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HEAT PIPE RADIATOR FOR SPACE POWER PLANTS

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HEAT PIPE RADIATOR FOR SPACE POWER PLANTS[*]

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ABSTRACT

A heat pipe radiator which forms the ternary loop of a Rankine power system and furnishes meteoroid protection and fluid isolation of the secondary loop is discussed. The radiator design is usable over a broad range of power and its fabrication is well within current technology. A representative value of specific weight which includes feed lines, return lines, manifolds and heat pipes is ~1.1 kg/KWe considered for a 20,000 hour mission at 1100 °K with a probability of no critical penetrating hits of 0.99

INTRODUCTION

The Lawrence Radiation Laboratory Space Electric Program has as its general program objective the development of the technology applicable to advanced liquid metal cooled reactors for space electric power generation in the 1980's. The power range of interest is from hundreds of kilowatts to megawatts. Emphasis is on reactor concepts for use in conjunction with Rankine cycle power conversion, although reactor concepts for use with other promising conversion devices are also considered

The immediate goals of the program are:

1. To define and provide a technological base for a space power reactor capability in the hundreds-of-kilowatts power range and the multi-megawatt power range for use with a Rankine cycle and,

2. To describe a reactor in the hundreds-of-kilowatts power range (which would incorporate as many of the design features of the ultimate manned electric propulsion reactor as appropriate) and be suitable for possible ground demonstration in the mid-seventies

The exploration of space, either manned or unmanned, for earth orbiting stations, lunar exploration, or interplanetary travel missions requires that whatever power supply is chosen to effect the mission and whatever system is elected for energy conversion adequate provision must be made for the disposal of the considerable quantity of waste heat that is generated.

Whether the energy conversion system is thermoelectric, thermionic, Brayton, Rankine or MHD it is evident that a radiator will be one of the principal components of the system. It is further evident that in a system which uses a nuclear reactor as its power source a radiation shield will be another principal component. The size and mass of the shield required will be influenced strongly by both the platform and the relative location of the radiator. How well one has been able to optimize the radiator has a direct bearing on shield mass and system specific weight

This paper will direct its attention to the Rankine cycle work and to the radiators that are connected with it. The power levels that the Rankine cycle studies encompassed were (a) a low to intermediate power of 50 to 300 KWE and, (b) high power system of about 1 to 10 MWe. Both are considered to operate at moderately high fuel temperatures of ~1500 °K. Emphasis will be placed on the lower power level radiator and all analysis will revolve around it. This is because the lower power application is more immediate

A space power reactor, one which was termed SPR-4 by the Lawrence Radiation Laboratory, is shown in Figure 1. The general constraints in the design of the main radiator coupled to this reactor were:

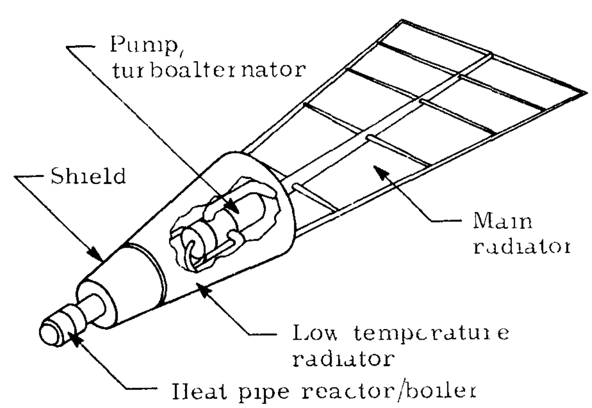

Fig. 1. SPR-4 Rankine Power System

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[*] Work performed under the auspices of the U S Atomic Energy Commission

[**] From the AEC Authorizing Legislation, 1968.

1. Mission time: 20,000 hours
2 Heat rejection: approx. 1500 kilowatts
3. Survival probability: at least 0.99
4. Rejection temperature: approx. 1000 K
5. Secondary loop fluid to to be cooled: Potassium

The Radiator Problem

Since conversion efficiencies for space power systems are at best around fifteen or twenty per cent, irrespective of the conversion method, the associated radiator must consist of large surface areas even with favorably high temperatures. For instance, at a radiator temperature of 1000 °Kelvin and an emissivity of about 0 9 maximum heat rejection rate to the zero degree space sink is only 5.0 watts for each square centimeter of radiator surface This is a small amount considering that the quantity of waste heat which is generated is many hundreds of kilowatts

The large surface area of the radiator plus its implied "thin skin", which is to be desired for efficient heat transfer, make it the most vulnerable to meteoroid damage of all the system components

The vulnerability of a radiator to meteoroid damage is, for the most part, due to the extensive exposed area of fluid filled passages or tubes from the turbine exhaust The consequence of fluid loss is loss of the mission

Conventional Radiators

The most elementary of radiator arrays (which could never be used in any practical case) would consist of a set of parallel, contiguous tubes through which the liquid from which heat is to be removed is caused to flow. Since the entire array contains fluid, the radiator has 100% vulnerability to critical meteoroid damage

To alleviate the vulnerability problem of full fluid radiators what is generally termed "conventional radiators" are introduced These employ thin, solid fins as extended surfaces between the fluid carrying tubes The purpose of the fins is to reduce the surface area occupied by flow passages and thus reduce the area vulnerable to critical penetration and loss of fluid The fins, as solid, metallic conductors, have limited effectiveness since, because of limited conductivity, they cannot be isothermal. This temperature degradation in the fins is a strong operator because of the fourth power relationship for radiant heat transfer Therefore, the total area of the radiator increases even though the fluid carrying area has been reduced The larger total platform area results in an increase in the nuclear shield size so the net gain is in doubt

A typical geometry for the meteoroid protected fin and tube radiator is shown in Figure 2 The dimensions are those generated for the high powered 10 MWe case in which the heat rejected was approximately 50 MW at an inlet fluid temperature of 1100 Kelvin. About 65% of the radiator weight was attributable to meteoroid protection Similar geometries apply to the lower power

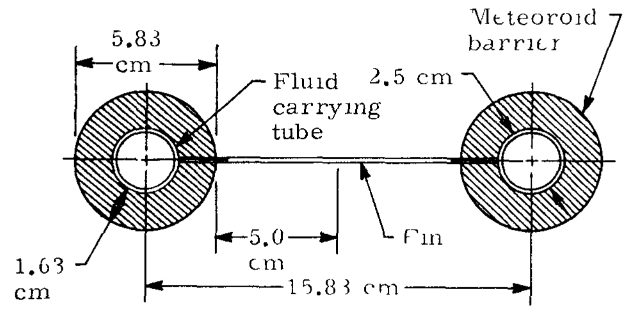

Fig. 2. Typical fin-tube geometry

Vapor Fin Radiators

To gain some weight saving over the fin and tube arrangement another approach has been proposed by NASA [1] This is one in which the solid, non-isothermal fin is replaced by what is termed a vapor fin A typical vapor fin geometry is shown in Figure 3.

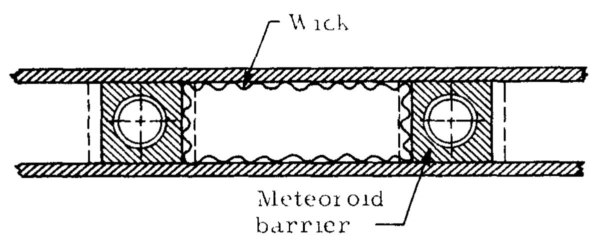

Fig. 3. Typical geometry of vapor fin radiator

This is a decided improvement over the solid fin since the chamber is isothermal There are, however, two principal heat transfer limits inherent in the vapor fin approach which seriously limits its applicability

1. The fluid carrying tubes are always separated from the vapor chamber by a meteoroid barrier thickness which is substantial and which creates a significant temperature drop between the fluid in the tube and the vapor in the fin. Since radiant heat rejection is a fourth power function of temperature, any temperature degradation between the fluid and the wall lowers the radiator effectiveness and creates a heavier system.

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[1] H. Haller, S Lieblein, and B Lindow, "Analysis and Evaluation of a Vapor Chamber Fin-Tube Radiator for High Power Rankine Cycles, "NASA-TN-D-2836, May 1965.

2. The area available for condensing heat transfer from the fluid to the tube is always limited to one half the tube circumference per vapor chamber. This seriously limits the total quantity of heat into the chamber and directly sets the maximum length of the vapor fin by the heat balance between condensing and radiant transfer Limiting the heat input cancels one of the vapor fins greatest potential virtues, that of being able to transport energy for long distances isothermally and thus provide large amplification of heat transfer area

The Heat Pipe Radiator

How does one resolve this dilemma of providing necessary meteoroid protection which does not sacrifice heat transfer capability? The solution lies in recognizing two important points First, meteoroid protection is inevitable and the thickness of the meteoroid barrier is a function of the total exposed area of all the fluid carrying pipes. Thus the thickness, t, of the meteoroid barrier is independent of the individual pipe diameter The fluid carrying pipe therefore might as well be sized so as to equate an internal pressure requirement for thickness with the meteoroid requirement. Pipes would thus tend to be large and carry large quantities of fluid Secondly, because a high mass flow can be accommodated in the pipe a large quantity of energy is available for removal. This suggests that instead of appending a vapor fin to the outside wall of the pipe that an evaporator section of a thin-walled heat pipe be placed inside the fluid carrying pipe The heat pipe will then be supplied with a large quantity of heat from a source which is well protected from meteoroids Heat transfer will be highly effective because of the thin wall heat pipe and at the same time the fluid carrying pipe will have a relatively low presented area because the mass flow of the fluid can increase as the square of the pipe diameter while the exposed area only increases linearly. This is truly a three loop system in which the secondary loop can be protected to whatever degree is necessary for the mission requirements without significant weight penalty and the ternary heat pipe loop can be designed around a highly redundant system approach

In this concept the inlet and outlet ducts originally used for vapor delivery and fluid return in the other systems become by suitable adjustment the primary leg of a heat exchanger One end of each of a large number of heat pipes is inserted into this ducting to a depth suitable for heat transfer (Figure 4) These heat pipes operate in parallel, functionally independent one from the other, and form the separate, secondary leg of the heat exchanger Because a large number of independent heat pipes are involved, they can be designed for a minimum weight for a specified probability that a certain fraction will remain unpunctured during the mission The ducting would carry the same amount of meteoroid protection as would be required for any radiator piping design This arrangement can be visualized in a first concept as a large U-tube out of which protrude the heat pipes in a plane array as shown in Figure 5

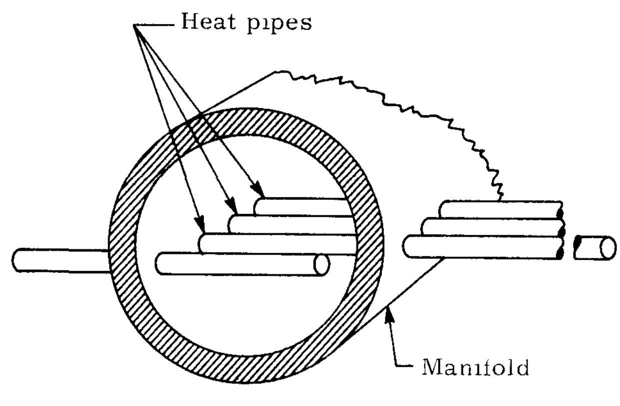

Fig 4 Heat pipes and duct

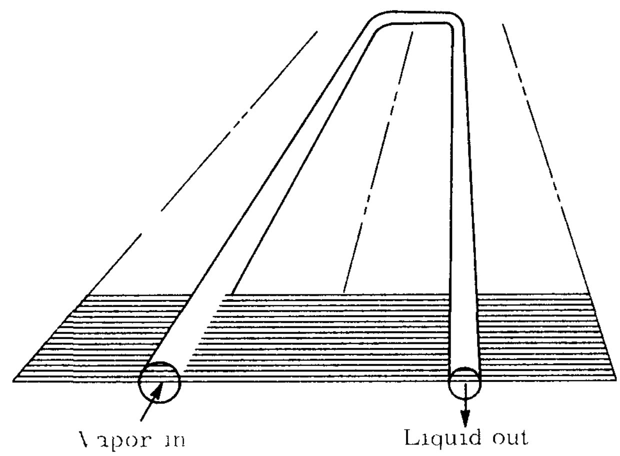

Fig 5 Heat pipe radiator concept

Since the thin-walled heat pipe is inserted directly into the meteoroid-shielded duct, the large temperature drop caused by conduction loss which was seen in the vapor fin is eliminated. Additionally, the area available for condensing heat transfer may now be set at whatever value is optimum

The only effect of the puncture of a single heat pipe is that it will cool and not contribute as a heat exchange member Since a large number are involved and redundancy can be incorporated, the loss of a considerable number can be tolerated

The Individual Heat Pipe Function and Equation

There has been a substantial amount of information published on heat pipes, how they are considered to function, and what some of their applications might be. The work of George Grover, who holds the patent on the heat pipe, and that of Ted Cotter, who did much of the original analytical investigations, are particularly valuable source materials. The heat pipe equation (1) which follows finds its origin in Ref. 2 and in Grover's heat pipe papers.[3]

A heat pipe, a nearly empty cavity, is a self contained thermal conductance device without moving parts which can transfer large quantities (kilowatts/cm^2) of heat as latent energy by evaporating a working fluid in a heating zone and condensing the vapor thus produced in a cooling zone. The heat pipe transports heat at "substantially" isothermal conditions. As shown in Figure 6 the

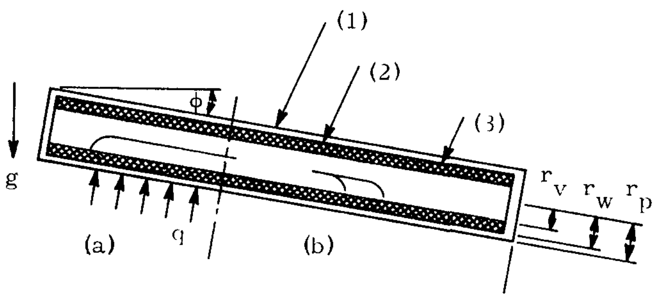

Fig. 6 Schematic of heat pipe

pipe has only three components (1) a container, (2) a capillary wick, and (3) a heat transfer fluid It has two principal regions (a) the evaporator, where heat is absorbed in the form of latent heat of vaporization, and (b) the condenser, where heat is rejected by condensation. It has one driving force (a) the pumping furnished by the capillary.

The necessary condition for heat pipe operation is that the capillary pumping force be equal to or greater than all the losses in the cycle. In general form:

Pressure rise due to capillary forces	≥	Pressure drop in the liquid
+ Pressure head in the liquid due to gravity	+	Pressure drop in the vapor

For pipes in which the capillary structure is a series of axial grooves of rectangular cross section covered with a single layer of mesh, the applicable equation is:

$$\frac{2\gamma\cos\theta}{r_c} \geq \frac{3\eta Q Z}{4b\pi L\rho_\ell r_v r_c^2} + \rho_\ell g Z \sin\phi + \frac{\left(1 - 4/\pi^2\right)Q^2}{8\rho_v r_v^4 L^2} \qquad (1)$$

Equation (1) assumes that the intervening wall between grooves is negligibly thin at the inner radius (the wall is of triangular rather than trapezoidal cross section), and that the groove half width is equal to the capillary pore radius, r_c.

Equation (1) can be expressed as

$$AQ^2 + D + \frac{BQ}{r_c^2} = \frac{C}{r_c} \qquad (2)$$

where

$$A = \frac{\left(1 - \frac{4}{\pi^2}\right)}{8\rho_v r_v^4 L^2}, \quad D = \rho_\ell g Z \sin\phi, \quad B = \frac{3\eta Z}{4b\pi L\rho_\ell r_v},$$

and $C = 2\gamma\cos\theta$

Solving for Q,

$$Q = \frac{-B \pm \sqrt{B^2 - 4Ar_c^2\left(Dr_c^2 - Cr_c\right)}}{2Ar_c^2}. \qquad (3)$$

Differentiating (2) with respect to r_c and setting dQ/dr_c equal to zero yields

$$\frac{B}{Ar_c^3}\left(B^2 - 4ADr_c^4 + 4ACr_c^3\right)^{1/2} \pm \frac{B}{Ar_c^3}\left(B + \frac{ACr_c^3}{B}\right) = 0, \qquad (4)$$

which may be rewritten

$$\left(r_c^3\right) + \left(\frac{4DB^2}{AC^2} r_c\right) - \left(\frac{2B^2}{AC}\right) = 0 \qquad (5)$$

For the case where the gravity term is zero (i e., in space or for $\phi = 0$), the optimum value for r_c is

$$r_{c,\,opt} = \left(\frac{2B^2}{AC}\right)^{1/3} \qquad (6)$$

and the axial power is

$$Q = \frac{-B + \sqrt{B^2 + 4Ar_c^3 C}}{2Ar_c^2} \qquad (7)$$

[2] T. P Cotter, "Theory of Heat Pipes." Los Alamos Scientific Laboratory, LA 3246-MA, 1965

[3] G. Grover, J Bohdansky, C Busse, "The Use of a New Heat Removal System in Space Thermionic Power Supplies"

For the special case of optimum r_c and maximum power,

$$Q_{max} = \frac{B}{A\left(r_{c,opt}\right)^2} . \quad (8)$$

The Heat Pipe Capability as a Radiator Element

Using these equations a computer code has been written which calculates various heat pipe properties, such as (a) axial heat flux, (b) optimum capillary radius, (c) Mach number of the vapor, (d) pressure drops, (e) mass flow rate, (f) vapor velocity, (g) radial and axial Reynolds number and (h) total power as a function of heat pipe length. The input data includes the desired operating temperature, the pipe diameter and the working fluid. A partial set of output curves for potassium at 1030 K generated by the code shown in Figure 7. This code is used as a precursor to the main HPRAD4 radiator design code which will be discussed subsequently As a utility tool this code provides some initial insight into the single heat pipe capability before integrating it into the full radiator package. Notice from Figure 7 for instance that a 1 5 cm diameter potassium fluid heat pipe is capable of transporting ~4 KW of power axially for a pipe length of 100 cm at a temperature of 1020 K The capillary radius r_c required to pump the 4 KW would be about 2.4×10^{-2} cm

COMPUTER DESIGN OF THE RADIATOR

The computer subroutine HPRAD4 is a fast-running code which designs a heat pipe radiator for a given set of input parameters Very little optimization is carried out internally, but optimum designs can be quickly found by calling the subroutine with different input data sets This discussion follows the order of calculations carried out by the code.

Input Data

The input data includes the mass flow rate of the working fluid to be condensed as well as its inlet temperature and quality and maximum allowable fractional pressure drop The input data also includes the mission time, the design survival probability, an overall condensing heat transfer coefficient for the working fluid, and the maximum allowable wall stress for the piping. The input data set is completed by specifying three heat pipe parameters: the inside diameter (including wicking structure), the heat pipe radial flux input, and the axial flux safety factor

Structural Material Properties

Structural material properties specified in the code are the density and modulus of elasticity for the heat pipe material, the manifold and supply line material, and the meteoroid barrier material. The thermal conductivity of the barrier material is also specified. These properties are assumed constant over the temperature range of the radiator.

Fluid Material Properties

The fluid properties required for computation are obtained from two fluid property subroutines developed at LRL· KVAP[4] and TRANP KVAP is a method of analytically correlating liquid metal properties in the saturated and superheated regions to experimental data by a least squares fit. The primary advantage of these correlations is that the relations are simple enough to preclude time consuming iterations for most of the properties The greatest RMS deviation of any of the correlations is 0.45 per cent. TRANP provides saturated liquid and vapor transport properties as a function of temperature for the principal liquid metals The continuous functions of TRANP were determined by fitting curves to data collected from several sources.

When the working fluid is specified these subroutines provide the following property data: inlet pressure and enthalpy, outlet pressure, temperature, and enthalpy; mean values, at $T_F = (T_{in} + T_{out})/2$, for the viscosity and specific volume of the saturated liquid, for the specific volume of the saturated vapor, and for the rate of change of the specific volume of the saturated vapor with respect to the pressure. Each heat pipe is considered to be exposed to the working fluid at its mean temperature T_F. The heat pipe temperature T_P (assumed to be essentially constant) is calculated as

$$T_P = T_F - \frac{Flux}{H}$$

where Flux is the heat pipe input flux and H is the condensing heat transfer coefficient. (The heat pipe input flux must be maintained below some value at which boiling in the wick structure becomes a danger. Potassium heat pipes have been successfully run at input fluxes in excess of 100 watts/cm^2 The input parameter "Flux" is generally chosen to be less than this number) After specifying the heat pipe fluid, the fluid property subroutines provide the following data: surface tension, heat of vaporization, pressure, viscosity and density of the saturated liquid, and viscosity and density of the saturated vapor.

Heat Pipe Design

The heat pipes are assumed to be right circular cylinders. The wick structure consists of axially directed rectangular grooves cut on the inside walls of the heat pipes The heat pipe equations used imply adjacent grooves (the groove separation is zero at the vapor duct radius) and a single layer of screen covering the grooves.

The ratio of the length of the heat pipe condenser section to the length of the evaporator section is calculated as

$$\frac{\ell_c}{\ell_e} = \frac{H\left(T_F - T_P\right)}{\sigma \epsilon F\, T_P^4} . \quad (9)$$

[4] N Brown and G Patraw, "Simple Relations for Thermodynamic Properties of Potassium Liquid and Vapor", Lawrence Radiation Laboratory, SPN-19, (unpublished)

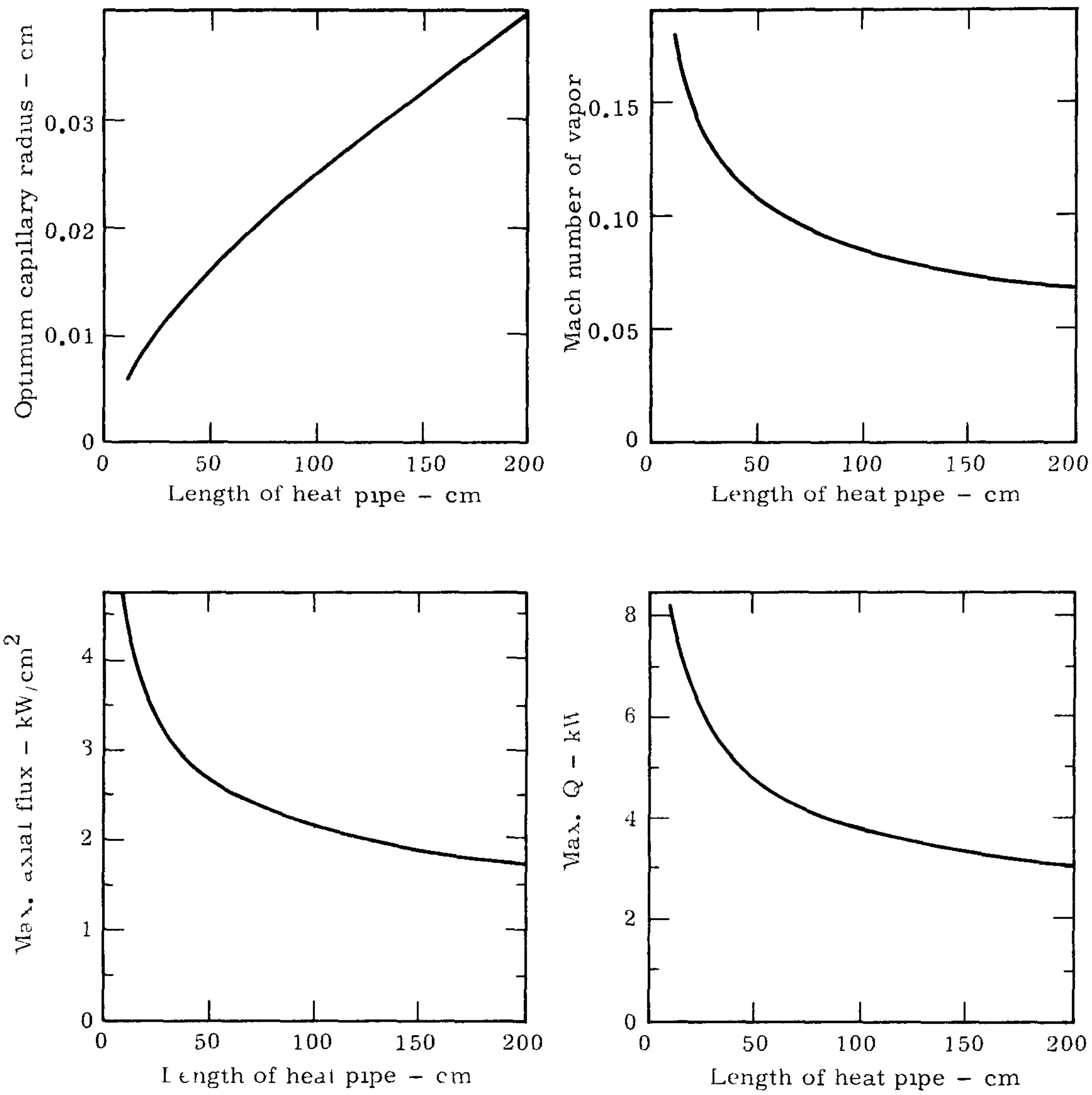

Fig. 7. Properties of a potassium filled pipe at 1020 K, 1.5 cm diam grooved pipe with one layer of mesh.

The vapor duct radius of the heat pipe, r_v, is chosen to be some fraction of the inside radius of the pipe, r_w. It can be shown by the differentiation of equation (8) with respect to r_v that an optimum heat pipe (in terms of maximum axial heat flow) is obtained for

$$r_v = \frac{5r_w}{6} . \qquad (10)$$

Because an off-optimum heat pipe may sometimes result in a system mass reduction, the r_v, r_w relation given by equation (10) should not be considered inviolable

An iterative procedure is carried out to determine the appropriate length for the heat pipe A length Z is guessed and the heat pipe equations (6) and (8) are used to calculate the optimum groove half-width, r_c, and the maximum axial heat transfer, Q_{max}. Q_{max} is then divided by the axial flux safety factor "SF."

$$Q = \frac{Q_{max}}{SF} \qquad (11)$$

Q is the design axial heat transfer for the heat pipe. SF has been varied between 1 and 4. An SF higher than unity serves two purposes. It results in a shorter heat pipe which reduces the meteoroid vulnerability, and it safeguards the heat pipe by selecting an operation level below the calculated maximum The heat into the heat pipe is now calculated as

$$Q_{in} = 2\pi r_w \ell_e H \left(T_F - T_P\right) . \qquad (12)$$

The evaporator length ℓ_e is obtained by combining equation (9) with the geometric relation

$$\ell_e + \ell_c = Z \,. \tag{13}$$

Q_{in} is now compared with the design axial heat transfer for the heat pipe Q. The guessed value for Z is revised until the two heats are identical. At this point the individual heat pipe is designed. The three heat transfer terms—convection to the pipe, axial transfer in the pipe, and radiation away from the pipe—are all equal.

The number of heat pipes necessary to condense all of the working fluid is calculated by dividing the total heat of condensation by the design heat transfer for the individual heat pipe.

The final heat pipe calculations are those concerned with meteoroid protection. As fully explained in the section on meteoroid criteria, the survival of the necessary number of heat pipes is assured through an appropriate tradeoff between individual pipe protection (by virtue of its structural wall) and the addition of redundant pipes.

Manifold Design

Having calculated the heat pipe dimensions as well as the total number of pipes N (including the redundant pipes), the code proceeds to design the manifold, in which the working fluid flows past the evaporator ends of the heat pipes The manifold is considered to be a right circular cylinder pierced by heat pipes as shown in Figure 8

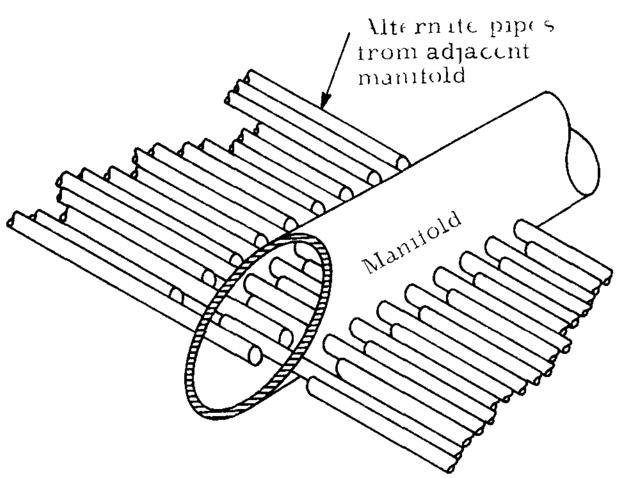

Fig 8 Manifold pierced by heat pipes

There is one heat pipe per axial plane in the manifold The condenser ends of the heat pipes (outside the manifold) form a plane. The evaporator ends of the pipes may be bent so as to present a staggered view to the working fluid flowing through the manifold. Given this arrangement, the manifold inside diameter and total length is straightforwardly calculated from the heat pipe diameter, heat pipe evaporator length, and total number of pipes

Next, the manifold is divided into parallel segments to reduce the pressure drop of the condensing working fluid. In order to insure symmetric radiator platforms only even numbers of segments are considered. The pressure drop in a manifold segment is calculated as described in the section on pressure drop using the working fluid properties, the dimensions of the manifold segment, and the mass flow rate per segment. The wetted perimeter S and the flow area A_c are calculated as

$$S = (\pi + 2)\, D_m \tag{14}$$

and

$$A_c = \frac{\pi}{4} D_m^2 - D_p D_m \tag{15}$$

where D_m is the inside diameter of the manifold and D_p is the outside diameter of the heat pipe. Manifold segmentation is continued until the calculated pressure drop is less than the maximum allowable value (The total allowable pressure drop is assumed to be equally divided between the manifolds, feed line, and return line)

Radiator Planform

The manifold segments are arranged as shown in Figure 9 and connected by feed and return lines In general, long narrow radiators are more desirable than short wide radiators because reactor shield size increases with radiator width. Therefore, in cases where the pressure drop criterion results in too few manifold segments for a longer than wide radiator, segmentation is continued until this geometric criterion is reached.

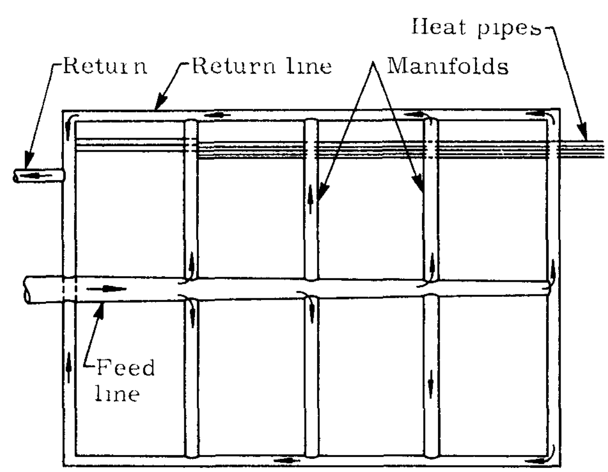

Fig. 9. Radiator planform schematic.

Feed and Return Line Design

The feed and return line lengths are easily calculated knowing the manifold and heat pipe dimensions. Then the lines are sized using the pressure drop criterion

1 The Feed Line

The feed line is assumed to carry the working fluid at a constant quality equal to its inlet quality. The pressure drop is calculated for a constant diameter pipe which carries the total flow rate of the working fluid. The diameter D_0 is increased until the pressure drop is less than the maximum allowable value.

After D_0 has been determined the feed line may be tapered since the mass flow rate decreases linearly as a function of position along its distribution length For a constant pressure gradient an approximate solution to the pressure drop relation for a linearly decreasing flow rate yields

$$D_{f\ell} = D_0\left(1 - \frac{x}{X}\right)^{0\ 375} \qquad (16)$$

where

$D_{f\ell}$ is the feed line diameter,
D_0 (as previously calculated) is the feed line diameter at the beginning of the distribution length,
x is the position along the distribution length,
X is the total distribution length.

2. The Return Line

The return line is assumed to carry the working fluid in an all-liquid state The pressure drop is calculated for a constant-diameter pipe Half the flow rate is assumed to travel the length of the radiator plus half its width The diameter of the line is increased until the pressure drop is sufficiently low. The return lines are not tapered because the resulting mass reduction would not be significant.

Line and Manifold Structural Wall Thicknesses

The line and manifold wall thicknesses t_i are calculated from the simple hoop stress relation using the inlet pressure of the working fluid P_{in}, the allowable stress σ as specified in the input, and the appropriate diameter D_i

$$t_i = \frac{P_{in}}{\sigma}\ \frac{D_i}{2} \qquad (17)$$

In some cases this calculation may result in an unrealistically low wall thickness. In such cases the thickness (and pipe mass) may be scaled up as desired

Meteoroid Protection of Lines and Manifolds

The lines and manifolds are now provided with a layer of armor of sufficient thickness to provide protection from meteoroids This application of the barrier thickness equations is discussed in the section on meteoroid criteria. In general, the meteoroid armor is considered to be of different material than the ducting it protects. However, the case of thick-walled, "self-protecting" ducting may be calculated by specifying equal material properties for the ducts and armor

Heat Rejection from Manifolds and Feed Line

The heat radiated directly to space from the armored manifolds and feed line is calculated The surface temperature of the armor used in this calculation is determined by considering three temperature drops· The working-fluid-to-pipe-wall temperature drop (using the condensing heat transfer coefficient), the pipe-wall-to-armor temperature drop (using a contact heat transfer coefficient), and the conduction temperature drop across the armor

The number of heat pipes may now be reduced because some of the heat (approximately 10%) is rejected by the ducting The code returns to the manifold design section and designs a radiator with an appropriately lower number of heat pipes. Revision of the number of pipes followed by a redesign of the manifolds and lines is repeated until the total heat rejection matches the total heat load.

Mass Calculations

The total radiator mass is calculated as the sum of the masses of six components· The heat pipes, the heat pipe fluid, the manifolds, the feed line, the return lines, and the meteoroid armor The working fluid inventory in the radiator is also calculated, but this mass is not included in the total radiator mass

A copy of a portion of the HPRAD4 code output which illustrates the design of a specific radiator is included in the appendix.

Discussion of Design Results

The heat pipe radiator considered in some detail was required to reject approximately 1 5 MW of heat. This heat rejection was accomplished through the condensation of an 0 96 kg/sec flow of potassium entering the radiator at a temperature of 1040 K and a quality of 85%. The radiator was designed for a 20,000 hour mission with a nominal survival probability of 99% The computer code HPRAD4 was used to investigate the effect of various parameters on heat pipe radiator mass. While the parameter variation was not exhaustive, enough has been done to specify a "near-optium" radiator design.

Heat Pipe Input Flux and Axial Flux Safety Factor

The heat pipe input flux and axial flux safety factor jointly determine the length of the individual heat pipe as well as its separation into evaporator and condenser sections. For a heat pipe of given length, higher input fluxes mean shorter evaporator sections and longer condenser sections. This can be advantageous because heat rejection goes up with heat pipe condenser area. Also, size and mass of the heat pipe manifold decrease as the heat pipe evaporator section becomes shorter. However, these advantages of a high input flux are eventually overtaken by the disadvantages: longer heat pipe condenser sections require thicker walls to provide the same meteoroid protection, and small manifolds require much segmentation which increases the amount of feed line piping. Thus there exists an optimum value for the heat pipe input flux. It can also be argued that there should be an optimum value for the axial flux safety factor. Given a ratio of condenser section length to evaporator section length, higher safety factors mean shorter heat pipes, but more of them. The shorter pipes may be constructed with thinner walls because of their reduced vulnerability to meteoroids. However, this mass reduction tends to be offset by the amount of manifolding required to accommodate the increased number of heat pipes. Thus there exists an optimum value for the axial flux safety factor.

Figure 10 shows the effect of the heat pipe input flux and axial flux safety factor on the total radiator mass. All of the results are for radiators using 0.75 cm-diam potassium-filled heat pipes. The piping is sized for a 5% pressure drop of the working fluid.

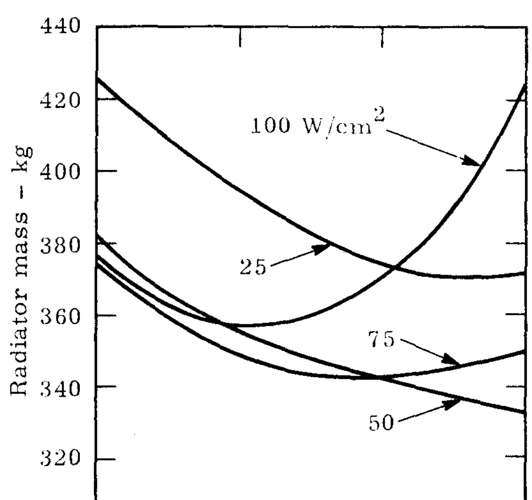

Fig. 10. The effect of radial flux and safety factor on radiator mass.

Heat Pipe Diameter

The total radiator mass is a strong function of the heat pipe diameter. As heat pipe diameter decreases the heat pipe length and required wall thickness increase. This results in a reduction in heat pipe mass, even though the number of heat pipes increases. Figure 11 shows the effect of heat pipe diameter on total radiator mass. The

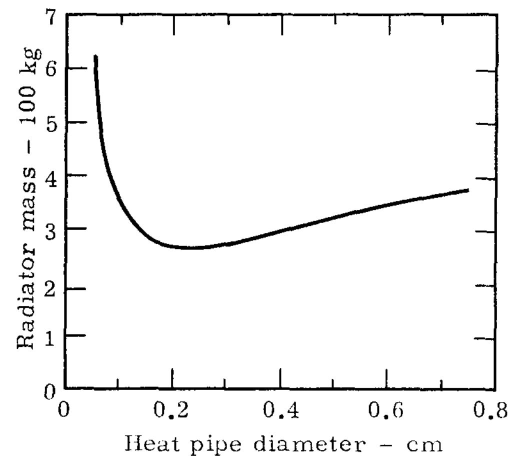

Fig. 11. The effect of heat pipe diameter on radiator mass (375 MWe case - 1568 kW rejected).

results shown are for potassium-filled heat pipes with an input flux of 50 W/cm^2 and an axial flux safety factor of 2 at the 375 kWe power level. It can also be shown that the heat pipe diameter minimum shifts to the right i.e., towards larger diameters, as power level increases. For example in the design of a much larger radiator, 10 MWe, the authors found a minimum radiator mass for a heat pipe diameter near 1.5 cm.

Heat Pipe Survival Probability

There is a tradeoff between individual heat pipe protection from meteoroids and the amount of redundancy included in the design. If N_s is defined as the number of heat pipes required for heat transfer then N_0 can be the number to start the mission. We select N_0 to have a value such that when the probability of no penetrating hits for other system components is set at some desired level such as 0.99, 0.999, etc. the probability of not exceeding the loss of $(N_0 - N_s)$ pipes will be the same or greater. That is, if $P(0)_{other\ comp} = 0.99$ then $P(X < (N_0 - N_s)) \geq 0.99$.

Radiator Survival Probability

The nominal value for overall radiator survival probability has been set at 0.99. Figure 12 shows the penalty which must be paid for survival probabilities in excess of 99%. The higher total mass at higher survival probabilities is due to the increased meteoroid armor on the manifolds and supply lines and the increased heat pipe wall thickness.

Pressure Drop Analysis

An influencing factor in the radiator planform is the allowable pressure drop in the manifolds which contain the heat pipes and in the feed and return lines. For this analysis it is assumed that there is two-phase condensing flow in the feed line and manifolds and that the return line fluid is all liquid.

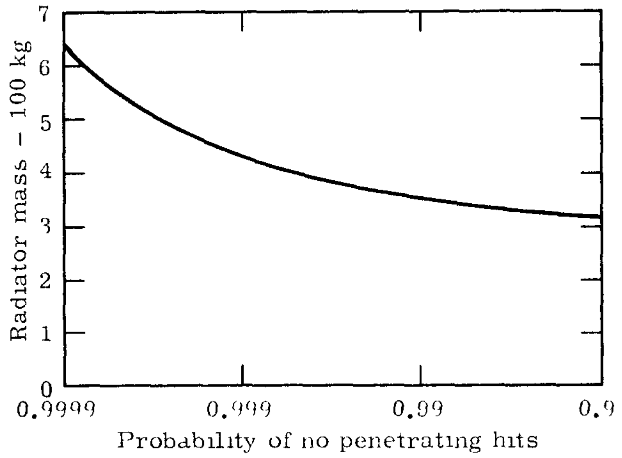

Fig. 12. The effect of survivial probability on the mass of the radiator.

The allowable pressure drop is taken as 5% of the inlet pressure, equally divided between feed and return lines and maniforlds. This is a somewhat arbitrary constraint based on the logic that the use of Owens' equation[5] for two-phase flow has greater validity the less the pressure drop.

METEOROID CRITERIA

The total mass of the radiator is the sum of the masses of the heat transport structure itself plus the not inconsiderable amount of meteoroid armor. The design of the space radiator is strongly influenced by the protection required against penetrating meteoroids. The data which are presently available on meteoroids are recognized as being imprecise, and the scaling laws which are used to determine penetration of structures by hypervelocity particles are estimates only and may be incorrect by a factor of 2 or 3

Flux Distribution

The latest meteoroid criteria for flux distribution are taken from a NASA report and two personal communications between NASA and LRL.[6,7] The meteoroid model used is a hybrid one consisting of the Whipple 1963A flux rate (which implicitly includes a meteoroid average density of 0.44 g/cc)[8] with modified values of meteoroid velocity and density. The latest NASA constants are

$$F_{>} = \alpha m^{-\beta} \tag{18}$$

where $\beta = 1.34$ and $\alpha = 5.30 \times 10^{-11}$ for $F_{>}$ in hits per square foot per day and m in grams, or $\alpha = 6.60 \times 10^{-15}$ for $F_{>}$ in hits per square meter per second and m in grams, $\overline{V} = 20$ km/sec, $\rho_p = 0.2$ g/cc

Penetration Equations

The general equation for expressing depth of penetration of hypervelocity particles into a plate of finite thickness can be expressed as

$$t = b\gamma d \left(\frac{\rho_p}{\rho_t}\right)^{\phi} \left(\frac{\overline{V}}{C}\right)^{\theta} . \tag{19}$$

An expression for thickness in terms of material properties is

$$t \propto E^{-1/3} \rho_t^{-1/6} . \tag{20}$$

Depth of penetration or required thickness can be expressed in terms of meteoroid mass and flux. Assuming a spherical meteoroid particle and substituting

$$d = \left(\frac{6}{\pi}\right)^{1/3} m^{1/3} \rho_p^{-1/3} \text{ cm} \tag{21}$$

into (19) gives

$$t = 1.5\gamma \left(\frac{6}{\pi}\right)^{1/3} \rho_p^{-1/3} \left(\frac{\rho_p}{\rho_t}\right)^{\phi} m^{1/3} \left(\frac{\overline{V}}{C}\right)^{\theta} \tag{22}$$

or

$$t = K m^{1/3} \left(\frac{\overline{V}}{C}\right)^{\theta} \tag{23}$$

where

$$K = 1.5\gamma \left(\frac{6}{\pi}\right)^{1/3} \rho_p^{-1/3} \left(\frac{\rho_p}{\rho_t}\right)^{\phi} \tag{24}$$

The average number of penetrating impacts, $\overline{H}$, which will occur on a vulnerable area A in mission time τ by the assumed flux distribution is:

$$\overline{H} = F_{>} A = \ln P(0). \tag{25}$$

Combining equations (23), (24), and (25) results in an expression for calculating the required thickness for vulnerable areas,

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[5]W. Owens, "The Phase Pressure Gradient," International Developments in Heat Transfer, Pt. II (American Society of Mechanical Engineers, New York, 1963), pp. 363-368.

[6]N. Clough and S. Lieblein, "Significance of Photographic Meteor Data in the Design of Meteoroid Protection for Large Space Vehicles", NASA-TN-D-2958, August 1965

[7]Letters from S. Lieblein, NASA, to C. Walter, LRL, Ref. 2720, December 10, 1965 and January 17, 1966

[8]F. L. Whipple, "On Meteoroids and Penetration", Interplanetary Missions Conference, 9th AAS Annual Meeting, Los Angeles, California, January 15, 1963

$$t = 1.5\gamma\left(\frac{6}{\pi}\right)^{1/3} \rho_p{}^{-1/3}\left(\frac{\rho_p}{\rho_t}\right)^{\phi}\left(\frac{\overline{V}}{C}\right)^{\theta}\left(\frac{\alpha A \tau}{\overline{H}}\right)^{1/4} . \quad (26)$$

Equation (26) as it is used in our computer code has been modified by substitution of the values of V, ρ_p, etc., which are known, and by change of units to produce

$$t = 6.15 \frac{1}{E^{1/3}\rho^{1/6}\overline{H}^{1/4}}\left(\frac{\tau}{10,000}\right)^{1/4} A^{1/4} . \quad (27)$$

Application of Barrier Thickness Equation

1. Heat Pipe Manifolds and Supply Lines

The heat pipe manifolds and the supply and return lines are protected from meteoroids by the addition of a layer of beryllium armor. The required armor thickness is calculated from equation (27) using as vulnerable area the total surface area of the manifolds and the supply and return lines. The barrier calculation is made without regard to the protection ability of the pipe walls themselves. The design survival probability P(0) is an input variable in the computer code.

2. Heat Pipes

It is undesirable to add a separate layer of meteoroid armor to the heat pipes because of the detrimental heat transfer effect. Thus, the heat pipes must depend on their wall structure to provide protection.

However, it is neither necessary nor desirable to require that the individual heat pipe survival probability P*(0) be as large as the overall design survival probability P(0). Instead, redundancy is introduced as a protection mechanism.

We calculate the number of heat pipes necessary for heat transfer purposes, NSUBS. This calculation will provide a first estimate of the vulnerable area A_s, and will furnish a thickness, t, of the heat pipe wall for meteoroid protection.

It will be assumed that we provide n extra heat pipes for redundancy and that we start the mission with (NSUBS + n) heat pipes. That is, we will let up to n pipes fail. Let $(N_s + n)$ = NSUBO.

We want n or NSUBO to have a value such that when the probability of no penetrating hits for other system components is set at some desired level such as 0.99, 0.999, etc., the probability of not exceeding the loss of n pipes will be the same or greater. That is, if $P(0)_{\text{other comp}} = 0.99$ then $P(<n) \geq 0.99$.

We could use the biomial distribution to calculate the required probabilities, i.e.;

$$1.0 - \sum_{x=0}^{n} b(x) = \binom{N}{x}\theta^x\left(1 - \theta^{N-x}\right) = \text{probability of} \quad (28)$$

having at least (N - n) left ≡ NSUBS left

However, the binomial is cumbersome to solve even for 1 case, much less a sum of perhaps several hundred. The normal distribution can be used to approximate the binomial with good accuracy for large NSUBS.

Let X = No. of survivors we actually have.

$$p = \text{probability of success} = e^{-\overline{H}}$$

and

$$P_r[X \geq \text{NSUBS}] \geq \gamma \quad (29)$$

We want to find the minimum NSUBO to satisfy (29).

Now the expectation of X is:

E(X) = NSUBO*p = expected number of survivors

and the standard deviation of X is:

$$\sigma(X) = \sqrt{\text{NSUBO*p*(1 - p)}}$$

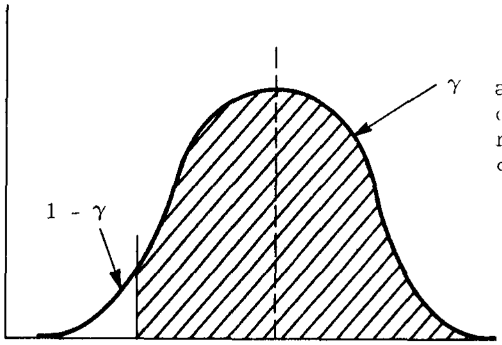

Fig. 13.

and the normal distribution equation is

$$\frac{\text{NSUBS - NSUBO}p}{\sqrt{\text{NSUBO}\,p\,(1 - p)}} = - \text{ZETA}. \quad (30)$$

If we solve for NSUBO

$$\text{NSUBO} = (\text{NSUBS} + \text{ZETA}\sqrt{\text{NSUBO*p(1 - p)}})p \quad (31)$$

ZETA = value from statistical tables which corresponds to the probability desired; for ex for 0.99 (0.990097) ZETA = 2.33 and for 0.999 ZETA = 3.1.

For computer use we set up the following:

$$\text{NSUBO} = (\text{NSUBS} + \text{ZETA(NSUBO*p(1 - p))}^{1/2})/p. \quad (32)$$

For an iteration technique we substitute v for the NSUBO on the right side of equation (32), then allow v to take on values starting at NSUBS and adjust until v = NSUBO which balances the equation.

Some Operating Limits on Heat Pipes

The equations which are used to calculate heat pipe properties make it possible to determine their capability analytically but say nothing about operational limits other than that of capillary pumping. Experimentally, heat pipes have shown promise of equaling or exceeding the calculated values of energy export. Figure 14 shows the good agreement between experimental and calculated values for heat pipe power achieved by J. Kemme of LASL.[9]

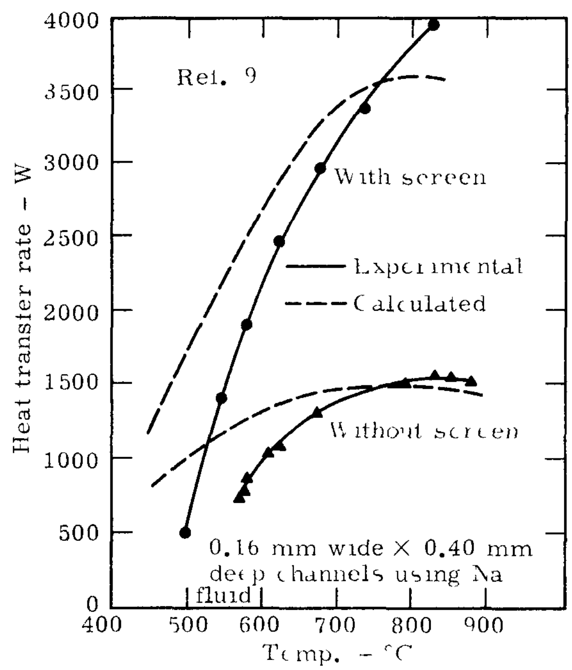

Fig. 14.

There are three operating limits in heat pipes, other than pumping, that must be recognized. These are:

1. Nucleate boiling in the wick structure producing burnout.

2. Reaching sonic velocity in the vapor, which will set an upper limit on the maximum axial flux, λ.

3. The existence of an interfacial shear stress between the liquid in the wick and the vapor such that as the counterflowing vapor velocity increases eventual entrainment of the liquid in the vapor will occur with probable catastrophic consequence to heat transfer.

[9] J. Kemme, "Heat Pipe Capability Experiments," Los Alamos Scientific Laboratory, LASL-3585-MS, October 1966.

Radial Flux Considerations in Heat Pipes

There is beginning to be built up a mythology about heat pipes and their isothermal qualities. Statements are made that temperature differences are "not discernible," "inconsequential," "of no significance," etc. This can be a seriously misleading concept. Temperature differences do exist and they exist by virtue of the fact that the heat into the pipe radially is by a conduction process across the tube wall and across the wick structure (see Figure 15). The driving force is by the temperature difference. The heat out, of course, is by the same process. Thus, although the vapor in the interior of the heat pipe may be isothermal, the external surface of the pipe cannot be. The greater the driving force the greater the ΔT. So, a caveat on "isothermalness" — in heat pipes — unless radial flux values are quite low, calculate the ΔT driving force. The significance of this lies in the fact that at some point this ΔT will be sufficiently large so that nucleate boiling will be initiated in the wick structure and failure of the heat pipe by burnout will occur.

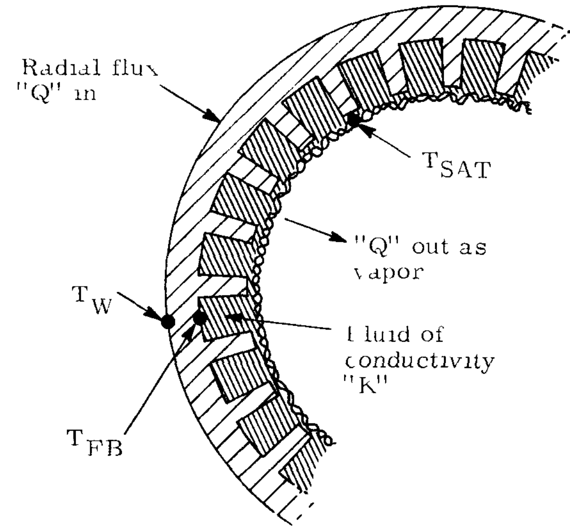

Fig. 15.

The onset of nucleate boiling is characterized by an equation of the form[10]

$$\Delta T = \frac{2\gamma T_{SAT}}{\rho_v L R_B} \qquad (29)$$

The equation represents an approximate expression for the superheat required ΔT for equilibrium of the bubble of radius R_B. Nuclei of radius greater than R_B should become bubbles and grow, those of smaller radius will collapse.

By setting R_B = 1 and solving equation (29) let us first look at the relative superheats for the liquid metals and water using water as a point of

[10] L. S. Tong, Boiling Heat Transfer and Two Phase Flow, J. Wiley & Son, 1965.

Liquid	BP-°K	Vapor Density gm/cm^3	Liquid Density gm/cm^3	Latent Heat of Vap. dyne cm/gm	Surface Tension dyne/cm	Thermal Cond. Liq. w/cm-K	Relative Super-heat	Relative δ
H_2O	373	$6*10^{-4}$	$9.6*10^{-1}$	$2.24*10^{10}$	$7*10$	$6.8*10^{-3}$	1	1
Potassium	1033	$4*10^{-4}$	$5.8*10^{-1}$	$1.94*10^{10}$	$6.6*10$	$3.5*10^{-1}$	4.5	232
Sodium	1165	$2*10^{-4}$	$6.6*10^{-1}$	$3.92*10^{10}$	$1.15*10^2$	$5.3*10^{-1}$	8.8	685
Rubidium	961	$1*10^{-3}$	1.25	$8.1*10^9$	$6.6*10$	$2.5*10^{-1}$	4.	14.7
Cesium	963	$3*10^{-4}$	1.4	$4.84*10^9$	$4.1*10$	$1.85*10^{-1}$	1.4	38.1
Lithium	1603	$6*10^{-5}$	$3.74*10^{-1}$	$1.94*10^{11}$	$2.41*10^2$	$6.8*10^{-2}$	17.2	1720

Table 1. Relative superheats for various heat pipe fluids

reference. The bubble size will not be considered since it is difficult to say what the distribution of nucleation centers and their size will be.

Table 1 is a compilation of data showing relative superheats. Aside from cesium it is apparent that any of the liquid metals can sustain much greater amounts of superheat than water.

The property data were evaluated at 1 bar.

For a given quantity of radial heat addition the relative thickness of the flow channel, δ_{rel}, will be a function of the relative superheat time the thermal conductivity. That is

$$\delta_{rel} = \frac{k * \Delta T_{rel}}{Q_{rad}}.$$

The last column of Table 1 shows this relationship. The very high values of relative δ for all metals except rubidium and cesium points out one of the major difficulties in using water as a fluid for heat pipes and hoping to correlate data from it with that of the liquid metals. It also makes it evident why sodium and lithium are such good heat pipe fluids.

The following Table 2 shows a comparison of surface superheat necessary for incipient boiling of sodium on stainless steel[11] for various surface conditions. Using data from Table 1, Table 2 has been extended to include the other liquid metals and water.

The function that is required to make use of this superheat limit in heat pipes can be developed as follows:

Let Q = axial power and λ = axial flux = Q/A then

$$\pi R\overline{V}^2\lambda = Q \qquad (32)$$

and the radial flux into the heat pipe is by heat balance

[11]Marto and Rohsenhow, "The Effect of Surface Conditions on Nucleate Pool Boiling Heat Transfer to Sodium," MIT Rpt. 5219-23, January 1965.

Surface Condition	Limiting Superheat - °CΔt Na	Cs	Rb	K	H_2O	Li
Mirror	70	11	32	36	8	136
Lap	40	6.3	18	20	4.5	78
Porous Weld	40	6.3	18	20	4.5	78
Doubly re-entrant Cavities	25	4.	11	13	2.8	49
Porous Coating	10	1.7	4.8	5.4	1.2	21

Table 2. Superheat necessary for incipient boiling.

$$Q_{RAD} = \frac{\pi RV^2\lambda}{2\pi RWZ_{EVAP}} \qquad (33)$$

It can be shown that the optimum RV/RW is 5/6 and by the conduction equation

$$Q_{RAD} = \frac{k\Delta T}{\delta} = \frac{5RV\lambda}{12Z_{EVAP}} \qquad (34)$$

or

$$\delta = \frac{12k\Delta T\ Z_{EVAP}}{5RV} * \frac{1}{\lambda} \qquad (35)$$

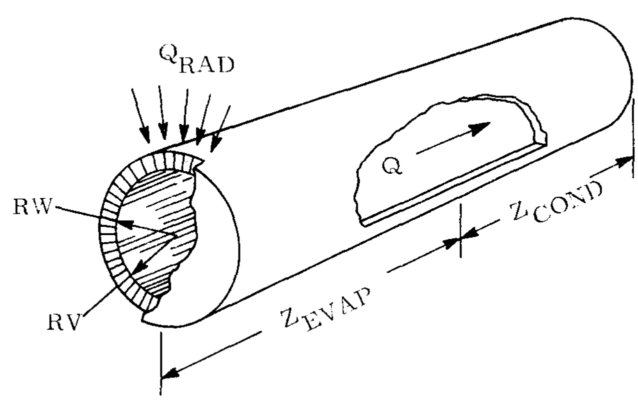

Fig. 16.

For the case where $\delta \longrightarrow 0.2$ RV as a limit, equation—can be rewritten as

$$\lambda = \frac{12KZ_{EVAP}}{RV^2} * \Delta T. \qquad (36)$$

Sonic Velocity in the Vapor

Consider next the vapor flow as limiting heat pipe operation. The maximum vapor flow rate will occur in the evaporator:

$$\dot{m}_\ell = \dot{m}_v = \frac{Q}{L}\rho_v V_v A_v = \rho_\ell V_\ell A_\ell \qquad (37)$$

and if we set V_v = c, the sonic velocity of the vapor, then the axial flux λ at Mach 1 is

$$\lambda = Q/A_V = L\rho_v c. \qquad (38)$$

It is interesting to calculate the values of λ for different candidate fluids at a temperature for a particular case of interest to radiator applications of heat pipes. Table 3 shows this relationship at ~1100 K.

	Sodium	Potassium
Density gm/cm^3	$1.5*10^{-4}$	$8*10^{-4}$
Latent heat of vapor dyne-cm/gm	$3.98*10^{10}$	$1.88*10^{10}$
Sonic velocity cm/sec	$5.72*10^{4}$	$4.55*10^{4}$
Axial flux KW/cm^2	34	68

Table 3. Axial heat flux at Mach 1 and constant temperature

Typical axial fluxes for currently anticipated radiator application of heat pipes are < 2.5 KW/cm^2 so it is evident that sonic velocity will never be reached under the conditions imposed in Table 3. However, the saturated vapor density is so strongly dependent upon temperature that one must be cautious against assuming that sonic velocity will never be a problem. Figure 17 shows a plot of axial flux vs temperature for constant vapor velocity equal to Mach 1. Lithium has been added as an interesting fluid at temperatures starting ~1400 K.

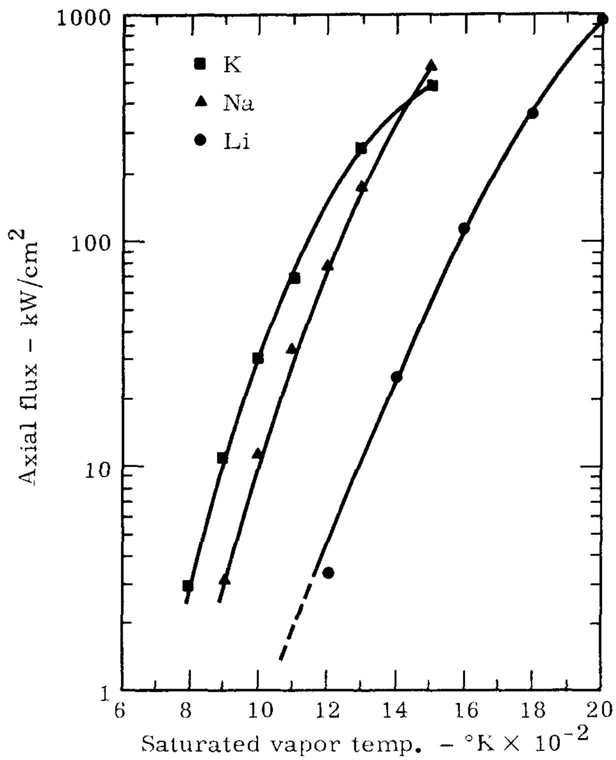

Fig. 17.

Entrainment of the Liquid in the Vapor

In the heat pipe system, which presupposes a countercurrent flow between liquid and vapor, it is possible that as the vapor velocity increases there will be a point reached at which the smooth laminar flow of the liquid film will begin to be distorted and have large waves created on it by the vapor velocity influence. Eventually, entrainment of the liquid will occur and concurrent flow will begin. This condition is known as flooding. There are available various empirical equations and experimental results which purport to allow calculation of flooding incidence.[12] For purposes of this report flooding incidence is assumed not to be a problem for two reasons. First, the single layer of mesh on top of the fluid channels provides a high degree of flow separation and secondly, the vapor velocities at the axial fluxes used are not particularly high. For a more quantitative approach the interested reader is referred to the Collier and Wallis reference.[12]

CONCLUSIONS AND RECOMMENDATIONS

The heat pipe radiator as proposed in this paper has a number of distinct advantages. These are:

1. It provides a genuine three loop system for space power application and allows the secondary, fluid carrying loop complete isolation and protection from meteoroids to whatever degree is necessary to satisfy the mission.
2. It provides extremely effective heat transfer because the heat pipes are directly inserted into the secondary loop heat exchanger.
3. The radiator has a very low specific weight over a broad power range.
4. All the materials that are used are well within current technology and fabrication can be done in modular units.
5. It has no moving parts and in modular units can be pre-tested easily.

[12] J.G. Collier & G.B. Wallis, Two Phase Flow Heat Transfer Notes for a Summer Coarse, July 24-Aug. 4, 1967, M.E. Dept., Stanford Univ.

NOMENCLATURE

A = vulnerable area, m^2 [ft^2 in equation (27)]

A_c = flow area, cm^2.

b = groove height - r_w - r_v, cm.

b = thin plate adjustment constant - 1.5

C_p = specific heat at constant pressure, W-sec/g-°K.

C = sonic velocity of target material

$$C = 12 \sqrt{\frac{E g_c}{\rho_t}} \text{ ft/sec,}$$

ρ_t inside sonic velocity term is in lbm/ft^3.

d = projectile diameter, cm.

D = pipe diameter, cm.

E = elastic modulus, lb/ft^2

f = Darcy-Weisbach friction factor.

F = heat pipe view factor ($2/\pi$ for close-packed tubes)

g = acceleration of gravity, cm/sec^2

g_c = gravitational constant.

G = mass flow rate per unit area, g/cm^2-sec.

H = condensing heat transfer coefficient, W/cm^2-°K.

$\overline{H}$ = -ln P(0), average number of penetrations on area A in time τ

K = thermal conductivity, W/cm-°K.

ℓ = axial pipe coordinate, cm

ℓ_c = length of heat pipe condenser section, cm

ℓ_e = length of heat pipe evaporator section, cm

L = latent heat of vaporization, dyne-cm/g

m = meteoroid mass, g

N = total number of heat pipes.

NSUBO = total number of heat pipes

NSUBS = number of heat pipes surviving

p = pressure, bars.

P(0) = survival probability.

p = probability of success

q, Q = heat flow, W or dyne-cm/sec depending on the units of the equation

r = radius, cm.

r_c = capillary radius (groove half-width), cm.

r_v = radius of vapor conduit, cm

r_w = internal radius of heat pipe including wicking, cm.

S = wetted perimeter, cm

t = thickness, cm [ft in equation (27)]

T = temperature, °K.

v_g = vapor specific volume, $cm^{3/g}$

v_ℓ = liquid specific volume, cm^3/g

$\overline{V}$ = mean meteoroid velocity, km/sec.

w = mass flow rate, g/sec

X = fluid quality.

Z = vapor fin tube length, cm

Z = heat pipe length, cm

ZETA = value of the normal standard deviation.

α = meteoroid flux constant, g^β/m^2-sec

β = constant in meteoroid equation = 4/3

γ = surface tension, dyne/cm

γ = material proportionality constant in meteoroid equation, ranging from 1.5 to 3.0

ϵ = emissivity

η = viscosity, g/cm-sec

θ = wetting angle. constant in heat pipe equation = 0

θ = constant in meteoroid equation = 2/3

μ = viscosity, g/cm-sec.

ρ = barrier material density, lbm/ft^3

ρ_ℓ = liquid density, g/cm^3

ρ_p = projectile density, g/cm^3

ρ_t = target density, g/cm^3, except lbm/ft^3 in sonic velocity term C

ρ_v = vapor density, g/cm^3

σ = Stefan-Boltzmann constant = 5 67 * 10^{12} W/cm^2-°K^4.

σ = stress, bars

τ = mission time, sec [hr in equation (27)]

ϕ = angle with horizontal

ϕ = constant in meteoroid equation = 1/2.

APPENDIX

```
        HPRAD4,HEAT PIPE RADIATOR

INPUT DATA
RADIATOR INLET TEMP,DEG K= 1.040E+03
FLOW RATE,G/SEC= 9.970E+02
INLET QUALITY= 8.200E-01
MAX FRACTIONAL P DROP OF PRIMARY= 5.000E-02
LIFETIME,HR= 2.000E+04
SURVIVAL PROB= 9.900E-01
HEAT COEFF,W/CM/CM/DEG K= 4.250E+00
MAX INPUT FLUX,W/CM/CM= 5.000E+01
AXIAL FLUX SAFETY FACTOR= 2.000E+00
HEAT PIPE ID INCLUDING WICKING,CM= 7.500E-01
PRIMARY PIPING STRESS,BAR= 3.500E+02
HPP,AVERAGE HITS PER HEAT PIPE= 1.000E-01

     END OF INPUT DATA


MATERIAL DENSITIES,G/CM**3
      HEAT PIPE= 8.560E+00
      MANIFOLD AND LINE= 8.560E+00
      BARRIER= 1.860E+00

MATERIAL MODULUS OF ELASTICITY,DYNE/CM**2
      HEAT PIPE= 1.100E+12
      BARRIER= 2.100E+12

BARRIER CONDUCTIVITY,W/CM/K= 6.000E-01
MANIFOLD-BARRIER INTERFACE HEAT COEFF,W/CM**2/K= 5.670E+00

AVERAGE PRIMARY FLUID PROPERTIES
LIQ VIS,G/SEC/CM= 1.161E-03
LIQ DENS,G/CM**3= 6.618E-01
VAP DENS,G/CM**3= 5.374E-04
DVGDP,CM**5/G/DYNE=-1.541E-03
ENTHALPY CHANGE,DYNE CM/G= 1.572E+10

HEAT PIPE FLUID PROPERTIES
SURF TENS,DYNE/CM= 1.297E+02
LIQ VIS,G/SEC/CM= 2.967E-03
VAP VIS,G/SEC/CM= 2.124E-04
LIQ DENS,G/CM**3= 7.716E-01
VAP DENS,G/CM**3= 8.074E-[illegible]
HEAT OF VAP,DYNE CM/G= 4.046E+10

HEAT REJECTION REQUIRED,KW= 1.567E+03

HEAT PIPE TEMP,DEG K= 1.025E+03
BARRIER SURFACE TEMP= 1.02[illegible]E+03

CALCULATED HEAT REJECTION,KW= 1.571E+03
HEAT REJECTION BREAKDOWN,PERCENTS
      HEAT PIPES= 90.67     MANIFOLDS=  6.61     FEED LINE=  2.71

PRESSURE IN BARS
HEAT PIPE       PRIMARY IN      PRIMARY OUT
 9.048E-01       1.112E+00       1.056E+00

D VAPOR          D WICK          TH PIPE         L EVAP          L COND          R CAP
 6.2[illegible]0E-01       7.500E-01       6.667E-02       5.232E+00       7.282E+01       1.029E-02

DI MAN           L MAN           GROOVES 95      PIPES 2203
 5.300E+00       1.946E+03

MASS FLOW IN HEAT PIPE,GM/SEC= 1.794E-01
RADIAL REYNOLDS NO= 2.570E+01     (ANALYSIS VALID ONLY IF MUCH GREATER THAN ONE)
```

MAX HEAT PIPE WALL STRESSES,BARS
COMPRESSION OF PUNCTURED PIPE= 7.364E+00
TENSION OF WHOLE PIPE= 5.089E+00

PARALLEL MANIFOLD SEGMENTATION

SEGMENTS	REY NO	FR P DROP	SEGMENT LENGTH
10	2.321E+04	-2.457E-02	1.946E+02

ORIENTATION,AXIAL FEED LINE,LATERAL MANIFOLDS

FEED LINE ID,CM= 1.472E+01   FR P DROP= 1.277E+02   TOTAL LENGTH,CM= 3.903E+02
RETURN LINE ID,CM= 3.000E+00   FR P DROP= 1.113E-02   TOTAL LENGTH,CM= 1.170E+03

NOTE,FEED LINE ID IS AT ENTRANCE,DISTRIBUTION LENGTH IS TAPERED AS (1-X/L)**0.375

PIPE WALL THICKNESSES,CM

MANIFOLD	FEED LINE	RETURN LINE
8.425E-03	2.337E-02	4.764E-03

METEOROID BARRIER THICKNESS,CM= 5.759E-01

RADIATOR MASSES IN KG

HEAT PIPES	H PIPE LIQ	MANIFOLDS	FEED LINE	RETURN LINES	BARRIER
2.707E+02	1.621E+01	2.343E+00	2.631E+00	4.503E-01	7.441E+01

TOTAL MASS= 3.667E+02

MASS PERCENTS

H PIPE	HP LIQ	MAN	FEED	RETURN	BARRIER
73.81	4.42	0.64	0.72	0.12	20.29

RAD LENGTH,CM= 4.684E+02   RAD WIDTH,CM= 4.099E+02

PRIMARY FLUID MASS,KG= 8.024E+00
DISTRIBUTION OF PRIMARY FLUID,MASS PERCENTS

FEED	MAN	RETURN
0.31	31.49	68.20

HT PIPES= 7.381E+01 PERCENT OF AREA

DISTRIBUTION

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