THERMACORE, INC.

METEOROID PROTECTION METHODS

FOR SPACECRAFT RADIATORS

USING HEAT PIPES

FINAL REPORT

CONTRACT

~~95437~~

955437

Donald M. Ernst

Principal Investigator

Prepared For

Jet Propulsion Laboratory

Pasadena, California

(NASA-CR-162545) METEOROID PROTECTION METHODS FOR SPACECRAFT RADIATORS USING HEAT PIPES Final Report (Thermacore, Inc.) 99 p HC A05/MF A01 CSCL 22B

N80-14185

Unclas

G3/18 46410

November 1979

TABLE OF CONTENTS

ABSTRACT

The work performed under JPL Contract 955437 was for a preliminary survey program to examine the various aspects of achieving a low mass heat pipe radiator for the NEP spacecraft. Specific emphasis was placed on a concept applicable to a closed Brayton cycle power sub-system.

Three aspects of inter-related problems were examined: the armor for meteoroid protection, emissivity of the radiator surface, and the heat pipe itself.

The study revealed several alternatives for the achievement of the stated goal, but a final recommendation for the best design requires further investigation.

NEW TECHNOLOGY

The following item of new technology was generated under the contract:

1. Segmented Heat Pipe Bumper for Protection Against Meteoroid Collisions - Donald M. Ernst

FIGURES

TABLES

SUMMARY AND CONCLUSIONS

This study program examined various aspects of achieving a low mass heat pipe radiator for the NEP spacecraft, with emphasis on a version using a Closed Brayton Cycle power sub-system. The mass of the radiator is a complex function of several variables. Thus three separate items were evaluated: the meteoroid armor, the emissivity of the surface and the heat pipe itself.

These three factors are inter-independent. However, they were analyzed separately in this preliminary survey program. A fully integrated analysis of a low mass heat pipe radiator would require considerably more effort than was permissible under the scope of this program.

The following conclusions can be drawn:

1. Small diameter pipes with the same wall thickness as larger diameter pipes will show a decreased penetration depth for a given meteoroid.
2. Interfaces between armor and underlying heat pipes are beneficial.
3. The armor must look homogeneous to the meteoroid. Taken in conjunction with the high total emissivity requirement of the surface, this consideration rules out the use of powder metallurgy armor.
4. Chevron fin armor is at this time impossible to evaluate completely. However, based on the comments of Southwest Research Institute it should be pursued further.
5. Segmented heat pipes used as bumpers on top of the radiator heat pipes look quite attractive and need additional evaluation.
6. A total emissivity is excess of 0.9 can be obtained by the use of geometrically produced effects in fins.

7. The mass of the CBC heat pipe without protection can be substantially reduced by going to small diameter heat pipes.

Finally one concludes that this study has just scratched the surface of the many possibilities for low mass radiators, and that there is ample and urgent reason for additional work on the design and evaluation of the various alternatives.

1. METEOROID PROTECTION

The Nuclear Electric Propulsion Spacecraft being considered for use in exploration and intensive study of the outer planets and the surrounds of the solar system will be subjected to the hazards of meteoroids during its journey through space. Accordingly, the spacecraft design must include some type of armor which will protect the vulnerable areas from catastrophic failure upon impact by these meteoroids.

Armor design is crucial to the success of a mission. Without it, missions could not be made. However, in order to achieve a high overall probability of mission success, the armor may be so massive that the system is no longer viable. Thus low mass armor is highly desirable.

In the 400kWe NEP designs currently being looked at, the total specific mass of the power sub-system is targeted at 20 kg/kWe, of which up to 35% (7 kg/kWe) may be necessary to achieve the required degree of protection from meteoroid impact. Accordingly, a reduction in the mass of the armor could be instrumental in the power sub-system achieving its targeted specific mass.

In the power sub-system, the majority of the armor which is required is for the protection of the power conversion heat rejection system which generally employs a matrix of heat pipes. These heat pipes may use a single element radiator for each conversion device: they may be a matrix of interconnecting heat pipes where several main heat pipes accept heat from many conversion devices for distribution to the radiator heat pipes, or the radiator heat pipes may be fed from a gas or liquid metal pumped loop. Whatever the design, the armor must be an integral part of the radiator elements which means that it must not act as radiation shield.

To arbitrarily design armor is not possible. In addition to its being integral to the radiator elements, armor design (and therefore mass) is a complex function of the overall system design. This is seen when essential criteria are established beginning with the mission which defines the flight time and path from which a meteoroid flux model can be generated. Additionally, an overall mission success probability must be defined from which sub-system and component probabilities are generated. These component probabilities are themselves a function of unrelated probabilities based on mechanical, thermal, electrical or meteoroid inflicted degradation or failure.

Armor is required to protect sensitive components from meteoroid puncture. Protection from meteoroids is a function of mission time, meteoroid flux, vulnerable area of the smallest component to be protected, the required probability of survival of that component which in turn is a function of the total number of components and the probability of survival of the collection of components. Therefore it becomes obvious that armor design is of primary importance to mission success.

Accordingly, in order to evaluate low mass armor, certain assumptions must be made in order to establish a base line design. For this purpose the CBC base line radiator heat pipe will be used. Section 3 establishes this base line as well as exploring other possible heat pipe designs.

This section looks at the basic phenomena of hypervelocity impact and four different types of meteoroid armor: solid metal, powder metallurgy, chevron fins, and heat pipes used as bumpers.

The initial basis for this study lay in concepts generated under JPL Contract 955100 (powder metallurgy and chevron fin armor). The intent was to evaluate the effectiveness of these armor designs. However, as information was received from new sources, it became apparent that additional theoretical and experimental work must be carried out to fully evaluate them.

Specifically, this study showed that powder metallurgy material, with its relatively low emissivity, is not well suited to the dual role of armor and radiating surface. Solid armor, with an interface between armor and heat pipe, may prove to be lower mass than originally though and is considerably less complex. The chevron fin armor showed promise but needs considerable additional investigation.

One new concept which was developed and showed several advantages as low mass armor is the idea of segmented heat pipes acting as bumpers to protect the underlying radiator heat pipe. This concept evolved late in the study and was not fully evaluated.

1.1 Hypervelocity Impact Phenomena

In examining hypervelocity impact, various books[1], and reports[2,3,4] were reviewed along with discussion with eminent professionals in the field.[5-8] These pointed out the marked differences in single plate armor, a thin shield protecting a backup plate, multiple shields, as well as the effects of velocity, mass and density of the meteoroid, and the effects of various materials.

Hickerson[9], in Kinslow's book[1], states "The hypervelocity impact of a projectile with a solid target results in an extremely complex phenomenon. A complete description of this behavior would involve consideration of all phases of continuum mechanics theory. In the initial high pressure phases of the impact, the material behaves essentially as an inviscid, compressible fluid since the pressures are high with respect to the maximum shear stresses that can be developed within the material. A crater forms which expands rapidly for a time, and a shock wave emanates from its surface. A stage of plastic deformation follows which apparently decays rapidly into a spherical elastic wave which continues through the target. A complete theory for the description of the hypervelocity impact phenomena would involve not only the above phases but also other situations such as melting and resolidification, vaporization and condensation, and the kinetics of phase change."

Accordingly, the evaluation of low density armor will be carried out by first looking at the mass of solid armor capable of protecting the CBC radiator followed by a narrative on several low density armors as best evaluated by the information available.

1.2 Solid Armor

The JPL supplied penetration equation for the NEP missions is

$$t = 0.0010144\ K \left[\frac{AT}{-\ln P}\right]^{0.2902} \qquad \text{Eq. 1.1}$$

t = Required armor thickness in cm to prevent penetration of the armor by the average meteoroid

K = Materials factor -- given as 1 for Lockalloy

A = Component vulnerable area - cm^2

T = Mission time in hours

P = Individual component survival probability

The equation predicts the required armor thickness to prevent penetration of Lockalloy at room temperature by the expected average meteoroid to be encountered during the NEP mission. In order to evaluate other armor material at elevated temperatures, additional information is required.

Examination of the various equations which have been theoretically and experimentally developed reveals much similarity in the basic equation. Accordingly, Equation 1.1 can be rewritten[3] in terms of the armor properties as:

$$t = \gamma_0 \, a(\rho a)^{-1/2}(v_s)^{-2/3} \left(\frac{Ta}{To}\right)^{1/6} K_1 \left[-\frac{A\Gamma}{\ell n P}\right]^{0.2902} \qquad \text{Eq. } 1.2$$

Where
γ_0 = room temperature cratering coefficient
a = rear surface damage factor
ρa = density of armor - gm/cc
v_s = velocity of sound in armor - cm/sec
Ta = temperature of armor $^{\circ}K$
To = room temperature $^{\circ}K$
K_1 = meteoroid flux constant

The cratering coefficient, γ_r and, the rear surface damage factor, A, vary for different materials as seen in Table 1.1. The three modes of damage by meteoroid impact are defined as follows:

1. Dimple - The impacted surface is physically dented but the integrity of the rear surface is not disrupted.

2. Spall - The impacted surface may be partially penetrated and spallation may occur from the rear surface; however, the complete thickness of the material is not perforated.

3. Perforation - The complete thickness of the impacted material is physically perforated.

The absence of a rear surface damage factor for Lockalloy makes it difficult to compare to other armor materials. However, since the rear surface factors for the listed materials are similar except for Nb-1% Zr, and the fact that Lockalloy is 38% aluminum, the aluminum factor will be used for Lockalloy.

From Equation 1.2 and Table 1.1, the materials factor K in Equation 1.1 can be calculated. Several values are seen in Table 1.2. The value of 0.67 for 316SS is higher than the 0.53 value as suggested by JPL. This discrepancy should be resolved in order to be able to fully evaluate SS as a possible armor.

Depending on the final design of the radiator heat pipes it is difficult to estimate whether dimpling or spallation will render the heat pipe inoperable. Thus, it was decided to use the perforation rear surface factor in order to evaluate the mass of the armor. Table 1.3 shows the perforation factor for the selected material at temperatures of interest for the NEP radiator.

Since Lockalloy and aluminum can not be used throughout the entire temperature range over which the CBC radiator must operate and are definitely not suitable for use in conjunction with the thermionic system, only the higher temperature materials will be evaluated and only at 700°K, the upper end of the CBC radiator temperature.

In discussing the required armor thickness, the thickness of the heat pipe wall must also be taken into consideration, as must the diameter of the heat pipe which defines the vulnerable area, and whether there are fins which can be utilized as armor. In order to reduce the number of variables so that the effects of the armor design could be evaluated, a base line heat pipe was established as seen in Table 1.4.

Table 1.1

CRATERING COEFFICIENT AND REAR SURFACE DAMAGE FACTOR
FOR SELECTED MATERIALS

Material	Cratering Coefficient γ_r	Rear Surface Damage Factor "a"		
		Dimple	Spall	Perforation
2024 AL	1.07	2.5	2.3	1.7
Lockalloy	2.06	-	-	1.7*
316 SS	2.19	2.4	1.9	1.4
A-286	1.77	2.4	1.9	1.4
Nb-1% Zr	1.81	4.5	4.0	1.7

*Estimated Value

Table 1.2

MATERIALS FACTOR K FOR SELECTED MATERIALS
AT ROOM TEMPERATURE

Material	Dimple	Spall	Perforation
2024 Al	1.68	1.54	1.14
Lockalloy	-	-	-
316 SS	1.15	0.91	0.67
A-286	0.93	0.73	0.54
Nb-1% Zr	2.22	1.98	0.84

Table 1.3

MATERIALS FACTOR K FOR PERFORATION OF SELECTED

MATERIALS AND TEMPERATURES

Material	K			
	300°K	500°K	700°K	900°K
2024 Al	1.14	1.24	1.31	1.37
Lockalloy	1.00	1.09	1.15	1.20
316 SS	0.67	0.73	0.77	0.80
A-286	0.54	0.59	0.62	0.65
Nb-1% Zr	0.84	0.91	0.97	1.00

Table 1.4

BASE LINE HEAT PIPE

O.D. - 2.54 (1")

Wall - 0.0254 cm (0.01")

Length - 262 cm (103")

Vulnerable area - 665 cm^2 (103 in^2)

Mass of SS heat pipe - 1.75 kg

Based on a mission time of 87,600 hours and a no-puncture probability of 0.9 for each heat pipe, the required armor thickness and mass for the selected materials at $700^{o}K$ is seen in Table 1.5.

Table 1.5

REQUIRED ARMOR THICKNESS AND MASS @ $700^{o}K$

Material	Thickness	Mass
316 SS	0.27 cm	2.26 kg
A-286	0.22 cm	1.83 kg
Nb-1% Zr	0.34 cm	3.05 kg

The mass of the armor was taken to be:

$$M_a = \frac{\pi D}{2} \; t \; \ell \; \rho a \qquad \text{Eq. 1.3}$$

M_a = Mass of the armor - grams

D = Heat pipe diameter - 2.54 cm

t = Armor thickness - cm

ℓ = Heat pipe length - 262 cm

ρa = Density of the armor - gm/cc

When solid armor is employed, there can be some advantage to having an interface between the armor and heat pipe. Ballistic tests of stainless tubes in aluminum armor show marked improvement over single aluminum tubes.[3] That is, in general, the integrity of the inner tube was not lost nor did spalling of the inner tube take place, even when the inner tube was completely closed. The rear surface thickness factor was found to decrease as a function of H/D. H is the inner tube dimple height and D is the tube diameter. Specific values are seen in Table 1.6.

Table 1.6

REAR SURFACE DAMAGE FACTOR A FOR ALUMINUM OVER STAINLESS STEEL FOR VARIOUS RATIO OF DIMPLE HEIGHT TO TUBE DIAMETER

A	H/D
2.5	0.0 (No Dimple)
2.0	0.13
1.7	0.22
1.5	0.32
1.4	0.34
1.0	0.60
0.9	0.75
0.8	1.0 (Inner Tube Closed)

By comparing the "A" factor for aluminum in Table 1 and 6, it is interesting to note that for an H/D of 0.22 or greater, the rear surface damage factor is less than that required to prevent perforation. If this same relationship holds for stainless on stainless, then the armor thickness required could possibly be reduced below that given in Table 1.5. However, the effects of the dimple on heat pipe operation would have to be taken into consideration.

A dimpled heat pipe can be affected in two ways. First, the liquid flow path could be interrupted or blocked. Second, the vapor flow path could be partially or totally blocked. However, since the armor thicknesses of Table 5 were based on the penetration rear surface damage factor, the heat pipes would have suffered some dimpling and spallation damage by those meteoroids which did not cause penetration. Thus the use of armor over a liner will have a definite advantage over a solid tube. However, this does imply that the armor thickness stands alone, i.e. the thickness of the heat pipe can not be used to reduce the armor thickness.

In order to make use of this interface effect, the validity of it with the materials of construction would have to be proven by tests, as well as establishing what effect a dimple in the heat pipe wall has on the heat pipe's performance. The reduction in heat pipe performance with dimples has to be considered anyway unless the armor is increased in thickness to prevent dimpling.

1.3 Low Mass Armor

Two types of low mass armor were investigated. They are powder metallurgy foam metal and a collection of thin plates. The investigation or evaluation of these armor types raised as many questions as were answered, which leads to the conclusion that additional work needs to be done in this area.

1.3.1 Powder Metallurgy Armor

In order to have a low mass armor either the real density or apparent density of the armor must be reduced. Materials of low density such as aluminum, beryllium, and Lockalloy are not useful at the higher temperatures of interest and they also present a bonding problem to a stainless heat pipe. Thus one look at low apparent density materials such as powder metallurgy foam metal.

From Equation 1.2, it is seen that the required armor thickness is proportional to the reciprocal of the square root of the density of the armor. Since mass is equal to the thickness times the density, the armor mass is proportional to the square root of the density. Thus 25% dense armor will be twice as thick and have 50% of the mass of solid armor. On the surface, this appears to be a good method by which to reduce the mass of the armor. However, for this method to be viable, the other physical properties of the armor cannot change with the apparent density. Also, the armor must appear to be a "solid" to the impinging meteoroids. That is, the diameter of the meteoroids must be at least 10 times the diameter of the particles making up the armor.

For a 50% dense armor made from 2×10^{-3} cm particles, the meteoroid must be at least 2×10^{-2}cm in diameter for the armor to behave as though it were solid. A particle of 2×10^{-2}cm diameter with a 0.5 gm/cc density will have a mass of 2.09×10^{-6}gm. JPL considers only those meteoroids with a mass in excess of 10^{-6}gm as being of concern to the radiator heat pipes. Therefore, if 50% dense armor is to be used it must be made with 2×10^{-3}cm or less diameter particles.

For densities less than 50%, non-spherical particles must be used to make up the armor, and the meteoroid size which will see the armor as being solid will increase accordingly.

The reduction in mass of armor by the use of porous material is based on the other physical properties of the armor being invariant of the density, which is not the case. For instance, the velocity of sound is equal to the square root of the modulus of elasticity divided by the density. The actual velocity of sound of the individual particles will remain the same. However, since the effective path length will be a tortuous one, and increases with decreasing density, the effective velocity of sound should be lower. Thus the required armor thickness and mass will increase as the sonic velocity decreases with decreasing density.

One physical property which definitely changes with the apparent density is the thermal conductivity. A high thermal conductivity is necessary for the armor so that the ΔT through it is low, thus keeping the radiating surface temperature as high as possible. At first one might think that the thermal conductivity is inversely proportional to the apparent density. However, for perfectly square packed spheres of the same diameter the theoretical packing density is 52% and the spheres are tangent to each other. Thus the thermal conductivity can not be 52% of the solid material since the particles only have point contact.

A theoretical treatment of the thermal conductivity of porous material should be carried out in a manner similar to that by which the permeability of porous material has been determined. This model should also take radiation into account, and be followed by experimental determination of the thermal conductivity of various porous materials.

As part of the determination of the effective emissivity of porous material (covered in Section 2) several tests were performed from which an effective thermal conductivity of 50% dense nickel at $1100^{o}K$ was calculated to be about 15% of that of solid nickel. These tests were not designed to measure thermal conductivity. Therefore, the accuracy is at best ±25% but it does indicate that indeed the thermal conductivity of porous metal is considerably less than the apparent density times the thermal conductivity of the base metal. Based on this marked reduction in the thermal conductivity of porous metal, its use as a low mass armor may be limited. The combined effect of reduced thermal conductivity and increased armor thickness for porous armor may increase the ΔT through the armor by an order of magnitude. At $700^{o}K$ the ΔT through solid SS armor is $2.2^{o}K$, thus the ΔT through porous armor may be as high as $22^{o}K$, which at $700^{o}K$ would require an increase in radiating surface area of 13.6% in order to dissipate the same amount of heat as compared to a 1.3% increase for the solid armor. Table 1.7 compares the mass of a $700^{o}K$ SS heat pipe with solid armor and 50% dense porous armor. The armor mass is assumed to be proportional to the square root of the apparent density (optimistic) and the thermal conductivity is 10% of the base material.

From Table 1.7, it appears that the 50% dense armor will have an overall lower mass than solid armor if the assumption about the porous armor material is correct. Further reduction in mass may be possible by going to 25% dense material. However, the effective thermal conductivity will probably decrease by another order of magnitude and the armor will start to look less like a solid surface to the impinging meteoroids.

Additionally, if the interface effect can be utilized, as discussed in 1.2, than the solid armor may be reduced in thickness such that its mass becomes comparable to that of the 50% dense armor.

Table 1.7

MASS COMPARISON

SOLID ARMOR VS. 50% DENSE ARMOR

Solid Armor		50% Dense Armor	
Heat Pipe	1.75 kg	Heat Pipe	1.75 kg
Solid Armor	2.26 kg	Armor (.707 x solid)	1.60 kg
Total	4.01 kg	Total	3.35 kg
1.3% increase in mass due to 22°K armor ΔT		13.6% increase in mass due to 22°K armor ΔT	
Total mass	4.06 kg	Total mass	3.81 kg

1.3.2 Thin Plate Armor

The evaluation of armor to this point has only considered stopping the meteoroid from puncturing the heat pipe wall with the use of solid or porous armor. This section will look at the use of single or multiple thin sheets as a possible means of achieving low mass armor.

Most of the thin shield work has been aimed at the concept of bumpers protecting an underlying armor. In this concept, the impinging meteoroids strike the thin shield causing the meteoroid to break up and spread radially over a large area such that the force per unit area which is observed at the underlying armor is substantially reduced.[3] This type of armor has from 35 to 50% of the mass of solid armor. The shield is approximately 10% as thick as solid armor and placed at least five solid armor thickness away from the underlying armor, which is from 25 to 40% of the thickness of solid armor. Thus the total thickness of a bumper and underlying armor is at least six times that of the solid armor and has 50% of the mass of the solid armor. Bumpers are not attractive for radiator service because they will act as radiation shields. However, in examining thin shields several interesting things were brought to light which were instrumental in arriving at the chevron armor design.

Gehring,[6] in Kinslow's book,[1] states for thin shields "the damage mechanisms to be considered are the breakup and dispersion of the projectile and shield debris at high velocities and the gross deformation, tensile failure, and spallation of the rear sheet."

"Upon striking a thin sheet, a particle or projectile may undergo a variety of processes depending upon impact conditions such as the particle velocity, the particle material and composition, the angle of impact, the material strength, and the thickness of the thin sheet. (A thin sheet as used herein will be defined as a sheet whose thickness is equal to orless than the diameter of the projectile). The particle may be stopped by the sheet, may pass through the sheet essentially undamaged, or may pass through the sheet fractured, molten, or vaporized. The last two cases are the cause of interest for meteoroid impacts as the velocities are sufficiently high to cause melting or vaporization."

"If the thin sheet is penetrated, the debris from the projectile and the shield then travel across the space between the sheets and strike a second sheet. Upon striking the second sheet a shock wave is generated within, and traverses, the second sheet. Depending upon the intensity and the structure of this shock an internal fracture or spall may form, resulting in some cases in complete detachment of some material from the surface of the sheet."

"In addition the second sheet will be given an impulsive load by the impact of the particle-shield debris. This load is applied over a very short period of time (a few microseconds) and results in a second sheet moving with some velocity. The sheet can then fail from this load by tensile failure or shear failure."

"The whole process of fracture of a projectile and a thin shield can be interpreted as a multiple spalling phenomenon that starts at the free surfaces. Hence, the significance of a shield is that it can fragment the projectile, spread the fragments radially and significantly reduce the velocity of many of the fragments below the velocity of the original projectile."

Summarizing, the following can be said about thin shield protecting backup plates.

1. For a shield to be effective, it must break up the particle into small pieces or cause melting to insure that no significant penetration of the second sheet will occur.
2. As the shield thickness is increased, the debris is spread out more.
3. The thickness of the backup shield to prevent failure is proportional to the mass of the projectile for high projectile velocity which assures that the projectile is sufficiently broken up and/or

vaporized.

4. The thickness of the backup shield to prevent failure is proportional to the projectile diameter for low velocity impacts.

5. The lower the melting temperature of the shield material, the lower the velocity of the meteoroid required to cause complete fragmentation of the particle and vaporization of the shield material. ($\simeq$ 6 km/sec for aluminum)

6. Based on thin shields and backup targets, the thickness of a shield to prevent fracture of the back plate for aluminum particles at 30° km/sec is:

$$t_t = \frac{M_p}{0.045} \left(\frac{5.08}{S}\right)^2 \left[\frac{0.0102}{(t_s/D)^2} + 0.079\right] \qquad \text{Eq. } 1.4$$

t_t = Backup target thickness = cm

M_p = Mass of projectile - gram

S = Shield to target spacing - cm

t_s = Thickness of shield - cm

D = Particle diameter - cm

This equation is valid for:

$S \geq 8D$ and

$0.1 \leq t_s/D \leq 1$

To use this equation with projectiles of materials other than aluminum, the t_s/D ratio must be computed on an equivalent mass of aluminum particle, i.e. $D = D_p \left(\frac{\rho_p}{\rho_{al}} \right)^{1/3}$

7. The t_s/D ratio should be as large as possible.

8. The inter-sheet spacing should be as large as possible.

9. The shield thickness scales with the projectile diameter.

10. The thickness of a single sheet armor scales with the projectile diameter.

11. All things taken into consideration, two sheets are better than one on an equal mass basis because any high velocity impact upon a shield results in the spread of the projectile - shield debris, the loss of energy and at most a slight increase in momentum per unit area are less.

12. Experiments with aluminum - aluminum and cadmium - cadmium impacts, supported the theoretical conclusion that two sheets provide more protection than a greater number sheets. These tests were at 7.4 km/sec for the aluminum and 6.4 km/sec for the cadmium.

13. Experimental results for an aluminum - aluminum system showed that if the back up shield could survive an impact of 10 km/sec, which causes the shield-projectile debris to be molten and/or vaporized, the same size shields could resist failure for all low velocity impacts. For aluminum below 7 km/sec the debris still contains fragments which inflict severe damage on the second sheet. In fact, the damage at 2.5 km/sec is the same as at 20 km/sec.

    However, since the average velocity of meteoroids is 20 km/sec the low velocity impact damage will not be critical.

14. The above statements about thin shields are for impacts which are normal to the shield. This is not the case with meteoroids. They will hit at all angles. The tests which have been carried out with

aluminum - aluminum impacts and cadmium - aluminum impacts show that as the angle from the normal is increased the particle-debris becomes more fragmentary and less molten or vaporized thus having the tendency to increase the damage inflicted on the rear sheet. However, the conclusion is that if a two-sheet structure can resist a low-velocity normal impact it can resist the fragment damage due to an oblique impact.

The significance of oblique impacts is in the fact that the shield debris comes off normal to the shield, while the particle debris appears to be spread in the angle between the direction of initial flight and the normal of the shield. It is this concentration of the particle debris that inflicts fragment damage to the second sheet. However, the important thing is that the integrated center of the debris emanating from the back side of the shield has experienced a shift in the direction of motion towards the normal to the shield. It is this change in direction which may be the key to low density chevron armor.

15. The optimum shield thickness to projectile diameter ratio is $t_s/D = 0.15$ and the total thickness of the shield and back up plate divided by the particle diameter is between 1 and 1.5. i.e.

$$1 \leq (t_s + t_b)/D \leq 1.5$$

These are the optimum values to prevent failure for high velocity oblique impact and low velocity normal impact.

16. For aluminum - aluminum impacts at 7.4 km/sec with a 5.08 cm spacing the following equation hold for non-optimum shields.

$$\frac{t_s + t_b}{D} = \frac{4}{5}\,\frac{t_s}{D} + 1 \qquad 5 \geq \frac{t_s}{D} \geq 0.15$$

$$\frac{t_s + t_b}{D} = 5 - 25.7\,\frac{t_s}{D} \qquad \frac{t_b}{D} \leq 0.15 \qquad \text{Eq. 1.5}$$

17. The depth of penetration of glass spheres into aluminum tubes, where the tube wall was held constant at 4.75 times the diameter of the projectile, decreased as the tube diameter decreased. The decrease in penetration as compared to that of an infinite diameter tube (flat plate) was 21.5% for a 2" ID, 32.7% for a 0.5" OD, and 41.7% for a 0.125" ID.

As can be seen from the above summary the amount of information directly applicable to the protection of the NEP radiator is limited, thus any conclusions which are drawn should be further examined by a more comprehensive review of the literature, discussion with current workers in the field and experimental verification of the design.

Having summarized the pertinent work in the field of thin shield protection, what can be said for protection of the NEP radiator? First, as mentioned earlier, the concept of the bumper can not be employed due to the radiation shielding effect. However, the use of multiple thin shields in a configuration shown in Section 2 to have a high intrinsic emissivity, may have some merit. For ease of analysis the shield was assumed to be protecting a flat plate rather than a tube and is seen in Figure 1.1. It is hard to say whether or not protecting a flat surface will be less massive than a curved surface.

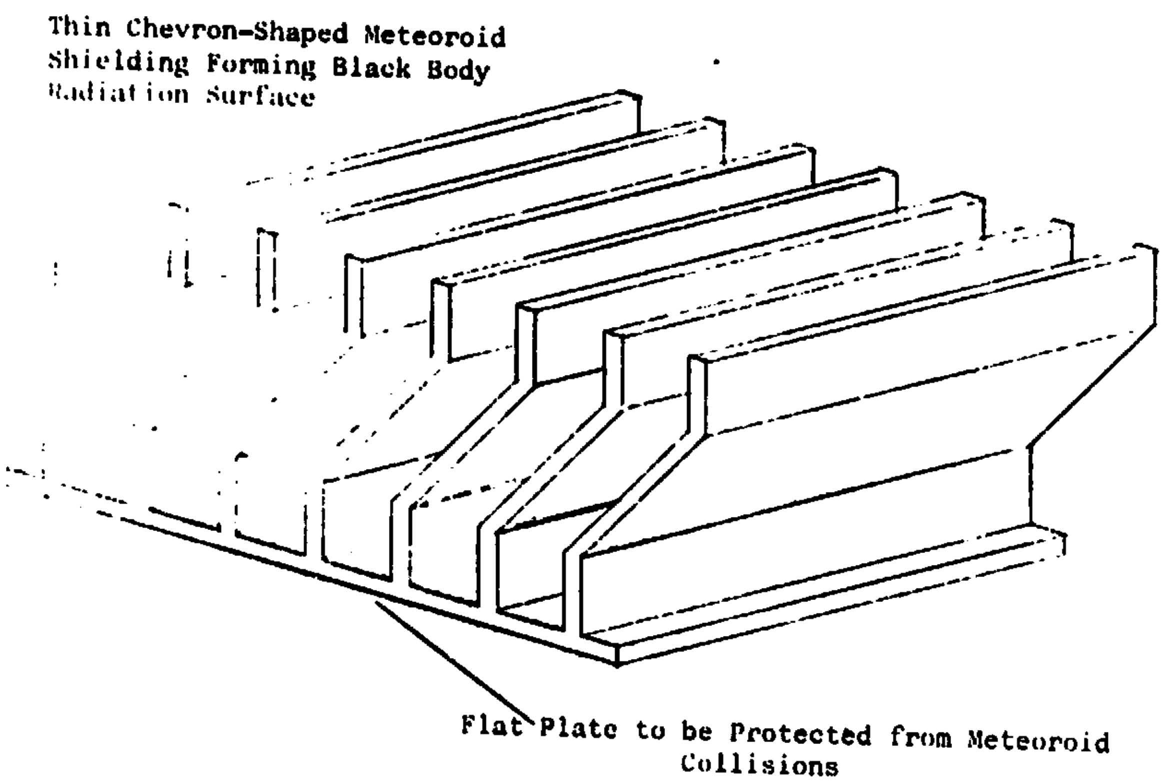


Figure 1.1

Chevron Armor Design for Flat Plate

As seen in 17 above, the smaller diameter tubes showed considerably less vulnerability than a flat plate. However, in order to contour the armor to a heat pipe additional mass may be required. It may also be possible to construct a heat pipe with one surface flat, such as a "D" shape, thus allowing for the use of the shield design, as shown in Figure 1.1.

The idea behind the armor design of Figure 1.1 is that in all probability most meteoroids will not be normal to the surface. Thus, the impact will be oblique to the outer portion of the shield and according to item 14 above, the debris should start to align itself into a path which is normal to the shield i.e. it will be parallel to the plate that the shield is protecting. Likewise, the offset in the shield will protect the underlying heat pipe from normal impacts and if the impact is such that debris gets close to the heat pipe wall, the portion of the shield which is perpendicular to the heat pipe should change its direction of flight to be parallel to the heat pipe wall.

An initial analysis of thin type of armor was carried out under JPL Contract 955100. This analysis was similar to that one in Section 1.3.1 where the mass of the armor was assumed to be equal to the apparent density of the armor to the one half power. There the apparent density is equal to the mass of the armor divided by the total of the volume of the fins plus the volume of the space between the fins.

This analysis is not valid since the density of this volume will not be homogeneous to an incoming meteoroid. Accordingly, the following analysis utilizes the thin shield approach summarized above.

From Table 1.5, the solid armor thickness for 316 SS was shown to be 0.27 cm with a mass of 2.26 kg. Therefore, this is obviously the upper limit for any low mass armor design. Accordingly, a 50% mass reduction was chosen as the target. Thus, the mass of the armor should not exceed 1.13 kg.

In order to evaluate the effectiveness of the armor certain assumptions must be made. First of all, if a 3-section shield is used, then the last third must exhibit a black body cavity effect as discussed in Section 2. For an emissivity of 0.9 to be achieved for a surface emissivity of 0.7, the minimum depth to width ratio is 2:1. (See Figure 2.1). However, the physical dimensions of the fins preclude the use of Equation 1.4 as they do not fall within the contraints of the equation. Accordingly, the critical mass can not be calculated for chevron armor.

Thus, the conclusion is that since the shield to shield spacing is small with respect to the shield thickness (See number 8 above) it is impossible without additional theoretical and experimental work to determine the effectiveness of chevron type armor. However, its potential is high as seen by the comments of James Rand of the Southwest Research Institue in his letter of August 2, 1979, following.

SOUTHWEST RESEARCH INSTITUTE

POST OFFICE DRAWER 28510 • 6220 CULEBRA ROAD • SAN ANTONIO, TEXAS 78284 • (512)684-5111

Department of Ballistics
and Explosives Sciences
August 2, 1979

Mr. Donald M. Ernst
Thermacore, Inc.
P. O. Box 135
Leola, Pennsylvania 17540

Dear Mr. Ernst:

This is in response to your letter of June 12 to Alex Wenzel and our conversation of July 31 pertaining to your proposed heat rejection system. The meteoroid protection system which you propose is quite unique and has definite promise. However, before a rational trade-off study can be performed on the advantages of the chevron armor over the solid armor, research will be necessary to establish the assumptions inherent in the design.

Although much work has been done on the penetration of thin plates at velocities of interest to the ballistics industry, only limited data exist in the hypervelocity regime which is necessary to simulate the meteoroid environment. The advantage of the chevron armor design is dependent on the assumption that the debris and ejecta will occur perpendicular to the fin. Unfortunately, this is only a qualitative observation since data exist which indicate that the projectile will continue on its original flight path while the spall or debris cloud is ejected perpendicular to the target. An experimental program will be necessary to define the limits of this mode of failure. A subsequent program to observe the synergistic effects of your particular design would then be necessary.

The Southwest Research Institute has a highly competent professional staff with experience in defining certain meteoroid impact effects for NASA. However, an estimate of the cost of a program in this area will naturally be dependent on the nature and scope of the work to be performed. I hope that you will plan to visit us when the time comes to prepare a formal proposal for this work.

Again, I would like to encourage you to pursue this concept. The criticism that you will undoubtedly receive regarding the inefficiency of spaced bumper shields is based on normal impact theory. Should the

SAN ANTONIO HOUSTON TEXAS AND WASHINGTON D C

projectile be completely arrested by the fin and if the debris cloud is normal to the fin, it does in fact seem possible to divert the cloud to a direction parallel to the pipe. Only a well defined ballistics program will confirm this.

If I can be of any further assistance, please do not hesitate to call.

Sincerely yours,

James L. Rand
Staff Engineer

JLR:jc

cc: A. B. Wenzel
J. S. Wilbeck

1.4 Heat Pipe - Heat Pipe Armor

One concept which grew out of the evaluation of the different types of armor is that of using a thin walled heat pipe to protect the underlying radiator heat pipe. Several possible design configurations are seen in Figure 2.1. These heat pipe designs could employ configuration pumping rather than conventional wicks.

The radiator heat pipe must be capable of axially transferring all of the required power. However, the bumper heat pipe requirement is a radial one with only enough axial capability to even out non-uniformities. Thus, the bumper heat pipe will not require as much wick structure as the radiator heat pipe and will therefore have less mass. In fact, the bumper heat pipe could have a knurled inside surface (x's) thus providing radial and axial grooves for liquid flow paths.

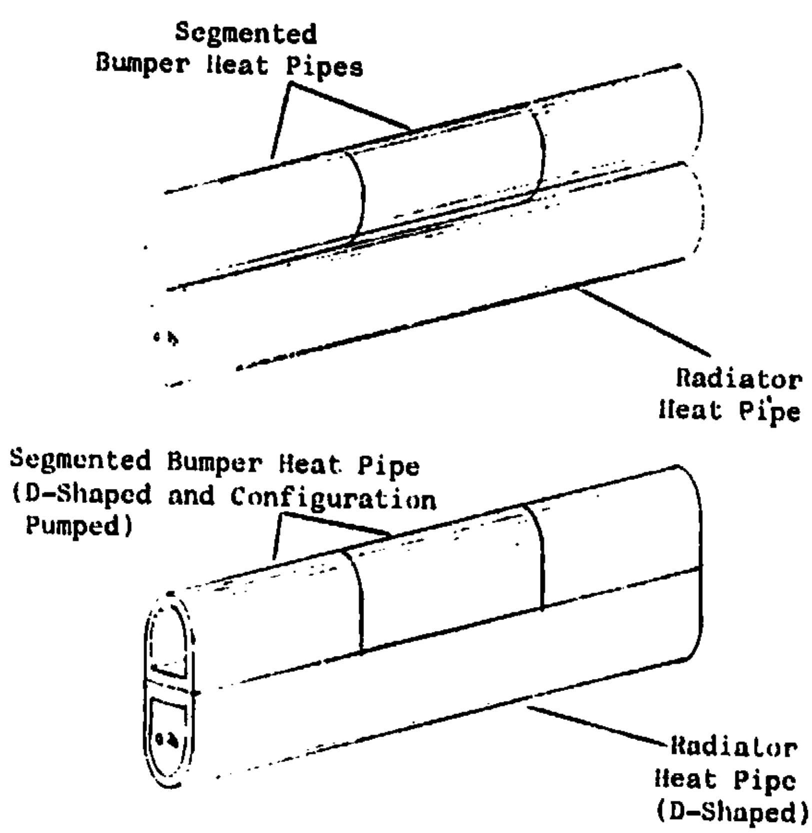


Figure 2.1
Segmented Bumper Heat Pipe

If the liquid inventory is small then a stainless steel circular bumper heat pipe could have a wall thickness of 0.135 cm and have 50% of the mass of the solid armor of Table 5.

Likewise, "D" shaped heat pipes with the bumper heat pipe having a thicker flat section, could make use of the interface effect as discussed in 1.2. In addition, the mass of a "D" shape is less than a complete circle of equal wall thickness and diameter.

One thing which has to be considered is what happens when the bumper heat pipe is punctured and the radiator heat pipe remains intact. The remains of the bumper heat pipe will act as a radiation shield and reduce the radiant heat transfer by up to 50%. However, the radiator heat pipe will remain intact. Thus, there arises a trade-off in the bumper heat pipe wall thickness and the radiation shield factor.

It may be possible to make the bumper heat pipe out of ten individal compartments. Thus, if one of the bumper heat pipes is penetrated, then the other nine can still dissipate the heat at a slightly higher overall temperature. For the CBC radiator, if the small diameter heat pipes evaluated in Section 3 were to be protected by large diameter bumper heat pipes with ten segments, the total mass of the system will be considerably reduced. In fact, the bumper heat pipe mass should be much less than that which would be required if a "T" bar bumper was used.

A complete evaluation of the bumper-heat pipe concept was not possible as its evolution as an idea came at the conclusion of the study. However, it does have enough merit to be considered along with the chevron fins to be studied in more detail.

2. HIGH EMISSIVITY SURFACES

The mass of a space vehicle's radiator is directly proportional to the effective total hemispherical emittance of the radiating surface. Additionally, the emitting surface must be thermally stable in the environs of space; i.e. it must not evaporate into the vacuum of space nor be affected by the slow but continuous ero sion by the micrometeoroids of 10^{-6} grams or less which will not otherwise damage the spacecraft. For minimum mass, the radiating surface should have an emissivity of 1. Thus, a minimum goal of 0.9 should be established for the radiating surfaces of a spacecraft radiator.

There are several ways in which an emissivity of 0.9 can be achieved. They include:

1. Use a material which has an emissivity of 0.9.
2. Chemically treat the surface to oxidize it or produce a compound of the base material which has a high emissivity.
3. Apply a coating which has a high emissivity.
4. Geometrically produced effects.

When one looks at each of these four possibilities, it becomes evident that most high emissivity materials are non-metals of poor thermal conductivity such as ceramics, porcelain, glass, marble, water, ice, and wood, and usually exhibit these properties below 200°C.

Likewise, if a metal such as 316 SS, A-286 or Nb-1% Zr is to be used as the heat pipe - armor material, the chemical treatment of these surfaces to produce a high emissivity compound or oxide is possible. However, reactive layers usually have sufficient vapor pressure in the range of interest such that they are not stable for ten years.

The remaining concepts are coatings and geometrically produced effects. Coatings are seen to be capable of producing emissivities of 0.9. Geometrically produced effects are shown to be a function of the geometry and surface emissivity with effective emissivities greater than 0.9 possible.

2.1 Coatings

Two coatings of interest have been shown to be thermally stable at 1000°K for 10,000 hours in a vacuum by Pratt and Whitney Company.[10] These were calcium titanate and iron titanate on 310 SS tubing, both of which exhibited emissivities of 0.9.

The extrapolation of 10,000 hours to 87,600 hours is not unreasonable. However, it is not known whether these coatings will be able to survive the ten or so thermal cycles which will be required for the multiple fabrication steps and ground level system check out, and then survive the shock and vibration of launch, plus the continuous erosion by the cosmic dust.

The high emissivity of these coatings is a function of at least two things: the normal high emittance of the titanates and the fact that the coatings are granular in composition which produces a high emissivity by geometric effects as is discussed in 2.2 below.

Coatings such as these are required if solid armor is to be used to protect the radiator heat pipe from meteoroids. However, the use of high emissivity coatings on the surface of low mass armor which also produces a geometric effect may provide emissivity greater than 0.9 which in turn will allow for further radiator mass reduction.

2.2 Black Body Effect

The artificial roughening of a surface is a known means of increasing its emissivity. This enhancement of emissivity by surface or geometric effects has been treated quite thoroughly by Sparrow.[11] The basis for this enhancement is the multi-reflections between surfaces which "see" each other, and is a function of the angular separation of a Vee-shaped cavity, the depth to width ratio for rectangular groove cavities and depth to radius ratio for cylindrical cavities. Additionally, the absolute value of the emissivity of the enclosing surfaces is important, as is the type of surface involved, which in turn defines the type of reflection, i.e., specular or diffuse.

Sparrow mathematically derived the effective emissivity of parallel plate or rectangular groove cavities for specular and diffuse reflecting surfaces. Figure 2.2 shows Sparrow's results from which one sees that emissivity enhancement is most dramatic for surfaces of low specular emissivity and low depth to width ratio.

This emissivity enhancement assumes that the enclosing surfaces are isothermal, of uniform emissivity and applies only to the projected surface area bound by the cavity and does not include the surfaces of the edges forming the cavity, i.e. fin tips for chevron armor in Figure 1.

Diffuse radiation denotes directional uniformity, i.e. the intensity of the radiation leaving a diffusely emitting and diffusely reflecting surface is uniform in all angular directions. Likewise, radiation arriving with uniform intensity at a surface is diffusely distributed. In other words, regardless of whether the incident radiation arrives as a beam directed along the surface normal, or as

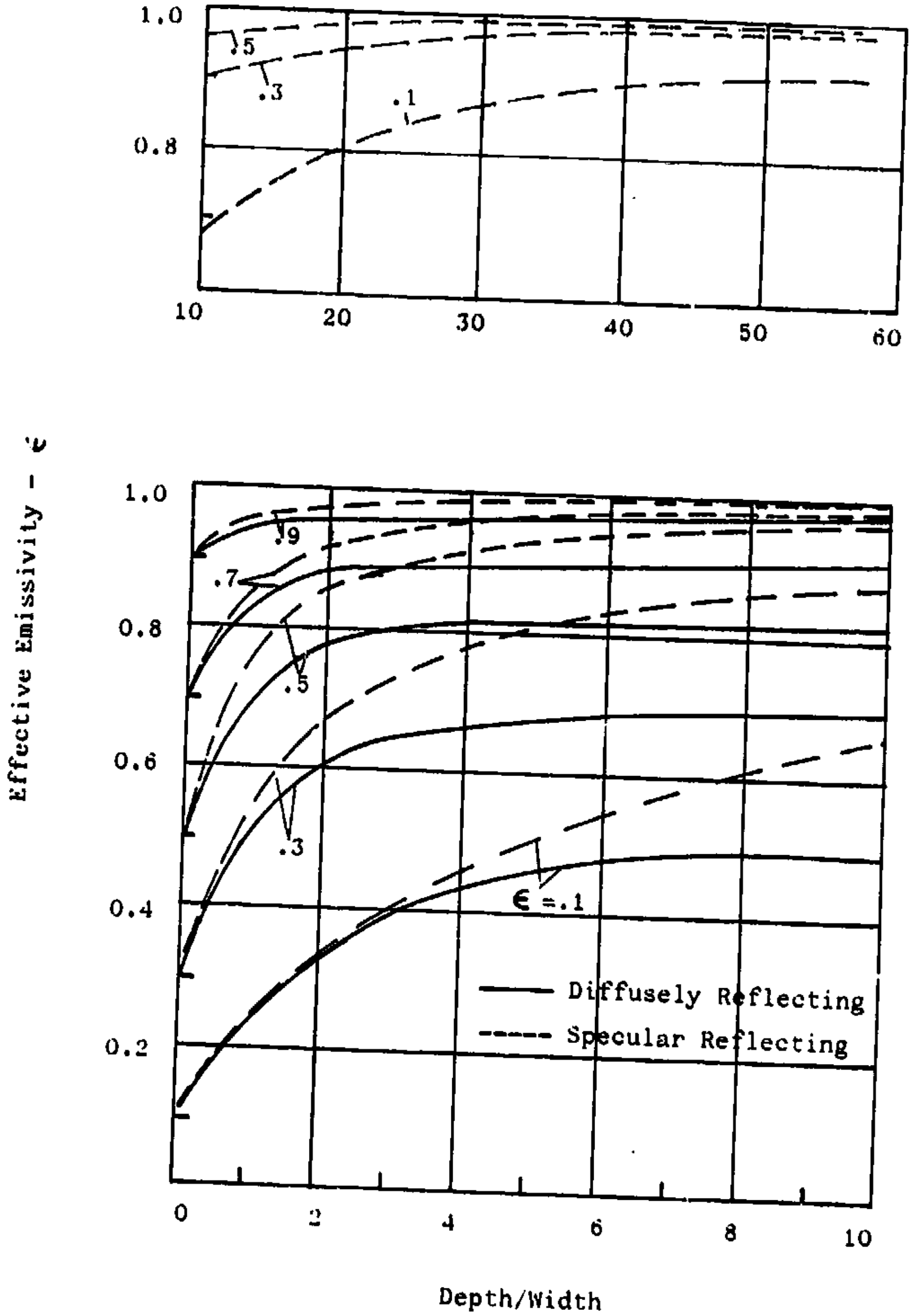


Figure 2.2

Effective Emissivity of Diffuse and Specular

Reflecting Rectangular Groove Cavities

a beam grazing the surface, or is uniformly distributed over the hemisphere, the radiation reflected from a diffuse surface is always of uniform intensity.

Specular or mirrorlike reflection maintains directional dependence, and a beam of radiation contained in a solid angle inclined at some angle to the normal of the surface will be reflected in the same solid angle on the opposite side of the normal at the same inclination.

Although a black body is a diffuse emmitter of energy it is obvious from Figure 2 that the surfaces of a rectangular cavity can not be diffuse reflectors in order to achieve a high emissivity which is desired.

There is no known material which is a perfectly diffuse reflector. However, it is interesting to note that the nonmetallic materials which have a high emissivity such as Al_2O_3, paper, wood, glass, and ice, also have a uniform emittance for inclination angles between 0 and 60^o before falling off to zero.

Conversely the emittance of the metals typically shows a very high degree of directional dependence with a peak around 80^o. Thus one concludes that metallic surfaces will behave more like a specular reflector and that nonmetallic surfaces will be more closely described as diffuse reflectors.

2.2.1 Powder Metallurgy Material

In an attempt to understand the properties of powder metallurgy material, Thermacore utilized some of its IR & D funds to construct a potassium heat pipe which had an annulus of sintered nickel powder around a portion of its condenser. This powder had eight holes in it. Four were 1 mm in diameter, four were 6 mm in diameter. Each set of holes had depth to diameter ratios of 3:1, 6:1, 7.5:1 and 10:1.

The results of the experiment were quite inconclusive with respect to measuring the effective emissivity of the material since it was observed that the thermal conductivity of the 50% nickel powder was so poor that the matrix could not be kept at a uniform temperature even with the application of radiation shields.

Qualitatively the following can be said of the experiment:

1. The smaller diameter holes appeared to have a lower effective emissivity then the larger holes.
2. The larger the depth to diameter ratio the higher the effective emissivity.

From 1, it can be concluded that there is a relationship between the diameter of the cavity and the roughness of the cavity walls which affects the effective emissivity of the cavity. (In the experiment, the cavity walls were both of identical material, 50% porous nickel).

Quantitatively it was concluded that the effective emissivity of the 50% nickel fell between 0.4 to 0.7. This is not at all unexpected, for if one looks at the actual emitting surface of the 50% dense material one sees that the surface is 80% nickel and 20% voids. Thus even if the 20% voids have an emissivity of 1, and the 80% nickel has an emissivity of 0.5 (oxidized nickel), and total effective emissivity of the surface would then be 0.6.

It can be concluded that for powder metallurgy to be used to achieve an emissivity of 0.9 or greater, the following is required. The density will have to be less than 50% so the projected surface area of the cavities is increased, and the material which makes up the powder metallurgy should have a high emissivity to begin with.

2.2.2 Fins

The evaluation of fins such as the chevron design of Figure 1.1 is considerably more straight forward. Based on the results of 2.2.1, a conservative approach is to assume that the surfaces of the fins do not have any roughness factor by which to enhance the surface emissivity. Also, based on the apparent high specular nature of metals, the surfaces will be assumed to be specular reflecting.

From Figure 2.1, it is seen that a depth/width ratio of 10:1 produces an effective emissivity of 0.9 for a material with a specular emissivity as low as 0.3. If the fins have an emissivity of 0.5 a ratio of only 4:1 is required to achieve 0.9.

If one assumes the use of iron titanate on the fin with an emissivity of 0.9, then at a ratio of 2:1 the effective emissivity will be in excess of 0.98. However, this may be risky based on the results of the powder metallurgy tests.

One thing that must be considered is what the total emissivity of the final structure is. If the fins represent 25% of the surface area the cavity represents 75% and the following equation can be written for the effective emissivity:

$$\epsilon_{eff} = \frac{\epsilon_m A_m + \epsilon_c (A - A_m)}{A}$$

ϵ_{eff} = Total effective emissivity of the radiator surface

ϵ_m = Emissivity of the edges of the fins

ϵ_c = Effective emissivity of the cavity

A_m = Area of the edges of the fin

A = Total area of radiator

Thus if we have 25% fin area and $\epsilon_{eff} = 0.9$ then

$$0.6 \leq \epsilon_m \leq 0.9 \quad \text{while} \quad 1 \geq \epsilon_c \geq 0.9$$

Therefore, a high emissivity fin material is required to increase the effective emissivity of the cavity as well as the edges of the fins.

Additionally, if the ΔT through the fin is taken into consideration it is seen that short stubby fins will perform thermally the best, but may not provide the required amount of meteoroid protection.

It is concluded that fins can produce a total surface emissivity in excess of 0.9. To achieve these high emissivities the surface area of the fin tips should be as low as possible and the surface emissivity as high as possible.

Additionally, it is seen that there will be a delicate balance in the protection afforded by the fins, a chevron armor, the ΔT in the fins and the associated mass increase and the effective emissivity of the radiating surface. Additional work is necessary to fully evaluate the total effectiveness of chevron fins to produce an effective low mass armor with a total effective emissivity in excess of 0.9.

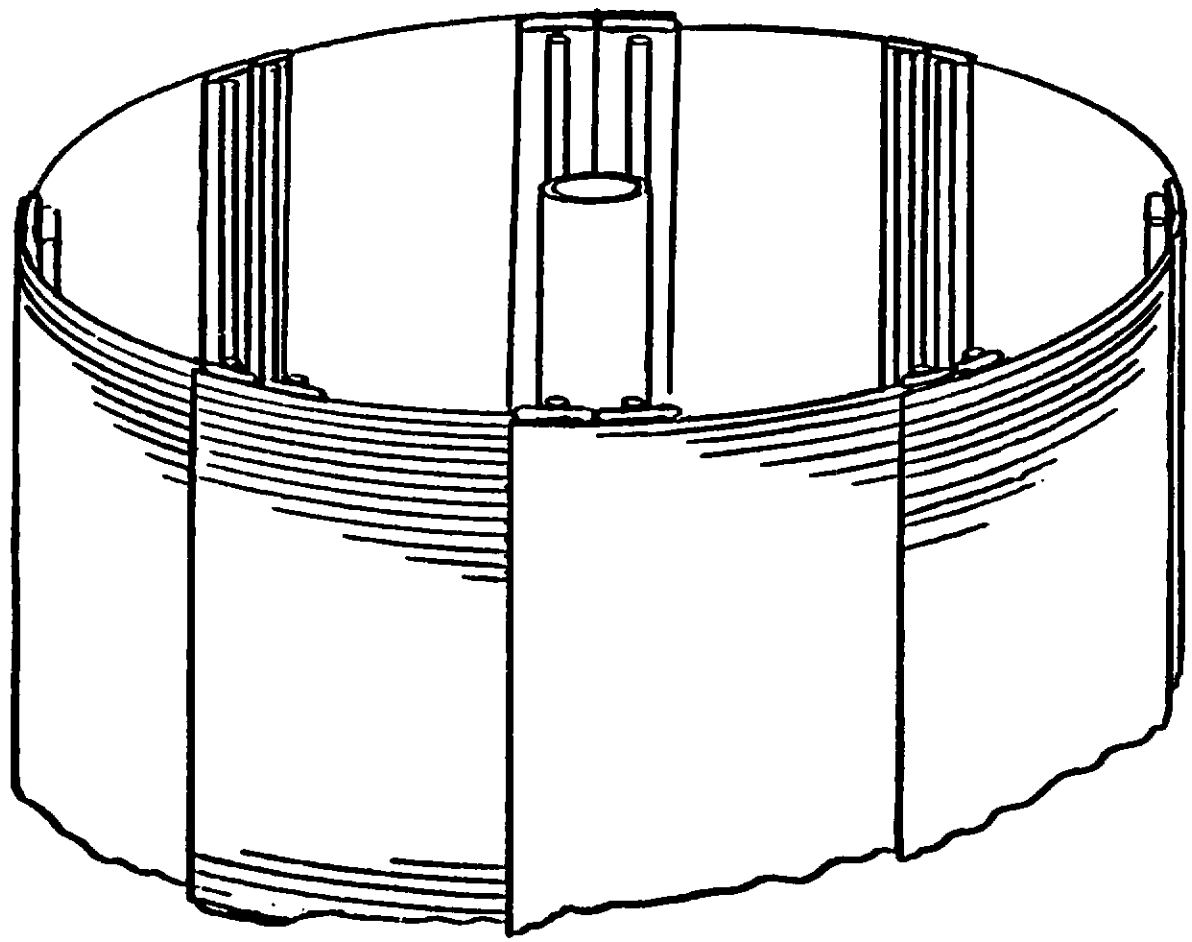
3. HEAT PIPE DESIGN FOR CBC RADIATOR

The 400 kW_e Closed Brayton Cycle power system for the Nuclear Electric Propulstion Spacecraft has been designed by Garrett AirResearch[12] to use heat pipes to achieve a thermally effective radiator which has a high survival probability. It is also anticipated that the heat pipe design will lead to a low specific mass. The heat pipe design evaluated in this work is for use in a cylindrical array as seen in Figure 3.1. This design has eight dual gas-to-radiator heat pipe heat exchangers fed from a dual central duct. The heat pipes are attached to both gas ducts over a length of 43 cm on each duct. Thus, the heat pipes provide armor protection for the gas ducts.

In normal operation, the total 86 cm length attachment over the heat pipes to the gas ducts will be used as heat pipe evapomtors. The condenser is 176 cm long. If either gas duct or engine should fail, then the whole power load will be transferred to the heat pipes through only one of the 43 cm attachments. Accordingly, for design considerations, the heat pipe must be sized as though it had a 43 cm evaporator, 43 cm adiabatic and 176 cm condenser.

Four different sets of heat pipe designs were analyzed with respect to mass and performance. However, no consideration was given to the required heat pipe armor and tradeoffs in the heat pipe diameter versus T-bar fins for total mass. The overall heat pipe cell dimension as designed by Garrett is 3.175 cm (1.25") and includes heat pipe and fins. All heat pipes discussed in the Sections 3.1 and 3.2 have computer printouts of their performance tabulated in Appendix 1.

FIGURE 3.1



CBC RADIATOR CONFIGURATION

3.1 Baseline Design

The total power to be dissipated is 1.1×10^6 watts. From the gas side of the radiator heat exchanger, heat pipe temperatures were calculated by Garrett AiResearch to range from $707^\circ K$ down to $492^\circ K$. The power levels are 720 watts per heat pipe at $707^\circ K$ and 169 watts per heat pipe at $492^\circ K$. Thus, $\sigma A \epsilon$ can be computed to be 2882×10^{-12} watts/$^\circ K^4$ from:

$$P = \sigma A \epsilon T^4 \qquad \text{Eq. 3.1}$$

where

P = Power radiated - watts

σ = Stefan Boltzman Constant = $5.67 \times 10^{-12} \dfrac{\text{watts}}{\text{cm}^2 - {}^\circ K^4}$

T = Heat pipe temperature - $^\circ K$

A = Individual heat pipe radiating area - cm^2

ϵ = Effective thermal emissivity

Table 3.1 shows the required heat pipe power for each of the end temperatures and each temperature divisible by $25^\circ K$.

Garrett AiResearch's baseline design is a 2.54 cm (1") O.D. heat pipe with a 0.0762 cm (.03") wall. The initial heat pipe designs under these conditions are seen in Table 3.2. Rubidium is the preferred heat pipe fluid from $707^\circ K$ down to $650^\circ K$. Below $650^\circ K$ Dowtherm A (DTA) is the preferred fluid. In both cases, a screen covered groove design is found to be the lowest mass system of those investigated. The rubidium heat pipes have a 1.75 kg mass. The DTA heat pipes have a 1.74 kg mass.

TABLE 3.1

Temperature		Req. Power
°K	°C	Watts
707	434	720
700	427	692
675	402	598
650	377	514
625	352	440
600	327	373
575	302	315
550	277	264
525	252	219
500	227	180
492	219	169

REQUIRED POWER PER HEAT PIPE AT ELEVEN DIFFERENT TEMPERATURES

TAB. 3.2

HEAT PIPE MASS & PERFORMANCE FOR BASELINE DESIGNS

Evaporator - 43 cm Adiabatic - 43 cm Condenser - 176 cm S = Sonic Limit C = Capillary Limit			Fluid: Rb O.D.: 2.54 cm # Grooves: 25	Vessel: 304 SS Wall: 0.0762 cm Groove Width: 0.2 cm			Fluid: DTA O.D.: 2.54 cm # Groove: 27	Vessel: 304 SS Wall: 0.0762 cm Groove Width:0.2 cm		
Temperature		Req. Power	ΔT @ Req. Power	Power Limit	Mass	Groove Depth	ΔT @ Req. Power	Power Limit	Mass	Groove Depth
°K	°C	Watts	°C	Watts	Kg	Cm	°C	Watts	Kg	Cm
707	434	720	2.56	1750-S	1.75	0.05				
700	427	692								
675	402	598								
650	377	514	6.44	608-S	1.75	0.05				
625	352	440					3.89	545-C	1.74	0.065
600	327	373								
575	302	315								
550	277	264								
525	252	219								
500	227	180								
492	219	169					1.73	710-C	1.74	0.065

Table 3.3 shows the same heat pipes, which have been, for the most part, optimized with respect to the number of grooves and their aspect ratio. The rubidium heat pipes have a 1.48 kg mass. The DTA heat pipes have a 1.55 kg mass.

The average mass reduction is 14%. Further groove optimization may result in an additional 1 or 2% mass reduction. However, far greater mass reduction can be realized by O.D. and/or wall thickness reduction.

Table 3.4 shows the 2.54 cm (1") heat pipe with a 0.025 cm (.01") wall. This wall thickness is 0.01 times the diameter and has been shown[13] to be acceptable for use as a heat pipe containment vessel where external buckling is the ultimate constraint, i.e., the internal pressure of the heat pipe was less than 14.7 psi, thus long term creep due to hoop stress was low.

The use of a wall thickness 0.01 times the diameter was developed for niobium, which has a modulus of elasticity of 15×10^6 psi. This includes a safety factor of 2. Stainless steels have moduli of about 28×10^6 psi which reduces the thickness/diameter ratio of about 0.008 with a safety factor of 2. However, the use of 0.01 as a thickness to diameter ratio will be used to assure success.

Examination of DTA at 625°K shows a fluid pressure of 85 psi which develops a hoop stress of 4250 psi. This stress is acceptable, since 316 SS will only creep 0.1% in 10^5 hours at 1100°F under a stress of 6000 psi.

The rubidium heat pipes have a mass of 0.69 kg and the DTA heat pipes have a mass of 0.78 kg.

TA : 3.3

OPTIMIZED HEAT PIPE MASS & PERFORMANCE - BASELINE DESIGN

Evaporator - 43 cm Adiabatic - 43 cm Condenser - 176 cm S = Sonic Limit C - Capillary Limit			Fluid: Rb O.D.: 2.54 cm # Grooves: 25 Vessel: 304 SS Wall: 0.0762 cm Groove Width: 0.275 cm				Fluid: DTA O.D.: 2.54 cm # Groove: 25 Vessel: 304 SS Wall: 0.0762 cm Groove Width: 0.275 cm			
Temperature		Req. Power	ΔT @ Req. Power	Power Limit	Mass	Groove Depth	ΔT @ Req. Power	Power Limit	Mass	Groove Depth
°K	°C	Watts	°C	Watts	Kg	Cm	°C	Watts	Kg	Cm
707	434	720	2.43	815-C	1.48	0.02				
700	427	692								
675	402	598								
650	377	514	5.83	640-S	1.48	0.02				
625	352	440					9.32	507-C	1.55	.055
600	327	373								
575	302	315								
550	277	264								
525	252	219								
500	227	180								
492	219	169					3.55	555-C	1.55	.055

TABLE [illegible].4

OPTIMIZED HEAT PIPE MASS & PERFORMANCE FOR THIN WALLED BASELINE DESIGN

Evaporator – 43 cm Adiabatic – 43 cm Condenser – 176 cm S = Sonic Limit C = Capillary Limit			Fluid: Rb O.D.: 2.54 cm # Grooves: 25		Vessel: 304 SS Wall: 0.0254 cm Groove Width: 0.275 cm		Fluid: DTA O.D.: 2.54 cm # Groove: 25		Vessel: 204 SS Wall: 0.0254 cm Groove Width: 0.275 cm	
Temperature		Req. Power	ΔT @ Req. Power	Power Limit	Mass	Groove Depth	ΔT @ Req. Power	Power Limit	Mass	Groove Depth
°K	°C	Watts	°C	Watts	Kg	Cm	°C	Watts	Kg	Cm
707	434	720	1.55	820–C	0.69	0.02				
700	427	692								
675	402	598								
650	377	514	4.56	705–C	0.69	0.02				
625	352	440					5.72	555–C	0.78	0.055
600	327	373								
575	302	315								
550	277	264								
525	252	219								
500	227	180								
492	219	169					2.49	710–C	0.78	0.055

Design Optimization

Examination of Tables 3.2, 3.3 and 3.4 reveals that a reduction in diameter of the rubidium heat pipes would soon result in the heat pipe becoming limited by sonic shock wave development in the vapor. However, the DTA pipes are capillary limited, thus a reduction in O.D. is possible. Accordingly, a higher pressure fluid, mercury, was used in small diameter pipes in place of rubidium. These results are seen in Table 3.5.

The mercury heat pipes are 0.635 cm (.250") in diameter with a wall to diameter ratio of 0.01. The mass of the mercury heat pipes are 0.45 kg and have a hoop stress of 625 psi at 707°K.

The DTA heat pipes are 0.9525 cm (.37") in diameter with a wall to diameter ratio of 0:01. They have 12 grooves 0.275 cm wide by a depth that varies from 0.075 cm down to 0.05 cm. Accordingly, their mass varies from 0.31 kg down to 0.27 kg. The DTA heat pipes at 625°K will have a hoop stress of 1600 psi.

The mercury heat pipes of Table 3.5 have eight grooves 0.2 cm wide by 0.02 cm deep. Optimizing the number of 0.275 cm wide by .02 cm deep grooves for different power levels results in a reduction in mass. At 707°K, a five-groove heat pipe has a mass of 0.29 kg. At 675°K, four grooves have a mass of 0.28 kg and at 550°K, three grooves have a mass of 0.27 kg. These results are seen in Table 3.6. Also shown in Table 3.6 is the thermal performance of two of the mercury heat pipes with 86 cm evaporators, which shows an increase in maximum power capability and a reduction in total ΔT.

Both the DTA heat pipes of Table 3.5 and the mercury heat pipes of Table 3.6 have a performance ΔT. Accordingly, it is important to assess the effect of this temperature loss in terms of increased

TABLE 3.5

OPTIMIZED HEAT PIPE MASS & PERFORMANCE - ALTERNATE DESIGN

Evaporator - 43 cm Adiabatic - 43 cm Condenser - 176 cm S - Sonic Limit C = Capillary Limit			Fluid: Hg O.D.: 0.9525 cm # Grooves: 8		Vessel: 304 SS Wall: .01 cm Groove Width: 0.200 cm		Fluid: DTA O.D.: 1.27 cm # Groove: 12		Vessel: 304 SS Wall: 0.0127 cm Groove Width: 0.275 cm	
Temperature		Req. Power	ΔT @ Req. Power	Power Limit	Mass	Groove Depth	ΔT @ Req. Power	Power Limit	Mass	Groove Depth
^oK	^oC	Watts	^oC	Watts	Kg	Cm	^oC	Watts	Kg	Cm
707	434	720	2.69	930-S	0.45	0.02				
700	427	692								
675	402	598								
650	377	514								
625	352	440	1.98	900-C	0.45	0.02	15.04	515-C	0.31	0.075
600	327	373					10.73	420-C	0.29	0.065
575	302	315					8.38	370-C	0.29	0.06
550	277	264	2.08	805-C	0.45	0.02	6.49	305-C	0.28	0.055
525	252	219					4.98	240-C	0.27	0.05
500	227	180								
492	219	169					4.05	215-C	0.27	0.05

TABLE 3.6

OPTIMIZED HEAT PIPE MASS & PERFORMANCE - ALTERNATE DESIGN

Evaporator - 43 cm Adiabatic - 43 cm Condenser - 176 cm S = Sonic Limit C = Capillary Limit			Fluid: Hg O.D.: 0.635 cm # Grooves: 3-5 Vessel: 304 SS Wall: 0.00635 cm Groove Width: 0.2 cm				Fluid: Hg O.D.: 0.635 cm # Groove: Vessel: 304 SS Wall: 0.00635 cm Groove Width: 0.2 cm Evaporator: 86 cm Condenser: 176 cm			
Temperature		Req. Power	ΔT @ Req. Power	Power Limit	Mass	Groove Depth	ΔT @ Req. Power	Power Limit	Mass	Groove Depth
°K	°C	Watts	°C	Watts	Kg	Cm	°C	Watts	Kg	Cm
707	434	720	4.13	770-C	0.24	(5) .02	2.5	1625-C	0.29	(5) .02
700	427	692								
675	402	598	3.62	610-C	0.28	(4) .02				
650	377	514								
625	352	440	3.30	445-C	0.27	(3) .02				
600	327	373								
575	302	315								
550	277	264	8.35	350-C	0.27	(3) .02	4.79	560-S	0.27	(3) .02
525	252	219								
500	227	180								
492	219	169								

mass (length of condenser) to be able to radiate the required power. Appendix 2 develops Equation 3.2 which is the increase in mass of heat pipe due to its ΔT.

$$dm = m \frac{l_c}{l_t} \left[(T_o/T)^4 - 1 \right] \qquad \text{Eq. 3.2}$$

Where

dm = Increase in mass

m = Initial mass of heat pipe

l_c = Length of heat pipe condenser

l_t = Total length of heat pipes

T_o = Desired operating temperature

T = Actual operating temperature

$T_o - T$ = ΔT down heat pipe

From Table 3.5 and 3.6, using the lowest mass heat pipes, the increase in mass was calculated using Equation 3.2 and is tabulated in Table 3.7. Therefore, to a first approximation, one can say that the heat pipes for the CBC radiator will have a mass of 0.3 kg each.

The performance of the mercury heat pipes is based on perfect wetting, that is, the wetting angle is zero (0). For long term stability, this may not be the case. Wetting angles from 0-60 degrees have been observed, with 30-60 degree angles the most common. Since the capillary force is a function of the cosine of the wetting angle, the mercury heat pipes may have a reduction of capillary force of up to 50% (cos 60 = .5). This reduction in performance will then require a reoptimization of the heat pipes with a small increase in mass.

Development work may be required to establish a reproducible wetting angle for mercury in heat pipe service.

3.3 Advanced Heat Pipe Concept

The grooved heat pipe designs of Sections 3.1 and 3.2 were optimized to an approximate mass of 0.3 kg per heat pipe, exclusive of fins and armor. This mass is quite low and may be acceptable in the overall system. However, there are several heat pipe design concepts which may offer further reduced mass with increased performance. These include but are not limited to arterial wick heat pipes and configuration pumped heat pipes. These wick structures were not available in Thermacore's computer library and were, therefore, not included in the analysis.

3.3.1 Artery/Wick Heat Pipes

There is a natural division in heat pipe fluids which takes place at approximately 600°K. Above 600°K, the liquid metals are useful working fluids. Below 600°K, one generally deals with non-metallic fluids and devises structures which compensate for their inferior physical properties. The low temperature fluids, taken as a class, have relatively low latent heats of vaporization, low surface tension, and low thermal conductivity. The consequences are that for a given heat transfer rate, heat pipes using these fluids must move relatively large quantities of liquid with unusually low pressure losses, yet must maintain very thin liquid films in the heat flow path. The arterial wick structures of Figure 3.2 have been used to offset these property limitations. The artery provides the primary liquid return

TABLE 3.7

INCREASE IN MASS DUE TO PERFORMANCE ΔT

TEMPERATURE ($^{\circ}K$)	POWER (W)	FLUID	MASS (Kg)	ΔT ($^{\circ}C$)	dM (Kg)	NEW MASS (Kg)
707	720	Hg	0.291	4.13	4.6×10^{-3}	0.296
700	692					
675	598	Hg	0.280	3.63	4.6×10^{-3}	0.284
650	514					
625	440	Hg	0.273	3.30	3.9×10^{-3}	0.277
600	373					
575	315	Hg	0.273	5.81	7.5×10^{-3}	0.280
550	264	DTA	0.280	6.49	1×10^{-2}	0.290
525	219	DTA	0.273	4.98	7.1×10^{-3}	0.280
500	180					
492	169	DTA	0.273	4.05	6.2×10^{-3}	0.279

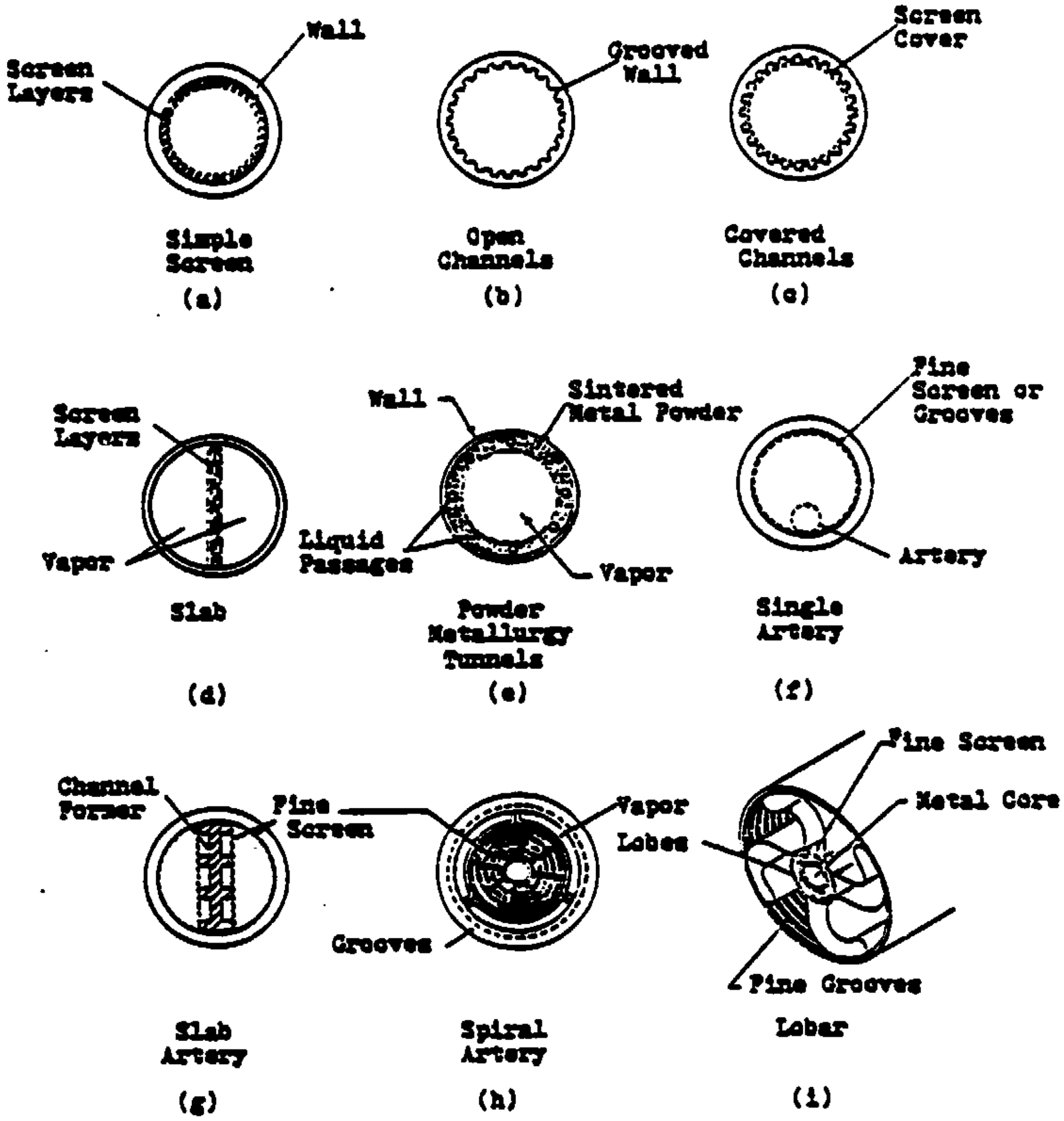


Figure 3.2 Representative Wick Geometries

to the evaporator. This passage has a large hydraulic radius and provides a very low drag path. In the evaporator and condenser, a thin film of liquid is distributed circumferentially. The distribution wick is often a thin layer of screen or circumferential grooves.

The artery is removed from the evaporator and condenser heat flow paths. The thin films provided by the circumferential wick prevent the development of excessive temperature gradients. Arterial wicks provide very high performance, sometimes even approaching that obtainable with liquid metals in more conventional wicks. Lengths in excess of ten meters have been reported. The primary limitations of arterial wicks lie in their difficulty of fabrication and their consequent lack of reproducible performance. The wick structures are quite difficult to form and to insert into the heat pipe vessel so as to maintain uniform close fit to the wall. There has been repeated difficulty with the priming of arteries, that is, the ability to fill an artery with fluid and keep it filled.

Two methods of priming are in use. Capillary priming, as the name implies, depends on capillary forces to maintain the fluid within the artery. The basic condition for capillary priming is that the largest single pore at the artery surface in the evaporator must provide sufficient capillary pressure to offset all counter forces including accelerations. Consequently, the evaporator ends of the arteries must be closed and there must be no single inadvertently large pore on the entire periphery of the enclosing surface. Due to the adverse effect of accelerations, capillary primed arteries can be more fractious during ground testing than in subsequent zero g operation. Yet ground testing is essential to establish the operability of the heat pipe.

If the artery is so located in the heat pipe temperature gradient that it always is the coldest spot, it will operate at a lower vapor pressure than the balance of the heat pipe. If the magnitude of the vapor pressure difference is sufficient, it will cause priming to take place. This is known as vapor pressure or Clapeyron priming. The process is highly temperature dependent. The pressure difference caused by a given temperature difference varies enormously with temperature. Thus, a heat pipe which primes reliably and quickly at high temperature (i.e. high pressure) may fail to prime at all at low temperature. It has also been reported that vibration has caused arteries to lose their prime and that subsequent re-priming can be unreliable.

In spite of their apparent drawbacks, the performance of arterial heat pipes is sufficiently high to justify further work to improve their reliability and reproducibility. In general, arterial wicks require less total mass of wicking material, and may also require less fluid inventory than conventional heat pipes. They are, therefore, serious candidates for use in space radiators.

3.3.2 Wickless (Configuration Pumped) Heat Pipes

A crevice has capillary properties. Therefore, if the wall of a non-round heat pipe is formed so as to produce longitudinal crevices, these may serve the purpose of wicks. That is, the configuration of the wall provides the capillary pumping force. Several potential configuration pumped heat pipe geometries are shown in Figure 3.3. Configuration pumped heat pipes have been built (Figure 3.4) and have

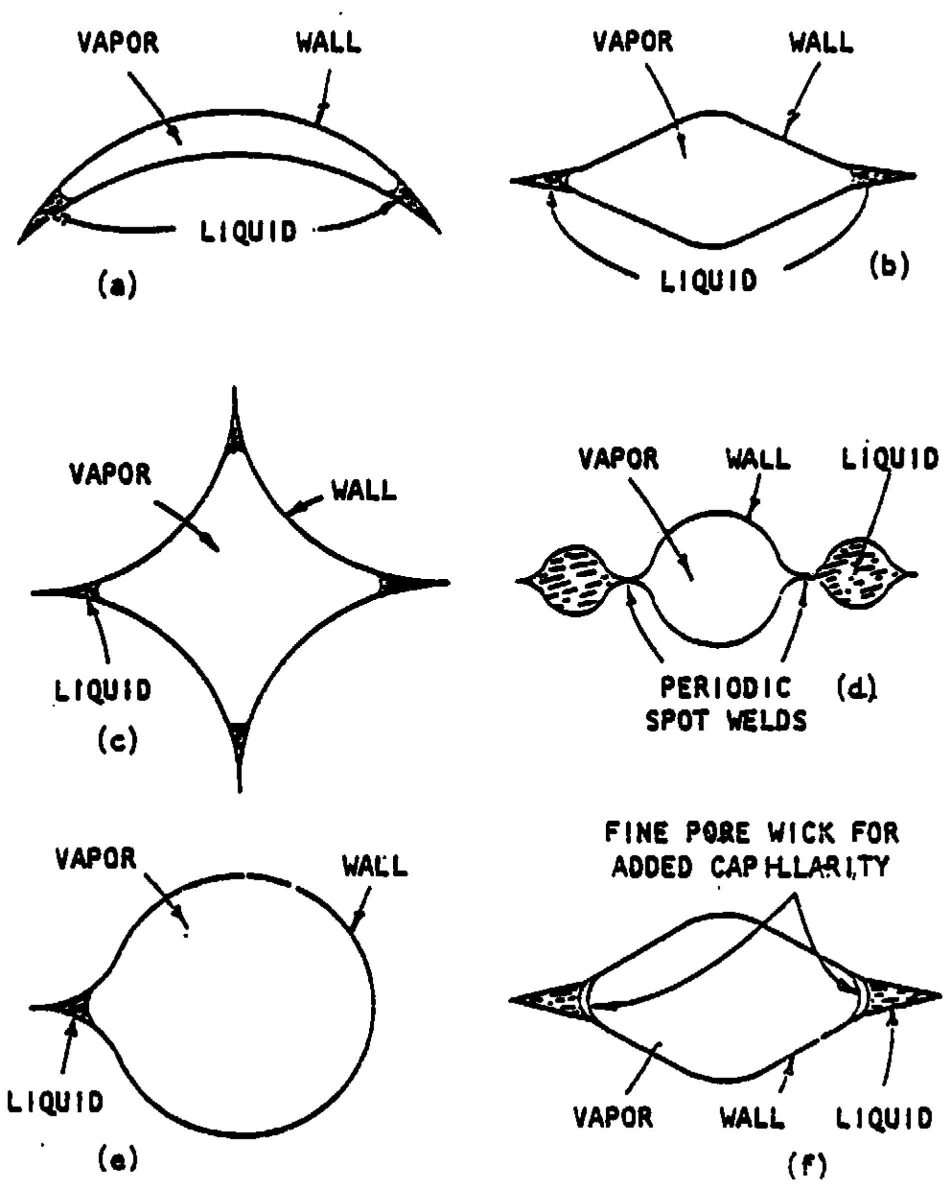


Figure 3.3 Configuration Pumped Geometries

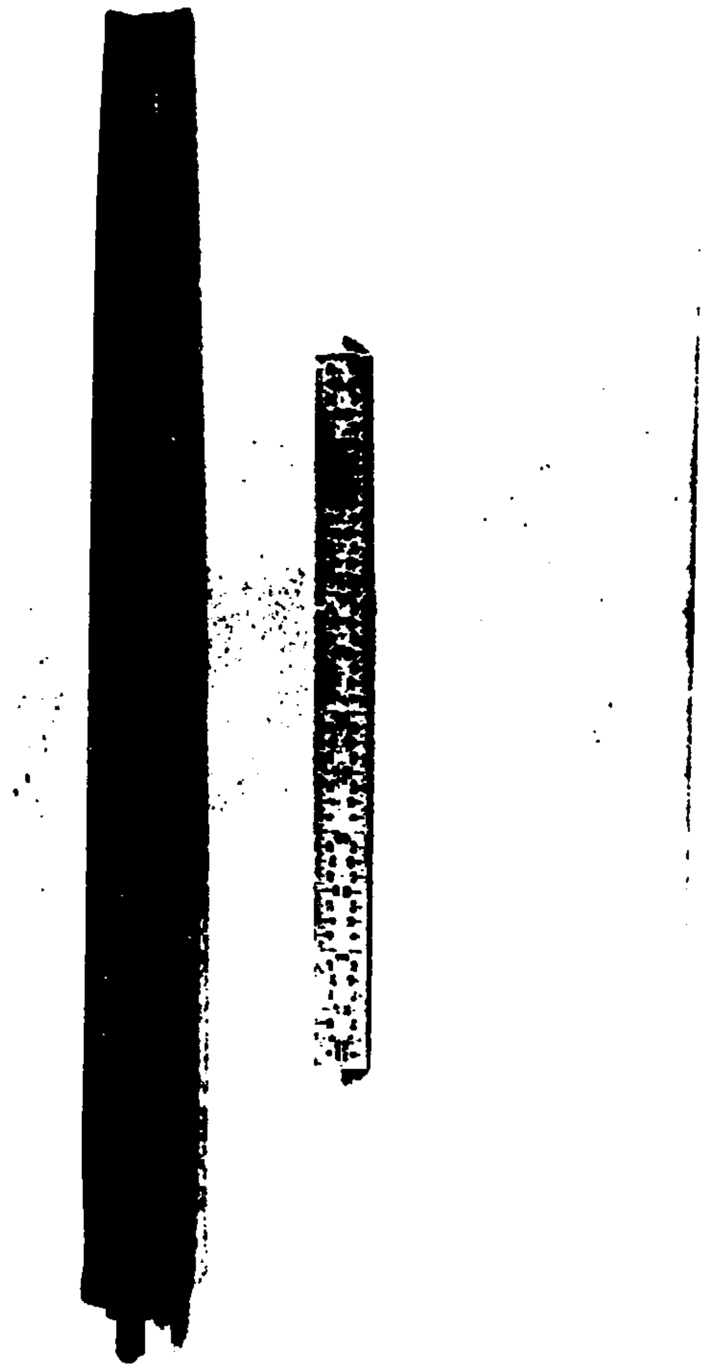


Figure 3.4

Photograph of a Configuration Pumped Heat Pipe been shown to operate. However, there has been very little work in the field, and the mathematical prediction of performance is incomplete.

The driving pressure difference which causes liquid flow in a heat pipe is determined by the surface tension and the difference in the radius of the liquid meniscus in the condenser and evaporator. Evaporation in the heat input section tends to depress the liquid level while condensation at the heat output end tends to increase the level. Thus, during operation, the liquid level in the evaporator of a configuration pumped heat pipe recedes into the crevice, increasing the pumping pressure but decreasing the flow area. The inverse occurs in the condenser. This makes for a delicate tradeoff of liquid fill versus power handling capability. The problem is somewhat alleviated in the configuration/artery geometry of Figure 3.3d and 3.3f.

Configuration pumped heat pipes tend by their nature to have relatively low capillary pumping forces and low liquid drag. They therefore lend themselves well to consideration as elements in low temperature space radiators where large radiating areas require long heat pipes. The liquid inventory requirementof configuration pumped heat pipes appears to be comparable to that of the arterial structures discussed previously. The complete absence of conventional wicks is a substantial mass reduction. However, the non-round shapes are relatively poor pressure vessels so that the gain in mass due to elimination of the wick may be at least partially offset by a thicker wall requirement unless fluid vapor pressures are kept relatively low. Thus the operating temperature range for a configuration pumped heat pipe of low mass may be narrower than that for other geometries.

The ability of configuration pumped heat pipes to hold their shape is a function of the creep strength of the heat pipe envelope. Thermacore[12] previously identified the iron alloy, A-286, which exhibits an exceptionally high creep strength, and may well serve as a containment for configuration pumped heat pipes. (A-286 has a 0.1% creep at $1100^{\circ}F$ in 10^{5} hours under a 38,000 psi stress load).

3.3.3 Hybrid Wick/Pumped Heat Pipes

Since the dissipating capacity of a space radiator declines as the fourth power of any temperature loss, there is a strong incentive to minimize losses. One of the principal advantages of the heat pipe is the low temperature loss it incurs while moving large amounts of heat. This low ΔT operation is characteristic of vapor heat transfer. There may, therefore, be reason to make use of vapor heat transfer even at power levels which cannot be sustained by capillary pumping alone. Alternative or hybrid pumping means are possible and deserve consideration. This may be true not only for the radiators themselves, but also for the primary loops feeding them. A practical hybrid system may use an alternative pumping means for liquid transport over appreciable distances with capillary pumping for local distribution and collection.

The heat transfer capability of a conventional heat pipe can be limited by entrainment of liquid from the walls by the high velocity, counterflowing vapor. Separation of the liquid and vapor passages will permit greater heat flow under these conditions. Figure 3.5 is a hybrid system where the liquid and vapor flow are in the same direction. Therefore, the vapor shear forces may aid rather than inhibit liquid flow.

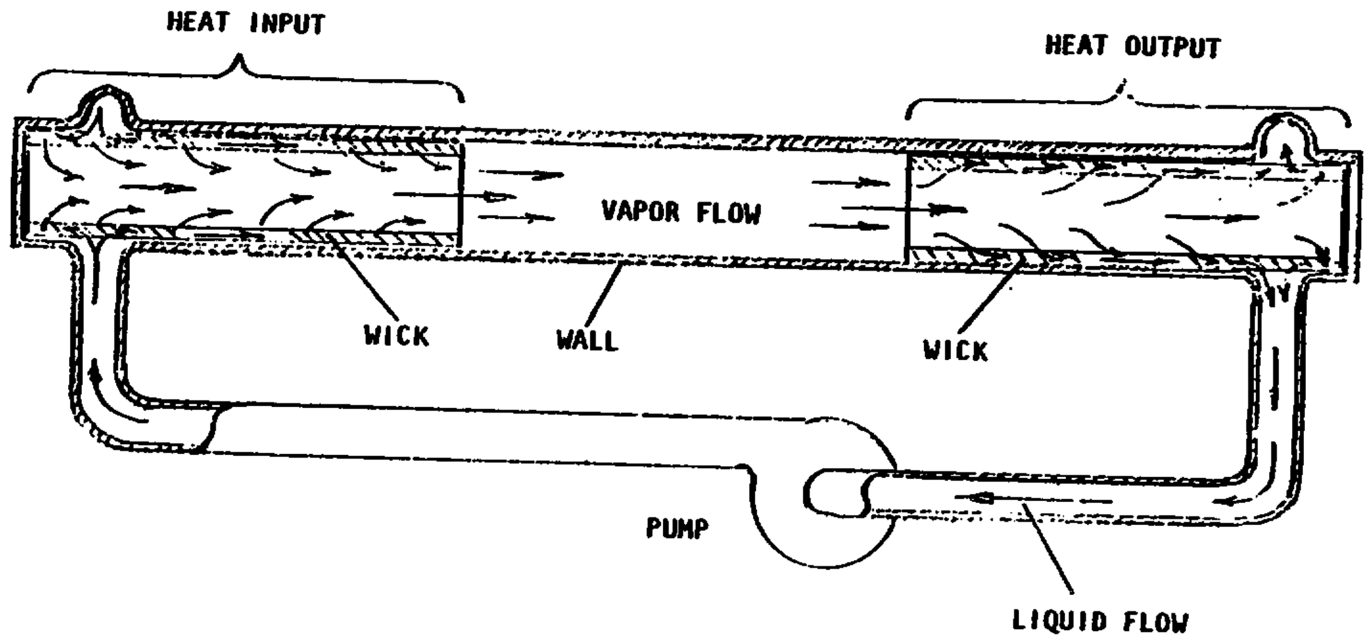


Figure 3.5 Mechanically Pumped Hybrid Heat Pipe

Hybrid heat pipes are directly analagous to two-pipe steam heating systems for buildings which use condensate pumps for liquid return. The principle has been extended to liquid metals by Philips Laboratories for use in Stirling engines.

The main disadvantages of the hybrid system are the increased probability of a leak at pump seals and joints and the dependence of operation on an external power source. For maximum redundancy, there should be a pump for each heat pipe, a serious penalty in complexity for a space radiator, making the approach seem more applicable to primary loops.

It may be possible to make use of the "heat of the radiator" to pump the liquid, much the same way that a capillary pump makes use of the "heat of the radiator."

Thermacore has recently begun the exploration of a "liquid piston pump" as part of its internal R & D effort. This pump uses a localized high heat flux, into the fluid, to develop a vapor bubble of sufficient pressure to push the liquid forward. Backward flow is prevented by the use of a check valve. A forward spring loaded valve permits regulation of the pressure at which the pump is activated.

Initial work to date has concentrated on gravity feed liquid systems with encouraging results. The extension of this concept to two phase systems with freedom from gravity will pose challenging work but may be worth a cursory investigation.

3.3.4 Other Concepts

There are numerous concepts which have been suggested as possible fluid pumping mechanisms for heat pipes and includes electro-magnetic, electrolytic, electrohydrodynamic and electrophoretic pumping. All of these are not suited for individual spacecraft radiator heat pipes. However, osmotic pumped heat pipes and artificial gravity are two possible mechanisms which are suited for spacecraft use.

If a spinning spacecraft can be so arranged that its centrifugal force will aid liquid return in heat pipes, it may be possible to eliminate pumping and depend entirely or predominantly on artificial gravity for this function. The result may be mass reduction (by wick elimination and, possibly, reduced fluid inventory) and an added degree of freedom in fluid selection (fluid need not have high surface tension).

Osmotic pressures can exceed capillary pressures by a factor of 100 to 1,000. An osmotically pumped heat pipe is feasible in principle. Several designs have been proposed, but only one hardware program has been reported. The proposed designs all make use of gravity in one way or another: to keep liquid in place, to redistribute salt by natural convection, etc. It may be possible to devise a geometry which will function in gravity-free space. If so, osmotic heat pipes may avoid entirely the capillary limitations on available pumping pressure.

Flow rates through semi-permeable membranes are low; i.e., large areas are required to permit useful heat flow. There is, however, an interesting factor which may favor further consideration for low temperature space radiators. These radiators also require large areas becuase of the low radiant power densities. The osmotic process is such that the membrane must be located at the condenser (heat dissipating) end of the system, which is the radiating surface of a radiator. At temperatures below about 900°K, the power density from a black body radiator is less than the power density sustainable by flow of the best fluids (e.g. water) through membranes. That is, below this temperature the unit liquid flow rate through a membrane is more than sufficient to support the unit radiant heat load from a radiator of equal area, and a basic condition of successful operation has been satisfied.

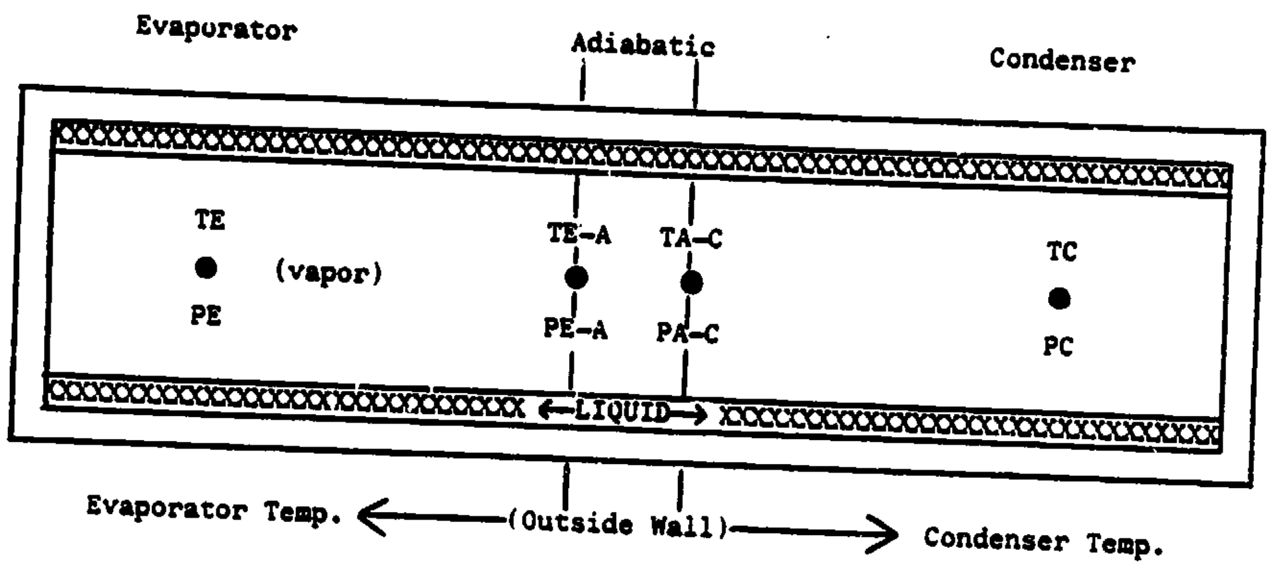
The geometries considered to date are relatively massive, having two walls and a large liquid inventory. Membranes do not exist for operation above about 400°K. However, since an osmotic heat pipe would need no auxiliary power (comparable to a capillary heat pipe), it deserves further consideration.

REFERENCES

1. Kinslow, Ray, High Velocity Impact Phenomena, Academic Press, NY, NY.
2. Liebelein, S. and Diedrich, J. H., Nuclear Applications, Volume 3, February 1967.
3. Dauterman, W. H. and Montgomery, L. D., Design and Analysis of the Radiator Structure for Space Power Systems, Atomics International Division, Rockwell International, June 1973, AI-AEC-13093/NASA-CR-121222.
4. Werner, Richard W. and Carlson, G. A., Heat Pipe Radiator for a 50MWt Space Power Plant, University of California, UCRL-50294, June 1967.
5. Private Communications, Kinslow, Ray, Tennessee Technological University, Cookeville, Tennessee.
6. Private Communications, Gehring, John W. Jr., Past Director, Impact Physics Laboratory, General Motors, Corporation.
7. Private Communications, Wenzel, Alex, Director, Department of Ballistics and Explosive Science, Southwest Research Institute, San Antonio, Texas.
8. Private Communications, Swift, Hal F., University of Dayton, Dayton, Ohio.
9. Hickenson, Norris L., Tennessee Technological University, Cookeville, Tennessee. (Section of Reference 1).
10. Emannelsen, R. C., "Determination of the Emissivity of Materials," NASA CR-54444, NASA 1965.
11. Sparrow, E. M. and Cess, R. D., Radiation Heat Transfer, Hemisphere Publishing Corporation.
12. Garrett AiResearch, Study of a Space Nuclear Power System for a Nuclear Electric Spacecraft; JPL Contract 955008.
13. Thermacore, Inc., Letter Progress Report #3, October, 1978, Heat Pipe Heat Rejection System and Demonstration Model for the Nuclear Electric Propulsion (NEP) Spacecraft; JPL Contract 955100.

APPENDIX 1

This appendix has complete performance printouts of all the heat pipes tabulated in Section 3.1 and 3.2. The heat pipe program used is Thermacore's GROOVE27. Figure A.1 depicts the placement and definitions of many of the symbols in the printout.



DPVE = Pressure drop in vapor in evaporator

DPLEG = Pressure drop in liquid in evaporator grooves

DPUA = Pressure drop in vapor in adiabatic

DPLAG = Pressure drop in liquid in adiabatic grooves

DPVC = Pressure drop in vapor in condenser - (+) means drop, (-) means recovery or increase

DPLGG = Pressure drop in liquid in condenser grooves

RUN CONDITIONS: 3:59 P.M. 4/ 5/79

FLUID = RUBIDIUM WALL MATL=304SS
EVAP TEMP = 434 VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00 WTG ANG = 0.00 DEG

EVAP LENGTH	16.9291	IN	43.0000	CM
ADB LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM

O.D.	1.0000	IN	2.5400	CM
WALL THKNSS	0.0300	IN	0.0762	CM
GROOVE WIDTH	0.0787	IN	0.2000	CM
GROOVE HEIGHT	0.0197	IN	0.0500	CM
LAND WIDTH	0.0344	IN	0.0875	CM

25 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ------------ 720 WATTS

------------ TOTAL DELTA-T = 2.56 DEG C
------------ TOTAL MASS = 1.749 KG

WANT PERFORMANCE DETAILS (Y OR N) ??

PE	PE-A	PA-C	PC	DYNES/CM2
31291.9	30261.1	29854.9	30641.5	

TE	TE-A	TA-C	TC	DEG C
432.253	431.046	430.502	431.72	

EVAP TEMP	COND TEMP	DELTA-T
434	431.439	2.56104

DPC= 18214 DPG= 0 DPC+DPG= 18214 DYNES/CM2

DPVE	DPLED	DPVA	DPLAD
1030	180	305	1115
DPVC	DPLCD		
-687	739		

SONIC LIMITS: EVAP= 2197 ADB= 2487 WATTS

Q/A'S=	EVAP	COND	AXIAL	WATTS/CM2
	2	0	142	

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
21	3244	109	3247	5

HOT FLUID CHARGE 129.61 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 84.6019 CM3

COLD FLUID CHARGE 151.636 GRAMS
98.9783 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 1696.9 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.326631	.1200064E-01	.6090364E-02	.300293	DEG C

VAPOR (E)	VAPOR (A)	VAPOR (C)
1.30042	.548806	-1.21729

CONDENSATION	COND MESH	COND LAG	COND WALL	
.073367	.1478646E-02	.2921012E-02	.20283	DEG C

POWER OF 1778 WATTS CAUSES ------------ ADB SONIC LIMIT

LAST NON-LIMITED POWER CALCULATION WAS AT ------------ 1750 WATTS

------------ TOTAL DELTA-T = 7.16 DEG C
------------ TOTAL MASS = 1.749 KG

```
RUN CONDITIONS:                                          4:38 P.M.    ?/22/79

FLUID = RUBIDIUM          WALL MATL=304SS
EVAP TEMP =  377          VAPOR DELTA-T =  50  DEG C
GRAV ANG =   0.00         WMB ANG =   0.00  DEG

EVAP  LENGTH    16.9291 IN    43.0000 CM
ADB   LENGTH    16.9291 IN    43.0000 CM
COND  LENGTH    69.2913 IN   176.0000 CM
TOTAL LENGTH   103.1500 IN   262.0000 CM

O.D.             1.0000 IN     2.5400 CM
WALL THICKS      0.0300 IN     0.0762 CM
GROOVE WIDTH     0.0787 IN     0.2000 CM
GROOVE HEIGHT    0.0197 IN     0.0500 CM
LAND WIDTH       0.0344 IN     0.0875 CM
 25    GROOVES (CLOSED) COVERED WITH 200 MESH


NO LIMIT ENCOUNTERED AT ----------        514 WATTS

---------- TOTAL DELTA-T =    6.44  DEG C
---------- TOTAL MASS    =    1.749 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE              PE-A            PA-C            PC              DYNES/CM2
 9906.91         8292.16         7774.8          8836.32

TE              TE-A            TA-C            TC              DEG C
 375.772         367.761         364.907         370.858

EVAP TEMP       COND TEMP       DELTA-T
 377             370.557         6.44312

DPO= 19141      DPG= 0        DPO+DPG= 19141  DYNES/CM2

DPVE            DPLEG           DPVA            DPLAG
 1613            132             517             821
DPVC            DPLCG
-1113            943

SONIC LIMITS:          EVAP=     749   ADB=     703   WATTS

Q/A'S=          EVAP            COND            AXIAL           WATTS/CM2
                 1               0               101

E R REY#        E A REY#        LIQ REY#        C A REY#        C R REY#
 16              2443            68              2436            3

HOT FLUID CHARGE                                132.019   GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE           86.1741   CM3

COLD FLUID CHARGE  151.636  GRAMS
                    98.9783  CM3

HEAT PIPE. (MESH) + 2 ENDCAPS  1806.8   GRAMS

DELTA-T VALUES:

EVAP WALL       EVAP L+G        EVAP MESH       EVAPORATION
 .516004         .712316E-02     .369768E-02     .000686         DEG C

VAPOR (E)       VAPOR (A)       VAPOR (C)
 8.01123         2.85362        -6.8602

CONDENSATION    COND MESH       COND L+G        COND WALL
 .146734         .868813E-03     .170840E-02     .181184         DEG C

POWER OF     608 WATTS CAUSED ---------- ADB  SONIC LIMIT

LAST NON-LIMITED POWER CALCULATION WAS AT ----------       608 WATTS

---------- TOTAL DELTA-T =    7.34  DEG C
---------- TOTAL MASS    =    1.749 KG
```

RUN CONDITIONS: 4:18 P.M. 3/22/79

FLUID = DOWTHERM A WALL MATL=304SS
EVAP TEMP = 352 VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00 WIG ANG = 0.00 DEG

EVAP LENGTH	16.9291	IN	43.0000	CM
ADB LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM

O.D.	1.0000	IN	2.5400	CM
WALL THKNSS	0.0300	IN	0.0762	CM
GROOVE WIDTH	0.0787	IN	0.2000	CM
GROOVE HEIGHT	0.0256	IN	0.0650	CM
LAND WIDTH	0.0247	IN	0.0627	CM

27 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ———————— 440 WATTS

———————— TOTAL DELTA-T = 3.89 DEG C
———————— TOTAL MASS = 1.744 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE	PE-A	PA-C	PC	DYNES/CM2
.546112E+07	.546109E+07	.546106E+07	.546106E+07	

TE	TE-A	TA-C	TC	DEG C
348.876	348.876	348.876	348.875	

EVAP TEMP	COND TEMP	DELTA-T
352	348.11	3.88989

DPC= 3160 DPG= 0 DPC+DPG= 3160 DYNES/CM2

DPVE	DPLEE	DPVA	DPLAG
19	233	7	1330
DPVC	DPLCG		
8	956		

SONIC LIMITS: EVAP= 156526 ADB= 188977 WATTS

Q/A'S=	EVAP	COND	AXIAL	WATTS/CM2
	1	0	86	

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
5	806	346	806	1

HOT FLUID CHARGE 114.987 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 107.637 CM3

COLD FLUID CHARGE 133.501 GRAMS
128.001 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 1610.42 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.536983	1.34221	.64413	.100098	DEG C

VAPOR (E)	VAPOR (A)	VAPOR (C)
.498281E-03	.498281E-03	.244141E-03

CONDENSATION	COND MESH	COND LAG	COND WALL	
.244587E-01	.157453	.460962	.131568	DEG C

POWER OF 550 WATTS CAUSES ———————— CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ———————— 540 WATTS

———————— TOTAL DELTA-T = 4.79 DEG C
———————— TOTAL MASS = 1.744 KG

RUN CONDITIONS:

8:41 A.M. 3/23/73

FLUID = DOWTHERM A   WALL MATL=304SS
EVAP TEMP = 219   VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00   HTG ANG = 0.00 DEG

EVAP LENGTH	16.9291	IN	43.0000	CM
ADB LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM
O.D.	1.0000	IN	2.5400	CM
WALL THKNSS	0.0300	IN	0.0762	CM
GROOVE WIDTH	0.0787	IN	0.2000	CM
GROOVE HEIGHT	0.0256	IN	0.0650	CM
LAND WIDTH	0.0247	IN	0.0627	CM

27 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ———— 160 WATTS

———— TOTAL DELTA-T = 1.73 DEG C
———— TOTAL MASS = 1.744 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE	PE-A	PA-C	PC	
402817	402802	402462	402432	DYNES/CM2

TE	TE-A	TA-C	TC	
217.618	217.618	217.612	217.609	DEG C

EVAP TEMP	COND TEMP	DELTA-T
219	217.271	1.72906

DPC= 7261   DPG= 0   DPC+DPG= 7261   DYNES/CM2

DPVE	DPLE	DPVA	DPLAG
24	151	21	868
DPVC	DPLCG		
29	619		

SONIC LIMITS: EVAP= 15075   ADB= 17120   WATTS

Q/A'S=	EVAP	COND	AXIAL	
	0	0	33	WATTS/CM2

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
2	328	42	328	0

HOT FLUID CHARGE   112.077 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE   106.784 CM3

COLD FLUID CHARGE 133.871 GRAMS
128.001 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 1610.42 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.228684	.780453	.272907	.100098	DEG C
VAPOR (E)	VAPOR (A)	VAPOR (C)		
.321962E-02	.292969E-02	.292969E-02		
CONDENSATION	COND MESH	COND LAG	COND WALL	
.244567E-01	.666962E-01	.190864	.068916	DEG C

POWER OF 715 WATTS CAUSES ———— CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ———— 710 WATTS

———— TOTAL DELTA-T = 6.87 DEG C
———— TOTAL MASS = 1.744 KG

RUN CONDITIONS: 10:57 A.M. 3/28/79

FLUID = RUBIDIUM WALL MATL=304SS
EVAP TEMP = 434 VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00 WTG ANG = 0.00 DEG

EVAP LENGTH	16.9291	IN	43.0000	CM
ADB LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM

O.D.	1.0000	IN	2.5400	CM
WALL THKNSS	0.0300	IN	0.0762	CM
GROOVE WIDTH	0.1083	IN	0.2750	CM
GROOVE HEIGHT	0.0079	IN	0.0200	CM
LAND WIDTH	0.0079	IN	0.0200	CM

25 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ---------- 720 WATTS

---------- TOTAL DELTA-T = 2.43 DEG C
---------- TOTAL MASS = 1.484 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE	PE-A	PA-C	PC	DYNES/CM2
31296.8	30370.4	30100.3	30716.1	

TE	TE-A	TA-C	TC	DEG C
432.861	431.242	430.762	431.349	

EVAP TEMP	COND TEMP	DELTA-T
434	431.57	2.42993

DPC= 18214 DPG= 0 DPC+DPG= 18214 DYNES/CM2

DPVE	DPLE	DPVA	DPLA
926	1196	269	9336
DPVC	DPLC		
-615	4900		

SONIC LIMITS: EVAP= 2314 ADB= 2631 WATTS

Q/A'S=	EVAP	COND	AXIAL	WATTS/CM2
	2	0	142	

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
21	3161	92	3163	5

HOT FLUID CHARGE 92.1796 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 60.1694 CM3

COLD FLUID CHARGE 107.824 GRAMS
70.3814 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 1375.82 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.644831	.362612E-02	.593373E-02	.300293	DEG C

VAPOR (E)	VAPOR (A)	VAPOR (C)
1.61914	.479736	-1.08667

CONDENSATION	COND MESH	COND LAG	COND WALL	
.073367	.143842E-02	.354732E-03	.202814	DEG C

POWER OF 820 WATTS CAUSES ---------- CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ---------- 815 WATTS

---------- TOTAL DELTA-T = 2.87 DEG C
---------- TOTAL MASS = 1.484 KG

RUN CONDITIONS:

4:24 A.M. 3/30/79

FLUID = RUBIDIUM WALL MATL=304SS
EVAP TEMP = 377 VAPOR DELTA-T = 60 DEG C
GRAV ANG = 0.00 WIG ANG = 0.00 DEG

EVAP LENGTH	16.9291	IN	43.0000	CM
ADB LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM
O.D.	1.0000	IN	2.5400	CM
WALL THKNSS	0.0300	IN	0.0762	CM
GROOVE WIDTH	0.1083	IN	0.2750	CM
GROOVE HEIGHT	0.0079	IN	0.0200	CM
LAND WIDTH	0.0079	IN	0.0200	CM

25 GROOVES (CLOSED) COVERED WITH 200 MESH

REPRODUCIBILITY OF THE ORIGINAL PAGE IS POOR

NO LIMIT ENCOUNTERED AT ---------- 514 WATTS

---------- TOTAL DELTA-T = 5.83 DEG C
---------- TOTAL MASS = 1.484 KG

WANT PERFORMANCE DETAILS (Y OR N) YY

PE	PE-A	PA-C	PC	
9928.78	8499.64	8061.89	9003.7	DYNES/CM2

TE	TE-A	TA-C	TC	
378.877	369.863	366.456	371.446	DEG C

EVAP TEMP	COND TEMP	DELTA-T
377	371.17	5.82983

DPC= 19139 DPG= 0 DPC+DPG= 19139 DYNES/CM2

DPVE	DPLEG	DPVA	DPLAG
1429	877	447	6867
DPVC	DPLCG		
-962	3699		

SONIC LIMITS: EVAP= 790 ADB= 765 WATTS

Q/A'S=	EVAP	COND	AXIAL	
	1	0	101	WATTS/CM2

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
16	2391	58	2392	3

HOT FLUID CHARGE 93.8783 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 61.2782 CM3

COLD FLUID CHARGE 107.824 GRAMS
70.3814 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 1375.82 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.615994	.206109E-02	.350566E-02	.500488	DEG C

VAPOR (E)	VAPOR (A)	VAPOR (C)
7.01489	2.40696	-4.3896

CONDENSATION	COND MESH	COND LAG	COND WALL	
.122278	.642339E-03	.500006E-03	.151117	DEG C

POWER OF 646 WATTS CAUSES ---------- ADB SONIC LIMIT

LAST NON-LIMITED POWER CALCULATION WAS AT ---------- 640 WATTS

---------- TOTAL DELTA-T = 7.13 DEG C
---------- TOTAL MASS = 1.484 KG

RUN CONDITIONS:

12: 2 P.M. 3/28/79

FLUID = DOWTHERM A WALL MATL=304SS
EVAP TEMP = 352 VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00 WTG ANG = 0.00 DEG

EVAP LENGTH	16.9291 IN	43.0000 CM
ADB LENGTH	16.9291 IN	43.0000 CM
COND LENGTH	69.2913 IN	176.0000 CM
TOTAL LENGTH	103.1500 IN	262.0000 CM

O.D.	1.0000 IN	2.5400 CM
WALL THKNSS	0.0300 IN	0.0762 CM
GROOVE WIDTH	0.1083 IN	0.2750 CM
GROOVE HEIGHT	0.0217 IN	0.0550 CM
LAND WIDTH	0.0044 IN	0.0112 CM

28 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ———— 440 WATTS

———— TOTAL DELTA-T = 9.23 DEG C
———— TOTAL MASS = 1.548 KG

WANT PERFORMANCE DETAILS (Y OR N) YY

PE	PE-A	PA-C	PC	
.511061E+07	.511059E+07	.511055E+07	.511054E+07	DYNES/CM2

TE	TE-A	TA-C	TC	
344.591	344.591	344.591	344.59	DEG C

EVAP TEMP	COND TEMP	DELTA-T
352	342.773	9.22729

DPO= 3294 DPG= 0 DPO+DPG= 3294 DYNES/CM2

DPVE	DPLE	DPVA	DPLA
18	219	7	1430
DPVC	DPLC		
7	896		

SONIC LIMITS: EVAP= 150576 ADB= 181298 WATTS

Q/A'S=	EVAP	COND	AXIAL	
	1	0	86	WATTS/CM2

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
5	797	291	797	1

HOT FLUID CHARGE 120.206 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 112.561 CM3

COLD FLUID CHARGE 141.404 GRAMS
132.401 CM3

HEAT PIPE. (MESH) & 2 ENDCAPS 1406.22 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.636083	6.13104	.640465	.100098	DEG C

VAPOR (E)	VAPOR (A)	VAPOR (C)
.244141E-03	.244141E-03	.244141E-03

CONDENSATION	COND MESH	COND LAG	COND WALL	
.244087E-01	.156547	1.80406	.132076	DEG C

POWER OF 575 WATTS CAUSES ———— CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ———— 570 WATTS

———— TOTAL DELTA-T = 11.92 DEG C
———— TOTAL MASS = 1.548 KG

RUN CONDITIONS: 11:50 A.M. 3/29/79

FLUID = DOWTHERM A WALL MATL=304SS
EVAP TEMP = 219 VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00 VTG ANG = 0.00 DEG

REPRODUCIBILITY OF THE ORIGINAL PAGE IS POOR

EVAP LENGTH	16.9291	IN	43.0000	CM
ADB LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM

O.D.	1.0000	IN	2.5400	CM
WALL THKNSS	0.0300	IN	0.0762	CM
GROOVE WIDTH	0.1083	IN	0.2750	CM
GROOVE HEIGHT	0.0197	IN	0.0500	CM
LAND WIDTH	0.0049	IN	0.0125	CM

25 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ———— 169 WATTS

———— TOTAL DELTA-T = 3.55 DEG C
———— TOTAL MASS = 1.535 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE	PE-A	PA-C	PC	DYNES/CM2
387562	387538	387508	387479	

TE	TE-A	TA-C	TC	DEG C
216.152	216.149	216.146	216.145	

EVAP TEMP	COND TEMP	DELTA-T
219	215.446	3.55353

DPO= 7307 DPG= 0 DPO+DPG= 7307 DYNES/CM2

DPVE	DPLEG	DPVA	DPLAG
24	179	20	1200
DPVC	DPLCG		
29	736		

SONIC LIMITS: EVAP= 14943 ADB= 16962 WATTS

Q/A'S=	EVAP	COND	AXIAL	WATTS/CM2
	0	0	33	

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
2	321	36	321	0

HOT FLUID CHARGE 111.829 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 104.709 CM3

COLD FLUID CHARGE 131.542 GRAMS
123.541 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 1402.85 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.228554	2.24923	.269646	.100098	DEG C

VAPOR (E)	VAPOR (A)	VAPOR (C)
.328064E-02	.292969E-02	.292969E-02

CONDENSATION	COND MESH	COND LAG	COND WALL	
.244567E-01	.566972E-01	.550474	.566992E-01	DEG C

POWER OF 569 WATTS CAUSES ———— CAPILLARY LIMIT. DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ———— 568 WATTS

———— TOTAL DELTA-T = 11.48 DEG C
———— TOTAL MASS = 1.535 KG

RUN CONDITIONS:

4: 7 P.M. 4/ 5/79

FLUID = RUBIDIUM WALL MATL=304SS
EVAP TEMP = 434 VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00 WTG ANG = 0.00 DEG

EVAP LENGTH	16.9291	IN	43.0000	CM
ADB LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM

O.D.	1.0000	IN	2.5400	CM
WALL THKNSS	0.0300	IN	0.0762	CM
GROOVE WIDTH	0.1083	IN	0.2750	CM
GROOVE HEIGHT	0.0079	IN	0.0200	CM
LAND WIDTH	0.0079	IN	0.0200	CM

25 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ----------- 720 WATTS

----------- TOTAL DELTA-T = 2.43 DEG C
----------- TOTAL MASS = 1.484 KG

WANT PERFORMANCE DETAILS (Y OR N) YY

PE	PE-A	PA-C	PC	
31296.8	30570.4	30100.3	30715.1	DYNES/CM2
TE	TE-A	TA-C	TC	
432.861	431.242	430.762	431.849	DEG C
EVAP TEMP	COND TEMP	DELTA-T		
434	431.57	2.42993		

DPC= 18214 DPG= 0 DPC+DPG= 18214 DYNES/CM2

DPVE	DPLE	DPVA	DPLAG
926	1196	269	9336
DPVC	DPLCG		
-615	4900		

SONIC LIMITS: EVAP= 2314 ADB= 2631 WATTS

Q/A'S=	EVAP	COND	AXIAL	
	2	0	142	WATTS/CM2

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
21	3161	92	3163	5

HOT FLUID CHARGE 92.1796 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 60.1694 CM3

COLD FLUID CHARGE 107.824 GRAMS
70.3814 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 1375.82 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.329831	.352612E-02	.593373E-02	.300293	DEG C
VAPOR (E)	VAPOR (A)	VAPOR (C)		
1.61914	.479736	-1.08667		
CONDENSATION	COND MESH	COND LAG	COND WALL	
.073367	.143842E-02	.854732E-03	.202811	DEG C

POWER OF 820 WATTS CAUSES ----------- CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ----------- 815 WATTS

----------- TOTAL DELTA-T = 2.37 DEG C
----------- TOTAL MASS = 1.484 KG

RUN CONDITIONS:

11: 3 A.M. 3/23/79


```
FLUID = RUBIDIUM              WALL MATL=304SS
EVAP TEMP =  377              VAPOR DELTA-T =  50  DEG C
GRAV ANG =   0.00             WTD ANG =   0.00  DEG

EVAP  LENGTH     16.9291 IN     43.0000 CM
ADB   LENGTH     16.9291 IN     43.0000 CM
COND  LENGTH     69.2913 IN    176.0000 CM
TOTAL LENGTH    103.1500 IN    262.0000 CM

O.D.              1.0000 IN      2.5400 CM
WALL THKNSS       0.0100 IN      0.0254 CM
GROOVE WIDTH      0.1083 IN      0.2750 CM
GROOVE HEIGHT     0.0079 IN      0.0200 CM
LAND WIDTH        0.0120 IN      0.0328 CM
 26    GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ----------        514 WATTS

---------- TOTAL DELTA-T =    4.56  DEG C
---------- TOTAL MASS    =    0.691 KG

WANT PERFORMANCE DETAILS (Y OR N) YY

PE               PE-A            PA-C            PC
 10019.5          9848.97         8496.41         9239.51        DYNES/CM2

TE               TE-A            TA-C            TC
 376.293          370.666         368.84          372.611        DEG C

EVAP TEMP        COND TEMP       DELTA-T
 377              372.438         4.56226

DPC= 19133       DPG= 0          DPC+DPG= 19133  DYNES/CM2

DPVE             DPLEG           DPVA            DPLAG
 1170             876             383             6860
DPVC             DPLCG
-746              3897

SONIC LIMITS:         EVAP=    867    ADB=    876    WATTS

Q/A'S=           EVAP            COND            AXIAL
                  1               0               101            WATTS/CM2

E R REY#         E A REY#        LIQ REY#        C A REY#        C R REY#
 16               2281            88              2290            3

HOT FLUID CHARGE                              96.36  GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE         62.5663   CM3

COLD FLUID CHARGE   110.102  GRAMS
                     71.8683  CM3

HEAT PIPE, (MESH) & 2 ENDCAPS   580.457  GRAMS

DELTA-T VALUES:

EVAP WALL        EVAP LAG         EVAP MESH        EVAPORATION
 .201125          .207981E-02      .336022E-02      .500438        DEG C

VAPOR (E)        VAPOR (A)        VAPOR (C)
 5.62695          1.82544         -3.77081

CONDENSATION     COND MESH        COND LAG         COND WALL
 .122278          .810208E-03      .501449E-03      .492989E-01    DEG C

POWER OF   710 WATTS CAUSES ----------- ADB  SONIC LIMIT

LAST NON-LIMITED POWER CALCULATION WAS AT -----------   705 WATTS

---------- TOTAL DELTA-T =    6.12  DEG C
---------- TOTAL MASS    =    0.691 KG
```

RUN CONDITIONS: 4:14 P.M. 4/ 5/79

FLUID = DOWTHERM A WALL MATL=304SS
EVAP TEMP = 352 VAPOR DELTA-T = 60 DEG C
GRAV ANG = 0.00 WTG ANG = 0.00 DEG

EVAP LENGTH	16.9291	IN	43.0000	CM
ADB LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM

O.D.	1.0000	IN	2.5400	CM
WALL THKNSS	0.0100	IN	0.0254	CM
GROOVE WIDTH	0.1083	IN	0.2750	CM
GROOVE HEIGHT	0.0217	IN	0.0550	CM
LAND WIDTH	0.0094	IN	0.0240	CM

25 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ----------- 440 WATTS

----------- TOTAL DELTA-T = 5.72 DEG C
----------- TOTAL MASS = 0.777 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE	PE-A	PA-C	PC	DYNES/CM2
.533874E+07	.533872E+07	.533871E+07	.533869E+07	

TE	TE-A	TA-C	TC	DEG C
347.403	347.402	347.402	347.402	

EVAP TEMP	COND TEMP	DELTA-T
352	346.275	5.72485

DPC= 3206 DPG= 0 DPC+DPG= 3206 DYNES/CM2

DPVE	DPLEG	DPVA	DPLAG
13	218	6	1426
DPVC	DPLCG		
4	894		

SONIC LIMITS: EVAP= 170527 ADB= 205679 WATTS

Q/A'S=	EVAP	COND	AXIAL	WATTS/CM2
	1	0	86	

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
5	764	297	764	1

HOT FLUID CHARGE 123.66 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 115.787 CM3

COLD FLUID CHARGE 142.992 GRAMS
133.888 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 634.002 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.176328	3.70969	.611878	.100098	DEG C

VAPOR (E)	VAPOR (A)	VAPOR (C)
.488281E-03	.244141E-03	0

CONDENSATION	COND MESH	COND LAG	COND WALL	
.244557E-01	.149562	.909027	.430151E-01	DEG C

POWER OF 560 WATTS CAUSES ----------- CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ----------- 555 WATTS

----------- TOTAL DELTA-T = 7.19 DEG C
----------- TOTAL MASS = 0.777 KG

RUN CONDITIONS: 3:49 A.M. 3/30/79

FLUID = DOWTHERM A   WALL MATL=304SS
EVAP TEMP = 219   VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00   WTG ANG = 0.00 DEG

EVAP LENGTH	16.9291	IN	43.0000	CM
ADB LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM

O.D.	1.0000	IN	2.5400	CM
WALL THKNSS	0.0100	IN	0.0254	CM
GROOVE WIDTH	0.1083	IN	0.2750	CM
GROOVE HEIGHT	0.0217	IN	0.0550	CM
LAND WIDTH	0.0094	IN	0.0240	CM

25 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ———— 160 WATTS

———— TOTAL DELTA-T = 2.49 DEG C
———— TOTAL MASS = 0.777 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE	PE-A	PA-C	PC	DYNES/CM2
396240	396219	396194	396170	

TE	TE-A	TA-C	TC	DEG C
217.008	217.006	217.003	217.001	

EVAP TEMP	COND TEMP	DELTA-T
219	216.514	2.48647

DPC= 7280   DPG= 0   DPC+DPG= 7280   DYNES/CM2

DPVE	DPLED	DPVA	DPLAG
20	141	17	926
DPVC	DPLCG		
24	560		

SONIC LIMITS: EVAP= 16600   ADB= 16738   WATTS

Q/A'S=	EVAP	COND	AXIAL	WATTS/CM2
	0	0	33	

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
2	308	36	308	0

HOT FLUID CHARGE 121.123 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 113.411 CM3

COLD FLUID CHARGE 142.992 GRAMS
133.886 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 634.002 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.746241E-01	1.55778	.259068	.100098	DEG C

VAPOR (E)	VAPOR (A)	VAPOR (C)
.27771E-02	.244141E-02	.244141E-02

CONDENSATION	COND MESH	COND LAG	COND WALL	
.244557E-01	.633176E-01	.381132	.182687E-01	DEG C

POWER OF 715 WATTS CAUSES ———— CAPILLARY LIMIT, DPL > DPT

LAST NON-LIMITED POWER CALCULATION WAS AT ———— 710 WATTS

———— TOTAL DELTA-T = 10.06 DEG C
———— TOTAL MASS = 0.777 KG

RUN CONDITIONS:

12:56 P.M. 3/27/79

FLUID = MERCURY WALL MATL=304SS
EVAP TEMP = 434 VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00 WTG ANG = 0.00 DEG

EVAP LENGTH	16.9291	IN	43.0000	CM
ADB LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM
O.D.	0.3750	IN	0.9624	CM
WALL THKNSS	0.0039	IN	0.0100	CM
GROOVE WIDTH	0.0787	IN	0.2000	CM
GROOVE HEIGHT	0.0079	IN	0.0200	CM
LAND WIDTH	0.0592	IN	0.1504	CM

8 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ———————— 720 WATTS

———————— TOTAL DELTA-T = 2.69 DEG C
———————— TOTAL MASS = 0.449 KG

WANT PERFORMANCE DETAILS (Y OR N) YY

PE	PE-A	PA-C	PC	
.386858E+07	.386703E+07	.386601E+07	.38667E+07	DYNES/CM2

TE	TE-A	TA-C	TC	
431.868	431.839	431.82	431.833	DEG C

EVAP TEMP	COND TEMP	DELTA-T
434	431.311	2.68872

DPC= 114106 DPG= 0 DPC+DPG= 114106 DYNES/CM2

DPVE	DPLEB	DPVA	DPLAG
1547	6864	1007	51200
DPVC	DPLCG		
-682	28087		

SONIC LIMITS: EVAP= 20097 ADB= 24059 WATTS

Q/A'S=	EVAP	COND	AXIAL	
	8	1	1010	WATTS/CM2

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
13	5074	330	5074	3

HOT FLUID CHARGE 281.891 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 20.8099 CM3

COLD FLUID CHARGE 290.48 GRAMS
21.444 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 158.595 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.284282	.732296	1.01444	.100098	DEG C

VAPOR (E)	VAPOR (A)	VAPOR (C)
.290827E-01	.192871E-01	-.129395E-01

CONDENSATION	COND MESH	COND LAG	COND WALL	
.244567E-01	.248109	.179159	.695814E-01	DEG C

POWER OF 935 WATTS CAUSES ———————— CAPILLARY LIMIT, DPL > DPT

LAST NON-LIMITED POWER CALCULATION WAS AT ———————— 930 WATTS

———————— TOTAL DELTA-T = 3.44 DEG C
———————— TOTAL MASS = 0.449 KG

```
RUN CONDITIONS:
                                                          10:41 A.M.    3/30/79

FLUID = MERCURY                WALL MATL=304SS
EVAP TEMP =  352               VAPOR DELTA-T =  50  DEG C
GRAV ANG =   0.00              HTG ANG =   0.00  DEG

EVAP  LENGTH    16.9291 IN    43.0000 CM
ADB   LENGTH    16.9291 IN    43.0000 CM
COND  LENGTH    69.2913 IN   176.0000 CM
TOTAL LENGTH   103.1500 IN   262.0000 CM

O.D.             0.3780 IN     0.9602 CM
WALL THICK       0.0039 IN     0.0100 CM
GROOVE WIDTH     0.0787 IN     0.2000 CM
GROOVE HEIGHT    0.0079 IN     0.0200 CM
LAND WIDTH       0.0592 IN     0.1500 CM
 8 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ----------       440 WATTS

---------- TOTAL DELTA-T =    1.98  DEG C
---------- TOTAL MASS    =    0.449 KG

WANT PERFORMANCE DETAILS (Y OR N) YY

PE              PE-A             PA-C             PC
 .116908E+07     .116769E+07      .116658E+07      .116651E+07      DYNES/CM2
TE              TE-A             TA-C             TC
 380.596         380.457          380.356          380.349          DEG C
EVAP TEMP       COND TEMP        DELTA-T
 382             380.023          1.97729

DPC= 120072     DPG= 0      DPC+DPG= 120072      DYNES/CM2

DPVE            DPLE             DPVA             DPLAG
 1926            4347             1313             38473
DPVC            DPLCG
-434             17796

SONIC LIMITS:          EVAP=   6806   ADB=   7012   WATTS

Q/A'S=          EVAP             COND             AXIAL
                 3                0                617               WATTS/CM2

E R REY#        E A REY#         LIQ REY#         C A REY#         C R REY#
 9               3479             187              3480             2
HOT FLUID CHARGE
ROOM TEMP. VOLUME OF HOT FLUID CHARGE        282.835   GRAMS
                                             20.6796   CM3
COLD FLUID CHARGE   290.5     GRAMS
                    21.4404   CM3

HEAT PIPE, (MESH) & 2 ENDCAPS  168.621  GRAMS

DELTA-T VALUES:

EVAP WALL       EVAP LIQ         EVAP MESH        EVAPORATION
 .184164         .472935          .66449           .100098          DEG C
VAPOR (E)       VAPOR (A)        VAPOR (C)
 .150635         .102539         -.336914E-01

CONDENSATION    COND MESH        COND LIQ         COND WALL
 .244557E-01     .160069          .116693          .460586E-01      DEG C

POWER OF    905 WATTS CAUSES ---------- CAPILLARY LIMIT, DPL > DPV
LAST NON-LIMITED POWER CALCULATION WAS AT ----------      900 WATTS
---------- TOTAL DELTA-T =    3.83  DEG C
---------- TOTAL MASS    =    0.449 KG
```

RUN CONDITIONS:

1:13 P.M. 3/27/79

```
FLUID = MERCURY           WALL MATL=304SS
EVAP TEMP =  277          VAPOR DELTA-T =  80  DEG C
GRAV ANG =   0.00         [illegible] ANG =   0.00  DEG

EVAP  LENGTH    16.[illegible] IN     42.0000 CM
ADB   LENGTH    16.[illegible] IN     42.0000 CM
COND  LENGTH    [illegible] IN    176.0000 CM
TOTAL LENGTH   10[illegible] IN    2[illegible].0000 CM

O.D.            0.[illegible] IN     0.[illegible] CM
WALL THICKS     0.[illegible] IN     0.0100 CM
GROOVE WIDTH    0.[illegible] IN     0.2000 CM
GROOVE HEIGHT   0.[illegible] IN     0.0[illegible] CM
LAND WIDTH      0.[illegible] IN     0.1[illegible] CM
 [illegible] GROOVES (CLOSED) COVERED WITH 200 MESH

[illegible] LIMIT ENCOUNTERED AT ----------    2[illegible] WATTS

---------- TOTAL DELTA-T =    2.0[illegible]  DEG C
---------- TOTAL MASS    =    0.[illegible] [illegible]

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE             PE-A           PA-C           PC
 [illegible]     [illegible]     [illegible]     [illegible]      [illegible]/CM2

TE             TE-A           TA-C           TC
 27[illegible]     27[illegible]     27[illegible]     27[illegible]      DEG C

EVAP TEMP      COND TEMP      DELTA-T
 277            27[illegible]     2.0[illegible]

DPC= 127000    D[illegible]= 0    DPC+D[illegible]= 127000    [illegible]/CM2

DPVE           DPLE           DPVA           DPLA[illegible]
 2[illegible]        2[illegible]         1[illegible]          20[illegible]
DPVC           DPLC
 [illegible]        11077

SONIC LIMITS:        EVAP= 1[illegible]   ADB= 1[illegible]   WATTS

Q/A'S=         EVAP           COND           AXIAL
               2              0              370            WATTS/CM2

[illegible] REY#    E A REY#       LIQ REY#       C A REY#       C R REY#
 6              2436           106            2041           1

HOT FLUID CHARGE                               284.166    GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE          20.9779    CM3

COLD FLUID CHARGE  290.48    GRAMS
                   21.464    CM3

HEAT PIPE, (MESH) & 2 ENDCAPS  106.596  GRAMS

DELTA-T VALUES:

EVAP WALL      EVAP LAG       EVAP MESH      EVAPORATION
 .116944        .300567        .416008        .100098        DEG C

VAPOR (E)      VAPOR (A)      VAPOR (C)
 .523926        .373779        .185547E-01

CONDENSATION   COND MESH      COND LAG       COND WALL
 .244657E-01    .101781        .736424E-01    .286171E-01    DEG C

POWER OF    310 WATTS CAUSES ---------- CAPILLARY LIMIT, DPL > DPT

LAST NON-LIMITED POWER CALCULATION WAS AT ----------    305 WATTS

---------- TOTAL DELTA-T =    8.33  DEG C
---------- TOTAL MASS    =    0.449 KG
```

REPRODUCIBILITY OF THE ORIGINAL PAGE IS POOR

```
RUN CONDITIONS:
                                                    2:07 P.M.    3/27/75

FLUID = DOWTHERM A          WALL MATL=304SS
EVAP TEMP =  342            VAPOR DELTA-T =  50  DEG C
GRAV ANG =   0.00           TIS ANG =   0.30  DEG

EVAP  LENGTH     16.9291 IN     43.0000 CM
ABS   LENGTH     16.9291 IN     43.0000 CM
COND  LENGTH     69.2913 IN    176.0000 CM
TOTAL LENGTH    103.1500 IN    262.0000 CM

O.D.              0.5000 IN      1.2700 CM
WALL THICKS       0.0050 IN      0.0127 CM
GROOVE WIDTH      0.1085 IN      0.2730 CM
GROOVE HEIGHT     0.0295 IN      0.0750 CM
LAND WIDTH        0.0046 IN      0.0116 CM
 12    GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ------------      440 WATTS

------------ TOTAL DELTA-T =    15.04  DEG C
------------ TOTAL MASS    =    0.306 KG

WANT PERFORMANCE DETAILS (Y OR N) YY

PE               PE-A             PA-C             PC
 .4746143E+07     .4747848E+07     .4747608E+07     .4747683E+07      DYNES/CM2

TE               TE-A             TA-C             TC
 330.956          330.931          330.930          330.928           DEG C

EVAP TEMP        COND TEMP        DELTA-T
 342              326.960          15.0408

DPC= 3430        DPG= 0           DPC+DPG= 3430    DYNES/CM2

DPVE             DPLEE            DPVA             DPLAS
 316              280              146              1330
DPVC             DPLCS
 60               905

SONIC LIMITS:             EVAP= 32604    ABS= 30214    WATTS

Q/A'S=           EVAP             COND             AXIAL              WATTS/CM2
                  2                0               347

E R REY#         E A REY#         LIQ REY#         C A REY#           C R REY#
 5                1656             862              1656                1

HOT FLUID CHARGE
ROOM TEMP. VOLUME OF HOT FLUID CHARGE          66.0229    GRAMS
                                               61.6192    CM3

COLD FLUID CHARGE   64.3628   GRAMS
                    60.0641   CM3

HEAT PIPE. (MESH) & 2 ENDCAPS   219.701   GRAMS

DELTA-T VALUES:

EVAP WALL        EVAP LAG         EVAP MESH        EVAPORATION
 .178328          10.482           1.33081          .100098            DEG C

VAPOR (E)        VAPOR (A)        VAPOR (C)
 .4394632E-02     .2127272E-02     .7324222E-03

CONDENSATION     COND MESH        COND LAG         COND WALL
 .2445671E-01     .326822          2.67382          .4331232E-01       DEG C

POWER OF    515 WATTS CAUSES ------------ CAPILLARY LIMIT, DPL > DPV
LAST NON-LIMITED POWER CALCULATION WAS AT ------------     510 WATTS
------------ TOTAL DELTA-T =    17.42  DEG C
------------ TOTAL MASS    =    0.306 KG
```

RUN CONDITIONS: 2:43 P.M.   3/27/79

FLUID = DOWTHERM A   WALL MATL=304SS
EVAP TEMP = 327   VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00   WTH ANG = 0.00 DEG

EVAP LENGTH	16.9291	IN	43.0000	CM
ADB LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM

O.D.	0.5000	IN	1.2700	CM
WALL THICKNESS	0.0050	IN	0.0127	CM
GROOVE WIDTH	0.1095	IN	0.2780	CM
GROOVE HEIGHT	0.0268	IN	0.0680	CM
LAND WIDTH	0.0066	IN	0.0168	CM

12 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ―――――― 373 WATTS

―――――― TOTAL DELTA-T = 10.73 DEG C
―――――― TOTAL MASS = 0.294 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE	PE-A	PA-C	PC	DYNES/CM2
.3320081E+07	.3320068E+07	.3320047E+07	.3320037E+07	

TE	TE-A	TA-C	TC	DEG C
316.304	316.300	316.288	316.284	

EVAP TEMP	COND TEMP	DELTA-T
327	316.27	10.7302

DPC= 4112   DPG= 0   DPC+DPG= 4112   DYNES/CM2

DPVE	DPLME	DPVA	DPLAG
283	274	160	1722
DPVC	DPLCG		
103	1126		

SONIC LIMITS: EVAP= 24980   ADB= 29626   WATTS

Q/A'S=	EVAP	COND	AXIAL	WATTS/CM2
	2	0	294	

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
4	1372	406	1372	1

HOT FLUID CHARGE   60.2462   GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE   56.4103   CM3

COLD FLUID CHARGE   77.4415   GRAMS
                    72.5108   CM3

HEAT PIPE, (MESH) & 2 ENDCAPS   216.768   GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.151407	7.22392	1.13017	.100098	DEG C

VAPOR (E)	VAPOR (A)	VAPOR (C)
.537109E-02	.292969E-02	.195313E-02

CONDENSATION	COND MESH	COND LAG	COND WALL	
.244587E-01	.275341	1.77523	.372096E-01	DEG C

POWER OF 421 WATTS CAUSES ―――――― CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ―――――― 420 WATTS

―――――― TOTAL DELTA-T = 12.08 DEG C
―――――― TOTAL MASS = 0.294 KG

RUN CONDITIONS: 2:31 P.M. 3/27/79

FLUID = DOWTHERM A	WALL MATL=304SS
EVAP TEMP = 302	VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00	TIS ANG = 0.00 DEG

EVAP LENGTH	16.9291 IN	43.0000 CM
ADB LENGTH	16.9291 IN	43.0000 CM
COND LENGTH	69.2913 IN	176.0000 CM
TOTAL LENGTH	103.1500 IN	262.0000 CM
O.D.	0.5000 IN	1.2700 CM
WALL THKNSS	0.0050 IN	0.0127 CM
GROOVE WIDTH	0.1083 IN	0.2750 CM
GROOVE HEIGHT	0.0236 IN	0.0600 CM
LAND WIDTH	0.0076 IN	0.0194 CM

12 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ———— 318 WATTS

———— TOTAL DELTA-T = 8.38 DEG C
———— TOTAL MASS = 0.288 KG

WANT PERFORMANCE DETAILS (Y OR N) YY

PE	PE-A	PA-C	PC	
.222297E+07	.222842E+07	.222821E+07	.222506E+07	DYNES/CM2

TE	TE-A	TA-C	TC	
296.287	296.279	296.274	296.271	DEG C

EVAP TEMP	COND TEMP	DELTA-T
302	293.623	8.37696

DPC= 4834 DPG= 0 DPC+DPG= 4834 DYNES/CM2

DPVE	DPLEB	DPVA	DPLAG
280	308	168	1966
DPVC	DPLCG		
160	1282		

SONIC LIMITS: EVAP= 17710 ADB= 20631 WATTS

Q/A'S=	EVAP	COND	AXIAL	
	1	0	248	WATTS/CM2

E R REY#	E A REY#	L/2 REY#	C A REY#	C R REY#
3	1161	283	1161	0

HOT FLUID CHARGE 57.6716 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 55.3096 CM3

COLD FLUID CHARGE 72.9806 GRAMS
68.3541 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 214.816 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.130299	6.52068	.962160	.100098	DEG C

VAPOR (E)	VAPOR (A)	VAPOR (C)
.756836E-02	.488281E-02	.366211E-02

CONDENSATION	COND MESH	COND LAG	COND WALL	
.244567E-01	.238245	1.35608	.032025	DEG C

POWER OF 378 WATTS CAUSED ———— CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ———— 370 WATTS

———— TOTAL DELTA-T = 9.82 DEG C
———— TOTAL MASS = 0.288 KG

RUN CONDITIONS: 2:24 P.M. 3/27/79

FLUID = DOWTHERM A WALL MATL=304SS
EVAP TEMP = 277 VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00 VTS ANG = 0.00 DEG

EVAP LENGTH	16.9291	IN	43.0000	CM
ADB LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM
O.D.	0.5000	IN	1.2700	CM
WALL THICKS	0.0050	IN	0.0127	CM
GROOVE WIDTH	0.1083	IN	0.2750	CM
GROOVE HEIGHT	0.0217	IN	0.0550	CM
LAND WIDTH	0.0087	IN	0.0220	CM

12 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ———————— 264 WATTS

———————— TOTAL DELTA-T = 6.49 DEG C
———————— TOTAL MASS = 0.281 KG

WANT PERFORMANCE DETAILS (Y OR N) YY

PE	PE-A	PA-C	PC	DYNES/CM2
.130919E+07	.130890E+07	.130867E+07	.130846E+07	

TE	TE-A	TA-C	TC	DEG C
271.81	271.799	271.791	271.784	

EVAP TEMP	COND TEMP	DELTA-T
277	270.511	6.48977

DPC= 5668 DPG= 0 DPC+DPG= 5668 DYNES/CM2

DPVE	DPLEB	DPVA	DPLAS
289	361	195	2296
DPVC	DPLCS		
208	1438		

SONIC LIMITS: EVAP= 11919 ADB= 13775 WATTS

Q/A'S=	EVAP	COND	AXIAL	WATTS/CM2
	1	0	208	

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
3	965	194	965	0

HOT FLUID CHARGE 55.1152 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 51.606 CM3

COLD FLUID CHARGE 68.5201 GRAMS
64.1575 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS, 212.537 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.111323	4.16533	.313467	.100098	DEG C
VAPOR (E)	VAPOR (A)	VAPOR (C)		
.109863E-01	.756836E-02	.732422E-02		
CONDENSATION	COND MESH	COND LAG	COND WALL	
.244507E-01	.19897	1.02143	.273363E-01	DEG C

POWER OF 310 WATTS CAUSES ———————— CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ———————— 305 WATTS

———————— TOTAL DELTA-T = 7.48 DEG C
———————— TOTAL MASS = 0.281 KG

RUN CONDITIONS: 2:12 P.M. 3/27/79

FLUID = DOWTHERM A WALL MATL=304SS
EVAP TEMP = 252 VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00 WTG ANG = 0.00 DEG

EVAP LENGTH	16.9291	IN	43.0000	CM
ADI LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM

O.D.	0.5000	IN	1.2700	CM
WALL THICKSS	0.0050	IN	0.0127	CM
GROOVE WIDTH	0.1083	IN	0.2750	CM
GROOVE HEIGHT	0.0197	IN	0.0500	CM
LAND WIDTH	0.0097	IN	0.0247	CM

12 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ———————— 219 WATTS

———————— TOTAL DELTA-T = 4.98 DEG C
———————— TOTAL MASS = 0.274 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE	PE-A	PA-C	PC	DYNES/CM2
837793	837486	837236	836962	

TE	TE-A	TA-C	TC	DEG C
248.035	248.017	248.004	247.999	

EVAP TEMP	COND TEMP	DELTA-T
252	247.018	4.98198

DPC= 6311 DPG= 0 DPC+DPG= 6311 DYNES/CM2

DPVE	DPLE	DPVA	DPLA
307	413	231	2761
DPVC	DPLC		
253	1693		

SONIC LIMITS: EVAP= 7617 ADI= 6733 WATTS

Q/A'S=	EVAP	COND	AXIAL	WATTS/CM2
	1	0	172	

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
2	806	131	806	0

HOT FLUID CHARGE 52.5281 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 49.1836 CM3

COLD FLUID CHARGE 64.0596 GRAMS
59.9808 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 209.931 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.941753E-01	3.09006	.381114	.100098	DEG C
VAPOR (E)	VAPOR (A)	VAPOR (C)		
.172119E-01	.131836E-01	.149926E-01		
CONDENSATION	COND MESH	COND LAG	COND WALL	
.244887E-01	.166808	.75716	.230996E-01	DEG C

POWER OF 245 WATTS CAUSES ———————— CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ———————— 240 WATTS

———————— TOTAL DELTA-T = 5.45 DEG C
———————— TOTAL MASS = 0.274 KG

```
RUN CONDITIONS:                                    9:36 A.M.     3/30/79

FLUID = MERCURY          WALL MATL=304SS
EVAP TEMP =  434         VAPOR DELTA-T =  50  DEG C
GRAV ANG =   0.00        WTG ANG =   0.00  DEG

EVAP  LENGTH     16.9291 IN     43.0000 CM
ADB   LENGTH     16.9291 IN     43.0000 CM
COND  LENGTH     69.2913 IN    176.0000 CM
TOTAL LENGTH    103.1500 IN    262.0000 CM

O.D.               0.2500 IN     0.6350 CM
WALL THKNSS        0.0025 IN     0.0064 CM
GROOVE WIDTH       0.1083 IN     0.2750 CM
GROOVE HEIGHT      0.0079 IN     0.0200 CM
LAND WIDTH         0.0358 IN     0.0909 CM
 8  GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ------------       720 WATTS

------------ TOTAL DELTA-T =    4.13  DEG C
------------ TOTAL MASS    =    0.290 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE               PE-A             PA-C              PC               DYNES/CM2
 .381508E+07      .380703E+07      .379924E+07       .380308E+07

TE               TE-A             TA-C              TC               DEG C
 430.875          430.712          430.564           430.637

EVAP TEMP        COND TEMP        DELTA-T
 434              429.872          4.12842

DPC= 114169      DPG= 0        DPC+DPG= 114169     DYNES/CM2

DPVE             DPLEG            DPVA              DPLAG
 8650             7271             7706              56096
DPVC             DPLCG
-3944             29763

SONIC LIMITS:          EVAP=   8444   ADB=  10069   WATTS

Q/A'S=           EVAP             COND              AXIAL             WATTS/CM2
                  8                2                 2273

E R REY#         E A REY#         LIQ REY#          C A REY#          C R REY#
 13               7785             393               7787              3

HOT FLUID CHARGE                                  206.068   GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE             15.2117   CM3

COLD FLUID CHARGE   213.036   GRAMS
                     15.7268  CM3

HEAT PIPE, (MESH) & 2 ENDCAPS   76.483   GRAMS

DELTA-T VALUES:

EVAP WALL        EVAP LAG         EVAP MESH         EVAPORATION
 .270618          1.19874          1.56487           .100098           DEG C

VAPOR (E)        VAPOR (A)        VAPOR (C)
 .163818          .147217         -.725098E-01

CONDENSATION     COND MESH        COND LAG          COND WALL
 .244557E-01      .380519          .293489           .663033E-01       DEG C

POWER OF    775 WATTS CAUSES ------------ CAPILLARY LIMIT, DPL > DPT

LAST NON-LIMITED POWER CALCULATION WAS AT ------------     770 WATTS

------------ TOTAL DELTA-T =    4.42  DEG C
------------ TOTAL MASS    =    0.290 KG
```

```
RUN CONDITIONS:
                                                      0:44 A.M.      3/30/79

FLUID = MERCURY                 WALL MATL=304SS
EVAP TEMP =  402                VAPOR DELTA-T =  50  DEG C
GRAV ANG =    0.00              WTG ANG =    0.00  DEG

EVAP  LENGTH     16.9291 IN     43.0000 CM
ADB   LENGTH     16.9291 IN     43.0000 CM
COND  LENGTH     69.2913 IN    176.0000 CM
TOTAL LENGTH    103.1500 IN    262.0000 CM

O.D.              0.2500 IN      0.6350 CM
WALL THKNSS       0.0025 IN      0.0064 CM
GROOVE WIDTH      0.1083 IN      0.2750 CM
GROOVE HEIGHT     0.0079 IN      0.0200 CM
LAND WIDTH        0.0718 IN      0.1825 CM
 4 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ----------         600 WATTS

---------- TOTAL DELTA-T =    3.62  DEG C
---------- TOTAL MASS    =    0.281 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE              PE-A            PA-C            PC
 .239459E+07     .238636E+07     .237868E+07     .238027E+07     DYNES/CM2

TE              TE-A            TA-C            TC
 399.401         399.153         398.924         398.316         DEG C

EVAP TEMP       COND TEMP       DELTA-T
 402             398.379         3.62061

DPC= 116291     DPG= 0          DPC+DPG= 116291    DYNES/CM2

DPVE            DPLEE           DPVA            DPLAG
 9242            7556            8462            89087
DPVC            DPLCC
-3408            31338

SONIC LIMITS:          EVAP=   5424   ADB=   6418   WATTS

Q/A'S=          EVAP            COND            AXIAL           WATTS/CM2
                 6               1               1883

E R REY#        E A REY#        LIQ REY#        C A REY#        C R REY#
 11              6726            307             6729            2

HOT FLUID CHARGE
ROOM TEMP. VOLUME OF HOT FLUID CHARGE    187.61     GRAMS
                                         13.8490    CMS

COLD FLUID CHARGE   193.616  GRAMS
                    14.2868  CMS

HEAT PIPE. (MESH) & 2 ENDCAPS  87.8669  GRAMS

DELTA-T VALUES:

EVAP WALL       EVAP LAG        EVAP MESH       EVAPORATION
 .229854         .960525         1.31773         .100098         DEG C

VAPOR (E)       VAPOR (A)       VAPOR (C)
 .248291         .229004        -.917969E-01

CONDENSATION    COND MESH       COND LAG        COND WALL
 .244587E-01     .322469         .232703         .562994E-01     DEG C

POWER OF    615 WATTS CAUSES ---------- CAPILLARY LIMIT, DPL > DPT

LAST NON-LIMITED POWER CALCULATION WAS AT ----------      610 WATTS

---------- TOTAL DELTA-T =    3.69  DEG C
---------- TOTAL MASS    =    0.281 KG
```

RUN CONDITIONS:

9:52 A.M. 3/30/79

FLUID = MERCURY WALL MATL=304SS
EVAP TEMP = 362 VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00 WTR ANG = 0.00 DEG

EVAP LENGTH	13.9291	IN	43.0000	CM
ADA LENGTH	16.9291	IN	43.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	262.0000	CM
O.D.	0.2500	IN	0.6350	CM
WALL THICKS	0.0025	IN	0.0064	CM
GROOVE WIDTH	0.1083	IN	0.2750	CM
GROOVE HEIGHT	0.0079	IN	0.0200	CM
LAND WIDTH	0.1316	IN	0.3346	CM

3 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ———— 440 WATTS

———— TOTAL DELTA-T = 3.30 DEG C
———— TOTAL MASS = 0.273 KG

WANT PERFORMANCE DETAILS (Y OR N) YY

PE	PE-A	PA-C	PC	
.116262E+07	.115185E+07	.114172E+07	.114423E+07	DYNES/CM2

TE	TE-A	TA-C	TC	
350.04	349.563	349.046	349.179	DEG C

EVAP TEMP	COND TEMP	DELTA-T
352	348.690	3.30103

DPC= 120118 DPG= 0 DPC+DPG= 120118 DYNES/CM2

DPVE	DPLEE	DPVA	DPLAG
10774	7684	10121	80918
DPVC	DPLCG		
-2503	31464		

SONIC LIMITS: EVAP= 2622 ADB= 2821 WATTS

Q/A'S=	EVAP	COND	AXIAL	
	5	1	1389	WATTS/CM2

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
9	5337	372	5346	2

HOT FLUID CHARGE 169.292 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 12.4076 CM3

COLD FLUID CHARGE 173.295 GRAMS
12.8448 CM3

HEAT PIPE. (MESH) & 2 ENDCAPS 99.2507 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.175328	.680505	1.00319	.100098	DEG C
VAPOR (E)	VAPOR (A)	VAPOR (C)		
.457031	.537354	-.133357		
CONDENSATION	COND MESH	COND LAG	COND WALL	
.344557E-01	.218536	.165508	.429384E-01	DEG C

POWER OF 450 WATTS CAUSES ———— CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ———— 440 WATTS

———— TOTAL DELTA-T = 3.34 DEG C
———— TOTAL MASS = 0.273 KG

```
RUN CONDITIONS:
                                                         9:38 A.M.    4/ 5/79

FLUID = MERCURY             WALL MATL=304SS
EVAP TEMP =  302            VAPOR DELTA-T =  50  DEG C
GRAV ANG =   0.00           VTG ANG =    0.00  DEG

EVAP   LENGTH     16.9291 IN     43.0000 CM
ADB    LENGTH     16.9291 IN     43.0000 CM
COND   LENGTH     69.2913 IN    176.0000 CM
TOTAL  LENGTH    103.1500 IN    262.0000 CM

O.D.               0.2500 IN      0.6350 CM
WALL THKNSS        0.0025 IN      0.0064 CM
GROOVE WIDTH       0.1083 IN      0.2750 CM
GROOVE HEIGHT      0.0079 IN      0.0200 CM
LAND WIDTH         0.1318 IN      0.3348 CM
 3  GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT -----------   318 WATTS

----------- TOTAL DELTA-T =     4.81  DEG C
----------- TOTAL MASS    =     0.273 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE            PE-A          PA-C          PC
 446637        432020        418926        419952         DYNES/CM2

TE            TE-A          TA-C          TC
 300.518       299.124       297.747       297.456        DEG C

EVAP TEMP     COND TEMP     DELTA-T
 302           297.493       4.50708

DPC= 124673   DPG= 0        DPC+DPG= 124673     DYNES/CM2

DPVE          DPLEG         DPVA          DPLAG
 13616         5634          13092         43963
DPVC          DPLCG
-1027          23093

SONIC LIMITS:       EVAP=  1062   ADB=  1146   WATTS

Q/A'S=        EVAP          COND          AXIAL              WATTS/CM2
               3             0             994

E R REY#      E A REY#      LIQ REY#      C A REY#      C R REY#
 7             4208          284           4232          1

HOT FLUID CHARGE                              169.39    GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE         12.5417   CM3

COLD FLUID CHARGE  173.996  GRAMS
                   12.8448  CM3

HEAT PIPE, (MESH) & 2 ENDCAPS  99.2807  GRAMS

DELTA-T VALUES:

EVAP WALL     EVAP LAG      EVAP MESH     EVAPORATION
 .130299       .[illegible]08764       .745609       .100098          DEG C

VAPOR (E)     VAPOR (A)     VAPOR (C)
 1.39453       1.37646      -.109375

CONDENSATION  COND MESH     COND LAG      COND WALL
 .244567E-01   .182729       .123971       .319433E-01     DEG C

POWER OF    390 WATTS CAUSES ----------- CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT -----------   388 WATTS

----------- TOTAL DELTA-T =     5.81  DEG C
----------- TOTAL MASS    =     0.273 KG

WANT PERFORMANCE DETAILS (Y OR N) ?N
```

RUN CONDITIONS: 12:29 P.M. 3/30/79

```
FLUID = MERCURY          WALL MATL=304SS
EVAP TEMP =  277         VAPOR DELTA-T =  50  DEG C
GRAV ANG =    0.00       VTG ANG =    0.00  DEG

EVAP  LENGTH    16.9291 IN    43.0000 CM
ADB   LENGTH    16.9291 IN    43.0000 CM
COND  LENGTH    69.2913 IN   176.0000 CM
TOTAL LENGTH   103.1500 IN   262.0000 CM

O.D.             0.2500 IN     0.6350 CM
WALL THKNSS      0.0025 IN     0.0064 CM
GROOVE WIDTH     0.1083 IN     0.2750 CM
GROOVE HEIGHT    0.0079 IN     0.0200 CM
LAND WIDTH       0.1318 IN     0.3348 CM
 3  GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ----------     264 WATTS

---------- TOTAL DELTA-T =    8.36  DEG C
---------- TOTAL MASS    =    0.273 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE          PE-A        PA-C        PC          DYNES/CM2
 256642      240593      224694      224698

TE          TE-A        TA-C        TC          DEG C
 276.719     272.545     268.961     268.962

EVAP TEMP   COND TEMP   DELTA-T
 277         268.646     8.35352

DPO= 127035    DPG= 0     DPC+DPG= 127035    DYNES/CM2

DPVE        DPLEE       DPVA        DPLAG
 16046       4781        15896       37334
DPVC        DPLCO
 -4          19636

SONIC LIMITS:      EVAP=     634   ADB=     630    WATTS

Q/A'S=      EVAP        COND        AXIAL       WATTS/CM2
             3           0           833

E R REY#    E A REY#    LIQ REY#    C A REY#    C R REY#
 6           3737        208         3601        1

HOT FLUID CHARGE                        170.187   GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE   12.5636   CM3

COLD FLUID CHARGE  173.996  GRAMS
                   12.8449  CM3

HEAT PIPE, (MESH) & 2 ENDCAPS  99.2507  GRAMS

DELTA-T VALUES:

EVAP WALL     EVAP LAG      EVAP MESH     EVAPORATION
 .111323       .432317       .637875       .100098       DEG C

VAPOR (E)     VAPOR (A)     VAPOR (C)
 3.17383       3.58326       -.732422E-03

CONDENSATION  COND MESH     COND LAG      COND WALL
 .244557E-01   .156789       .105314       .273751E-01   DEG C

POWER OF   350 WATTS CAUSES ---------- CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ----------   346 WATTS

---------- TOTAL DELTA-T =   12.83  DEG C
---------- TOTAL MASS    =    0.273 KG
```

RUN CONDITIONS: 12:46 P.M. 3/30/72

FLUID = MERCURY WALL MATL=304SS
EVAP TEMP = 434 VAPOR DELTA-T = 60 DEG C
HEAT ANG = 0.00 WTG ANG = 0.00 DEG

EVAP LENGTH	33.0683	IN	84.0000	CM
ADB LENGTH	0.0000	IN	0.0000	CM
COND LENGTH	69.2913	IN	176.0000	CM
TOTAL LENGTH	103.1500	IN	232.0000	CM

O.D.	0.2504	IN	0.6360	CM
WALL THKNSS	0.0025	IN	0.0064	CM
GROOVE WIDTH	0.1083	IN	0.2750	CM
GROOVE HEIGHT	0.0079	IN	0.0200	CM
LAND WIDTH	0.0360	IN	0.0916	CM

6 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ———— 720 WATTS

———— TOTAL DELTA-T = 2.50 DEG C
———— TOTAL MASS = 0.290 KG

WANT PERFORMANCE DETAILS (Y OR N) ?Y

PE	PE-A	PA-C	PC	DYNES/CM2
.389669E+07	.388724E+07	.388721E+07	.389094E+07	

TE	TE-A	TA-C	TC	DEG C
432.39	432.216	432.216	432.285	

EVAP TEMP	COND TEMP	DELTA-T
434	431.498	2.50244

DPC= 114073 DPG= 0 DPC+DPG= 114073 DYNES/CM2

DPVE	DPLEG	DPVA	DPLAG
9361	14632	0	0
DPVC	DPLCG		
-3725	29742		

SONIC LIMITS: EVAP= 8646 ADB= 10334 WATTS

Q/A'S=	EVAP	COND	AXIAL	WATTS/CM2
	4	2	2266	

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
6	7758	394	7759	3

HOT FLUID CHARGE 206.234 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 15.2247 CM3

COLD FLUID CHARGE 213.233 GRAMS
15.7414 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 76.7047 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LEG	EVAP MESH	EVAPORATION	
.135096	.598018	.775944	.100098	DEG C

VAPOR (E)	VAPOR (A)	VAPOR (C)
.174072	.483281E-03	-.693359E-01

CONDENSATION	COND MESH	COND LEG	COND WALL	
.489113E-01	.379483	.292541	.661243E-01	DEG C

POWER OF 1650 WATTS CAUSES ———— CAPILLARY LIMIT, DPL > DPV

LAST NON-LIMITED POWER CALCULATION WAS AT ———— 1625 WATTS

———— TOTAL DELTA-T = 5.47 DEG C
———— TOTAL MASS = 0.290 KG

RUN CONDITIONS: 12:54 P.M. 3/30/79

FLUID = MERCURY WALL MATL=304SS
EVAP TEMP = 277 VAPOR DELTA-T = 50 DEG C
GRAV ANG = 0.00 VTG ANG = 0.00 DEG

EVAP LENGTH 33.8583 IN 86.0000 CM
ADB LENGTH 0.0000 IN 0.0000 CM
COND LENGTH 69.2913 IN 176.0000 CM
TOTAL LENGTH 103.1500 IN 262.0000 CM

O.D. 0.2500 IN 0.6350 CM
WALL THKSS 0.0025 IN 0.0064 CM
GROOVE WIDTH 0.1083 IN 0.2750 CM
GROOVE HEIGHT 0.0079 IN 0.0200 CM
LAND WIDTH 0.1316 IN 0.3344 CM
3 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ———— 264 WATTS

———— TOTAL DELTA-T = 4.79 DEG C
———— TOTAL MASS = 0.273 KG

WANT PERFORMANCE DETAILS (Y OR N) YY

PE	PE-A	PA-C	PC	DYNES/CM2
259831	240821	240817	240617	

TE	TE-A	TA-C	TC	DEG C
276.309	272.592	272.592	272.55	

EVAP TEMP	COND TEMP	DELTA-T
277	272.21	4.78955

DPC= 126975 DPG= 0 DPC+DPG= 126975 DYNES/CM2

DPVE	DPLEG	DPVA	DPLAG
19010	9859	0	0
DPVC	DPLCG		
199	19601		

SONIC LIMITS: EVAP= 641 ADB= 682 WATTS

Q/A'S=	EVAP	COND	AXIAL	WATTS/CM2
	1	0	833	

E R REY#	E A REY#	LIQ REY#	C A REY#	C R REY#
3	3731	208	3766	1

HOT FLUID CHARGE 170.18 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE 12.5631 CM3

COLD FLUID CHARGE 173.995 GRAMS
12.8448 CM3

HEAT PIPE, (MESH) & 2 ENDCAPS 99.2507 GRAMS

DELTA-T VALUES:

EVAP WALL	EVAP LAG	EVAP MESH	EVAPORATION	
.556614E-01	.216152	.318761	.100098	DEG C
VAPOR (E)	VAPOR (A)	VAPOR (C)		
3.7168	.732422E-03	.419922E-01		
CONDENSATION	COND MESH	COND LAG	COND WALL	
.469113E-01	.156322	.106007	.272991E-01	DEG C

POWER OF 500 WATTS CAUSES ———— ADB SONIC LIMIT

LAST NON-LIMITED POWER CALCULATION WAS AT ———— 495 WATTS

———— TOTAL DELTA-T = 9.24 DEG C
———— TOTAL MASS = 0.273 KG

APPENDIX 2

This appendix develops Equation 3.2 which shows how the mass of a radiator heat pipe increases with the performance T of the heat pipe.

T_o = desired heat pipe temperature
ΔT = temperature drop down heat pipe
$T = T_o - \Delta T$, actual heat pipe radiating temperature
A_o = radiating area of heat pipe at T_o
$A = A_o + da$, actual heat pipe radiator area required at T
Q = power to be radiated from heat pipe

$$\frac{da}{dt} = \frac{\text{increase in surface area}}{\text{decrease in temperature}}$$

$$\frac{da}{dt} = \frac{A - A_o}{T_o - T} \qquad \text{Eq. A.1}$$

but $\qquad A = \dfrac{Q}{\varepsilon\sigma(T_o - T)^4} \qquad$ and $\qquad A_o = \dfrac{Q}{\varepsilon\sigma T_o^{\,4}}$

therefore, with substitution into Equation A.1 and proper rearranging,

$$\frac{da}{dt} = \frac{A_o}{\Delta T}[(T_o/T)^4 - 1] \qquad \text{Eq. A.2}$$

Now, since area is a function of length, we have

$$dl = l_c[(T_o/T)^4 - 1] \qquad \text{Eq. A.3}$$

where l_c = condenser $\qquad$ but $\dfrac{dl}{l_t} = \dfrac{dm}{m}$ where l_t = total heat pipe length, m = mass, we obtain with substitution and rearrangement -

$$dm = \frac{m l_c}{l_t}[(T_o/T)^4 - 1] \qquad \text{Eq. A.4}$$

which is Equation 3.2.