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LIVERMORE

HEAT PIPE RADIATOR DESIGN

UNCLASSIFIED

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~~J. Hadley / File~~

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ABSTRACT

The design of a heat pipe radiator for SPR-6 is presented. A wide range of parameters may be quickly evaluated through the use of the computer code HPRAD4. A "near-optimum" design yields a 46 Mw radiator with a mass of 12,000 kg.

INTRODUCTION

The heat pipe radiator has been proposed as a solution to the problem of providing adequate meteoroid protection to the fluid carrying parts of a space radiator without sacrificing heat transfer capability (see SPN-198, Werner). The heat pipe radiator consists of an appropriate length of fluid carrying manifold into which one end of each of a large number of heat pipes is inserted to a depth suitable for heat transfer. The heat from the condensing manifold fluid is transferred almost isothermally along the heat pipes and radiated to space. Because a large number of independent heat pipes are involved, they can be designed for a minimum mass by an appropriate tradeoff between individual pipe protection and redundancy[1]. The manifolding carries the same amount of meteoroid protection as would be required for any radiator piping design.

HEAT PIPE OPERATION AND HEAT TRANSFER

The advantages of introducing the heat pipe principle to produce space radiators of low mass and minimum planform area lies in its unique feature of being a lightweight, geometrically adaptable, isothermal device capable of transporting large quantities of energy.

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[1] R. English and D. Guentert, "Segmenting of Radiators for Meteoroid Protection", ARSJ Vol. 31, No. 8, August 1961, pp. 1162-1164.

A heat pipe, a nearly empty cavity, is a self-contained, thermal conductance device devoid of moving parts which transfers 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. The 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. Fluid return is via the wick and capillary pumping.

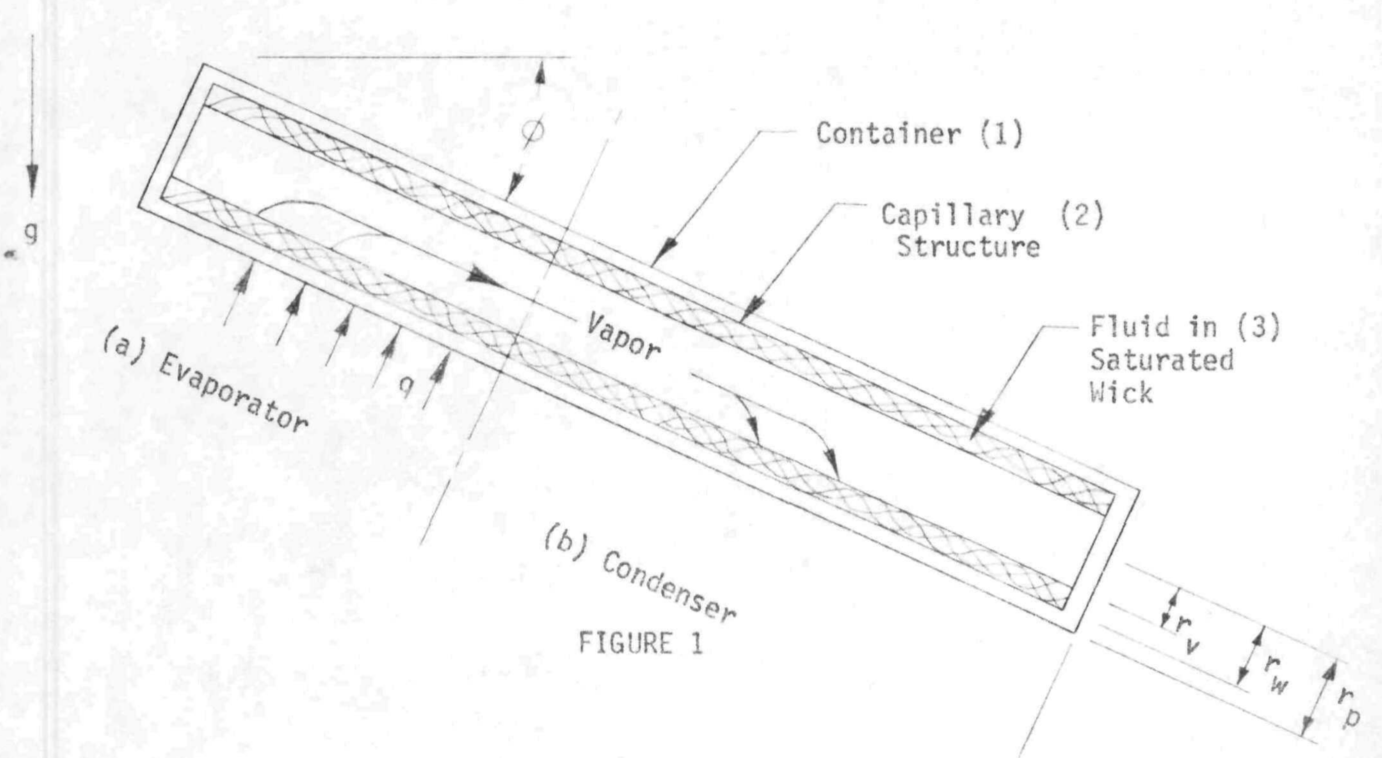

FIGURE 1

The necessary condition for heat pipe operation is that the capillary pumping force be equal 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 LIQ. DUE TO GRAVITY	+	PRESSURE DROP IN THE VAPOR

For pipes in which the capillary structures is a series of axial grooves or channels covered with a single layer of mesh the applicable equation is:[2]

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

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).

Equation (1) can be expressed as

$$AQ^2 + D + \frac{BQ}{r_c^2} = \frac{C}{r_c} \qquad (1A)$$

where

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

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[2]G. M. Grover, J. Bohdansky, and C. A. Busse, "The Use of a New Heat Removal System in Space Thermionic Power Supplies", EURATOM EUR-2229e (1965).

Solving for Q

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

Differentiating (1A) with respect to r_c and setting $\frac{dQ}{dr_c} = 0$ yields

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

which may be rewritten

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

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

$$r_{c_{opt}} = \left(\frac{2B^2}{AC} \right)^{1/3} \qquad (4)$$

and in the absence of gravity the axial power is

$$Q = \frac{- B + \sqrt{B^2 + 4Ar_c^3 C}}{2Ar_c^2} \qquad (5)$$

For the special case of optimum r_c and max power

$$Q_{max} = \frac{B}{Ar_{c_{opt}}^2} \qquad (6)$$

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 input flux, and the axial flux safety factor. These last two variables will be explained in the heat pipe design discussion.

Material Properties

Material properties are required for the structural materials and for the working fluid and heat pipe fluid. The structural material properties are assumed to be essentially constant over the temperature range of the radiator. The properties required 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. The fluid properties are obtained from two fluid property subroutines, KVAP and TRANP. After specifying the working fluid these subroutines provide the following property data: inlet pressure and enthalpy; outlet pressure, temperature and enthalpy; mean values $\left(\text{at } T_F = \left(T_{in} + T_{out}\right)/2\right)$ 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

$$T_p = T_F - \frac{\text{Flux}}{H} \tag{7}$$

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. Heat pipes have been successfully run at input fluxes in excess of 100 watts/cm^2.[3] 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

$$\frac{\ell_c}{\ell_e} = \frac{H\left(T_f - T_p\right)}{\sigma \varepsilon F\, T_p^4} \tag{8}$$

[3] J. E. Kemme, "Heat Pipe Capability Experiments", LASL 3585-MS, Oct. 1966.

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 that an optimum heat pipe (in terms of maximum axial heat flow) is obtained for

$$r_v = \frac{5r_w}{6} \tag{9}$$

Because an off-optimum heat pipe may sometimes result in a system mass reduction, the r_v, r_w relation given by equation (9) 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 (4) and (6) are used to calculate the optimum groove half width r_c and the maximum axial heat transfer Q_{max}. (The only variable in these equations which has not yet been mentioned is the wetting angle θ. In the absence of any exact information θ has so far been considered to be zero.) Q_{max} is then divided by the axial flux safety factor SF.

$$Q = \frac{Q_{max}}{SF} \tag{10}$$

Q is the design heat transfer for the heat pipe. SF has been varied between 1 and 5. A higher than unity SF 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

$$Q_{in} = 2\pi r_w \ell_e H \left(T_F - T_P \right) \tag{11}$$

The evaporator length ℓ_e is obtained by combining equation (8) with the geometric relation

$$\ell_e + \ell_c = Z \tag{12}$$

Q_{in} is now compared with the design 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 Appendix A 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 2.

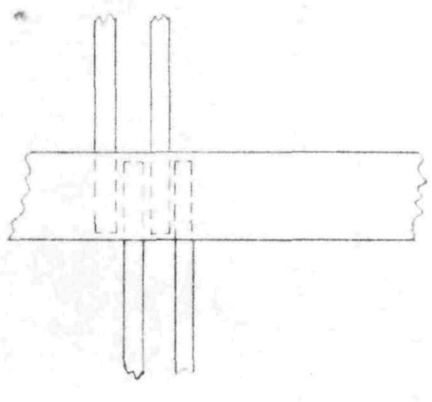

FIGURE 2

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 planforms only even numbers of segments are considered. The pressure drop in a manifold segment is calculated as described in Appendix B 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

$$S = \left(\pi + 2\right) D_m \tag{13}$$

and

$$A_c = \frac{\pi}{4} D_m^2 - D_p D_m \tag{14}$$

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 3 and connected by feed and return lines. Figure 3 shows the radiator in an axial feed line orientation. Figure 4 shows the same planform in a lateral feed line orientation. The computer code chooses whichever orientation results in a radiator whose axial dimension is greater than its lateral dimension. This restriction can be important to the total system because the shield size increases with radiator width. The simplicity of this orientation selection does have some drawbacks. In the almost square case the lateral feed line orientation will result in a heavier system because of the extra feed line leg. The lateral feed line orientation also results in the shadowing of some of the heat pipes by the feed line. This complication is ignored in the code. However, these drawbacks are academic for the 45 - 50 Mw radiator because the optimum radiator designs have been found to be in the axial feed line orientation. If the code is to be used to design much smaller heat pipe radiators (with fewer manifold segments), it may be necessary to change the planform and orientation restrictions.

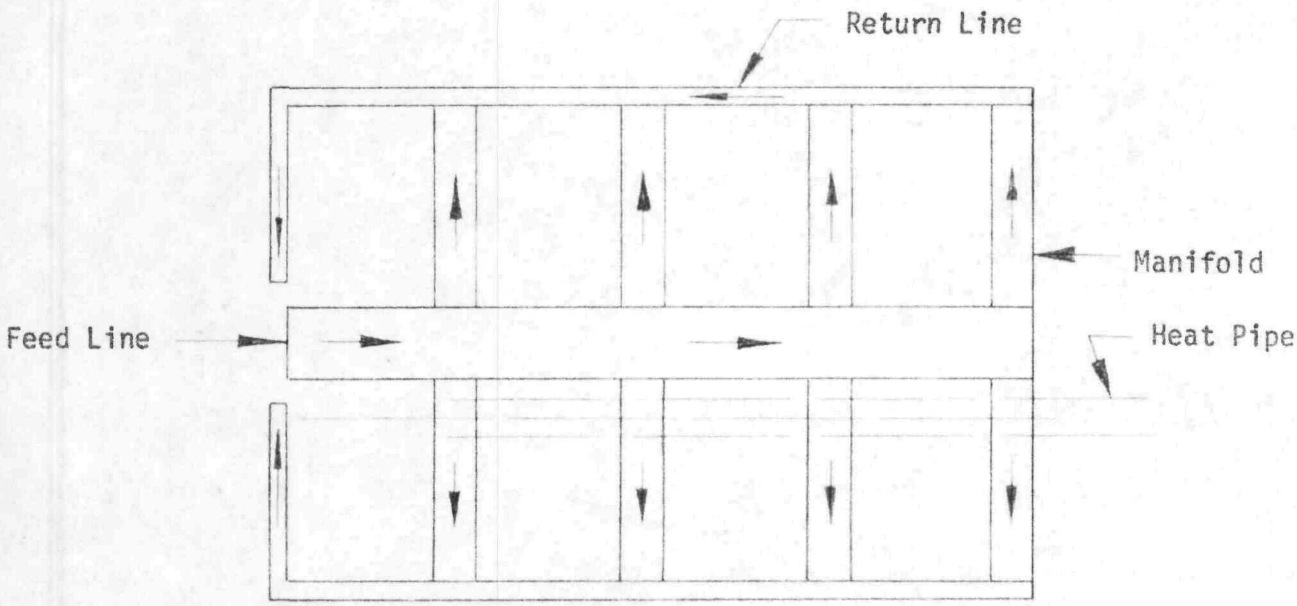

FIGURE 3 ORIENTATION 1 AXIAL FEED LINE

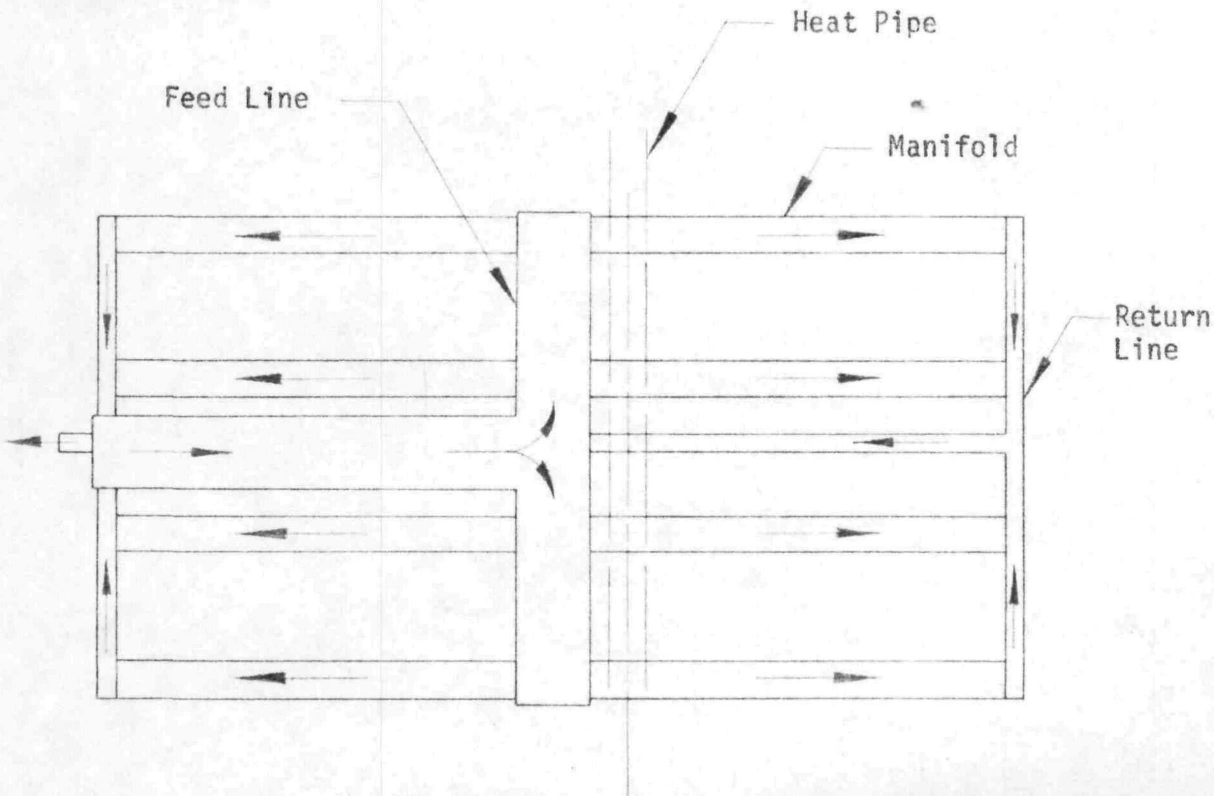

FIGURE 4 ORIENTATION 2 LATERAL FEED LINE

Feed and Return Line Design

The feed and return line lengths are easily calculated knowing the manifold and heat pipe dimensions and the orientation of the radiator. 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 (see Appendix B) for a constant diameter pipe which carries the total flow rate of the working fluid. The pipe length used in the calculation depends on the radiator orientation. In the axial feed line case the radiator length is used; in the lateral feed line case half the length plus half the width is used. The diameter D_o is increased until the pressure drop is less than the maximum allowable value.

After D_o 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

$$D_{f\ell} = D_o \left(1 - \frac{x}{X} \right)^{0.375} \tag{15}$$

where

$D_{f\ell}$ is the feed line diameter

D_o (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. In both radiator orientations 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 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.

$$t_i = \frac{P_{in}}{\sigma} \frac{D_i}{2} \tag{16}$$

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 equation is discussed in Appendix A. 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 continued 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.

The following is a copy of a portion of the HPRAD4 code output which illustrates the design of a specific radiator for SPR-6. A listing of the code appears in Appendix C.

1

HPRAD4, HEAT PIPE RADIATOR

```
INPUT DATA
RADIATOR INLET TEMP,DEG K= 1.100E+03
FLOW RATE,G/SEC= 2.850E+04
INLET QUALITY= 8.500E-01
MAX FRACTIONAL P DROP OF PRIMARY= 5.000E-02
LIFETIME,HR= 1.000E+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= 1.500E+00
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                               (Niobium)
     MANIFOLD AND LINE= 8.560E+00                       (Niobium)
     BARRIER= 1.860E+00                                 (Beryllium)

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                        (Potassium)
LIQ VIS,G/SEC/CM= 1.085E-03
LIQ DENS,G/CM**3= 5.487E-01
VAP DENS,G/CM**3= 8.227E-04
DVGDP,CM**5/G/DYNE=-5.782E-04
ENTHALPY CHANGE,DYNE CM/G= 1.602E+10

HEAT PIPE FLUID PROPERTIES                              (Potassium)
SURF TENS,DYNE/CM= 6.126E+01
LIQ VIS,G/SEC/CM= 1.100E-03
VAP VIS,G/SEC/CM= 2.005E-04
LIQ DENS,G/CM**3= 5.543E-01
VAP DENS,G/CM**3= 7.541E-04
HEAT OF VAP,DYNE CM/G= 1.888E+10

HEAT REJECTION REQUIRED,KW= 4.566E+04

HEAT PIPE TEMP,DEG K= 1.085E+03
BARRIER SURFACE TEMP= 1.081E+03

CALCULATED HEAT REJECTION,KW= 4.567E+04
HEAT REJECTION BREAKDOWN,PERCENTS
     HEAT PIPES= 89.59     MANIFOLDS= 7.96     FEED LINE= 2.45

PRESSURE IN BARS
HEAT PIPE        PRIMARY IN       PRIMARY OUT
 1.648E+00        1.863E+00        1.770E+00
```

D VAPOR	D WICK	TH PIPE	L EVAP	L COND
1.250E+00	1.500E+00	6.746E-02	7.872E+00	8.740E+01

DI MAN	L MAN	R CAP	GROOVES	PIPES
8.040E+00	3.677E+04	2.043E-02	96	22489

MASS FLOW IN HEAT PIPE,GM/SEC= 1.071E+00
RADIAL REYNOLDS NO= 1.080E+02 (ANALYSIS VALID ONLY IF MUCH GREATER THAN ONE)

MAX HEAT PIPE WALL STRESSES,BARS
COMPRESSION OF PUNCTURED PIPE= 2.258E+01
TENSION OF WHOLE PIPE= 1.832E+01

PARALLEL MANIFOLD SEGMENTATION

SEGMENTS	REY NO	FR P DROP	SEGMENT LENGTH
44	1.098E+05	1.211E-02	8.356E+02

ORIENTATION,AXIAL FEED LINE,LATERAL MANIFOLDS

FEED LINE ID,CM= 5.836E+01 FR P DROP= 1.554E-02
TOTAL LENGTH,CM= 2.096E+03
RETURN LINE ID,CM= 1.300E+01 FR P DROP= 1.452E-02
TOTAL LENGTH,CM= 5.863E+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
2.140E-02	1.554E-01	3.461E-02

METEOROID BARRIER THICKNESS,CM= 1.087E+00

RADIATOR MASSES IN KG

HEAT PIPES	H PIPE LIQ	MANIFOLDS
7.001E+03	5.824E+02	1.706E+02
FEED LINE	RETURN LINES	BARRIER
3.731E+02	7.112E+01	3.560E+03

TOTAL MASS= 1.176E+04

MASS PERCENTS

H PIPE	HP LIQ	MAN	FEED	RETURN	BARRIER
59.54	4.95	1.45	3.17	0.60	30.28

RAD LENGTH,CM= 2.191E+03 RAD WIDTH,CM= 1.756E+03

PRIMARY FLUID MASS,KG= 5.205E+02
DISTRIBUTION OF PRIMARY FLUID,MASS PERCENTS

FEED.	MAN	RETURN
0.60	17.35	82.05

HPRAD4 DESIGNS FOR SPR-6 RADIATOR

Introduction

The SPR-6 power system requires a radiator capable of rejecting approximately 45 megawatts of heat. This heat rejection is accomplished through the condensation of a 28.5 kg per sec flow of potassium, which enters the radiator at a quality of 85%. The radiator is to be designed for a 10,000 hour mission with a 99% probability of survival. The computer code HPRAD4 has been used to investigate the effect of various parameters on heat pipe radiator mass. While the parameter variation has not been exhaustive, enough has been done to specify a "near-optimum" 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 given length heat pipe 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, heat pipe manifold size and mass decreases as the heat pipe evaporator section becomes shorter. However, these advantages to 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 5 shows the effect of the heat pipe input flux and axial flow safety factor on the total radiator mass. All of the results are for radiators using 1.5 cm diameter potassium-filled heat pipes. The piping is sized for a 5% pressure drop of the working fluid. A "near-optimum" radiator is seen to result for an input flux of 50 w/cm^2 and an axial flux safety factor of 2. (The 75 w/cm^2 safety factor unity case is not chosen because it is considered undesirable to operate the heat pipes at maximum capacity.)

Heat Pipe Diameter

The total radiator mass is a strong function of the heat pipe diameter. As heat pipe diameter decreases the required wall thickness decreases. This results in a reduction in heat pipe mass, even though the number of heat pipes increases. At some point, however, this mass decrease is negated by the increased amount of piping required to accommodate the large number of small heat pipes. Figure 6 shows the effect of heat pipe diameter on total radiator mass. The 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. The optimum heat pipe diameter is seen to be between 1.0 and 1.5 cm.

Heat Pipe Survival Probability

As stated previously there is a tradeoff between individual heat pipe protection from meteoroids and the amount of redundancy included in the design. $\overline{H}$ has been defined as $-\ln P(o)$ where $P(o)$ is the survival probability of the individual heat pipe. $\overline{H}$ is the average number of damaging hits per heat pipe. A low value for $\overline{H}$ means a high degree of protection for the individual heat pipe and little redundancy. It is shown theoretically in Appendix A that $\overline{H}$ = 0.20 results in the minimum total heat pipe mass. Because the heat pipes do not make up the total radiator, however, it is to be expected that the minimum total radiator mass will occur for a somewhat lower $\overline{H}$. In Figure 7 it is seen that the optimum value of $\overline{H}$ is 0.10. The results shown are for 1.5 cm diameter potassium-filled heat pipes with an input flux of 50 w/cm^2 and an axial flux safety factor of 2.

Radiator Survival Probability

It has been specified that the radiator piping carry enough meteoroid armor to ensure a 99% probability of survival. Figure 8 shows the penalty which must be paid for survival probabilities in excess of 99%. The higher total mass at higher survival probabilities is due entirely to the increase in armor thickness. The armor thickness increases from 1.09 cm at P(o) = 0.99 to 3.77 cm at P(o) = 0.9999. Note that "adding a 9" to the survival probability approximately doubles the thickness and mass of the armor.

Pressure Drop of Working Fluid

Higher allowable pressure drops result in lighter radiators because manifold segmentation decreases and feed and return lines decrease both in length and diameter. Figure 9 shows the effect of varying the allowable pressure drop from 5% to 20%. The mass reduction resulting from the increased allowable pressure drop is seen to be rather small. In a system study this effect must be balanced with the increased pumping required by the higher pressure drops. The results shown are for 1.5 cm diameter potassium-filled heat pipes with an input flux of 50 w/cm^2 and an axial flux safety factor of 3.

Heat Pipe Fluid

It can be argued that sodium is a better heat pipe fluid than potassium. However, the use of sodium-filled heat pipes for the SPR-6 radiator does not necessarily result in a lighter radiator. Figure 10 presents the same parameter variation as Figure 5 except that the heat pipes are sodium rather than potassium-filled. The minimum mass radiator of Figure 10 (at an input flux of 50 w/cm^2 and a safety factor of 3) is actually 150 kg heavier than the minimum mass radiator of Figure 5. In one sense, however, the sodium heat pipe radiator is "better" than the potassium radiator. At an input flux of 50 w/cm^2, the sodium design reaches its minimum mass at a higher axial flux safety factor than the potassium design. Thus, the sodium pipes operate at a lower fraction of their capacity than the potassium pipes. Unfortunately, this "advantage" does not appear as a reduction in radiator mass.

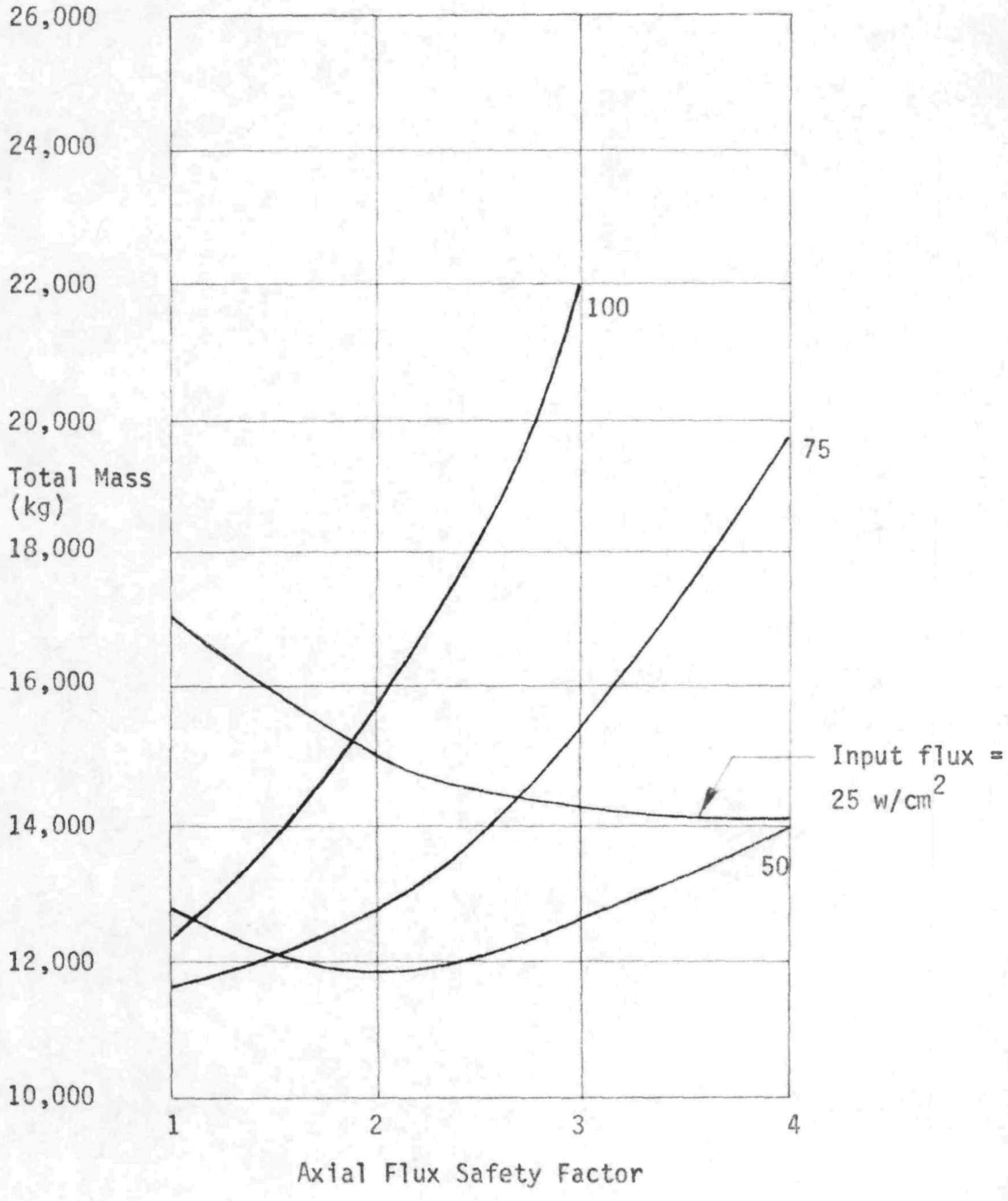

FIGURE 5 TOTAL MASS vs. AXIAL FLUX SAFETY FACTOR

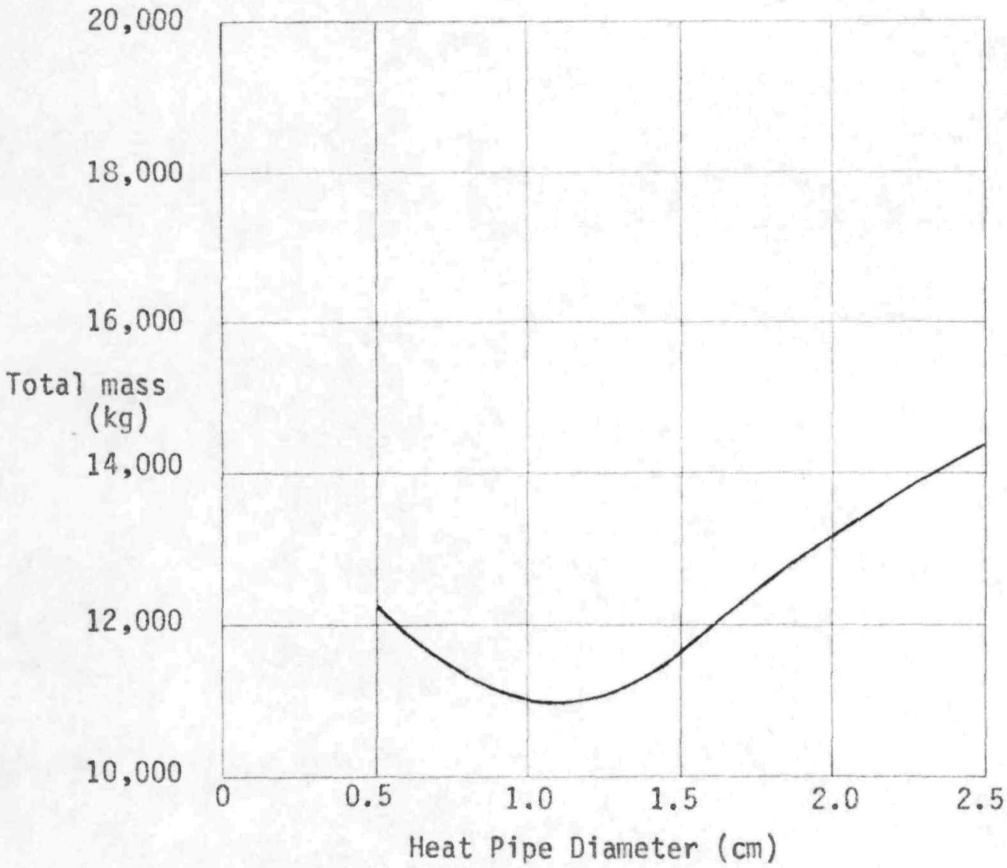

FIGURE 6 TOTAL MASS vs. HEAT PIPE DIAMETER

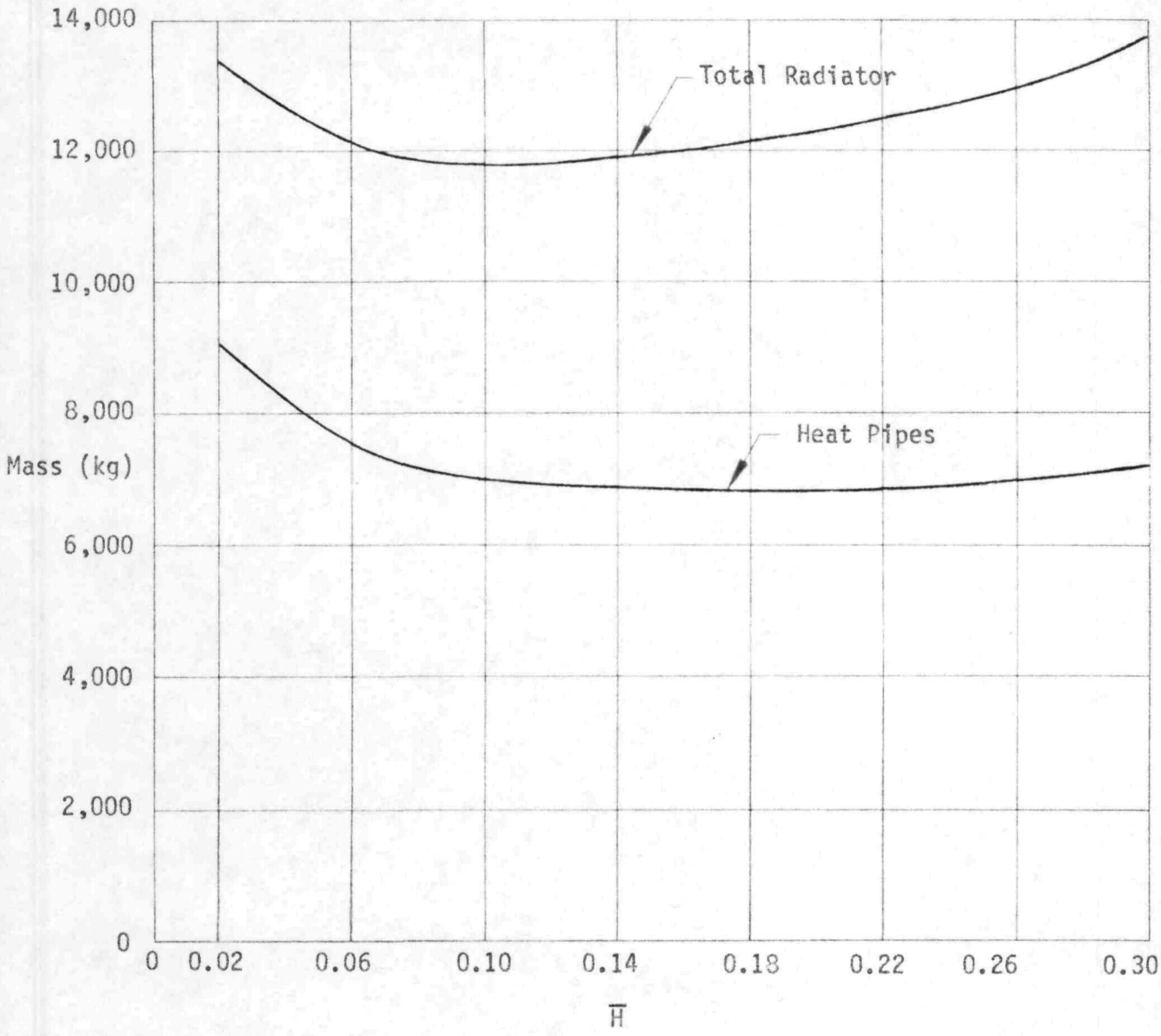

FIGURE 7 MASS vs. $\overline{H}$

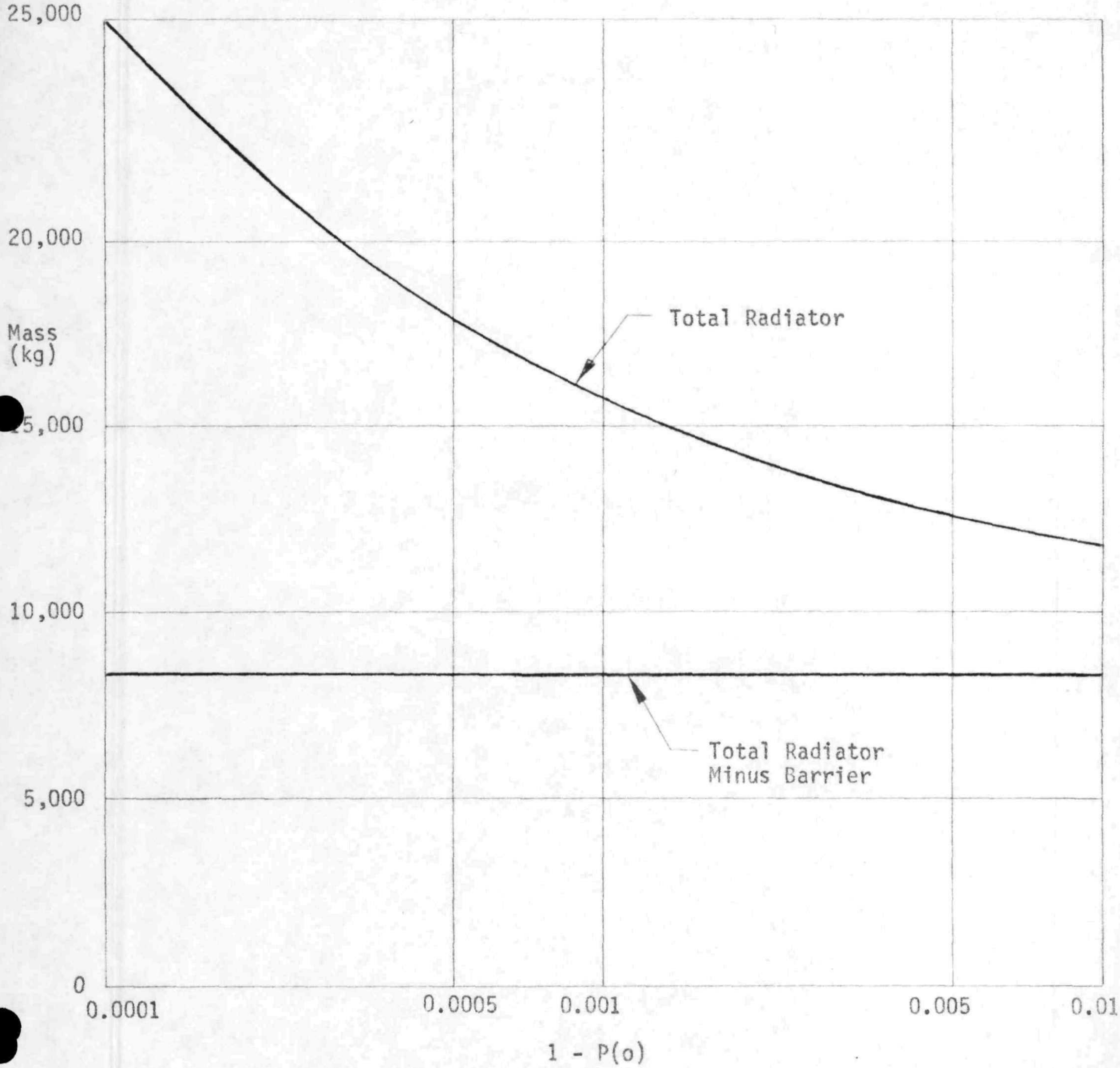

FIGURE 8 MASS vs. P(o)

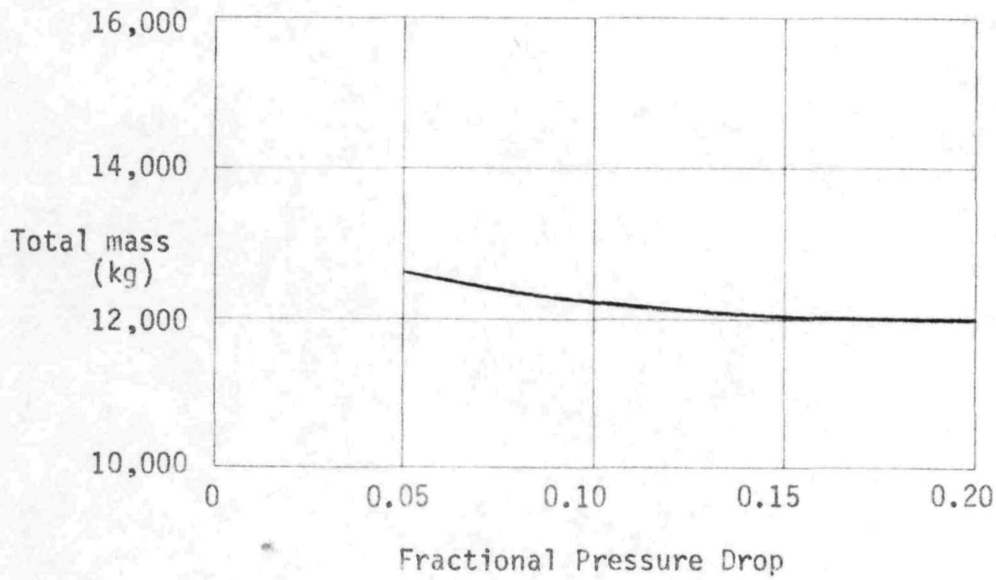

FIGURE 9 TOTAL MASS vs. PRESSURE DROP

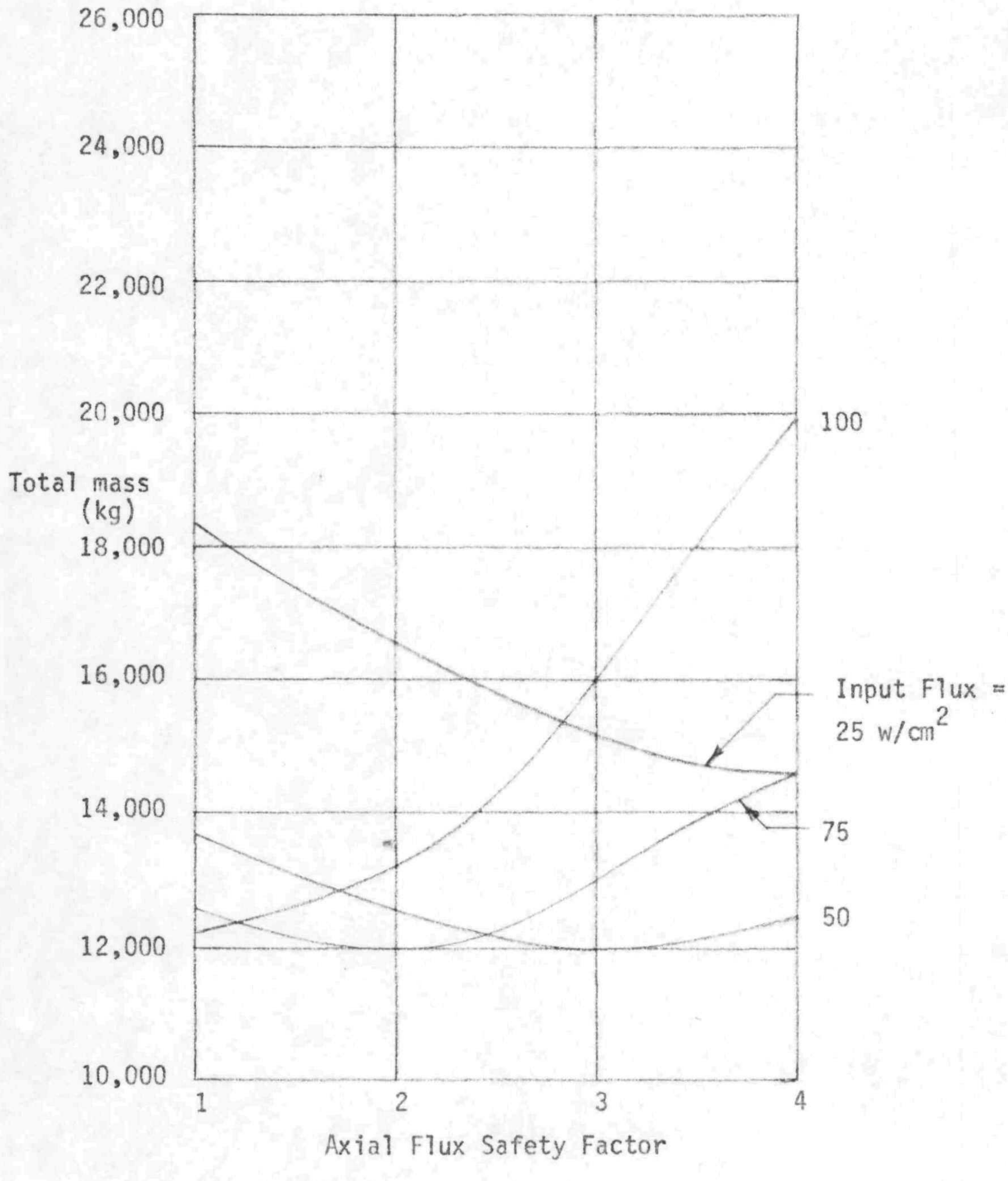

FIGURE 10 TOTAL MASS vs. AXIAL FLUX SAFETY FACTOR

COMMENTS AND RECOMMENDATIONS

This section calls attention to two of the problem areas which have been recognized, but not thoroughly investigated.

Subcooling

It is necessary to subcool the working fluid in the radiator so that the unavoidable entropy increase in the pumps does not force the fluid under the vapor dome. In previous LRL radiator analyses the amount of subcooling has been rather arbitrarily set at 100 R (55.6 K).

While not explicity considered in HPRAD4, subcooling will occur due to the presence of the redundant heat pipes. An upper limit on the amount of subcooling, ΔT, is given by

$$\Delta T = \frac{q_{sub}}{wC_p} \tag{17}$$

For 10% heat pipe redundancy it is clear that q_{sub} will be less than 10% of the heat of condensation. Thus, for SPR-6,

$$q_{sub} < 4.57 \text{ Mw}$$

$$w = 28.5 \text{ kg/sec}$$

$$C_p \cong 0.8 \text{ w sec/g-K}$$

So, $\Delta T < 203\ ^{o}K$

ΔT will be less than the limiting value because the temperature of the heat pipes in the subcooling section will be lowered considerably as a result of (a) the liquid metal heat transfer coefficient which is less than the condensing heat transfer coefficient and (b) the lowered source temperature itself.

For a constant mass flow rate the amount of subcooling will decrease with time as a result of heat pipe punctures. Control of the amount of subcooling has not been investigated.

An Additional Meteoroid Hazard

Leakage from a manifold via a doubly-punctured heat pipe (one puncture outside the manifold, the other inside) is a meteoroid hazard that has not been quantitatively considered. It is felt, however, that this hazard is not serious.

The double wall thickness of heat pipe provides considerable protection. As shown by equation (A.1) the average number of penetrations per heat pipe is inversely proportional to the fourth power of the wall thickness. Thus the average number of double penetrations per heat pipe is a factor of 16 lower than the number of single penetrations. The probability of a double penetration is further decreased by the bumper screen effect.[4]

The probability of a double penetration that provides a manifold leakage path is much lower than the overall double penetration probability because only meteoroids with particular flight directions can effect such a "critical" double penetration. For the case of a 1.5 cm diameter heat pipe piercing a manifold covered with a 1.0 cm thick armor, only those meteoroids which pierce the heat pipe at an angle greater than 34^{0} from the normal can possibly exit inside the manifold. This "critical" angle increases as meteoroid hits further from the manifold are considered. In addition to the reduced meteoroid flux which need be considered for "critical" double penetrations, the inclination of the "critical" meteoroid paths reduces the penetration because (a) the normal component of meteoroid velocity is proportional to the cosine of the angle from the normal, and (b) as indicated by oblique hypervelocity impact data, for angles greater than 50^{0} penetration is less than that predicted by the normal component of velocity.[4]

---

[4] Loeffler, I. J., Lieblein, Seymour, and Clough, Nestor, "Meteoroid Protection for Space Radiators", ARS Paper #2543-62, September 1962.

The "critical" inclination angle may be further increased by the addition of a short collar around the heat pipe at the manifold barrier surface. For the case previously mentioned the addition of a 1.0 cm long collar would increase the "critical" angle to 53^0 from the normal. The mass of such collars is already implicitly included in the computer design code because the barrier mass is calculated for a cylindrical shell with no correction made for the heat pipe entry holes.

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NOMENCLATURE

A = vulnerable area, ft^2

A_c = flow area, cm^2

b = groove height = $r_w - r_v$, cm

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

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

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

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

N = total number of heat pipes

P = pressure, bar

$P(o)$ = survival probability

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

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 A.1)

T = temperature, K

v_g = vapor specific volume, cm^3/g

v_ℓ = liquid specific volume, cm^3/g

w = mass flow rate, g/sec

X = fluid quality

Z = heat pipe length, cm

γ = surface tension, dyne/cm

ε = emissivity

η = viscosity, g/cm sec

θ = wetting angle

μ = viscosity, g/cm sec

ρ = barrier material density, lbm/ft^3

ρ_ℓ = liquid density, g/cm^3

ρ_v = vapor density, g/cm^3

σ = Stefan-Boltzmann constant = 5.67×10^{-12} $W/cm^2\text{-}K^4$

σ = stress, bar

τ = mission time, hr

ϕ = angle with horizontal

APPENDIX A

METEOROID PROTECTION

The meteoroid criterion used is the NASA-recommended model as reported in SPN-36 by Walter. The required barrier thickness is calculated from the following equation:

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

where $\overline{H} = -\ln P(o)$ and $P(o)$ is the design survival probability.

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 (A.1) 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(o) is set at 0.99.

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. Also, redundancy may be introduced as a protection mechanism because each heat pipe is a separate entity. For a single heat pipe the average number of penetrating hits is given from equation (A.1) as

$$\overline{H} = \frac{C}{t^4} \qquad (A.2)$$

where C is a constant and t is the wall thickness. If $\overline{H}$ is less than unity and the total number of heat pipes N is very large then $\overline{H}N$ is the number of punctured pipes. If N_s is the number of heat pipes required for the completion of the mission it follows that

$$N = \frac{N_s}{1 - \overline{H}} \tag{A.3}$$

If M_w is the total mass of the heat pipe walls then

$$M_w \propto Nt = \frac{N_s}{1 - \frac{C}{t^4}} t = \frac{N_s t^5}{t^4 - C} \tag{A.4}$$

Minimizing M_w yields $\overline{H}_{OPT} = 0.20$ and $N = 1.25\ N_s$. (Because M_w may or may not be the dominant radiator mass term $\overline{H}_{OPT}$ is carried as a variable in the computer code.) The heat pipe wall thickness as calculated from equation (A.1) using $\overline{H} = \overline{H}_{OPT}$ has been found to be more than adequate from the fluid pressure standpoint.

Equation (A.3) assumes that the number of heat pipes is large such that the probability S of survival of N_s or more out of N pipes is sufficiently high. This probability may be calculated from the relation[1]

$$S = \sum_{n = N_s}^{N} \frac{N!}{n!\left(N - n\right)!} \left(1 - P(o)\right)^{N-n} P(o)^n \tag{A.5}$$

where $-\ln P(o) = \overline{H}$. For $\overline{H} = 0.20$ and $N_s = 0.8\ N$ the probability S is greater than 0.99 for $N > 2500$ and greater than 0.999 for $N > 4000$. The number of heat pipes in a typical 45 Mw radiator is considerably greater than 4000; therefore equation (A.3) is a valid redundancy relation.

[1] R. English and D. Guentert, "Segmentation of Radiators for Meteoroid Protection", ARSJ, Vol. 31, No. 8, August 1961, pp. 1162-1164.

APPENDIX B

PRESSURE DROP

An influencing factor in the radiator planform is the allowable pressure drop in the duct which contains the heat pipes and in the supply and return headers. For this analysis it is assumed: that there is two phase flow, that the vapor quality is equal to that at the turbine exhaust, and that the return fluid is all liquid with subcooling as a consequence of carrying excess heat pipes for redundancy.

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

The pressure drop of the condensing working fluid in the radiator is obtained by integrating the two phase pressure gradient derived by the method of W. L. Owens.[5] Owens' pressure gradient is derived for a two phase mixture flowing in a constant area circular pipe with the assumption of equal velocities for the two phases (the fog flow model). The result is:

$$\frac{dP}{d\ell} = \frac{-\frac{fG^2 v_\ell}{2Dg_c}\left[1 + X\left(\frac{v_g}{v_\ell} - 1\right)\right] + \frac{G^2 v_\ell}{g_c}\left(\frac{v_g}{v_\ell} - 1\right)\frac{dx}{d\ell} + \frac{\sin\phi}{v_\ell\left[1 + X\left(\frac{v_g}{v_\ell} - 1\right)\right]}}{1 + X\frac{dv_g}{dP}\frac{G^2}{g_c}} \quad (B.1)$$

[5] W. L. Owens, "Two-Phase Pressure Gradient", International Developments in Heat Transfer, Pt II, ASME, pp. 363-368.

Following Owens, the pressure gradient for two phase flow in a constant area non-circular duct (or a circular duct with internal frictional surfaces) may be derived as

$$\frac{dP}{d\ell} = \frac{-\dfrac{fG^2 v_\ell S}{8 g_c A_c}\left[1 + x\left(\dfrac{v_g}{v_\ell} - 1\right)\right] + \dfrac{G^2 v_\ell}{g_c}\left(\dfrac{v_g}{v_\ell} - 1\right)\dfrac{dx}{d\ell} + \dfrac{\sin\phi}{v_\ell\left[1 + X\left(\dfrac{v_g}{v_\ell} - 1\right)\right]}}{1 + X\dfrac{dv_g}{dP}\dfrac{G^2}{g_c}} \qquad (B.2)$$

where

S = wetted permiter

A_c = flow area

Neglecting the gravity term and substituting for G in terms of the mass flow rate w and the flow area A_c yields

$$\frac{dP}{d\ell} = \frac{-\dfrac{w^2}{g_c A_c^2}\left[\left(v_g - v_\ell\right)\dfrac{dx}{d\ell} + \dfrac{fS}{8A_c}\left(\left(1 - X\right)v_\ell + xv_g\right)\right]}{1 + \dfrac{w^2}{A_c^2 g_c}\dfrac{dv_g}{dP} X} \qquad (B.3)$$

Now assume that

1. v_g, v_ℓ, and $\frac{dv_g}{dP}$ are essentially constant over the range of integration.

2. Quality varies linearly with position, $X = X_1 + \left(X_2 - X_1\right)\frac{\ell}{L}$, where L = length of duct.

Substituting for X and integrating yields

$$\frac{P_2 - P_1}{P_1} = \frac{w^2 v_g}{g_c A_c^2 P_1}\int_0^L \frac{a\ell + b}{c\ell + d}\, d\ell \qquad (B.4)$$

where

$$a = \frac{fS}{8A_cL}\left(X_1 - X_2\right)\left(1 - \frac{v_\ell}{v_g}\right)$$

$$b = \frac{1}{L}\left(X_1 - X_2\right)\left(1 - \frac{v_\ell}{v_g}\right) - \frac{fS}{8A_c}\left(\left(1 - X_1\right)\frac{v_\ell}{v_g} + X_1\right)$$

$$c = \frac{w^2}{g_cA_c^2L}\frac{dv_g}{dP}\left(X_2 - X_1\right)$$

$$d = 1 + \frac{w^2}{g_cA_c^2}\frac{dv_g}{dP}X_1$$

The integral may be evaluated exactly to yield

$$\frac{P_2 - P_1}{P_1} = \frac{w^2v_g}{g_cA_c^2P_1}\left(\frac{aL}{c} + \frac{bc - ad}{c^2}\ln\left|\frac{cL}{d} + 1\right|\right) \qquad (B.5)$$

If, as is often the case, $\left|\frac{cL}{d}\right| << 1$, then the result reduces to

$$\frac{P_2 - P_1}{P_1} = \frac{w^2v_g}{g_cA_c^2P_1}\left(\frac{aL^2}{2d} + \frac{bL}{d}\right) \qquad (B.6)$$

The above solution reduces to the circular pipe case with the substitution

$$S = \pi D$$

and

$$A_c = \frac{\pi D^2}{4}$$

It has been demonstrated by Owens that the two phase friction factor should be chosen to be the same as that for single phase liquid flow. For turbulent flow in a smooth-walled circular pipe, the Darcy-Weisbach friction factor is often given as

$$f = 0.184 \; Re^{-0.2} \tag{B.7}$$

where Re is the Reynolds number. Writing the Reynolds number in terms of the flow area,

$$f = 0.184 \left(\frac{2w}{\mu \sqrt{\pi A_c}} \right)^{-0.2} \tag{B.8}$$

Following Owens, f is calculated using the liquid viscosity for μ. For the circular pipe case the above relation for f reduces to the more familiar

$$f = 0.184 \left(\frac{4w}{\pi D \mu} \right)^{-0.2}$$

A listing of PDROP2, the computer subroutine which calculates the pressure drop, appears in Appendix C.

APPENDIX C

```
*      FORTRAN              HPRAD04
       SUBROUTINE HPRAD4(TIN,WDOT,QUAL,DPR,TAU,PZERO,H,FLUX,SF,DW,STRESS,
      2HPP)
       COMMON/KIFL/MZ
C      INPUT   UNITS              EXPLANATION
C       TIN    DEG K              INLET TEMP
C       WDOT   G/SEC              FLOW RATE
C       QUAL                      INLET QUALITY
C       DPR                       FRACTIONAL P DROP
C       TAU    HOURS              LIFETIME
C       PZERO                     SURVIVAL PROBABILITY
C       H      W/CM**2/DEG K      CONDENSING HEAT TRANSFER COEFF
C       FLUX   W/CM**2            HEAT PIPE INPUT FLUX
C       SF                        HEAT PIPE AXIAL FLUX SAFETY FACTOR
C       DW     CM                 HEAT PIPE DIAMETER
C       STRESS BARS               PIPING STRESS
C       HPP                       AVERAGE HITS PER HEAT PIPE
       WRITE OUTPUT TAPE 3,1
     1 FORMAT(1H1)
       WRITE OUTPUT TAPE 3,201,TIN,WDOT,QUAL,DPR,TAU,PZERO
   201 FORMAT(10X,28H   HPRAD4,HEAT PIPE RADIATOR//5X,10HINPUT DATA/5X,
      226HRADIATOR INLET TEMP,DEG K=,E10.3/5X,16HFLOW RATE,G/SEC=,
      3E10.3/5X,14HINLET QUALITY=,E10.3/5X,33HMAX FRACTIONAL P DROP OF PR
      4IMARY=,E10.3/5X,12HLIFETIME,HR=,E10.3/5X,14HSURVIVAL PROB=,E10.3)
       WRITE OUTPUT TAPE 3,701,H,FLUX,SF,DW,STRESS,HPP
   701 FORMAT(5X,25HHEAT COEFF,W/CM/CM/DEG K=,E10.3/5X,23HMAX INPUT FLUX,
      2W/CM/CM=,E10.3/5X,25HAXIAL FLUX SAFETY FACTOR=,E10.3/5X,34HHEAT PI
      3PE ID INCLUDING WICKING,CM=,E10.3/5X,26HPRIMARY PIPING STRESS,BAR=
      4,E10.3/5X,31HHPP,AVERAGE HITS PER HEAT PIPE=,E10.3//10X,17HEND OF
      5INPUT DATA///)
C      HEAT PIPE MATERIAL PROPERTIES
C      DENSITIES IN G/CM**3
C      ELASTIC MODULI IN DYNE/CM**2
       RHOP=8.56
       EP=1.1E12
C      MANIFOLD AND FEED LINE MATERIAL PROPERTIES
       RHOM=8.56
C      BARRIER MATERIAL PROPERTIES
       RHOB=1.86
       EB=2.1E12
       CONDB=0.6
C      CONDB IS THERMAL CONDUCTIVITY OF BARRIER IN W/CM/K
       HRES=5.67
C      HRES IS MANIFOLD-BARRIER INTERFACE HEAT COEFF IN W/CM**2/K
       WRITE OUTPUT TAPE 3,702,RHOP,RHOM,RHOB,EP,EB
   702 FORMAT(5X,26HMATERIAL DENSITIES,G/CM**3/10X,10HHEAT PIPE=,E10.3/
      210X,18HMANIFOLD AND LINE=,E10.3/10X,8HBARRIER=,E10.3//5X,41HMATERI
      3AL MODULUS OF ELASTICITY,DYNE/CM**2/10X,10HHEAT PIPE=,E10.3/10X,
      48HBARRIER=E10.3//)
       WRITE OUTPUT TAPE 3,550,CONDB,HRES
   550 FORMAT(5X,28HBARRIER CONDUCTIVITY,W/CM/K=,E10.3/5X,48HMANIFOLD-BAR
      2RIER INTERFACE HEAT COEFF,W/CM**2/K=,E10.3//)
```

```
C     FLUID PROPERTIES ARE OBTAINED FROM SUBROUTINES KVAP AND TRANP
C     PRIMARY FLUID PROPERTIES
C     PICK PRIMARY FLUID
C     FLUID CODE         J       MZ
C         POTASSIUM      3        1
C         SODIUM         2        2
C         CESIUM         5        3
      J=3
      MZ=1
      TINR=1.8*TIN
      CALL KVAP(1,TINR,PIN,QUAL,HIN,X1,X2)
      PIN=PIN/0.9869
      HIN=HIN*2.32E7
      POUT=(1.0-DPR)*PIN
      POUTA=POUT*0.9869
      CALL KVAP(4,TOUT,POUTA,0.0,HOUT,X1,X2)
      TOUT=TOUT/1.8
      HOUT=HOUT*2.32E7
      DELH=HIN-HOUT
C     DELH IS ENTHALPY OF CONDENSATION FOR PRIMARY FLUID
      TF=(TIN+TOUT)/2.0
C     TF IS MEAN TEMPERATURE OF PRIMARY FLUID
      CALL VISL(J,TF,VISLF)
      CALL RHOL(J,TF,RHOLF)
      VL=1.0/RHOLF
      CALL RHOG(J,TF,RHOGF)
      VG=1.0/RHOGF
      PF=(PIN+POUT)/2.0
      PF1=PF-0.05
      PF1A=PF1*0.9869
      CALL KVAP(4,X1,PF1A,1.0,X2,X3,VV1)
      VV1=VV1/0.01602
      PF2=PF+0.05
      PF2A=PF2*0.9869
      CALL KVAP(4,X1,PF2A,1.0,X2,X3,VV2)
      VV2=VV2/0.01602
      DVG=(VV2-VV1)/1.0E5
      P2=(1.0-DPR)**0.333*PIN
      P3=(1.0-DPR)**0.333*P2
      DPR3=1.0-(1.0-DPR)**0.333
C     DPR3 IS MAX FRACTIONAL P DROP IN EITHER MANIFOLD,FEED LINE,OR
C     RETURN LINE
      WRITE OUTPUT TAPE 3,400,VISLF,RHOLF,RHOGF,DVG,DELH
  400 FORMAT(5X,32HAVERAGE PRIMARY FLUID PROPERTIES/5X,17HLIQ VIS,G/SEC/
     2CM=,E10.3/5X,17HLIQ DENS,G/CM**3=,E10.3/5X,17HVAP DENS,G/CM**3=,
     3E10.3/5X,19HDVGDP,CM**5/G/DYNE=,E10.3/5X,26HENTHALPY CHANGE,DYNE C
     4M/G=,E10.3//)
C     HEAT PIPE FLUID PROPERTIES
C     PICK HEAT PIPE FLUID
      J=3
      MZ=1
      TP=TF-FLUX/H
```

```
C     TP IS HEAT PIPE OPERATING TEMP
      H=H*1.0E7
      CALL TENS(J,TP,GAM)
      CALL VISL(J,TP,VISL)
      CALL VISG(J,TP,VISG)
      CALL RHOL(J,TP,RHOL)
      CALL RHOG(J,TP,RHOG)
      TPR=9.0*TP/5.0
      CALL KVAP(1,TPR,P,0.0,HL,S,V)
      CALL KVAP(1,TPR,P,1.0,HG,S,V)
      P=P/0.9869
      HVAP=HG-HL
      HVAP=HVAP*2.32E7
C     HVAP IS HEAT OF VAPORIZATION FOR HEAT PIPE FLUID
      WRITE OUTPUT TAPE 3,401, GAM,VISL,VISG,RHOL,RHOG,HVAP
  401 FORMAT(5X,26HHEAT PIPE FLUID PROPERTIES/5X,18HSURF TENS,DYNE/CM=,
     2E10.3/5X,17HLIQ VIS,G/SEC/CM=,E10.3/5X,17HVAP VIS,G/SEC/CM=,E10.3/
     35X,17HLIQ DENS,G/CM**3=,E10.3/5X,17HVAP DENS,G/CM**3=,E10.3/5X,
     422HHEAT OF VAP,DYNE CM/G=,E10.3//)
C     CONSTANTS AND PRELIMINARY CALCULATIONS
      QTOT=DELH*WDOT*1.0E-10
C     QTOT IS TOTAL HEAT TO BE REJECTED IN KW
      PI=3.14159
      SIG=5.6697E-5
C     SIG IS STEFAN-BOLTZMANN CONSTANT IN ERG/CM**2/SEC/K**4
      F=2.0/PI
C     F IS HEAT PIPE VIEW FACTOR
      EPS=0.9
C     EPS IS EMISSIVITY
      ELRAT=H*(TF-TP)/(SIG*EPS*F*TP**4)
C     ELRAT IS RATIO OF H PIPE CONDENSER LENGTH TO EVAPORATOR LENGTH
      C1=0.75
C     C1 IS CONSTANT IN H PIPE EQS (GROVER SAYS IT SHOULD BE PI/4)
      C2=6.15
C     C2 IS CONSTANT IN METEOROID EQ
      COSTHE=1.0
C     COSTHE IS COSINE OF WETTING ANGLE
      RW=DW/2.0
      RV=5.0*RW/6.0
C     RV IS VAPOR RADIUS OF HEAT PIPE
      DV=2.0*RV
      A=(1.0-4.0/PI/PI)/(8.0*RHOG*RV**4*HVAP**2)
      C=2.0*GAM*COSTHE
C     A AND C ARE CONSTANTS IN THE HEAT PIPE EQS
C     BEGINNING OF ROUTINE TO DESIGN HEAT PIPE
C     GUESS EL AND THP
C     EL IS H PIPE LENGTH,THP IS WALL THICKNESS
      II=1
      EL=5.0*DW
      THP=0.0
  340 DP=DW+2.0*THP
C     DP IS OUTSIDE PIPE DIAMETER
```

```
C     CALCULATE HEAT PIPE LENGTH FROM HEAT TRANSFER CONSIDERATIONS
   10 B=C1*VISL*EL/(PI*RHOL*RV*(RW-RV)*HVAP)
C     B IS A CONSTANT IN THE HEAT PIPE EQS
      ROPT=(2.0*B*B/A/C)**0.333
      QMAX=B/A/ROPT/ROPT
C     ROPT IS OPTIMUM GROOVE HALF WIDTH,QMAX IS AXIAL HEAT TRANSFER
      ELE=EL/(1.0+ELRAT)
      ELC=EL-ELE
C     ELE IS EVAPORATOR LENGTH,ELC IS CONDENSER LENGTH
      QIN=PI*DP*ELE*H*(TF-TP)
C     QIN IS HEAT INTO PIPE
      ELOLD=EL
C     CALCULATE NEW EL TO BALANCE QMAX AND QIN
      EL=ELOLD*QMAX/QIN/SF
C     COMPARE OLD AND NEW H PIPE LENGTHS
      IF (ABSF((ELOLD-EL)/ELOLD)-0.0001)11,11,10
   11 IF (II-1) 350,350,351
  350 II=2
C     CALCULATE HEAT PIPE WALL THICKNESS FROM METEOROID EQUATION
C     METEOROID EQ IS IN FUNNY ENGLISH UNITS
      THP=0.0
C     THP IS H PIPE WALL THICKNESS
      C3=C2/((EP*1.0E-6*0.9869*14.7*144.0)**0.333*(RHOP*62.43)**0.167*
     2HPP**0.25)*(TAU/10000.0)**0.25
      ELPFT=ELC/2.54/12.0
  330 DPFT=DW/2.54/12.0+2.0*THP
      AVUN=2.0*DPFT*ELPFT
C     AVUN IS VULNERABLE AREA OF SINGLE HEAT PIPE
      TH2=C3*AVUN**0.25
C     TH2 IS NEW H PIPE WALL THICKNESS
C     COMPARE OLD AND NEW WALL THICKNESSES,REPEAT AVUN CALCULATION
C     UNTIL AGREEMENT IS REACHED
      IF (ABSF(THP-TH2)/TH2-0.0001) 310,310,300
  300 THP=TH2
      GO TO 330
  310 THP=TH2*12.0*2.54
C     THP IS FINAL VALUE FOR H PIPE WALL THICKNESS
C     RETURN TO HEAT TRANSFER CALCULATIONS WITH CORRECT WALL THICKNESS
      GO TO 340
  351 CONTINUE
      EMDOT=QMAX/SF/HVAP
C     EMDOT IS MASS FLOW RATE IN SINGLE PIPE
      RER=EMDOT/(2.0*PI*ELE*VISG)
C     RER IS RADIAL REYNOLDS NUMBER IN H PIPE. THE H PIPE EQS USED
C     IMPLY THAT RER BE MUCH GREATER THAN UNITY.
      QMAX=QMAX*1.0E-10
      NPIPE=QTOT*SF/QMAX/(1.0-HPP)
C     NPIPE IS NUMBER OF H PIPES,INCLUDING REDUNDANCY
      NGROOV=PI*DV/2.0/ROPT
C     NGROOV IS NUMBER OF GROOVES PER PIPE
      H=H*1.0E-7
C     END OF HEAT PIPE DESIGN
```

```
  753 CONTINUE
C     CALCULATE MANIFOLD SIZE TO ACCOMODATE HEAT PIPES
      ELMAN=NPIPE*DP
      DIMAN=SQRTF(ELE**2+DP**2)
C     ELMAN IS TOTAL MANIFOLD LENGTH,DIMAN IS INSIDE DIAMETER
C     CALCULATE HEAT PIPE WALL STRESS
      STR1=PIN*DP/2.0/THP
      STR2=P*DW/2.0/THP
C     STR1 IS MAX COMPRESSION,STR2 IS MAX TENSION
C     DIVIDE MANIFOLD INTO PARALLEL SEGMENTS TO REDUCE PRESSURE DROP
      P22=P2*1.0E6
      S=PI*DIMAN+2.0*DIMAN
      AC=PI*DIMAN**2/4-DP*DIMAN
C     S IS WETTED PERIMETER,AC IS FLOW AREA
      NSEG=0
  800 NSEG=NSEG+2
      ENSEG=NSEG
      WDOTS=WDOT/ENSEG
      ELMANS=ELMAN/ENSEG
C     WDOTS IS FLOW RATE PER SEGMENT,ELMANS IS SEGMENT LENGTH
C     GUARD AGAINST SONIC EXIT FROM PDROP SUBROUTINE
      GARD=1.0+(WDOTS/AC)**2*DVG*QUAL
      IF (GARD-0.0) 800,800,805
  805 CALL PDROP2(WDOTS,P22,QUAL,0.0,VG,VL,DVG,VISLF,S,AC,ELMANS,DELP)
      DELP=-DELP
C     CHECK PRESSURE DROP CRITERION,INCREASE NSEG IF NECESSARY
      IF (DELP-DPR3) 803,803,800
  803 CONTINUE
      REMAN=2.0*WDOTS/VISLF/SQRTF(PI*AC)
C     DETERMINE DESIRABLE RADIATOR ORIENTATION
C     ORIENTATION IS SUCH THAT RADIATOR LENGTH IS GREATER THAN WIDTH
C     KONFIG=1 IMPLIES AXIAL FEED LINE,KONFIG=2 IMPLIES LATERAL FEED
      TEST1=2.0*ELMANS
      TEST2=(ENSEG/2.0+1.0)*EL
      IF (TEST1-TEST2) 500,500,501
  500 KONFIG=1
      GO TO 504
  501 KONFIG=2
  504 CONTINUE
C     CALCULATE DIMENSIONS OF FEED LINE
      GO TO (505,506),KONFIG
  505 ELFL=(ENSEG/2.0)*EL
      GO TO 507
  506 ELFL=(ENSEG/2.0-1.0)*EL/2.0+ELMANS
C     ELFL AT THIS POINT IS FLOW LENGTH FOR FEED LINE
  507 WDOTFL=WDOT
C     WDOTFL IS FLOW RATE IN FEED LINE
      DFL=DIMAN-1.0
C     DFL IS FEED LINE ID
C     GUARD AGAINST SONIC EXIT FROM PDROP SUBROUTINE
      DFLMIN=(-16.0*WDOTFL**2/PI**2*DVG*QUAL)**0.25
      DFL=MAX1F(DFL,DFLMIN)
```

```
   80 DFL=DFL+1.0
      PIN2=PIN*1.0E6
      S=PI*DFL
      AC=PI*DFL**2/4.0
      CALL PDROP2(WDOTFL,PIN2,QUAL,QUAL,VG,VL,DVG,VISLF,S,AC,ELFL,DPFL)
      DPFL=-DPFL
C     CHECK P DROP CRITERION,INCREASE DFL IF NECESSARY
      IF (DPFL-DPR3) 81,81,80
   81 CONTINUE
      GO TO (508,509),KONFIG
  508 ELFL1=0.0
      ELFL2=(ENSEG/2.0)*EL
      ELFL3=0.0
      GO TO 510
  509 ELFL1=ELMANS
      ELFL2=(ENSEG/2.0-1.0)/2.0*EL
      ELFL3=ELFL2
  510 ELFL=ELFL1+ELFL2+ELFL3
C     ELFL IS NOW TOTAL FEED LINE LENGTH
C     CALCULATE DIMENSIONS OF THE RETURN LINE
      GO TO (511,512),KONFIG
  511 ELRL=(ENSEG/2.0)*EL+ELMANS
      GO TO 513
  512 ELRL=(ENSEG/2.0-1.0)*EL/2.0+2.0*ELMANS
C     ELRL AT THIS POINT IS FLOW LENGTH FOR RETURN LINE
  513 WDOTRL=WDOT/2.0
C     WDOTRL IS FLOW RATE IN RETURN LINE
      DRL=1.0
C     DRL IS RETURN LINE ID
   82 DRL=DRL+1.0
      P32=P3*1.0E6
      S=PI*DRL
      AC=PI*DRL**2/4.0
      CALL PDROP2(WDOTRL,P32,0.0,0.0,VG,VL,DVG,VISLF,S,AC,ELRL,DPRL)
      DPRL=-DPRL
C     CHECK P DROP CRITERION,INCREASE DRL IF NECESSARY
      IF (DPRL-DPR3) 83,83,82
   83 CONTINUE
      GO TO (611,612),KONFIG
  611 ELRL=ENSEG*EL+2.0*ELMANS
      GO TO 613
  612 ELRL=(ENSEG/2.0-1.0)*EL*2.0+2.0*ELMANS
C     ELRL IS NOW TOTAL RETURN LINE LENGTH
  613 CONTINUE
C     CALCULATE MANIFOLD AND LINE WALL THICKNESSES BASED ON STRESS
      THMAN=(PIN/STRESS)*(DIMAN/2.0)
      THFL=(PIN/STRESS)*(DFL/2.0)
      THRL=(PIN/STRESS)*(DRL/2.0)
C     THMAN,THFL,THRL ARE WALL THICKNESSES FOR MANIFOLDS,FEED LINE,AND
C     RETURN LINE,RESPECTIVELY
C     CALCULATE METEOROID BARRIER THICKNESS
C     METEOROID EQ IS IN FUNNY ENGLISH UNITS
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```
      TH=0.0
C     TH IS BARRIER THICKNESS
      HP=LOGF(1.0/PZERO)
      C3=C2/((EB*1.0E-6*0.9869*14.7*144.0)**0.333*(RHOB*62.43)**0.167*
     2HP**0.25)*(TAU/10000.0)**0.25
      ELMFT=ELMAN/2.54/12.0
      ELFL1F=ELFL1/2.54/12.0
      ELFL2F=ELFL2/2.54/12.0
      ELFL3F=ELFL3/2.54/12.0
      ELFLFT=ELFL/2.54/12.0
      ELRLFT=ELRL/2.54/12.0
      DMFT=(DIMAN+2.0*THMAN)/2.54/12.0
      DFLFT=DFL/2.54/12.0
      THFLFT=THFL/2.54/12.0
      DRLFT=(DRL+2.0*THRL)/2.54/12.0
   33 AVUN=PI*((DMFT+2.0*TH)*ELMFT+
     2DFLFT*(ELFL1F+ELFL2F/1.375+ELFL3F/1.375)+2.0*(THFLFT+TH)*ELFLFT+
     3(DRLFT+2.0*TH)*ELRLFT)
C     AVUN IS VULNERABLE AREA OF MANIFOLDS,TAPERED FEED LINE,AND RETURN
C     LINE
      TH2=C3*AVUN**0.25
C     TH2 IS NEW BARRIER THICKNESS
C     COMPARE OLD AND NEW BARRIER THCIKNESSES,REPEAT AVUN CALCULATION
C     UNTIL AGREEMENT IS REACHED
      IF (ABSF(TH-TH2)/TH2-0.0001) 31,31,30
   30 TH=TH2
      GO TO 33
   31 TH=TH2*12.0*2.54
C     TH IS FINAL VALUE FOR BARRIER THICKNESS
C     GUESS TEMP DROP THROUGH BARRIER
      DELT1=10.0
  750 DELT2=CONDB/TH/HRES*DELT1
C     DELT2 IS T DROP ACROSS MANIFOLD-BARRIER INTERFACE
      DELT3=CONDB/TH/H*DELT1
C     DELT3 IS T DROP BETWEEN CONDENSING FLUID AND MANIFOLD WALL
      TBARR=TF-DELT1-DELT2-DELT3
C     TBARR IS BARRIER SURFACE TEMP
      Q1=CONDB/TH*DELT1
      Q2=SIG*1.0E-7*EPS*TBARR**4
C     CALCULATE REVISED DELT1,CHECK CONVERGENCE
      DELT1=Q2/CONDB*TH
      IF (ABSF(Q1-Q2)/Q2-0.001) 751,751,750
  751 CONTINUE
      QPIP=QMAX/SF*NPIPE*(1.0-HPP)
C     QPIP IS HEAT REJECTED FROM HEAT PIPES
      QMAN=QPIP/ELRAT*(TBARR/TP)**4
C     QMAN IS HEAT REJECTED FROM MANIFOLDS
      QFL=QMAN*(DFL/1.375/ELE)*(ELFL/ELMAN)
C     QFL IS HEAT REJECTED FROM FEED LINE
      QREJ=QPIP +QMAN+QFL
C     QREJ IS TOTAL HEAT REJECTED
      QRAT1=QPIP /QREJ*100.0
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      QRAT2=QMAN/QREJ*100.0
      QRAT3=QFL/QREJ*100.0
C     COMPARE QREJ WITH QTOT
      IF (ABSF(QREJ-QTOT)/QTOT-0.005) 540,540,541
  541 NPIPE=NPIPE*QTOT/QREJ
C     RETURN TO MANIFOLD DESIGN WITH REVISED NUMBER OF HEAT PIPES
      GO TO 753
  540 CONTINUE
      WRITE OUTPUT TAPE 3,209,QTOT,TP,TBARR,QREJ,QRAT1,QRAT2,QRAT3
  209 FORMAT(5X,27HHEAT REJECTION REQUIRED,KW=,E10.3//5X,21HHEAT PIPE TE
     2MP,DEG K=,E10.3/5X,21HBARRIER SURFACE TEMP=,E10.3//5X,29HCALCULATE
     3D HEAT REJECTION,KW=,E10.3/5X,33HHEAT REJECTION BREAKDOWN,PERCENTS
     4/10X,11HHEAT PIPES=,F6.2,5X,10HMANIFOLDS=,F6.2,5X,10HFEED LINE=,
     5F6.2//)
      WRITE OUTPUT TAPE 3,305,P,PIN,POUT
  305 FORMAT(5X,16HPRESSURE IN BARS/5X, 9HHEAT PIPE,6X,10HPRIMARY IN,5X,
     211HPRIMARY OUT/3(5X,E10.3)//)
      WRITE OUTPUT TAPE 3,202,DV,DW,THP,ELE,ELC,ROPT,NGROOV,NPIPE
  202 FORMAT(5X,7HD VAPOR,8X,6HD WICK,10X,7HTH PIPE,8X,6HL EVAP,9X,
     26HL COND,9X,5HR CAP,10X,7HGROOVES,8X,5HPIPES/6(5X,E10.3),
     32(5X,I10)//)
      WRITE OUTPUT TAPE 3,203,DIMAN,ELMAN
  203 FORMAT(5X,6HDI MAN,9X,5HL MAN/2(5X,E10.3)//)
      WRITE OUTPUT TAPE 3,206,EMDOT,RER
  206 FORMAT(5X,30HMASS FLOW IN HEAT PIPE,GM/SEC=,E10.3/5X,19HRADIAL REY
     2NOLDS NO=,E10.3,5X,46H(ANALYSIS VALID ONLY IF MUCH GREATER THAN ON
     3E)//)
      WRITE OUTPUT TAPE 3,360,STR1,STR2
  360 FORMAT(5X,32HMAX HEAT PIPE WALL STRESSES,BARS/5X,30HCOMPRESSION OF
     2 PUNCTURED PIPE=,E10.3/5X,22HTENSION OF WHOLE PIPE=,E10.3//)
      WRITE OUTPUT TAPE 3,802
  802 FORMAT(5X,30HPARALLEL MANIFOLD SEGMENTATION//5X,8HSEGMENTS,4X,
     26HREY NO,7X,9HFR P DROP,5X,14HSEGMENT LENGTH/)
      WRITE OUTPUT TAPE 3,801,NSEG,REMAN,DELP,ELMANS
  801 FORMAT(5X,I5,3(5X,E10.3))
      GO TO (530,531),KONFIG
  530 WRITE OUTPUT TAPE 3,502
  502 FORMAT(//5X,45HORIENTATION,AXIAL FEED LINE,LATERAL MANIFOLDS)
      GO TO 532
  531 WRITE OUTPUT TAPE 3,503
  503 FORMAT(//5X,45HORIENTATION,LATERAL FEED LINE,AXIAL MANIFOLDS)
  532 CONTINUE
      WRITE OUTPUT TAPE 3,84,DFL,DPFL,ELFL,DRL,DPRL,ELRL
   84 FORMAT(//5X,16HFEED LINE ID,CM=,E10.3,5X,10HFR P DROP=,E10.3,5X,
     216HTOTAL LENGTH,CM=,E10.3/5X,18HRETURN LINE ID,CM=,E10.3,5X,
     310HFR P DROP=,E10.3,5X,16HTOTAL LENGTH,CM=,E10.3///)
      WRITE OUTPUT TAPE 3,600
  600 FORMAT(5X,81HNOTE,FEED LINE ID IS AT ENTRANCE,DISTRIBUTION LENGTH
     2IS TAPERED AS (1-X/L)**0.375///)
      WRITE OUTPUT TAPE 3,1000,THMAN,THFL,THRL
 1000 FORMAT(5X,24HPIPE WALL THICKNESSES,CM/5X, 8HMANIFOLD,7X, 9HFEED LI
     2NE,6X,12HRETURN LINE /3(5X,E10.3)///)
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      WRITE OUTPUT TAPE 3,1005,TH
 1005 FORMAT(5X,31HMETEOROID BARRIER THICKNESS,CM=,E10.3//)
C     CALCULATE MASSES
C     MASS BREAKDOWN
C          WTP    HEAT PIPES
C          WTL    HEAT PIPE LIQUID
C          WTMAN  MANIFOLDS
C          WTFL   FEED LINE
C          WTRL   RETURN LINE
C          WTB    BARRIER
C          WT     TOTAL RADIATOR MASS WITHOUT PRIMARY FLUID
      VOLL=NPIPE*NGROOV*2.0*ROPT*(RW-RV)*EL
      VOLP=NPIPE*PI*((RW+THP)**2-RV**2)*EL-VOLL
      WTP=RHOP*VOLP/1000.0
      WTL=RHOL*VOLL/1000.0
      WTMAN=RHOM*ELMAN*PI*(DIMAN+THMAN)*THMAN/1000.0
      WTFL=RHOM*PI*(DFL*(ELFL1+ELFL2/1.375+ELFL3/1.375)+THFL*ELFL)*
     2THFL/1000.0
      WTRL=RHOM*ELRL*PI*(DRL+THRL)*THRL/1000.0
      WTB=RHOB*(AVUN*929.0)*TH/1000.0
      WT=WTP+WTL+WTMAN+WTFL+WTRL+WTB
      WRITE OUTPUT TAPE 3,35,WTP,WTL,WTMAN,WTFL,WTRL,WTB,WT
   35 FORMAT(5X,21HRADIATOR MASSES IN KG/5X,10HHEAT PIPES,5X,10HH PIPE L
     2IQ,5X, 9HMANIFOLDS,6X, 9HFEED LINE,6X,12HRETURN LINES,3X, 7HBARRIE
     3R/6(5X,E10.3)/5X,11HTOTAL MASS=,E10.3//)
      PCT1=WTP/WT*100.0
      PCT2=WTL/WT*100.0
      PCT3=WTMAN/WT*100.0
      PCT4=WTFL/WT*100.0
      PCT5=WTRL/WT*100.0
      PCT6=WTB/WT*100.0
      WRITE OUTPUT TAPE 3,100,PCT1,PCT2,PCT3,PCT4,PCT5,PCT6
  100 FORMAT(5X,13HMASS PERCENTS/8X,6HH PIPES,4X,6HHP LIQ,7X,3HMAN,6X,
     24HFEED,4X,6HRETURN,3X,7HBARRIER/5X,6(F10.2)//)
C     CALCULATE OVERALL RADIATOR SIZE
C     ELRAD IS RADIATOR LENGTH,WIDRAD IS WIDTH
      GO TO (514,515),KONFIG
  514 ELRAD=TEST2
      WIDRAD=TEST1+DFL+2.0*DRL
      GO TO 516
  515 ELRAD=TEST1+DFL+2.0*DRL
      WIDRAD=TEST2
  516 CONTINUE
      WRITE OUTPUT TAPE 3,36,ELRAD,WIDRAD
   36 FORMAT(5X,14HRAD LENGTH,CM=,E10.3,5X,13HRAD WIDTH,CM=,E10.3//)
C     PRIMARY FLUID INVENTORY
C          WTFLD1  FEED LINE
C          WTFLD2  MANIFOLDS
C          WTFLD3  RETURN LINE
C          WTFLD   TOTAL MASS OF PRIMARY FLUID IN RADIATOR
      VOLFL=PI*DFL*DFL/4.0*(ELFL1+ELFL2/1.75+ELFL3/1.75)
      VOLM=PI*DIMAN*DIMAN/4.0*ELMAN-NPIPE*PI*DP*DP/4.0*ELE
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      VOLRL=PI*DRL*DRL/4.0*ELRL
      WTFLD1=VOLFL/(VL+QUAL*(VG-VL))/1000.0
      WTFLD2=(1.0-HPP)*VOLM/(QUAL*(VG-VL))*LOGF((VL+QUAL*(VG-VL))/VL)/
     21000.0+HPP*VOLM/VL/1000.0
      WTFLD3=VOLRL/VL/1000.0
      WTFLD=WTFLD1+WTFLD2+WTFLD3
      PCT7=WTFLD1/WTFLD*100.0
      PCT8=WTFLD2/WTFLD*100.0
      PCT9=WTFLD3/WTFLD*100.0
      WRITE OUTPUT TAPE 3,520,WTFLD,PCT7,PCT8,PCT9
  520 FORMAT(5X,22HPRIMARY FLUID MASS,KG=,E10.3/5X,43HDISTRIBUTION OF PR
     2IMARY FLUID,MASS PERCENTS/11X,4HFEED,7X,3HMAN,4X,6HRETURN/5X,
     33(F10.2)//)
      RETURN
      END
```

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*       FORTRAN              PDROP2
        SUBROUTINE PDROP2(WDOT,P1,X1,X2,VG,VL,DVGDP,VIS,S,AC,EL,DELP)
C       THE SUBROUTINE PDROP2 CALCULATES THE FRACTIONAL PRESSURE DROP OF A
C       CONDENSING OR VAPORIZING FLUID FLOWING IN A CONSTANT AREA DUCT.
C       THE FOLLOWING ASSUMPTIONS ARE MADE. SMOOTH WALL DUCT,TURBULENT
C       FLOW,FOG FLOW MODEL (NO SLIP BETWEEN PHASES),NO GRAVITY EFFECT,
C       QUALITY CHANGES LINEARLY WITH POSITION,VG AND VL AND DVGDP ARE
C       ESSENTIALLY CONSTANT.
C
C       INPUT    UNITS            EXPLANATION
C       WDOT     G/SEC            MASS FLOW RATE
C       P1       DYNE/CM**2       INLET PRESSURE
C       X1                        INLET QUALITY
C       X2                        OUTLET QUALITY
C       VG       CM**3/G          SPECIFIC VOL OF VAPOR PHASE
C       VL       CM**3/G          SPECIFIC VOL OF LIQUID PHASE
C       DVGDP    CM**5/G/DYNE     DERIVATIVE OF VG WITH RESPECT TO PRESSURE
C       VIS      G/SEC/CM         VISCOSITY OF LIQUID PHASE
C       S        CM               WETTED PERIMETER
C       AC       CM**2            FLOW AREA
C       EL       CM               LENGTH OF PIPE
C
C       OUTPUT   UNITS            EXPLANATION
C       DELP                      FRACTIONAL PRESSURE CHANGE=(P2-P1)/P1
        PI=3.14159
C       CALCULATE FRICTIONAL AREA
        AW=S*EL
C       CALCULATE REYNOLDS NUMBER
        REN=2.0*WDOT/VIS/SQRTF(PI*AC)
        RENIN=1.0/REN
C       CALCULATE FANNING FRICTION FACTOR (ONE QUARTER OF DARCY-WEISBACH)
        FW=0.046*RENIN**0.20
        Z1=(WDOT/AC)**2*VG/2.0/P1
        A1=FW*AW/EL/AC*((X1-X2)/EL)*(1.0-VL/VG)
        B1=2.0*(1.0-VL/VG)*(X1-X2)/EL-FW*AW/EL/AC*((1.0-X1)*VL/VG+X1)
        C1=(WDOT/AC)**2*DVGDP*(X2-X1)/EL
        D1=1.0+(WDOT/AC)**2*DVGDP*X1
        IF (D1-0.0) 60,61,62
     60 WRITE OUTPUT TAPE 3,1
      1 FORMAT(//5X,42HSUPERSONIC FLOW,PDROP SUBROUTINE NOT VALID//)
        GO TO 75
     61 WRITE OUTPUT TAPE 3,2
      2 FORMAT(//5X,37HSONIC FLOW,PDROP SUBROUTINE NOT VALID//)
        GO TO 75
     62 CONTINUE
        IF (ABSF(C1*EL/D1)-0.001) 70,70,71
     70 DELP=Z1*(A1*EL**2/2.0/D1+B1*EL/D1)
        GO TO 75
     71 DELP=Z1*(A1*EL/C1+(B1*C1-A1*D1)/C1/C1*LOGF(ABSF(C1*EL/D1+1.0)))
     75 RETURN
        END
```