, and de Vries, relies only on the volume fractions of the phases, the number of conditions were reduced to volume fractions of 4, 6, and 8% to reflect this.

Using Equation [4], which was developed in Chapter 2, the time to freeze can be predicted for an interlaced system by substituting the conductivities of the fluid and solid phases of the material with the effective

Table 11. Predicted Foam+Water Effective Thermal Conductivities Based On Four Equations

Pore size (mm)	Volume Fraction Solid %	Asaad (W/mK)	Hadley (W/mK)	De Vries (W/mK)	Experimental Regression (W/mK)	Average (W/mK)
2	4	1.41	1.07	1.23	1.47	1.295
4	4	1.41	1.07	1.23	1.28	1.198
2	6	1.55	1.32	1.55	1.86	1.571
4	6	1.55	1.32	1.55	1.28	1.426
2	8	1.71	1.57	1.89	2.26	1.827
4	8	1.71	1.57	1.89	1.48	1.663

thermal conductivities predicted by the above models for a multi-component system. The numerical data is presented in Table 13 and the same data are shown in graphical format in Figure 38. The system parameters of container radius and start and end temperatures were held constant. These values are listed with the tabulated data in Table 13.

From this data, the time to freeze is seen to reduce with increasing volume fraction. At volume fractions of about 7 to 8% solid material, the time to freeze drops below the eight hour mark. This compares to the estimated time to freeze of a container of water-only to be over 17 hours. This

Table 12. Estimated Time To Freeze For Selected Foam+Water Systems Based On Predictions Using Models By Asaad, Hadley, And de Vries

Volume Fraction Solid, %	Asaad (hrs.)	Hadley (hrs.)	De Vries (hrs.)	Average (hrs.)
4	8.86	10.33	10.39	9.86
6	8.28	8.74	8.85	8.62
8	7.70	7.60	7.70	7.67

$T_{\text{start}}=4.4^{\circ}\text{C}$

$T_{\text{end}}=-17.8^{\circ}\text{C}$

$T_{\text{freezer}}=-23^{\circ}\text{C}$

Container radius = 16 cm

point is significant because it suggests the use of an aluminum foam in the Direct Interface Fusible Heat Sink may reduce regeneration times significantly— by as much as 50%.

While the reduction in regeneration time estimated here is not proven with direct experiment, the potential for significant time savings is encouraging. Future work in this area should concentrate on quantifying the true time savings when using an interlaced material over water alone.

In the previous two sections the results obtained by experiment have been presented. The effective thermal conductivities of the foam+water specimens using the Transient Hot Strip method and the results of the

Table 13. Predicted Effective Thermal Conductivities Using Three Models For Various Volume Fractions Of Solid Phase With Either Liquid Water Or Ice

Volume Fraction Solid, %	Asaad (W/mK)		Hadley (W/mK)		De Vries (W/mK)	
	Water	Ice	Water	Ice	Water	Ice
4	1.41	4.18	1.07	3.57	1.23	3.47
6	1.55	4.49	1.32	4.29	1.55	4.14
8	1.71	4.83	1.57	5.02	1.89	4.83

volume fraction and pore size measurements were discussed. From these results it was concluded that the effective thermal conductivity increases with increases in the volume fraction of the foam phase. Changes in conductivity with pore size, on the other hand, were not statistically substantiated by the data. This effect can be attributed to the large spread in pore size within the samples and the small size of the test population. A much larger sampling base may substantiate an effect of pore size on the effective conductivity.

The empirical models taken from the literature were compared to the experimental data and three models, authored by Asaad, Hadley, and de Vries, were found to predict the data to within the confidence intervals used. From these models, predictions of the regeneration, or freezing, rate were generated which indicated that the addition of a high conductivity

Figure 38. Estimated Time To Freeze As A Function Of Volume Fraction
Of Added Aluminum Foam

material could decrease the regeneration time by as much as one-half. The actual time to freeze of a full-sized DIFHS would be required to substantiate these findings.

7. Conclusion

The effect of volume fraction and pore size on the effective thermal conductivity of an aluminum foam and water system were examined in this study. Nine specimens of aluminum foam representing a matrix of three volume fractions (4-8% by vol.) and three pore sizes (2-4mm) were tested with water as the matrix fluid, to determine relationships with the effective thermal conductivity. It was determined that increases in volume fraction of the aluminum phase leads to increases in the effective thermal conductivity. It was not statistically possible to prove that changes in pore size of the aluminum foam correlated to changes in the effective thermal conductivity. However, interaction effects between the volume fraction and pore size of the foam were statistically significant. Ten theoretical models were selected from the published literature to compare against the experimental data. Models by Asaad, Hadley [1986] and de Vries provided effective thermal conductivity predictions that were within a 95% confidence interval.

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8. References

1. G. Vogt, Ed., *Suited for Spacewalking*, NASA publication EP-279, NASA-Johnson Space Center, Houston, TX, March 1992
2. G. Vogt, Ed., *Information Summaries: Wardrobe for Space*, NASA publication PMS-033, NASA-Johnson Space Center, Houston, TX, July, 1989
3. <http://www.ksc.nasa.gov/shuttle/technology/sts-newsref/sts-apu.html#sts-eva>
4. M. Diener and M. Balashov, Maintenance Practices on the EVA Suit 2000, *Aerospace Engineering*, SAE Publications, November 1994, pp. 7-10
5. P. Stone, J. Miles, S. Nair, C. Lomax, and B.W. Webbon, *Tubular Evaporator Development for the PLSS*, 1996 ICES Publication 961486, International Conference of Environmental Systems 1996, Monterey, CA 1996
6. J.L. Williams, B.W. Webbon, and R.J. Copeland, *Advanced Extravehicular Protective Systems (AEPS) Study: Detailed Technical Notes*, NASA-Ames Research Center, Moffett Field, CA, March 1972
7. C. Lomax and B.W. Webbon, *A Direct-Interface Fusible Heat Sink For Astronaut Cooling*, NASA Technical Memorandum 102835, NASA-Ames Research Center, Moffett Field, CA, May 1990
8. C. Lomax and B. Kader, *Direct-Interface Fusible Heat Sink Performance Tests*, ICES Publication 941384, International Conference On Environmental Systems 1994, Friedrichshafen, Germany, 1994
9. *Extravehicular Activity Systems Project Plan: Human Support Technology Thrust Exploration Technology Program*, NASA-Ames Research Center, Moffett Field, CA, April 1991
10. J. Moskito, C. Lomax, and G. Selvaduray, *Effective Thermal Conductivity of Interlaced Materials*, ICES Publication # 951636, International Conference On Environmental Systems 1995, San Diego, CA, 1995

11. W. Woodside and J.H. Messmer, Thermal Conductivity Of Porous Media I. Unconsolidated Sands, *Journal Of Applied Physics*, vol. 32, no. 9, September 1961, pp. 1688-1698
12. P.G. Collinshaw and J.R.G. Evans, An Assessment of Expressions for the Apparent Thermal Conductivity of Cellular Materials, *Journal of Materials Science*, vol. 25, no. 8, 1994, pp. 486-497
13. <http://www.ksc.nasa.gov/shuttle/technology/sts-newsref/sts-apu.html#sts-wsb>
14. M. Dubois, B. Mullender, S. Van Oost, J. Druart, W. Supper, and D. Titterton, *Space Qualification of High Capacity Grooved Heat Pipes*, ICES Publication 951551, International Conference On Environmental Systems 1995, San Diego, CA, 1995
15. B. Moschetti and E. Voyer, *Development of a High Performance Heat Pipe (HPHP) for a Satellite Application*, ICES Publication 951552, International Conference On Environmental Systems 1995, San Diego, CA, 1995
16. J. Moskito, *Analysis of the Vapor Flow in a Tubular Boiler Heat Exchanger*, Unpublished, 1995
17. K. Lundsford, *Flow Across and Inside of a Membrane Tube*, MS Thesis, MAE Dept., University of Missouri-Columbia, 1997
18. W. Schneider, Vapor Flow Through a Porous Membrane- A Throttling Process with Condensation and Evaporation, *Acta Mechanica*, vol. 47, 1983, pp. 15-25
19. D. Pitts and L. Sissom, *Theory and Problems of Heat Transfer*, McGraw-Hill Publishers, New York, 1977
20. Duocel Ceramic Foam, Sales Literature, Energy Research and Generation, Inc., Oakland, CA, 1995
21. Reticulated Vitreous Carbon, Sales Literature, Energy Research and Generation, Inc., Oakland, CA, 1995
22. D. A. Zumbrennen, R. Viskanta, and F.P. Incropera, Heat Transfer Through Porous Solids with Complex Internal Geometries, *International Journal of Heat and Mass Transfer*, vol. 29, no. 2, pp. 275-283

23. Duocel Aluminum Foam, Sales Literature, Energy Research and Generation, Inc., Oakland, CA, 1995
24. B. Leyda, ERG Inc., Private Conversation September 19, 1996
25. F.B. Nimick and J.R. Leith, A Model for Thermal Conductivity of Granular Porous Media, *Journal of Heat Transfer*, 114, May 1992, pp. 505-508
26. G.R. Hadley, Thermal Conductivity Of Packed Metal Powders, *International Journal Of Heat and Mass Transfer*, vol. 29, no. 6, 1986, pp. 909-920
27. Y. Asaad, *A Study of the Thermal Conductivity of Fluid-Bearing Porous Rocks*, Ph.D. Thesis, University of California, Berkeley, CA, 1955
28. A.V. Luikov, A.G. Shashkov, L.L. Vasiliev, and Y. E. Fraiman, Thermal Conductivity Of Porous Systems, *International Journal Of Heat And Mass Transfer* , vol.. 11, 1968 pp. 117-140
29. G.N. Dulnev, D.P. Volkov, and V.I. Malarev, Thermal Conductivity of Moist Porous Materials, *Journal of Engineering Physics (English translation of Inzhenerno-Fizicheskii Zhurnal)*, vol. 56, no. 2, February, 1989, pp. 198- 206
30. H. Bjurstrom, E. Karawacki, and B. Carlsson, Thermal Conductivity of a Microporous Particulate Medium: Moist Silica Gel, *International Journal of Heat and Mass Transfer*, vol. 27, no. 11, 1984, pp. 2025-2036
31. R. P. Tye, *Thermal Conductivity*, 2nd Ed., Academic Press, London, 1969, pg. 398
32. *Standard Test Method For Thermal Conductivity Of Liquids*, ASTM D2717-90, American Society for Testing and Materials, Philadelphia, PA, 1990
33. *Standard Test Method For Thermal Conductivity Of Solids By Means Of The Guarded Comparative Longitudinal Heat Flow Technique*, ASTM E1225-87, American Society for Testing and Materials, Philadelphia, PA, 1987

34. ASTM E1530 Guarded Heat Flow Meter Method, Technical Note 50, Anther Corp., Pittsburgh, PA, 1995
35. G. Sheffield and J.R. Schorr, Comparison of Thermal Diffusivity and Thermal Conductivity Methods, *Ceramic Bulletin*, vol. 70, no. 1, 1991, pp. 102-106
36. T. Ashworth, E. Ashworth, and S.F. Ashworth, *A New Apparatus for Use with Materials of Intermediate Conductivity*, International Thermal Conductivity Conference (21st), Lexington, KY, 15-18 Oct 1989 pp. 229-241
37. N. Moore, H. Wolf, and F.K. Deaver, *Simultaneous Heat and Mass Transfer in Moist Porous Media*, University of Arkansas, Fayetteville, AR, 1972
38. R. Singh, N.S. Saxena, and D.R. Chaudhary, Simultaneous Measurement Of Thermal Conductivity And Thermal Diffusivity Of Some Building Materials Using The Transient Hot Strip Method., *Journal Of Physics D: Applied Physics*, vol. 18, 1985, pp. 1-8
39. S.E. Gustafsson, E. Karawacki, and M.N. Khan, Transient Hot-Strip Method For Simultaneously Measuring Thermal Conductivity And Thermal Diffusivity Of Solids And Fluids. *Journal Of Physics D: Applied Physics*, vol. 12, 1979 pp. 1411-1421
40. S.E. Gustafsson, E. Karawacki, and M.A. Chohan, Thermal Transport Studies Of Electrically Conducting Materials Using The Transient Hot-Strip Technique, *Journal Of Physics D; Applied Physics*, vol. 19 1986 pp. 727-735
41. P. Andersson and G. Backstrom, Thermal Conductivity Of Solids Under Pressure By The Transient Hot Wire Method, *Review Of Scientific Instruments*, vol. 47, no. 2, Feb. 1976, pp. 205-209
42. C.W. Kistler and J. Homeny, *Hot Wire Thermal Conductivity Test System, Presentation Notes*, Orton Refractories Testing and Research Center, Westerville, OH, 1993
43. *Standard Test Method For Thermal Conductivity Of Refractories by Hot Wire (Platinum Resistance Thermometer Technique)* ASTM C1113-90, American Society for Testing and Materials, Philadelphia, PA, 1990

44. J.C. Wilshee, Comparison of Thermal Conductivity Methods, *Proceedings of the British Ceramic Society*, 29, 1980, pp. 153-173
45. D.G. Cahill and R.O. Pohl, Thermal Conductivity Of Amorphous Solids Above The Plateau, *Physical Review B*, vol. 35, no. 8, 1987, pp. 4067-4073
46. H.G. Jensen, Kolrausch Heat Conductivity Apparatus for Intermediate or Advanced Laboratory, *American Journal of Physics*, vol. 38, no. 7, 1970, pp. 870-874
47. K. Bala, P.R. Pradhan, N.S. Saxena, and M.P. Saksena, Effective Thermal Conductivity Of Copper Powders, *Journal of Physics D, Applied Physics*, vol. 22, 1989, pp. 1068-1072
48. N.S. Saxena, M.A. Chohan, and S.E. Gustafsson, Effective Thermal Conductivity Of Loose Granular Materials, *Journal Of Physics D: Applied Physics*, vol. 19, 1986, pp. 1625-1630
49. L. Berkstein, Ed. *CRC of Materials Science*, 64th ed., CRC Press, New York, 1992, pg. 1287
50. M. Abramowitz and I.A. Stegun, *Handbook of Mathematical Functions with Formulas, Graphs, and Mathematical Tables*, National Bureau of Standards, Applied Mathematics Series 55, 1964 pg. 231
51. *Standard Test Method For Determining Average Grain Size*, ASTM E112-88, American Society for Testing and Materials, Philadelphia, PA, 1988
52. E.E. Underwood, Quantitative Metallography, *ASM Metals Handbook, Volume 9, 9th Ed.*, ASM, Metals Park, OH, 1989, pp. 124-134
53. T. Log and M. Metallinou, Thermal Conductivity Measurements Using A Short Transient Hot Strip Method, *Review Of Scientific Instruments*, vol. 63, no. 8 1992, pp. 3966-3971
54. J. Taple, V. Bekeris, M. Sanchez, and G. Simonelli, Reduction Of Radiation Errors In Thermal Conductivity Measurements Above 100 K, *Review Of Scientific Instruments*, vol. 62, no. 4, April 1991, pp. 1054-1056
55. W. Mendenhall and T. Sincich, *Statistics for Engineers and the Sciences*, 4th ed., Prentice -Hall, Englewood Cliffs, New Jersey, 1995, pp. 569-582