METEOROID PROTECTION METHODS
FOR SPACECRAFT RADIATORS
USING HEAT PIPES

FINAL REPORT
CONTRACT
95437
955437

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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
RC A05/MF A01
CSCL 22B
03/18 46410

November 1979
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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
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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. Segment-d heat pipes used as bumpers on top of the radiator heat pipes look quite attractive and need additional evaluation.
6. A total emissivity in 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 27 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, and reports were reviewed along with discussion with eminent professionals in the field. 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, in Kinslow’s book, 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

\[ t = 0.0010144 \, K \left[ \frac{A \times T}{-1nP} \right]^{0.2902} \]  \hspace{1cm} \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 - cm²
- \( T \) = Mission time in hours
- \( P \) = Individual component survival probability

6
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 in terms of the armor properties as:

\[ t = Y_0 a(\xi a)^{-\frac{2}{3}} \left( \frac{v_s}{v_s} \right)^{-2/3} \left[ \frac{T_a}{T_o} \right]^{1/4} K_1 \left[ 0.02902 \right] \]

Where
- \( Y_0 \) = room temperature cratering coefficient
- \( a \) = rear surface damage factor
- \( \xi a \) = density of armor - gm/cc
- \( v_s \) = velocity of sound in armor - cm/sec
- \( T_a \) = temperature of armor °K
- \( T_o \) = room temperature °K
- \( K_1 \) = meteoroid flux constant

The cratering coefficient, \( Y_r \), 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°F, 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

<table>
<thead>
<tr>
<th>Material</th>
<th>Cratering Coefficient $a_T$</th>
<th>Rear Surface Damage Factor &quot;a&quot;</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Dimple</td>
</tr>
<tr>
<td>2024 Al</td>
<td>1.07</td>
<td>2.5</td>
</tr>
<tr>
<td>Lockalloy</td>
<td>2.06</td>
<td>-</td>
</tr>
<tr>
<td>316 SS</td>
<td>2.19</td>
<td>2.4</td>
</tr>
<tr>
<td>A-286</td>
<td>1.77</td>
<td>2.4</td>
</tr>
<tr>
<td>Nb-1%Pr</td>
<td>1.81</td>
<td>4.5</td>
</tr>
</tbody>
</table>

*Estimated Value

Table 1.2
MATERIALS FACTOR K FOR SELECTED MATERIALS
AT ROOM TEMPERATURE

<table>
<thead>
<tr>
<th>Material</th>
<th>Dimple</th>
<th>Spall</th>
<th>Perforation</th>
</tr>
</thead>
<tbody>
<tr>
<td>2024 Al</td>
<td>1.68</td>
<td>1.54</td>
<td>1.14</td>
</tr>
<tr>
<td>Lockalloy</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>316 SS</td>
<td>1.15</td>
<td>0.91</td>
<td>0.67</td>
</tr>
<tr>
<td>A-286</td>
<td>0.93</td>
<td>0.73</td>
<td>0.54</td>
</tr>
<tr>
<td>Nb-1%Pr</td>
<td>2.22</td>
<td>1.98</td>
<td>0.84</td>
</tr>
</tbody>
</table>
Table 1.3
MATERIALS FACTOR K FOR PERFORATION OF SELECTED MATERIALS AND TEMPERATURES

<table>
<thead>
<tr>
<th>Material</th>
<th>300°C</th>
<th>500°C</th>
<th>700°C</th>
<th>900°C</th>
</tr>
</thead>
<tbody>
<tr>
<td>2024 Al</td>
<td>1.14</td>
<td>1.24</td>
<td>1.31</td>
<td>1.37</td>
</tr>
<tr>
<td>Lockalloy</td>
<td>1.00</td>
<td>1.09</td>
<td>1.15</td>
<td>1.20</td>
</tr>
<tr>
<td>316 SS</td>
<td>0.67</td>
<td>0.73</td>
<td>0.77</td>
<td>0.80</td>
</tr>
<tr>
<td>A-286</td>
<td>0.54</td>
<td>0.59</td>
<td>0.62</td>
<td>0.65</td>
</tr>
<tr>
<td>Nb-1% Zr</td>
<td>0.84</td>
<td>0.91</td>
<td>0.97</td>
<td>1.00</td>
</tr>
</tbody>
</table>

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 cm² (103 in²)
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°K is seen in Table 1.5.

Table 1.5
REQUIRED ARMOR THICKNESS AND MASS @ 700°K

<table>
<thead>
<tr>
<th>Material</th>
<th>Thickness</th>
<th>Mass</th>
</tr>
</thead>
<tbody>
<tr>
<td>316 SS</td>
<td>0.27 cm</td>
<td>2.26 kg</td>
</tr>
<tr>
<td>A-286</td>
<td>0.22 cm</td>
<td>1.83 kg</td>
</tr>
<tr>
<td>Nb-1% Zr</td>
<td>0.34 cm</td>
<td>3.05 kg</td>
</tr>
</tbody>
</table>

The mass of the armor was taken to be:

\[ m_a = \frac{\pi D}{2} t \cdot \sigma_a \]  

\[ m_a = \text{Mass of the armor - grams} \]
\[ D = \text{Heat pipe diameter - 2.54 cm} \]
\[ t = \text{Armor thickness - cm} \]
\[ \cdot = \text{Heat pipe length - 262 cm} \]
\[ \sigma_a = \text{Density of the armor - gm/cc} \]

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. 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

<table>
<thead>
<tr>
<th>A</th>
<th>H/D</th>
</tr>
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<tr>
<td>2.5</td>
<td>0.0 (No Dimple)</td>
</tr>
<tr>
<td>2.0</td>
<td>0.13</td>
</tr>
<tr>
<td>1.7</td>
<td>0.22</td>
</tr>
<tr>
<td>1.5</td>
<td>0.32</td>
</tr>
<tr>
<td>1.4</td>
<td>0.34</td>
</tr>
<tr>
<td>1.0</td>
<td>0.60</td>
</tr>
<tr>
<td>0.9</td>
<td>0.75</td>
</tr>
<tr>
<td>0.8</td>
<td>1.0 (Inner Tube Closed)</td>
</tr>
</tbody>
</table>

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 $2 \times 10^{-3}$ cm particles, the meteoroid must be at least $2 \times 10^{-2}$ cm in diameter for the armor to behave as though it were solid. A particle of $2 \times 10^{-2}$ cm diameter with a 0.5 gm/cc density will have a mass of $2.09 \times 10^{-6}$ 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 $2 \times 10^{-3}$ 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°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°K the ΔT through solid SS armor is 2.2°K, thus the ΔT through porous armor may be as high as 22°K, which at 700°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°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>
<thead>
<tr>
<th>Table 1.7</th>
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</thead>
<tbody>
<tr>
<td><strong>MASS COMPARISON</strong></td>
</tr>
<tr>
<td><strong>SOLID ARMOR VS. 50% DENSE ARMOR</strong></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Solid Armor</th>
<th>50% Dense Armor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Pipe</td>
<td>Heat Pipe</td>
</tr>
<tr>
<td>1.75 kg</td>
<td>1.75 kg</td>
</tr>
<tr>
<td>Solid Armor</td>
<td>Armor (.707 x solid)</td>
</tr>
<tr>
<td>2.26 kg</td>
<td>1.60 kg</td>
</tr>
<tr>
<td>Total</td>
<td>Total</td>
</tr>
<tr>
<td>4.01 kg</td>
<td>3.35 kg</td>
</tr>
<tr>
<td>1.3% increase in mass due to 220K armor $\Delta T$</td>
<td>13.6% increase in mass due to 220K armor $\Delta T$</td>
</tr>
<tr>
<td>Total mass</td>
<td>Total mass</td>
</tr>
<tr>
<td>4.06 kg</td>
<td>3.81 kg</td>
</tr>
</tbody>
</table>

**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. 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, in Kinslow's book, 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 or less 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. (≈ 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:

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

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:

\[ S \geq 8D \text{ and } 0.1 \leq t_s/D \leq 1 \]

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. \( D = \frac{D}{P} \left( \frac{E}{T_{\text{al}}} \right)^{1/3} \)

7. The \( t_a/D \) 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 
\[ \frac{t_s}{D} = 0.15 \] and the total thickness of the shield and back up plate divided by the particle diameter is between 1 and 1.5. i.e. 
\[ 1 \leq \frac{(t_s + t_h)}{D} \leq 1.5 \]

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.
\[
\frac{t_s + t_b}{D} = \frac{4}{5} \frac{t_s}{D} + 1 \quad 5 \geq \frac{t_s}{D} \geq 0.15
\]

\[
\frac{t_s + t_b}{D} = 5 - 25.7 \frac{t_s}{D} \quad \frac{t_b}{D} \leq 0.15
\]

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 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 Institute in his letter of August 2, 1979, following.
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
Mr. Donald M. Ernst  
Thermacore, Inc.  
August 2, 1979

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
(D-Shaped and Configuration Pumped)
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 individual 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 erosion 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°C for 10,000 hours in a vacuum by Pratt and Whitney Company. 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 37,600 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. 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 emitter 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 Al₂O₃, paper, wood, glass, and ice, also have a uniform emittance for inclination angles between 0 and 60° 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°. 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 than 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 straightforward. 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:

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

$\epsilon_{\text{eff}}$ = Total effective emissivity of the radiator surface
$\epsilon_m$ = Emissivity of the edges of the fins
$\epsilon_c$ = 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 $\epsilon_{\text{eff}} = 0.9$ 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 $\Delta T$ 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 $\Delta T$ 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 Propulsion Spacecraft has been designed by Garrett AirResearch\textsuperscript{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 evaporators. 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.
3.1 Baseline Design

The total power to be dissipated is $1.1 \times 10^6$ 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, $\sigma A \varepsilon$ can be computed to be $2882 \times 10^{-12}$ watts/$^\circ K^4$ from:

$$P = \sigma A \varepsilon T^4$$

Eq. 3.1

where

$P$ = Power radiated - watts

$\sigma$ = Stefan Boltzmann Constant = $5.67 \times 10^{-12}$ watts $cm^{-2} \cdot ^\circ K^4$

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

$A$ = Individual heat pipe radiating area - $cm^2$

$\varepsilon$ = 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

<table>
<thead>
<tr>
<th>Temperature</th>
<th>Req. Power</th>
</tr>
</thead>
<tbody>
<tr>
<td>°K</td>
<td>°C</td>
</tr>
<tr>
<td>707</td>
<td>434</td>
</tr>
<tr>
<td>700</td>
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<td>600</td>
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<td>575</td>
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<td>252</td>
</tr>
<tr>
<td>500</td>
<td>227</td>
</tr>
<tr>
<td>492</td>
<td>219</td>
</tr>
</tbody>
</table>

**REQUIRED POWER PER HEAT PIPE AT ELEVEN DIFFERENT TEMPERATURES**
### Table 3.2
Heat Pipe Mass & Performance for Baseline Designs

<table>
<thead>
<tr>
<th>Temperature (°K)</th>
<th>Temperature (°C)</th>
<th>Required Power (Watts)</th>
<th>ΔT &amp; Req. Power (°C)</th>
<th>Power Limit (Watts)</th>
<th>Mass (Kg)</th>
<th>Groove Depth (cm)</th>
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</thead>
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<tr>
<td>707</td>
<td>434</td>
<td>720</td>
<td>2.56</td>
<td>1750-S</td>
<td>1.75</td>
<td>0.05</td>
</tr>
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<td>427</td>
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<tr>
<td>550</td>
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<td>492</td>
<td>219</td>
<td>189</td>
<td></td>
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<td></td>
<td></td>
</tr>
</tbody>
</table>

- Evaporator: 43 cm
- Adiabatic: 43 cm
- Condenser: 176 cm
- S = Sonic Limit
- C = Capillary Limit

Fluid: H²O  Vessel: 304 SS
- O.D.: 2.54 cm
- Wall: 0.0762 cm
- # Grooves: 25
- Groove Width: 0.2 cm

Fluid: DTA  Vessel: 304 SS
- O.D.: 2.54 cm
- Wall: 0.0762 cm
- # Groove: 27
- Groove Width: 0.2 cm
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 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 $15 \times 10^6$ psi. This includes a safety factor of 2. Stainless steels have moduli of about $28 \times 10^6$ 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°F 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.
### Optimized Heat Pipe Mass & Performance - Baseline Design

<table>
<thead>
<tr>
<th>Temperature (°K)</th>
<th>Temperature (°C)</th>
<th>Watts</th>
<th>Req. Power</th>
<th>ΔT @ Req. Power</th>
<th>Power Limit</th>
<th>Mass</th>
<th>Groove Depth</th>
<th>ΔT @ Req. Power</th>
<th>Power Limit</th>
<th>Mass</th>
<th>Groove Depth</th>
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<tr>
<td>700</td>
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<tr>
<td>625</td>
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<td>169</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td>3.55</td>
<td>555-C</td>
<td>1.55</td>
<td>.055</td>
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</tbody>
</table>

- Evaporator: 43 cm
- Adiabatic: 43 cm
- Condenser: 176 cm
- Fluid: Rb
- Vessel: 304 SS
- O.D.: 2.54 cm
- Wall: 0.0762 cm
- # Grooves: 25
- Groove Width: 0.275 cm

- Fluid: DTA
- Vessel: 304 SS
- O.D.: 2.54 cm
- Wall: 0.0762 cm
- # Groove: 25
- Groove Width: 0.275 cm

S = Sonic Limit
C = Capillary Limit
<table>
<thead>
<tr>
<th>Evaporator - 43 cm</th>
<th>Adiabatic - 43 cm</th>
<th>Condenser - 476 cm</th>
<th>C = Capillary Limit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid: Hb Vessel: 304 SS</td>
<td>Fluid: DTA Vessel: 204 SS</td>
<td>Fluid: DTA Vessel: 204 SS</td>
<td># Grooves: 25 Groove Width: 0.275 cm</td>
</tr>
<tr>
<td>Temperature</td>
<td>Req. Power</td>
<td>Groove Power Limit</td>
<td>Mass Groove Depth</td>
</tr>
<tr>
<td>°K</td>
<td>Watts</td>
<td>°K</td>
<td>Watts</td>
</tr>
<tr>
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<td>434</td>
<td>720</td>
<td>1.55</td>
</tr>
<tr>
<td>700</td>
<td>427</td>
<td>692</td>
<td>1.55</td>
</tr>
<tr>
<td>675</td>
<td>402</td>
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<td>4.56</td>
</tr>
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<td>377</td>
<td>514</td>
<td>4.56</td>
</tr>
<tr>
<td>625</td>
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<td>252</td>
<td>219</td>
<td>4.56</td>
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<tr>
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<td>227</td>
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<td>4.56</td>
</tr>
<tr>
<td>492</td>
<td>219</td>
<td>169</td>
<td>4.56</td>
</tr>
</tbody>
</table>

**TABLE 4.4**

Optimized Heat Pipe Mass & Performance for Thin Walled Baseline Design.
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>
<thead>
<tr>
<th>Temperature</th>
<th>Req. Power</th>
<th>ΔT @ Req. Power</th>
<th>Power Limit</th>
<th>Mass</th>
<th>Groove Depth</th>
<th>ΔT @ Req. Power</th>
<th>Power Limit</th>
<th>Mass</th>
<th>Groove Depth</th>
</tr>
</thead>
<tbody>
<tr>
<td>707 K</td>
<td>434 °C</td>
<td>720 Watts</td>
<td>930-2</td>
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<td>0.02</td>
<td>15.04</td>
<td>515-2</td>
<td>0.31</td>
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</tr>
<tr>
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<td>692 Watts</td>
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<td>0.05</td>
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<td>215-2</td>
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<td>0.05</td>
</tr>
<tr>
<td>492 K</td>
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<td>169 Watts</td>
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<td>Temperature</td>
<td>Req. Power</td>
<td>$\Delta T @$ Req. Power</td>
<td>Power Limit</td>
<td>Mass</td>
<td>Groove Depth</td>
<td>$\Delta T @$ Req. Power</td>
<td>Power Limit</td>
<td>Mass</td>
<td>Groove Depth</td>
</tr>
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</tr>
</tbody>
</table>
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 $\Delta T$.

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

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

$T_o - T = \Delta T$ 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 AT**

<table>
<thead>
<tr>
<th>Temperature (°K)</th>
<th>Power (W)</th>
<th>Fluid</th>
<th>Mass (Kg)</th>
<th>ΔT (°C)</th>
<th>ΔM (Kg)</th>
<th>New Mass (Kg)</th>
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<td>675</td>
<td>598</td>
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<td>0.280</td>
<td>3.63</td>
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</tr>
<tr>
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<td>514</td>
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<td></td>
<td></td>
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<tr>
<td>625</td>
<td>440</td>
<td>Iig</td>
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<td></td>
<td></td>
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<td></td>
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<tr>
<td>575</td>
<td>315</td>
<td>Iig</td>
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<td>5.81</td>
<td>7.5 x 10^{-3}</td>
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<tr>
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<tr>
<td>500</td>
<td>180</td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>492</td>
<td>169</td>
<td>DTA</td>
<td>0.273</td>
<td>4.05</td>
<td>6.2 x 10^{-3}</td>
<td>0.279</td>
</tr>
</tbody>
</table>
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 requirement of 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\textsuperscript{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°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 $\Delta T$ 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.
Hybrid heat pipes are directly analogous 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 because 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

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.
9. Hickenson, Norris L., Tennessee Technological University, Cookeville, Tennessee. (Section of Reference 1).
APPENDIX I

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.

Evaporator

Adiabatic

Condenser

Evaporator Temp. ← (Outside Wall) → Condenser Temp.

<table>
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<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
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<td>TE</td>
<td>(vapor)</td>
</tr>
<tr>
<td>PE</td>
<td></td>
</tr>
<tr>
<td>TE-A</td>
<td></td>
</tr>
<tr>
<td>TA-C</td>
<td></td>
</tr>
<tr>
<td>PE-A</td>
<td></td>
</tr>
<tr>
<td>PA-C</td>
<td></td>
</tr>
<tr>
<td>TC</td>
<td></td>
</tr>
<tr>
<td>PC</td>
<td></td>
</tr>
</tbody>
</table>

**Variables:**

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:**

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<th>Fluid</th>
<th>Wall Matl.</th>
<th>ETAP TEMP</th>
<th>VAPOR DELTA-T</th>
<th>GRAY ABS</th>
<th>VTO ABS</th>
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<table>
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<th>COLD LENGTH</th>
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0.00

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<th>VOT LENGTH</th>
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<th>GROOVE LENGTH</th>
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<th>Grooves (Closed) Covered with 500 Mesh</th>
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**NO LIMIT ENCOUNTERED AT**

720 WATTS

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<th>TC</th>
<th>DEB C</th>
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<table>
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<th>1110 A RATE</th>
<th>3 A RATE</th>
<th>G A RATE</th>
<th>C R RATE</th>
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<td>329.46</td>
<td>103</td>
<td>324.77</td>
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</tbody>
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| HOT FLUID CHARGE | 123.0495 | GRAMS |
| HOT TEMP | VOLUME OF HOT FLUID CHARGE | 104.4919 | CH3 |

| COLD FLUID CHARGE | 138.664 | GRAMS |
| COLD-FLUID | 29.763 | CH3 |

| ETAP PIPE (LBS) | 2 BAGS/10 | 10.06 | GRAMS |

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**POWER OF 1770 WATTS CAUSES**

1720 WATTS
REPRODUCIBILITY OF THE ORIGINAL PAGE IS POOR
RUN CONDITIONS:

FLUID = BOVULUX A
VAP MAST = 20400
EVP TEMP = 345
VAPOR DELTA-T = 60 DBR C
STAT ANSI = 0.00
VAP ANSI = 0.00 DBR

EVP LENGTH 13.68661 IN 45.0000 CM
SHE LENGTH 16.69061 IN 42.3000 CM
COLD LENGTH 20.30313 IN 51.5000 CM
TOTAL LENGTH 103.1500 IN 262.0000 CM

-DBR-
VAP MASSE 0.0000 IN 0.0000 CM
COLD MASSE 0.0000 IN 0.0000 CM
EVAP VAPOR 0.0000 IN 0.0000 CM
COLD VAPOR 0.0000 IN 0.0000 CM
WIN VAPOR 0.0000 IN 0.0000 CM
WIN COLD 0.0000 IN 0.0000 CM

EVAP (CLOSED) COVERED WATER 200 MESH

NO LIMIT ENCOUNTERED AT 440 VAPES

TOTAL DELTA-T = 3.88 DBR C
TOTAL MASSE = 1.7764 KG

VAPOR PERFORMANCE DETAILS (T OR H) YT

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ERG = 31.60
ERG = 0
P70+P80 = 31.60
NHEX/CM

NITR EFLD = DFL
ERG = 153

NITR DFL = DFL
ERG = 958

SONIC LIMITS:

HAT- 1.29225
AB= 1.29977
WAVES

C/A= EVAP
COLD
AXIAL WAVE/CM

2.5 ERT
N 4 RT
L4 RT
G A RT
G 2 RT
G 0 RT

645
634
605
1

EAT FLUID CHAMBER
ENUM TEMP. VOLUME OF HOT FLUID CHAMER 107-007 CM3

COLD FLUID CHAMER 125-001 CM3
125-001 CM3

ERG YEP: (MMK) A 2 EDGES 1020-42 GRAMS

ERG T-VALUES:

HAT VALL 1.34221
HAT AG 1.94513
HAT MINE 1.00098

ERG VAPOR (CQ)
ERG (MMK)
ERG (GQ)

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POWER OF 650 VAPES CAUSES CAPILLARY LIMIT: DFL > DFL

LAST NO-LIMITED POWER CALCULATION WAS AT 3-10 VAPES

TOTAL DELTA-T = 4.79 DBR C
TOTAL MASSE = 1.7764 KG
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**NO LIMIT ENCOUNTERED AT**

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**TOTAL DELTA-T = 1.73 DEG C**

**TOTAL MASS = 1.746 LB**

**HEAT TRANSFER DETAILS (T or K)**

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**EVAPE TEMP**

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**SHELL LIMITS:**

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**INPUTS:**

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<th>C R</th>
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<td>2</td>
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<td>2</td>
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**HOT FLUID VOLUME:**

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**COLD FLUID VOLUME:**

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<td>125.00 grams</td>
</tr>
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**HEAT PIPE: (HEAT) & 2 INTERCPS 10.00 GRAMS**

**DELTA-T VALUES:**

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</table>

**VAPOR (E):**

<table>
<thead>
<tr>
<th>VAPOR (E)</th>
<th>VAPOR (A)</th>
<th>VAPOR (C)</th>
</tr>
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<tbody>
<tr>
<td>271.6</td>
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**CONDENSATION:**

<table>
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<tr>
<th>COMB HESS</th>
<th>COMB LAG</th>
<th>COMB HALL</th>
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<tbody>
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**POW r**

<table>
<thead>
<tr>
<th>POWER</th>
<th>715 WATTS</th>
</tr>
</thead>
<tbody>
<tr>
<td>CAPILLARY LIMIT</td>
<td>710 WATTS</td>
</tr>
</tbody>
</table>

**LAST NON-LIMITED POWER CALCULATION WAS AT**

---

**TOTAL DELTA-T = 0.87 DEG C**

**TOTAL MASS = 1.746 LB**
RHE CONDITIONS: 3/22/78

FLUID = RURIDIAN WALL MATT=60%SS
EVAP TEMP = 434 VAPOR DELTA-T = 50 DEG C
GAS AMB = 0.00 VTD AMB = 0.00 DEG
EVAP LENGTH 16.000 IN 45.0000 GN
ARD LENGTH 16.000 IN 45.0000 GN
COND LENGTH 62.000 IN 176.0000 GN
TOTAL LENGTH 103.000 IN 268.0000 GN

0-O 1.0000 IN 2.0000 GN
WALL THICK 0.0700 IN 0.0700 GN
GROOVE WIDTH 0.0583 IN 0.2750 GN
GROOVE HEIGHT 0.0200 GN
LINED WIDTH 0.0200 IN 0.0200 GN
25 GROOVES (CLOSSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT 720 WATTS

TOTAL DELTA-T = 2.63 DEG C
TOTAL MASS = 1.406 LBS

VAPOR PERFORMANCE DETAILS (T OR H)T

PE PE-A PE-G PC
31294.6 30270.4 30100.3 30712.1
PE T
332.251 431.942 450.703 431.949
EVAP TEMP
COND TEMP DELTA-T
434 431.949 2.63 DEG C

DPC 16214 DPH 0 DPC-DPH= 16214 OUTD/CM2

DPH 1106 255 9536
DPC 0.615 6900

SONIC LIMITS: EVAP= 2216 ADD= 2203 WATTS
G/A'S EVAP COND AXIAL WATTS/CM2
2 0 163

R & R BENT 2 & RENT LIG RENT G & RENT G & RENT
21 3101 92 3103 9

HOT FLUID CHARGE 82.1796 Grams
HOT TEMP VOLUME OF HOT FLUID CHARGE 80.1694 CM3
COND FLUID CHARGE 107.824 Grams
COND TEMP 70.1614 CM3

HEAT PIPE PER KNEE & 2 BENDS 17.7682 Grams

DELTA-T VALUES:

EVAP WALL EVAP LAG EVAP MESH EVAPABORATION
-0.0031 -0.0052 -0.0052 -0.0052
VAPOR (3) VAPOR (12) VAPOR (C)
1.03586 -0.00756 -1.03586
CONDENSATION COND MESH COND LAG COND WALL
0.073507 0.14348-02 0.305730-03 0.305730-03

POWER OF 720 WATTS CAUSES CAPILLARY LIMIT DPL = DPH
LAST NON-LIMITED POWER CALCULATION WAS AT 415 WATTS

TOTAL DELTA-T = 2.63 DEG C
TOTAL MASS = 1.406 LBS
HON CONDITIONS: 3/30/79

FLUID = NITRILE
WALL MAT = 304 MS
EVAP TEMP = 377
VAPOR DELTA-T = 0.0 DEG C
GRAY ABS = 0.90
WET ABS = 0.00 DEG C

EVAP LENGTH 18.6291 IN 63.0000 CM
ADD LENGTH 18.6291 IN 63.0000 CM
GORE LENGTH 59.2653 IN 176.0000 CM
TOTAL LENGTH 108.1500 IN 274.0000 CM
0.1 1.0000 IN 2.5400 CM
WALL THICK 0.0300 IN 0.0762 CM
GROOVE WIDTH 0.1063 IN 0.0270 CM
GROOVE HEIGHT 0.0079 IN 0.0200 CM
LAND WIDTH 0.0079 IN 0.0200 CM
25 GROOVS (CLOSED) COVERED WITH 200 HAZE

NO LIMIT ENCOUNTERED AT ********** 514 WATTS

********** TOTAL DELTA-T = 5.83 DEG C
********** TOTAL MASS = 1.484 HB

VANT PERFORMANCE DETAILS (IT OR XI) YI

| FE | FE-A | FE-C | FE | DETER/CM2
|---|---|---|---|---|
| 0062.79 | 0060.06 | 0061.59 | 0063.7

| TE | TE-A | TE-C | TE | DEG C
|---|---|---|---|
| 371.497 | 368.683 | 365.436 | 371.445

<table>
<thead>
<tr>
<th>EVAP TEMP</th>
<th>COOL TEMP</th>
<th>DELTA-T</th>
</tr>
</thead>
</table>
| 377 | 371.17 | 5.83983

| DPF | 19138 | DPF = 0 | DPF+DPF = 19138 | DETER/CM2
|---|---|---|---|---|
| DNF | DNF | DNF | DNF | DNF
| DPH | 628 | 624 | 624 | 624
| DPF | 624 | 624 | 624 | 624

SONIC LIMITS: EVAP = 780 AND = 705 WATTS

<table>
<thead>
<tr>
<th>O/A' S</th>
<th>EVAP</th>
<th>COOL</th>
<th>AXIAL</th>
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<th>LIG KEY</th>
<th>C A KEY</th>
<th>C R KEY</th>
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<tr>
<td>10</td>
<td>2029</td>
<td>2029</td>
<td>2029</td>
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</table>

HOT FLUID CHANGE 93.6783 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHANGE 01.2782 CM3

COLD FLUID CHANGE 107.824 GRAMS
70.2914 CM3

EVEIT PIPES (HNESS) & 2 ENDCAPS 1375.82 GRAMS

DELTA-T VALUES:

<table>
<thead>
<tr>
<th>EVAP WALL</th>
<th>EVAP LAD</th>
<th>EVAP HESS</th>
<th>EVAP EVAPORATION</th>
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<tbody>
<tr>
<td>+016064</td>
<td>+205028.62</td>
<td>+350453.62</td>
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<th>VAPOR (C)</th>
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<td>+01489</td>
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<table>
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<th>COND LAD</th>
<th>COND WALL</th>
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<tr>
<td>-182278</td>
<td>-0423353.03</td>
<td>+5000058.03</td>
<td>-191117</td>
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</table>

POWER OF 648 WATTS CAUSES ********** ADD SONIC LIMIT

LAST NON-LIMITED POWER CALCULATION WAS AT ********** 040 WATTS

********** TOTAL DELTA-T = 7.13 DEG C
********** TOTAL MASS = 1.904 HB
REU CONDITIONS

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

FLUID = DOWTHERM A  VAPE MALT=30453
EVAP TEMP = 332  VAPE DELTA-T = 50 DEG C
GRN AM = 0.00  WTR AM = 0.00 DEG

EVAP LENGTH 16.9521 IN 65.0000 CM
LAB LENGTH 16.9521 IN 65.0000 CM
COND LENGTH 09.2913 IN 178.0000 CM
TOTAL LENGTH 103.1500 IN 262.0000 CM

WALL THICK 0.0050 IN 0.0762 CM
GROOVE WIDE 0.1083 IN 0.2750 CM
GROOVE DEPTH 0.0217 IN 0.0550 CM
LAD WIDE 0.0044 IN 0.0112 CM
25 3 HOLES (CLOSED) COVERED W/T 200 MESH

NO LIMIT ENCOUNTERED AT 460 WATTS

TOTAL DELTA-T = 9.23 DEG C
TOTAL MASS = 1.548 E2

VAPE PERFORMANCE DETAILS (E OR N) TT

TAPE PA-A PA-G PA-C PC DEGES/CM2
.811066E+07 .811065E+07 .811068E+07 .811068E+07 .811068E+07

TAPE PA-A PA-G DEG C
544.681 544.681 544.681 544.681

EVAP TEMP 332 342.773 232.773

DPCT 3264 0 3264 00000001

DWF = DUAL DWF = DUAL
DWF = DUAL 1500 DWF = DUAL

SOEIC LIMITS:

EVAP= 160576 ADD= 181288 WATTS

C/A/ S= EVAP COND AXIAL WATTS/CM2
1 0 10

E R EKT = LIG EKT G A EKT G EKT EKT
5 797 281 797

SOFT FLUID GRAMS:

ROOM TEMP VOL OF SOFT FLUID GRAMS
130.205

COLD FLUID GRAMS:

167.606

EVAPE PIPE (MEAS) & 2 HEADS 1604.22 GRAMS

DELTA-T VALUES:

EVAP VALL EVAP LAG EVAP MESS EVAPARATION DEX C
.529683 0.12106 .040618 .100608 DEX C

VAPE (3) VAPE (4) VAPE (5)
.244612E-05 .244612E-05 .244612E-05

COND KREAT COND LAG COND WALL
.04607E-01 .15604 .153078 DEX C

POWER OF 475 WATTS CAUSES CAPILLARY LIMIT; DPF = DPF

LAST NON-LIMITED POWER CALCULATION WAS AT 370 WATTS

TOTAL DELTA-T = 11.62 DEG C
TOTAL MASS = 1.548 E2
AUX CONDITIONS:

4:7 P.M.  6/5/79

FLUID = RUBIDIUM  WALL MAT=304SS
EVAP TEMP = 534  VAPOR DELTA-T = 50 DEG C
GRAV AM = 0.00  VTS AM = 0.00 DEG

EVAP LENGTH 16.9294 IN  43.0000 CM
ADD LENGTH 16.9294 IN  43.0000 CM
COND LENGTH 69.2213 IN  176.0000 CM
TOTAL LENGTH 103.1000 IN  262.0000 CM

0-D.  1.0000 IN  2.5400 CM
WALL THICK 0.0300 IN  0.0762 CM
GROOVE WIDTH 0.1063 IN  0.2700 CM
GROOVE HEIGHT 0.0079 IN  0.2000 CM
LAND WIDTH 0.0079 IN  0.2000 CM
20 GROOVES (CLOSED) COVERED VTH X 300 WHEE

NO LIMIT ENCOUNTERED AT ------------- 720 VATTS

------------- TOTAL DELTA-T = 2.43 DEG C
------------- TOTAL MASS = 1.486 ED

VAPOR PERFORMANCE DETAILS (Y OR M) TT

F6 31296.8 30379.4 30106.0 30735.1 DINES/CM2
T2 432.251 431.242 430.752 431.849 DEG C

T6 EVAP TEMP 434 COND TEMP DELTA-T
431.07 2.43 DEG C

DPC=18214 DPH=0 DPC+DPH=18214 DINES/CM2

DPVT DPLT DPLM DPTA DPL4 DPL46 DPL3 DPL30
-415 4000

SONIC LIMITS: 3161 2491 2531 WATTS
2/1/2 8/1/2 168 168

E/R KEY 0 R & KEY LIQ KEY 0 & KEY 0 R KEY
12 3161 22 3163 3163 2

HOT FLUID GRAM 92.1706 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID GRAM 40.1306 CM3
COLD FLUID GRAM 107.824 GRAMS
70.3616 CM3

HEAT PIPE: (XERS) & 2 ENDCAPS 1375.82 GRAMS

DELTA-T VALUES:

EVAP WALL 0.00531 0.32821 0.36383 0.30283 DINES C
EVAP LAM 0.32821 0.36383 0.30283 0.36383 DINES C
EVAP XER 0.36383 0.30283 0.36383 0.36383 DINES C

VAPOR (E) 1.01916 0.79730 1.02607
VAPOR (A) 479730 1.02607
VAPOR (C) 1.02607 479730

CONDENSATION COND XER COND LAM COND WALL
1.06546 1.06546 1.06546 1.06546

POWER OF 620 WATTS CAUSES ----------- CAPILLARY LIMIT, DPL = DPVT
LAST NON-LIMITED POWER CALCULATION WAS AT -------------- 315 WATTS

------------- TOTAL DELTA-T = 2.37 DEG C
------------- TOTAL MASS = 1.486 ED
FLUID = ETHANOL
WALL MAT = 304SS
EVAP TEMP = 377
VAPOR DELTA-T = 50 DEG C
GRAY ABS = 0.00
WTO ABS = 0.00 DEG

EVAP LENGTH 16-9291 IN 42-0000 CM
ADD LENGTH 16-9291 IN 42-0000 CM
CORD LENGTH 66-2913 IN 175-0000 CM
TOTAL LENGTH 103-1500 IN 258-0000 CM

O.D. 1.0000 IN 25.4000 CM
WALL THICK 0.0100 IN 0.2540 CM
GROOVE WIDTH 0.1083 IN 2.7500 CM
GROOVE HEIGHT 0.0079 IN 0.2000 CM
LARD WIDTH 0.0120 IN 0.3050 CM
26 GROOVES (CLOSED) COVERED WITH 500 MESH

NO LIMIT ENCOUNTERED AT ----------------- 916 WATTS

TOTAL DELTA-T = 4.56 DEG C
TOTAL MASS = 0.891 LB

WATT PERFORMANCE DETAILS IT OR HI TT

<table>
<thead>
<tr>
<th>PR</th>
<th>FR-A</th>
<th>PA-C</th>
<th>RG</th>
<th>OTHERS/GM2</th>
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<td>10019.5</td>
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<table>
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<tr>
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<th>TV-A</th>
<th>TA-C</th>
<th>RC</th>
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<td>767.225</td>
<td>375.285</td>
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|urdy = 101 | EVA = 377 | ADD = 876 WATT |

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<tr>
<th>0/A's</th>
<th>EVAP</th>
<th>CORD</th>
<th>AXIAL</th>
<th>VERT WATTS/GM2</th>
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<th>3 A RPM</th>
<th>LIG RPM</th>
<th>G A RPM</th>
<th>G R RPM</th>
<th>C R RPM</th>
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<td>15</td>
<td>2881</td>
<td>99</td>
<td>2390</td>
<td>3</td>
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HOT FLUID GRAMS 89.35 GRAMS
HOT TEMP VOLUME OF HOT FLUID GRAMS 42.5658 CM3

COLD FLUID GRAMS 110.122 GRAMS
71.6693 CM3

HEAT PIPES (INNER) & 2 HEXCAPS 880.457 GRAMS

DELTA-T VALUES:

<table>
<thead>
<tr>
<th>EVAP WALL</th>
<th>EVAP LGS</th>
<th>EVAP KEZ</th>
<th>EVAP VAP</th>
<th>EVAP DEP</th>
<th>DEG C</th>
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<tbody>
<tr>
<td>0.20125</td>
<td>0.207966</td>
<td>0.338838</td>
<td>0.292453</td>
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<table>
<thead>
<tr>
<th>VAPOR (B)</th>
<th>VAPOR (A)</th>
<th>VAPOR (C)</th>
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</thead>
<tbody>
<tr>
<td>5.6286</td>
<td>4.6286</td>
<td>3.770912</td>
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<thead>
<tr>
<th>COMBINATION</th>
<th>CORD KEZ</th>
<th>COMB LGS</th>
<th>COMB WALL</th>
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<tr>
<td>+122275</td>
<td>+0102552</td>
<td>+5102552</td>
<td>+4929292</td>
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</table>

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

LAST NON-LIMITED POWER CALCULATION WAS AT ----------------- 706 WATTS

TOTAL DELTA-T = 6.12 DEG C
TOTAL MASS = 0.691 LB

75
**FLUID = DOWNTEM A**
**WALL MATT=30455**
**EVAP TEMP = 352**
**VAPOR DELTA-T = 50 DEG C**
**DRY AM = 0.00**
**WGT AM = 2.00 DEG**

**EVAP LENGTH** 15.2331 IN 43.0000 CM
**ADS LENGTH** 15.2231 IN 43.0000 CM
**COND LENGTH** 66.2133 IN 170.0000 CM
**TOTAL LENGTH** 103.1500 IN 262.0000 CM

- OD: 1.0000 IN 2.5400 CM
- WALL THICK: 0.0100 IN 0.0254 CM
- GROOVE WIDTH: 0.1060 IN 0.2700 CM
- GROOVE HEIGHT: 0.0217 IN 0.0550 CM
- LANE WIDTH: 0.0094 IN 0.0240 CM
- 25 GROOVES (C10000) COVERED WITH 200 MESH

**NO LIMIT ENCOUNTERED AT** ———— 440 WATTS
————— TOTAL DELTA-T = 5.72 DEG C
————— TOTAL MASS = 0.077 EG

**WAX PERFORMANCE DETAILS** (Y OR 1)?
- PB: PA-A
- PA: PA-C
- PC: PC
- ETHYS: CM2

- TE: TE-A
- TA-C
- TC

- ETHY: 3200
- ETHA: 3200
- ETHAS: CM2

- 13: 219
- 6: 1638

**SODIC LIMITS:**
**EVAP = 170328 AND = 203379 WATTS**

- ETH: 17.74
- ETHA: 0
- ETHAS: CM2

**LET# =**

- 3: 704
- 287
- 704
- 1

**HOT FLUID CHARGE:**
**123.66 GRAMS**

**COLD FLUID CHARGE:**
**142.992 GRAMS**
**153.888 CM3**

**HEAT PIPE (MESH) & 2 BENDCAPS 654.002 GRAMS**

**DELTA-T VALUES:**

- EVAP WALL: 3.175328
- EVAP LEG: 3.070939
- EVAP MESH: 0.012878
- EVAP ANG: 0.100000

**VEPR (B):**
- 0.602833
- 264613.63

**COND MESH:**
- 0.218033
- 0.090000

**COND WALL:**
- 0.149562
- 0.050000

**POWER OF 550 WATTS CAUSES** ———— CAPILLARY LIMIT: DPL = DFR

**LAST NOX-LIMITED POWER CALCULATION WAS AT** ———— 565 WATTS
————— TOTAL DELTA-T = 7.19 DEG C
————— TOTAL MASS = 0.777 EG
**Reproducibility of the Original Page is Poor**

### Run Conditions:

- **Fluid:** Dowtherm A
- **Wall Mat.:** 30-425
- **Evap Temp.:** 219
- **Vapor Delta-T:** 0.0 Deg C
- **Gray Avg:** 0.00
- **Wet Avg:** 0.00 Deg C

### Dimensions:

- **Evap Length:** 10-09261 in 43.0000 cm
- **Adv Length:** 10-09261 in 43.0000 cm
- **Cond Length:** 50-09213 in 176.0000 cm
- **Total Length:** 103-1500 in 262.0000 cm

### No Limit Encountered At: 139 Watts

<table>
<thead>
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<th>Total Delta-T</th>
<th>Total Mass</th>
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<tr>
<td>2.49 Deg C</td>
<td>0.777 lb</td>
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### Watt Performance Details (T or K) 

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### Evap Temp:

- **Cond Temp:** 219
- **Delta-T:** 219 - 218.516 = 0.4847

### delta T Values:

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<thead>
<tr>
<th>Evap Value</th>
<th>Evap Low</th>
<th>Evap High</th>
<th>Evaporation</th>
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<td>+100392</td>
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### Power of 715 Watts Causes: Capillary Limit, DPL > DPF

**Last Non-Limited Power Calculation Was At:** 710 Watts

<table>
<thead>
<tr>
<th>Total Delta-T</th>
<th>Total Mass</th>
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</thead>
<tbody>
<tr>
<td>10.06 Deg C</td>
<td>0.777 lb</td>
</tr>
<tr>
<td>0.777 lb</td>
<td>0.777 lb</td>
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</table>
RUN CONDITIONS:

FLUID = MERCURY
EVAP TEMP = 434
VAPOR DELTAT = 70-80 DBS C
GLOW ANG = 0.00
VET ANG = 0.00 DBS

EVAP LENGTH = 18.0281 IN 45.8000 CM
ADD LENGTH = 16.0291 IN 41.1500 CM
COOL LENGTH = 69.2013 IN 176.2000 CM
TOTAL LENGTH = 103.1300 IN 262.0000 CM

G/D = 0.3756 IN 0.9534 CM
WALL THICK = 0.0032 IN 0.0081 CM
GROOVE WIDTH = 0.0779 IN 0.0000 CM
GROOVE DEPTH = 0.0079 IN 0.0000 CM
LAMD WIDTH = 0.0006 IN 0.0000 CM
G GROOVES (CLOSED) COVERED WITH 300 MESH

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

------------- TOTAL DELTA-T = 2.69 DBS C
------------- TOTAL MASS = 0.449 LB

WATT PERFORMANCE DETAILS (Y OR R) ??

PR .308686E+07 .306686E+07 .306686E+07 .306686E+07
PG .306686E+07 .306686E+07 .306686E+07 .306686E+07


DPC = 114106 DPM = 0
DPM+RPM = 114106 DPM/GMM
DPTE 1507 GPM 1007 152600
DPVC 1507 GPM 152600

SONIC LIMITS:

0/9 = 20097 ADD = 24099 WATTS

G/A = EVAP COMB AXIAL WATTS/GMM
3 1 1010

R ENTR L PIC EYTR G A ENT C R ENT 320 330 320

HOT FLUID CHARGE
281.891 GRAMS
ROOM TEMP. VALUE OF HOT FLUID CHARGE 20.8099 CMM
COLD FLUID CHARGE 280.48 GRAMS 21.664 CMM

HEAT PIPE (MESH) & 2 EJGAPS 150.000 GRAMS

DELTA-T VALUES:

EVAP VALL .292832
EVAP LAG .722206
EVAP MESS .106008
EVAPULATION .106008 DBS C

VAPOR (E) .292832-.01 .722206-.01 .106008-.01 .106008-.01

CONDENSE .292832-.01 .722206-.01 .106008-.01 .106008-.01

POWER OF 936 WATTS CAUSES -------- CAPILLARY LIMIT: DPL > DPV
LADT NON-LIMITED POWER CALCULATION WAS AT ------------ 930 WATTS

------------- TOTAL DELTA-T = 3.64 DBS C
------------- TOTAL MASS = 0.449 LB
**RHE CONDITIONS:**

| FLUID = MERCURY | FALL MELT = 30°C | ELEVATION = 705.8
| VAPOR TEMPERATURE = 192 | VAPOR PRESSURE = 450 DEG C | WATER AMOUNT = 0.00 | VAPOR AMOUNT = 0.00 DEG |

| EVAP LENGTH | 19.00 | 43.0000 CM |
| ADD LENGTH | 23.00 | 43.0000 CM |
| COLD LENGTH | 115.00 | 175.0000 CM |
| TOTAL LENGTH | 155.00 | 258.0000 CM |
| O.D. | 0.0625 | 0.0625 CM |
| INNER DIAMETER | 0.0524 | 0.0524 CM |
| GROOVE WIDTH | 0.0705 | 0.0705 CM |
| GROOVE DEPTH | 0.0661 | 0.0661 CM |
| LAND WIDTH | 0.0682 | 0.0682 CM |
| 8 GROOVES (CLOSED) COVERED WITH 500 PAR T|

**SO LIMIT ENCOUNTERED AT: 300 WATTS**

---

**TOTAL DELTA-T:** 1.28 DEG C

---

**TOTAL MASS:** 0.448 KG

---

**VAPOR PERFORMANCE DETAILS (X OR H) TT**

<table>
<thead>
<tr>
<th>X</th>
<th>X = A</th>
<th>X = B</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.11000000</td>
<td>0.11000000</td>
<td>0.11000000</td>
</tr>
<tr>
<td>0.10000000</td>
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<table>
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<tr>
<th>T2</th>
<th>T2 = A</th>
<th>T2 = B</th>
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<tbody>
<tr>
<td>300.0000</td>
<td>300.0000</td>
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<tr>
<td>300.0000</td>
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</tr>
<tr>
<td>300.0000</td>
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</tbody>
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---

**EVAP TEMPERATURE RANGES**

<table>
<thead>
<tr>
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</thead>
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<tr>
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<tr>
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</tbody>
</table>

---

**HOT FLUID VOLUME**

**RANGE VOLUME OF HOT FLUID**

<table>
<thead>
<tr>
<th>X</th>
<th>X = A</th>
<th>X = B</th>
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<tbody>
<tr>
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<td>0.11000000</td>
</tr>
<tr>
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</tbody>
</table>

---

**COLD FLUID MASS**

**MAG-MASS**

<table>
<thead>
<tr>
<th>X</th>
<th>X = A</th>
<th>X = B</th>
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</thead>
<tbody>
<tr>
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<td>0.11000000</td>
</tr>
<tr>
<td>0.10000000</td>
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<td>0.10000000</td>
</tr>
</tbody>
</table>

---

**DELTAT-T VALUES**

<table>
<thead>
<tr>
<th>X</th>
<th>X = A</th>
<th>X = B</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.11000000</td>
<td>0.11000000</td>
<td>0.11000000</td>
</tr>
<tr>
<td>0.10000000</td>
<td>0.10000000</td>
<td>0.10000000</td>
</tr>
</tbody>
</table>

---

**POWER UP 305 WATTS CAUSES: JAPILLARY LIMIT; DEL = DEP**

---

**LAST NON-LIMITED POWER CALCULATION WAS AT 300 WATTS**

---

**TOTAL DELTA-T:** 3.08 DEG C

---

**TOTAL MASS:** 0.448 KG
**DEW CONDITIONS**

**315 F.C.** 3/27/79

**FLUID** = LATEX

**STAB TEMP** = 277

**VAPOR DELTA-T** = 80 229 C

**2 K X 3 = 0.00**

**F L X 1** = 0.00

---

**STAB L.G.**

**L.G.**

**COLD L.G.**

**TOTAL L.G.**

---

**0.00**

**0.00**

**0.00**

**0.00**

---

**0.00**

**30.00**

**60.00**

---

**204 V A T T S**

**TOTAL DELTA-T = 2.00**

**TOTAL MASS = 0.449**

---

**FLUID PERFORMANCE DETAILS** (T or K) **Y**

<table>
<thead>
<tr>
<th>Y</th>
<th>Y+</th>
<th>Y-</th>
<th>Y</th>
</tr>
</thead>
<tbody>
<tr>
<td>2000</td>
<td>0</td>
<td>0</td>
<td>2000</td>
</tr>
</tbody>
</table>

---

**SONIC LIMITS**

**EVAP** = 1200

**AR** = 1200

**V A T T S**

**G A S**

---

**COLD FLUID CHANGES**

**VOLUME OF COLD FLUID CHANGES**

**COLD FLUID CHANGES**

---

**HEAT PIPE** (H.E.K) & 2 B.R.CAPS 108.000 OR 100.000

**DELTA-T VALUES**

<table>
<thead>
<tr>
<th>EVAP FALL</th>
<th>STAB L.G.</th>
<th>EVAP X.S.E.</th>
<th>VAPORATION</th>
</tr>
</thead>
<tbody>
<tr>
<td>-110.064</td>
<td>-30.0967</td>
<td>-100.068</td>
<td>100.068</td>
</tr>
</tbody>
</table>

---

**POWER OF 315 V A T T S CAUSES**

**CAPILLARY LIMIT**

---

**LAST NON-LIMITED POWER CALCULATION WAS AT**

**305 V A T T S**

**TOTAL DELTA-T = 8.33**

**TOTAL MASS = 0.449**
<table>
<thead>
<tr>
<th>FLUID</th>
<th>HOTTEST A</th>
<th>TANK MASS: 50.6 KG</th>
</tr>
</thead>
<tbody>
<tr>
<td>START TEMP</td>
<td>355</td>
<td>TANK STEM: 60 KBG</td>
</tr>
<tr>
<td>JAT ANS</td>
<td>0.00</td>
<td>HAT ANS: 0.00 KBG</td>
</tr>
<tr>
<td>FLOWS LENGTH</td>
<td>14.0983</td>
<td>43.0000 CM</td>
</tr>
<tr>
<td>AND LENGTH</td>
<td>16.0681</td>
<td>43.0000 CM</td>
</tr>
<tr>
<td>CHILL LENGTH</td>
<td>22.0313</td>
<td>17.0200 CM</td>
</tr>
<tr>
<td>TOTAL LENGTH</td>
<td>102.1800</td>
<td>295.0000 CM</td>
</tr>
<tr>
<td>G.C.</td>
<td>0.06</td>
<td>TANK VOLUME: 1.3700 KG</td>
</tr>
<tr>
<td>COLD VOLUME</td>
<td>0.0080</td>
<td>0.0127 KG</td>
</tr>
<tr>
<td>COLD VOLUME</td>
<td>0.0080</td>
<td>0.0127 KG</td>
</tr>
<tr>
<td>COLD WALL VOLUME</td>
<td>0.0080</td>
<td>0.0127 KG</td>
</tr>
<tr>
<td>TOTAL VOLUME</td>
<td>0.0080</td>
<td>0.0127 KG</td>
</tr>
</tbody>
</table>

**NO LIMIT REACHED AT ——— 460 WATTS**

**TOTAL REAEL: 15.0 KG**

**TOTAL MASS: 0.0309 KG**

**FLUID PERFORMANCE DETAILS (T OR H)**

<table>
<thead>
<tr>
<th>FLUID</th>
<th>T</th>
<th>H</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>FLOW</td>
<td>339.00</td>
<td>339.00</td>
<td>339.00</td>
</tr>
<tr>
<td>SAT TEMP</td>
<td>355</td>
<td>355</td>
<td>355</td>
</tr>
<tr>
<td>SAT PRESS</td>
<td>355</td>
<td>355</td>
<td>355</td>
</tr>
<tr>
<td>PRESS</td>
<td>355</td>
<td>355</td>
<td>355</td>
</tr>
<tr>
<td>NOT FLUID CHARGE</td>
<td>0.0309 KG</td>
<td></td>
<td></td>
</tr>
<tr>
<td>ROOM TEMP: VOLUME OF HOT FLUID CHARGE</td>
<td>0.0127 KG</td>
<td></td>
<td></td>
</tr>
<tr>
<td>COLD FLUID CHARGE</td>
<td>0.0309 KG</td>
<td></td>
<td></td>
</tr>
<tr>
<td>EXTR FIVE: (KNEE)</td>
<td>2 INCHES</td>
<td>219.701 GRAMS</td>
<td></td>
</tr>
</tbody>
</table>

**DELTA-T VALUES:**

| FLOW | 355 | 355 | 355 |
| SAT TEMP | 10.062 | 10.062 | 10.062 |
| SAT PESSure | 1.3500 | 1.3500 | 1.3500 |
| VAPOR (H) | 1.000000 | 1.000000 | 1.000000 |
| COND (H) | 2.073025 | 2.073025 | 2.073025 |
| PROP | 0.351233 | 0.351233 | 0.351233 |

**POWER OF 518 WATTS CAUSES ———— CAPILLARY LIMIT: DFL = DFN**

**LAST NON-LIMITED POWER CALCULATION WAS AT ———— 510 WATTS**

**TOTAL DELTA-T | 17.42 KBG**

**TOTAL MASS | 0.0309 KG**

---

81
**SHEET CONDITIONS**

- **EVAP FLUID**: DOWTHERM A
- **COLD cassette**: 
- **EVAP TEMP**: 38°F
- **COLD TEMPERATURE**: 50°F
- **EVAP HEMP**: 0.00
- **COLD HEMP**: 0.00
- **EVAP LENGTH**: 16.50000 IN
- **COLD LENGTH**: 42.30000 IN
- **TOTAL LENGTH**: 1.382.0000 IN
- **FLUID HEMP**: 0.30000 IN
- **COLD HEMP**: 0.30000 IN
- **COLD WALL**: 0.30000 IN
- **LAD SIZED**: 0.00000 IN
- **12 GROOVED (DOWTHERM) COVERED WITH 200 MESH

**NO LIMIT RECONCILATION AT ** 375 WATTS

---

**THERMAL PERFORMANCE DETAILS (Y or H) Y**

<table>
<thead>
<tr>
<th></th>
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<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>31.5-316</td>
<td>31.5-316</td>
<td>31.5-316</td>
<td>31.5-316</td>
<td>31.5-316</td>
<td>31.5-316</td>
<td>31.5-316</td>
<td>31.5-316</td>
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<td>31.5-316</td>
<td>31.5-316</td>
<td>31.5-316</td>
<td></td>
</tr>
</tbody>
</table>

**Sonic Limits:**

- **ETAP**: 26000
- **ADD**: 28000
- **ETAP**: 29000
- **ADD**: 29000

**GAS FLOW MEASUREMENTS:**

<table>
<thead>
<tr>
<th>C/A</th>
<th>STAP</th>
<th>COMB</th>
<th>AXIAL</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>80</td>
<td>394</td>
<td></td>
</tr>
</tbody>
</table>

**HEAT FLUID CHARGE**: 326000 GRAMS

**VOLUME OF HOT FLUID CHARGE**: 0.4153 CM3

**COLD FLUID CHARGE**: 77.66400 CM3

**HEAT PIPE**: (MESH) @ 2 ENDCAPS 216.7 FOR GRAMS

**DELTA-T VALUES:**

- **EVAP FLUID**: 0.101407
- **EVAP LAD**: 0.123922
- **EVAP HEMP**: 1.251217
- **EVAP EVAPORATION**: 1.052998
- **EVAP VAPOR (B)**: 0.0971258-02
- **EVAP VAPOR (C)**: 0.0839578-02
- **COLD CONDensation**: 0.373138-02
- **COLD CONDensation**: 0.2445878-01
- **COLD CONDensation**: 0.2795343
- **COLD CONDensation**: 1.779523
- **COLD CONDensation**: 0.3771958-01
- **COLD CONDensation**: 0.3771958-01

**POWERS OF 42A WATTS CAUSES**: CAPILLARY LIMIT: DPL = DPF

**LAST NON-LIMITED POWER CALCULATION WAS AT**: 420 WATTS

---

- **TOTAL DELTA-T**: 12.08 DBUS C
- **TOTAL MASS**: 0.286 HE
FLUID = DOWTHERM A  WALL HATL = 304853
EVAP TEMP = 302  VAPOR DELTA-T = 00 DEG C
SAT ANG = 0.00  VAP ANG = 0.00 DEG
EVAP LENGTH 14.6281 IN 43.0000 CM
ADD LENGTH 16.8289 IN 43.0000 CM
COND LENGTH 28.6253 IN 73.0000 CM
TOTAL LENGTH 103.1900 IN 263.0000 CM

0.9
WALL THICK = 0.0200 IN 0.0508 CM
GROOVE WIDE = 0.0088 IN 0.0223 CM
JOGGERS DEPTH = 0.00256 IN 0.0065 CM
LAM V'VEX = 0.0195 IN 0.0049 CM
12 GROOVES (CLOSED) COVERED WITH 300 MESH

NO LIMIT ENCOUNTERED AT ~~~~~~~~~~~~~~~~~ 318 WATTS

~~~~~~~~~~~~~~~~ TOTAL DELTA-T = 8.28 DEG C
~~~~~~~~~~~~~~~~ TOTAL MASS = 0.328 IN

VAPOR PERFORMANCE DETAILS (T OR R) IT T
FE  TB-A  TB-C  TB-D  TB-E
CHRE/CM
TE  TB-A  TB-C  TB-D  TB-E
286.277  286.277  286.277  286.277
CMB/CM
EVAP TEMP  COND TEMP DELTA-T
302  85.0  217

INF= 4636 INF= 0 INF+INF= 4636 CHRE/CM

DPM 305 

DPM 120 

SONIC LIMITS: EVAP= 57710 AND= 20634 WATTS

C/A'S=
EVAP COND AXIAL WATTS/CHRE
1 0 248

E R HES 2 A RESP. L/2 RESP. C A RESP. G 2 RESP
3 1151 283 1151 0

HOT FLUID GRAMS 67.0316 GRAMS
ROOM TEMP. VOLUME OF HOT FLUID GRAMS 65.0000 CM3

COLD FLUID GRAMS 72.0968 GRAMS
83.3351 CM3

HEAT PIPE: (MESH) & 2 MESH & 216.956 GRAMS

DELTA-T VALUES:

EVAP WALL EVAP LEG EVAP MESH EVAP EVAP VAPORIZATION
-13.5069 6.02608 0.52159 -10.0084 DEG C
VAPOR (H) VAPOR (A) VAPOR (G)
-7.06263+02 -8.63215+02 -3.63215+02

CONDEN. MESH MESH MESH MESH MESH
-24.2857+01 -3.53559 1.35559 -0.0219 DEG C

POWER OF 376 WATTS CAUSES ~~~~~~~~~~~~~~~~~ CAPILLARY LIMIT: DPL > DPV
LAST NON-LIMITED POWER CALCULATION WAS AT ~~~~~~~~~~~~~~~~~ 370 WATTS

~~~~~~~~~~~~~~~~ TOTAL DELTA-T = 9.82 DEG C
~~~~~~~~~~~~~~~~ TOTAL MASS = 0.268 IN
**USE CONDITIONS:**

**ILLUM = DOWNFACING A**
**WALL MAT = 304 SS**
**EVAP TEMP = 277**
**VAPOR DELTA-T = 80 DEG C**
**GRAY AND = 0.00**
**VTS AND = 0.00**
**DEV**

**EVAP LENGTH**
15.0000 IN. 43.0000 CM
**AIR LENGTH**
16.0000 IN. 43.0000 CM
**COIL LENGTH**
16.0000 IN. 40.0000 CM
**TOTAL LENGTH**
106.0000 IN. 269.0000 CM

0+50
0.0000 IN. 1.2700 CM
**WALL THICK**
0.0050 IN. 0.0127 CM
**GROOVE VMENTS**
0.0050 IN. 0.0127 CM
**GROOVE EXTRAN**
0.0050 IN. 0.0127 CM
**LAND VMENTS**
0.0050 IN. 0.0127 CM
**12 GROOVES (G2039Y) COVERED WITH 300 MESH**

**NOS LIMIT ENCOUNTERED AT**

---------------
**TOTAL DELTA-T**

---------------
**TOTAL MARS**

**LIST PERFORMANCE DETAILS (TH=19F) TH**

<table>
<thead>
<tr>
<th>TH</th>
<th>TH-A</th>
<th>TH-C</th>
<th>TH-E</th>
<th>DIVES/CM</th>
</tr>
</thead>
<tbody>
<tr>
<td>110</td>
<td>1100</td>
<td>1100</td>
<td>1100</td>
<td>1100</td>
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</tbody>
</table>

**DP**

**DP**

**DP**

**DP**

**DP**

**TOTAL DELTA-T**

**TOTAL MARS**

**SONIC LIMITS**

**TOTAL DELTA-T**

**TOTAL MARS**

**HOT FLUID GRAMS**

**COOL FLUID GRAMS**

**HOT FLUID GRAMS**

**COOL FLUID GRAMS**

**DELTA-T VALUES**

**EVAP WALL**

**EVAP Lg**

**EVAP MESH**

**EVAPORATION**

**VAPOR (E)**

**VAPOR (A)**

**VAPOR (G)**

**CONDENSATION**

**COND MESH**

**COND WALL**

**100 WATTS CAUSES**

**CAPILLARY LIMIT: DPL > DP**

**LAST NON-LIMITED POWER CALCULATION**

**TOTAL DELTA-T**

**TOTAL MARS**

---

**REPRODUCIBILITY OF THE ORIGINAL PAGE IS FORC**
**FLUID = SOUTHERN A**
**VAIL MATL = 304SS**
**EVAP TIME = 288**
**VAPOR DELTA-T = 60**

**GRAV AMR = 0.00**
**VOL AMR = 0.00**

**EVAP LENGTH** = 14.62211 in 45.0000 cm
**ADD LENGTH** = 18.89071 in 48.0000 cm
**COOL LENGTH** = 20.38015 in 178.0000 cm
**TOTAL LENGTH** = 183.28605 in 2282.0000 cm

- **0.2**
- **0.5000 in 12.7000 cm**
- **0.0000 in 0.00 cm**
- **0.0148 in 0.3780 cm**
- **0.0000 in 0.00 cm**
- **0.0000 in 0.00 cm**

12 GROOVES (GROOVE COVERED WITH 500 HSS)

**NO LIMIT ENCOUNTERED AT**

<table>
<thead>
<tr>
<th><strong>TOTAL DELTA-T</strong></th>
<th><strong>3.49 DEG C</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>TOTAL MASS</strong></td>
<td><strong>0.075 lb</strong></td>
</tr>
</tbody>
</table>

**VAPOR PERFORMANCE DETAILS (T or 3) 4C**

<table>
<thead>
<tr>
<th>T</th>
<th>T-3</th>
<th>4C</th>
</tr>
</thead>
<tbody>
<tr>
<td>288.0000</td>
<td>288.0000</td>
<td>288.0000</td>
</tr>
<tr>
<td>248.0000</td>
<td>248.0000</td>
<td>248.0000</td>
</tr>
</tbody>
</table>

**DPO** = 0.006
**DPE** = 0.006

**DPT** = 0.006
**DPT** = 0.006

**SONIC LIMITS:**

**EVAP** = 707.7
**ADD** = 6733.8

**Q/A** =

**NAT** =

**AXIAL** =

**VATTS/CH2**

**HOT FLUID CHARGE**

<table>
<thead>
<tr>
<th><strong>ROOM TEMP</strong></th>
<th><strong>VOLUME OF HOT FLUID CHARGE</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>58.3281</td>
<td>46.1836 GMS</td>
</tr>
</tbody>
</table>

**COLD FLUID CHARGE**

| **54.0000** |
| **GMS**     |
| **56.3280** |
| **GMS**     |

**HEAT PITA (HEAT) & 2 PEUDCAPS 209.031 GMS**

**DELTA - T VALUES:**

<table>
<thead>
<tr>
<th><strong>EVAP WALL</strong></th>
<th><strong>NAT WALL</strong></th>
<th><strong>EVAP MESH</strong></th>
<th><strong>EVAP MESH</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0317</td>
<td>0.0300000</td>
<td>0.0000</td>
<td>1.0000</td>
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</tbody>
</table>

**VAPO (1)**

<table>
<thead>
<tr>
<th><strong>VAPO (1)</strong></th>
<th><strong>VAPO (1)</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1230</td>
<td>0.1230</td>
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</table>

**CONDENSATION**

<table>
<thead>
<tr>
<th><strong>COND MESH</strong></th>
<th><strong>COND LOG</strong></th>
<th><strong>COND WALL</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0000</td>
<td>0.0000</td>
<td>0.0000</td>
</tr>
</tbody>
</table>

**POWER OF 245 WATTS CAUSES**

**CAPILLARY LIMIT: DPL > DPF**

**LACT NON-LIMITED POWER CALCULATION WAS AT**

<table>
<thead>
<tr>
<th><strong>TOTAL DELTA-T</strong></th>
<th><strong>3.49 DEG C</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>TOTAL MVE</strong></td>
<td><strong>0.075 lb</strong></td>
</tr>
</tbody>
</table>
### Test Conditions:

**FLUID** = MERCURY  
**VAPOR MAST=50GSS**  
**EVAP TEMP** = 634  
**VAPOR DELTA-T** = 50 DEG C  
**GAS AMB** = 0.00  
**VHR AMB** = 0.00 DEG

<table>
<thead>
<tr>
<th>PART</th>
<th>CODE</th>
<th>IN</th>
<th>IN</th>
<th>IN</th>
<th>IN</th>
</tr>
</thead>
<tbody>
<tr>
<td>EVAP LENGTH</td>
<td>16.0291</td>
<td>63.0000 CM</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>LLS</td>
<td>16.9291</td>
<td>63.0000 CM</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>GCV</td>
<td>63.2915</td>
<td>176.0000 CM</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TOTAL LENGTH</td>
<td>103.1400</td>
<td>226.0000 CM</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

- **O.D.** = 0.3250  
- **WALL THICKNESS** = 0.0000  
- **GROOVE WIDTH** = 0.0103  
- **31000 HOLE** = 0.0097  
- **LAD WHT** = 0.0000  
- **5 GROOVES (CLOVER)** = 0.0000

**NO LIMIT ENCOUNTERED AT**  
720 WATTS

**TOTAL DELTA-T** = 4.13 DEG C  
**TOTAL HRS** = 0.2800 HRS

### Watt Performance Details

**PE**  
-391523.07  
**PF-A**  
271.96  
**PC**  
-391523.07

**FE**  
620.07  
**FE-A**  
271.96  
**FG**  
620.07

**EVA TEMP**  
424  
**COND TEMP**  
620  
**DELTA-T**  
21

**EPO** = 114100  
**DUN** = 0  
**DPC** = 114100  
**DVER** = 0  
**EVPL** = 7771  
**EVTA** = 7771  
**DPLA** = 50000  
**DPTU** = 50000

**SOFT LIMITS:**  
**EVA** = 6464  
**AMM** = 10000 WATTS

**G/A** =  
**EVAP**  
13  
**COND**  
7777  
**AXIAL**  
8  
**WATT/CAM**  
2773

**HOT FLUID CHARGE**  
206.062 GRAMS  
**ROOM TEMP. VOLUME OF HOT FLUID CHARGE**  
15.5117 CM3

**COLD FLUID CHARGE**  
213.000 GRAMS  
15.7295 CM3

**HEAT PIPES (MESH)** & 2 EDCAPS  
76.483 GRAMS

### Delta-T Values:

<table>
<thead>
<tr>
<th>PART</th>
<th>CODE</th>
<th>IN</th>
<th>IN</th>
<th>IN</th>
<th>IN</th>
</tr>
</thead>
<tbody>
<tr>
<td>EVAP VALL</td>
<td>270618</td>
<td>5.1597</td>
<td>1.35007</td>
<td>100007</td>
<td>99.00 DEG C</td>
</tr>
<tr>
<td>VAPOR (A)</td>
<td>158519</td>
<td>5.4725</td>
<td>-7200000-01</td>
<td></td>
<td></td>
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<tr>
<td>CONDENSATION</td>
<td>2645872-01</td>
<td>350101</td>
<td>2954698</td>
<td>6630003-01</td>
<td>99.00 DEG C</td>
</tr>
</tbody>
</table>

**POWER OF 770 WATTS CAUSES**  
CAPILLARY LIMIT, DPL > DTY

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

**TOTAL DELTA-T** = 4.42 DEG C  
**TOTAL HRS** = 0.2800 HRS
RBM CONDITIONS

FLUID = AMERCURY
WALL MATT = 30453
EVAP TEMP = 600
VAPOR DELTA-T = 50 DEG C
GRAV. AM = 0.06
VTS AM = 0.00 DEG

EVAP LENGTH 16.93911 IN 43.00000 CM
AHX LENGTH 16.93911 IN 43.00000 CM
COOL LENGTH 0.923253 IN 1.840000 CM
TOTAL LENGTH 103.10000 IN 2568.00000 CM

0.40
0.25000 IN 0.63500 CM
WALL THICKNESS 0.10000 IN 0.25400 CM
GROOVE WIDE 0.1085 IN 0.27500 CM
GROOVE DEPT 0.0075 IN 0.19050 CM
LASH WIDE 0.0718 IN 0.18200 CM
6 GROOFS (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AT ------- 500 WATTS

TOTAL DELTA-T = 3.68 DEG C
TOTAL MASS = 0.281 KG

VAP PERFORMANCE DETAILS IT OR IT TT

<table>
<thead>
<tr>
<th>FE</th>
<th>PA</th>
<th>PC</th>
<th>IT T T</th>
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<tr>
<td>FE</td>
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<td>.230658E+07</td>
<td>.230658E+07</td>
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<tr>
<td>PA</td>
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<td>.230658E+07</td>
<td>.230658E+07</td>
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<td>PC</td>
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<td>.230658E+07</td>
<td>.230658E+07</td>
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</tbody>
</table>

IT = .08 
T T = .00 
T T = .00 

GROVE 4000 IN 0.00000 CM
3.60000 CM

SOON LIMITS

EVAP = 0.4544 AND = 0.4544 WATTS

C/R = 5
C/R = 5
C/R = 5

E R N Y F L O W

BOOM TEMP: VOLUME OF HOT FLUID CHARGE 12,4982 GRAMS
COLD FLUID CHARGE 122102 GRAMS
COLD FLUID CHARGE 122102 GRAMS
COLD FLUID CHARGE 122102 GRAMS
COLD FLUID CHARGE 122102 GRAMS

HEAT PIPE (XMM) & 2 MERCAPS 97.8000 GRAMS

DELTA-T VALUES:

<table>
<thead>
<tr>
<th>EVAP VAIL</th>
<th>EVAP LAE</th>
<th>EVAP MEE</th>
<th>EVAP FERRATION</th>
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<tbody>
<tr>
<td>.2525854</td>
<td>.2525854</td>
<td>.2525854</td>
<td>.2525854</td>
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<tr>
<td>.100896</td>
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VAPOR (A) | VAPOR (A) | VAPOR (A) |
<table>
<thead>
<tr>
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</tr>
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<tr>
<td>.2525854</td>
<td>.2525854</td>
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</table>

GOLDUXERATION CORD MEE | CORD LAD | CORD VAIL |
<table>
<thead>
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<th></th>
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</thead>
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<tr>
<td>.2525854</td>
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</table>

POWXY OF 618 WATTS CAUSE ----- CAPILLARY LIMIT, DPP = DPP

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

TOTAL DELTA-T = 3.68 DEG C
TOTAL MASS = 0.281 KG
### Reproducibility of the Original Page is Poor

#### Performance Details

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value 1</th>
<th>Value 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Delta-T</td>
<td>( \Delta T )</td>
<td>3.30 DEG C</td>
<td>0.20 DEG C</td>
</tr>
<tr>
<td>Total Mass</td>
<td>( m )</td>
<td>0.270 KG</td>
<td>0.270 KG</td>
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</tbody>
</table>

#### Heat Performance Details

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</thead>
<tbody>
<tr>
<td>Fluid</td>
<td>( F )</td>
<td>11.00929 + 0.06</td>
<td>11.01229 + 0.06</td>
</tr>
<tr>
<td>Delta-T</td>
<td>( \Delta T )</td>
<td>1.30 DEG C</td>
<td>1.30 DEG C</td>
</tr>
<tr>
<td>Total Mass</td>
<td>( m )</td>
<td>0.270 KG</td>
<td>0.270 KG</td>
</tr>
</tbody>
</table>

### Sonic Limits

<table>
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</tr>
</thead>
<tbody>
<tr>
<td>Delta-T</td>
<td>( \Delta T )</td>
<td>2.622 WATTS</td>
<td>2.621 WATTS</td>
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<tr>
<td>Total Mass</td>
<td>( m )</td>
<td>2.622 WATTS</td>
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###热液容量

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<td>油液温度</td>
<td>( T )</td>
<td>120.6976 DEG C</td>
<td>120.6976 DEG C</td>
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<tr>
<td>冷液容量</td>
<td>( m )</td>
<td>12.6222 GRAMS</td>
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### Delta-T Values

<table>
<thead>
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<th>Parameter</th>
<th>Symbol</th>
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<tbody>
<tr>
<td>Evaporation</td>
<td>( \Delta H_{evap} )</td>
<td>1.00025 DEG C</td>
<td>1.00025 DEG C</td>
</tr>
<tr>
<td>Condensation</td>
<td>( \Delta H_{cond} )</td>
<td>1.00025 DEG C</td>
<td>1.00025 DEG C</td>
</tr>
<tr>
<td>Vapor</td>
<td>( \Delta H_{vapor} )</td>
<td>1.00025 DEG C</td>
<td>1.00025 DEG C</td>
</tr>
<tr>
<td>Condensation</td>
<td>( \Delta H_{cond} )</td>
<td>1.00025 DEG C</td>
<td>1.00025 DEG C</td>
</tr>
<tr>
<td>Evaporation</td>
<td>( \Delta H_{evap} )</td>
<td>1.00025 DEG C</td>
<td>1.00025 DEG C</td>
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</table>

### Power of 450 Watts Causes

<table>
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<th>Parameter</th>
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<tbody>
<tr>
<td>Total Mass</td>
<td>( m )</td>
<td>0.270 KG</td>
<td>0.270 KG</td>
</tr>
</tbody>
</table>

88
FLUID = MERCURY
WALL MAT = 304SS
ATAP TEMP = 308
VAPOR DELTA-T = 50
R1333 C
GRAY AMM = 0.0
YTH AMM = 0.00

EVAP LENGTH 16.0291 IN 40.6000 CM
ADD LENGTH 16.0291 IN 40.6000 CM
CORD LENGTH 66.3913 IN 170.6000 CM
TOTAL LENGTH 108.1806 IN 275.0000 CM

0.5
0.0
VALL THEKES 0.0550 IN 0.0630 CM
GROOVE WIDTH 0.1020 IN 0.2590 CM
GROOVE HEIGHT 0.0540 IN 0.1370 CM
LAND WIDTH 0.1530 IN 0.3890 CM
3 GROOVES (CLOSED) COVERED WITH 200 MESH

NO LIMIT ENCOUNTERED AS 310 WATTS

TOTAL DELTA-T = 4.51 R1333 C
TOTAL MASS = 0.273 KG

VAPOR PERFORMANCE DETAIL (Y OR X) TK

PE
PE-A
PE-C
PE
64565P
440060
419962
419962

TE
TE-A
TE-C
21 C
TC
199.86
297.86

EVAP TEMP CORD TEMP DELTA-T
308
297.463
11.397

DPC= 124673
DPM= 0
DPC+DPM= 124673
DPM= 0

DPLS
8735 
DPM= 15082
43963

ETC
DPM= 123963

-1987
22906

SONIC LIMITS:
EVAP= 1052
ADD= 1146 WATTS

G/A= 3
EVAP
CORD
AXIAL
WATTS/KM2

3
0
996

K X REV
S A REV
LIQ REV
C A REV
C R REV

7
4232
284
4232

1

HOT FLUID CHARGE
159.29
GRAMS
ROOM TEMP. VOLUME OF HOT FLUID CHARGE
12.8447
CM3

COLD FLUID CHARGE
173.6905
GRAMS
12.8448
CM3

HEAT PIPE (MESH) & 2 END CAPS
99.2907
GRAMS

DELTA-T VALUES:

EVAP VALL
1535929

EVAP LAG
0.009704

EVAP HESE
7426609

EVAPORATION
-1000000

R1333 C

VAPOR (E)
1.515463

VAPOR (A)
1.515463

VAPOR (G)
-100375

CONDENSATION
CORD HESE
-244584.01
-192729

CORD LAG
-120971

CORD WALL
-3194333.01

R1333 C

POWER OF 390 WATTS CAUSES CAPILLARY LIMIT.

LAST R1333 LIMITED POWER CALCULATION WAS AT 388 WATTS

TOTAL DELTA-T = 5.41 R1333 C
TOTAL MASS = 0.273 KG

VAPT PERFORMANCE DETAIL (Y OR X) TK

89
**FLUID** = MERCURY  
**WALL MATT = 304 SS**  
**STAP TEMP = 277**  
**TAPOR DELTA-T = 50 DEG C**  
**GAS AXE = 0.00**  
**WTS AXE = 0.00 DEG**

<table>
<thead>
<tr>
<th>STAP LENGTH</th>
<th>14.925 IN</th>
<th>43,00000 CN</th>
</tr>
</thead>
<tbody>
<tr>
<td>ADD LENGTH</td>
<td>16.92925 IN</td>
<td>43,000 CN</td>
</tr>
<tr>
<td>CORD LENGTH</td>
<td>68.2813</td>
<td>175,0000 CN</td>
</tr>
<tr>
<td>TOTAL LENGTH</td>
<td>105.1500 IN</td>
<td>252,0000 CN</td>
</tr>
</tbody>
</table>

- 0.2#
  
- 0.2500 ED 0.6500 CN
- YALL TRIM 0.0085 ED 0.0085 G1
- GROOVE WIDTH 0.1085 ED 0.2780 CN
- GROOVE HEIGHT 0.0979 ED 0.0200 CN
- LADE WIDTH 0.1310 ED 0.2360 CN

| 3 GROoves (CLOSED) COVER1D W1TH 200 A1N|  |

**NO LIMIT ENCOUNTERED AT** 264 WATTS

**TOTAL DELTA-T** 8.35 DEG C

**TOTAL MASS** 0.275 LBS

**VAPOR PERFORMANCE DETAILS**

<table>
<thead>
<tr>
<th>PR</th>
<th>TB-A</th>
<th>21-3</th>
<th>PC</th>
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</thead>
<tbody>
<tr>
<td>268.646</td>
<td>240.555</td>
<td>224.896</td>
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<tr>
<td>273.819</td>
<td>273.365</td>
<td>266.560</td>
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<tr>
<td>277.266</td>
<td>268.646</td>
<td>8.25382</td>
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</table>

**DEG** 127.036  
**TB=** 0  
**DFP+DCF=** 127.036  
**DF1S/CH2**

**L1MTIC LIMITS**

<table>
<thead>
<tr>
<th>TB</th>
<th>G/1V</th>
<th>C/1V</th>
<th>T1</th>
<th>T2</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>654</td>
<td>ADD</td>
<td>650 WATTS</td>
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**G/A/V**

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<tr>
<th>8</th>
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<tbody>
<tr>
<td>308</td>
<td>3661</td>
<td>1</td>
</tr>
</tbody>
</table>

**HOT FLUID CHARGE** 170.167 GRAMS  
**COLD FLUID CHARGE** 17.034 GRAMS  
**VOLUME OF HOT FLUID CHARGE** 12.2816 CM3  
**WATER/CM2**

**BEAT PIPE (MESH) & 2 EDGAPS 99.297 GRAMS**

**DELTA-T VALUES**

<table>
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<th>TB AF</th>
<th>TB AF</th>
<th>TB AF</th>
<th>TB AF</th>
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<td>0.10098</td>
</tr>
<tr>
<td>TB AF</td>
<td>TB AF</td>
<td>TB AF</td>
<td>TB AF</td>
</tr>
<tr>
<td>1.1333</td>
<td>0.33217</td>
<td>0.33775</td>
<td>0.10098</td>
</tr>
<tr>
<td>TB AF</td>
<td>TB AF</td>
<td>TB AF</td>
<td>TB AF</td>
</tr>
<tr>
<td>1.1333</td>
<td>0.33217</td>
<td>0.33775</td>
<td>0.10098</td>
</tr>
</tbody>
</table>

**POWER OF 350 WATTS CAUSES**

**CAPILLARY LIMIT**  
**DEL = DFV**

**LAST NON-LIMITED POWER CALCULATION WAS AT** 340 WATTS

**TOTAL DELTA-T** 12.38 DEG C

**TOTAL M1ASS** 0.273 LBS
FLUID = MERCURY  
VALL WAT=304855 
VAP TMP = 434  
VAPOR DELTA-T = 50 DEG C 
VAP TML = 0.000  
VAP AMG = 0.000 CMB

VAP LENGTH 33.5686 IN  88.0000 CN 
AIR LENGTH 0.0000 IN  0.0000 CN 
GUID LENGTH 66.2135 IN  174.0000 CN 
TOTAL LENGTH 103.8000 IN  232.0000 CN 

NO LIMIT REACHED AT  
720 VOLTS 

TOTAL DELTA-T  2.80 Deg C 
TOTAL MASS  0.2200 Lb

VAPOR PERFORMANCE DETAILS (T OR H) ??

PS  PS= A  PA=G  TG  DPTHS/CM2
.3687213+07  .3687213+07  .3687213+07

TH  TH= A  TH=G  TG  DPTHS
432.39  432.216  432.216  432.216

VAP TEMP  CORR TEMP  DELTA-T
434  431.435  2.904

DPC= 114673  DPM  = 0  DPH=DPH= 114673  DPTHS/CM2

DFV= 16338  DFV=  DFV=  DFV=0
G331  16338  0  0

DFVG  DFVG  DFVG  DFVG
-3725  37745

SONIC LIMITS:  
VAP= 6946  A=10384  V=51534  VOLTS

G/A'9=  

3 R REF = 3 A REF  LIQ REF  C A REF  C R REF
G 7738  394  7738  3

HOT FLUID CHAMBER  
208.519  GRAMS 

COLD TEMP. VOLUME OF HOT FLUID CHAMBER 15.2267  CM3

COLD FLUID CHAMBER 213.553  GRAMS  
15.7416  CM3

HEAT PIPE (MESH) & 2 ENDGAPS 74.7047  GRAMS

DELTAT VALUES:

VAP TML  VAP LAG  VAP MESH  VAPORATION
-150896  .598019  .775944  .100998  DEG C

VAPOR (G)  VAPOR (A)  VAPOR (C)
.477073  .482831E-03  .483359E-01

COLD MESH  COLD LAG  COLD VALL
-.659133E-01  .379433  .232541  .561243E-01  DEG C

POWER OF 1650 VOLTS CAUSES  
CAPILLARY LIMIT: DPH > DFV

LAST NON-LIMITED POWER CALCULATION WAS AT  
1625 VOLTS

TOTAL DELTA-T = 5.47 DEG C 
TOTAL MASS = 0.2200 Lb
**XML CONDITIONS**

<table>
<thead>
<tr>
<th>FLUID</th>
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<tbody>
<tr>
<td>WALL MAT</td>
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<td>NAP TEMP</td>
<td>277</td>
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<tr>
<td>VAPOR DELTA-T</td>
<td>0.34</td>
</tr>
<tr>
<td>GRAY ARC</td>
<td>0.00</td>
</tr>
</tbody>
</table>

| EVAP LENGTH | 53.000 IN | 64.0000 CN |
| ADD LENGTH | 0.0000 IN | 0.0000 CN |
| COLD LENGTH | 26.0000 IN | 17.8000 CN |
| TOTAL LENGTH | 103.0000 IN | 225.0000 CN |
| VAP TEMP | 272.592 |
| COLD TEMP | 272.21 |
| DELTA-T | 0.38 |
| VAP-123075 | 0 | VAP-123075 | 123075 |
| COLD-0 | 0 | COLD-0 | 0 |
| NXT-2CH | 1 | NXT-2CH | 1 |
| H A H | 100 |
| R | 0 |
| Z | 3763 |
| 1 | 208 |
| 2 | 3765 |

**NO LIMIT ENCOUNTERED AT **

- TOTAL DELTA-T = 0.24 | DEG C
- TOTAL MASS = 0.27 | KG

**VAP PERFORMANCE DETAILS (T OR H) VT**

<table>
<thead>
<tr>
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<th>P-A</th>
<th>P-A</th>
<th>P</th>
<th>BURRELL/GME</th>
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<table>
<thead>
<tr>
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<th>P-A</th>
<th>P</th>
<th>BURRELL/GME</th>
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<td>272.592</td>
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<td></td>
</tr>
</tbody>
</table>

**SONIC LIMITS:**

- EVAP = 0.00 |
- ADD = 0.00 |
- VAP = 0.00 |
- COLD = 0.00 |
- AXIAL = 0.00 |
- 833 |

**TOTAL VOLUME:**

- 170.19 | GRAMS
- 12.035 | GRAMS
- 98.998 | GRAMS

**EVAP PIPE (NCHE) & 2 EHCAPS 98.998 GRAMS**

**DELTA-T VALUES:**

<table>
<thead>
<tr>
<th>DELTA-T</th>
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<tbody>
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<tr>
<td>LAG DELTA-T</td>
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<tr>
<td>MEE DELTA-T</td>
<td>0.24</td>
</tr>
<tr>
<td>EVAP DELTA-T</td>
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<td>H</td>
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<tr>
<td>0</td>
<td></td>
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<tr>
<td>2</td>
<td></td>
</tr>
</tbody>
</table>

**POWER OF 500 VATTS CAUSES**

- 300 | VATTS |
- 200 | VATTS |
- 100 | VATTS |

**TOTAL DELTA-T = 0.24 | DEG C**

**TOTAL MASS = 0.27 | KG**

92
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.

- $T_0 =$ desired heat pipe temperature
- $\Delta T =$ temperature drop down heat pipe
- $T = T_0 - \Delta T,$ actual heat pipe radiating temperature
- $A_0 =$ radiating area of heat pipe at $T_0$
- $A = A_0 + da,$ actual heat pipe radiator area required at $T$
- $Q =$ power to be radiated from heat pipe

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

\[
\frac{da}{dt} = \frac{A - A_0}{T_0 - T} \quad \text{Eq. A.1}
\]

but

\[
A = \frac{Q}{\varepsilon \sigma (T_0 - T)^4} \quad \text{and} \quad A_0 = \frac{Q}{\varepsilon \sigma T_0^4}
\]

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

\[
\frac{da}{dt} = \frac{A_0}{\Delta T} [(T_0/T)^4 - 1] \quad \text{Eq. A.2}
\]

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

\[
dl = l_c [(T_0/T)^4 - 1] \quad \text{Eq. A.3}
\]

where $l_c =$ condenser

but \( \frac{dl}{l_t} = \frac{dm}{m} \) where $l_t =$ total heat pipe length,

$m =$ mass, we obtain with substitution and rearrangement -

\[
\frac{dm}{l_t} = \frac{m l_c}{l_t} [(T_0/T)^4 - 1] \quad \text{Eq. A.4}
\]

which is Equation 3.2.