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SNAP PROGRAM

M-3679 (33rd Ed.)

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PERFORMANCE
ANALYSIS

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ABSTRACT

The most significant steady-state thermodynamic and fluid dynamic analyses of the SNAP 2 Nuclear 3 to 5 kw power supply is presented. In addition to the general system and design point analysis, the detailed analysis of the thermal and fluid dynamic performance of the Radiator-Condenser and the thermal performance of the instrument compartment are presented.

Special potential problem areas, including condenser stability, and the effects of reactor hydrogen leakage into the NaK system, are analyzed and discussed in detail.

Experimental programs involving, single tube condensing mercury pressure drop and stability, braze diffusion effects on the thermal conductivity of copper, and hydraulic loss coefficients of the boiler, pipe welds and bends, and the condensing tube inlet, are described and the available results presented.

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I. INTRODUCTION

The present SNAP 2 Nuclear 3 to 5 kw power supply is fundamentally the same as that originally conceived. However, a considerable number of changes in hardware and hardware design have occurred due to detailed analysis and experimental development, the results of which suggested improvements in performance, reductions in weight, and uncovered problem areas that could not be resolved without some system modification.

The performance analysis task has been directed toward determining all system thermal and hydraulic characteristics where two or more components are involved as well as detailed component thermal and/or hydraulic performance of the Radiator-Condenser and instrument compartment.

The primary task has been analytical, however, experimental work has been performed or proposed where detail support of the analysis is required. These detailed tests have included boiler NaK side pressure drop, braze diffusion effects on the copper thermal conductivity, entrance loss coefficients for mercury condensing tubes, hydrogen diffusion through the condenser tubes, mercury condensing pressure drop and stability, NaK piping weld and bend hydraulic losses, and vacuum thermal contact resistance of spring type pipe clamps.

This performance analysis report outlines the methods presently being used to establish the SNAP 2 performance requirements. These are by no means complete, and are constantly kept up to date through evaluation of component and system test data. The SNAP 2 flight design will be established by the techniques given in this performance analysis as verified by developmental system tests.

A. DESCRIPTION OF THE SYSTEM

The SNAP 2 auxiliary power unit utilizes a mercury vapor rankine cycle to generate electrical energy from a nuclear power source. The SNAP unit is composed primarily of a nuclear reactor heat source and a power conversion system (PCS), all contained in a radiator-condenser (RC) and nose cap which forms the Agena satellite nose cone (see Figure 1). Energy is produced in the nuclear reactor by the fission of U^{235} . A sodium potassium alloy (NaK), a liquid metal, serves as the heat transfer fluid and is circulated through the reactor core and a mercury boiler by a thermoelectric pump, a pump with no moving parts. The

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Figure 1. Agena Satellite Nose Cone

reactor heat is transferred from the NaK to the mercury in the boiler. The liquid mercury is converted into superheated vapor, which is expanded through a turbine, the resulting turbine mechanical power is converted to electrical power by an alternator. The mercury vapor exhaust from the turbine is condensed in a radiator-condenser and pumped back to the boiler. All the power conversion system rotating components are mounted on a single, common shaft, the combined rotating unit (CRU). The PCS, therefore, has only one moving part. The SNAP unit is a two-loop system, one loop is provided to circulate the NaK and the second to circulate the mercury. The SNAP unit is shown schematically in Figure 2.

B. SUMMARY OF SYSTEM PERFORMANCE

The SNAP 2 system primary NaK loop flow is 70 lb/min with a reactor outlet temperature of $1209^{\circ}\text{F} \pm 25^{\circ}\text{F}$. In the mercury loop, the turbine flow is 20 lb/min at 115 psia and 1150°F while the mean condensing temperature is 610°F . The minimum net electrical power output requirement is 3.00 kw. The present design has a minimum net power output capability of 3.72 kw with a reactor power generation of 55.37 kw. This results in an overall system efficiency of 6.7%. The total energy transferred to the mercury is 51.37 kw resulting in a 7.2% mercury cycle efficiency. A more complete summary of flows, temperatures, pressures, and energy exchange is presented in Tables I and II.

The maximum power capability of the combined rotating units (CRU's) is significantly larger than the design value, limited only by the allowable temperature of the alternator bore seal. An early design (CRU IV M) has demonstrated a gross output capability of 6.5 kw at 25.9 lb/min and 141 and 6.5 psia turbine inlet and exhaust pressure respectively. The measured performance of the advanced CRU (CRU V) when extrapolated indicates 6.0 kw can be obtained with a weight flow of 26.5 lb/min and 152 and 9.0 psia CRU inlet and exhaust pressures respectively. Also the same power could be obtained with higher mercury flow rates and lower turbine inlet pressures with a slight change in the turbine blading. Therefore, higher powers are attainable with only small changes in the turbine, boiler and radiator condenser.

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Figure 2. SNAP 2 Schematic

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TABLE I

SNAP 2 DESIGN POINT FLOW AND PRESSURE DISTRIBUTION

Component	Flow In (lb/min)	Flow Out (lb/min)	Pressure In (psia)	Pressure Out (psia)	Pressure Change (psia)
NaK System					
Reactor	70	70	19.80	19.60	-0.2
Thermoelectric Pump	70	70	19.34	20.39	1.05
Boiler	70	70	20.33	20.05	-0.28
Mercury System (Primary)					
Mercury Pump	47.05	45.05	6.0 ± 0.1	525 ± 20	519 ± 20
Bearing Heat Exchanger	20	20	525	524	-1
Thermoelectric Pump	20	20	519	517	-1
Boiler	20	20	180	119	-101
CRU	20	19.8	115	8.65 ± 0.35	-106
Radiator-Condenser	19.8	19.8	8.45 ± 0.35	6.20 ± 0.1	-2.25 ± 0.25
Mercury System (Bearings)					
Thrust Bearing	8.0	9.0	156	6.2 ± 0.1	-150
Alternator	8.7	9.8	218	6.2 ± 0.1	-212
Turbine Bearing	8.35	8.45	218	6.2 ± 0.1	-212
Bearing Heat Exchanger	27.25	27.25	6.2 ± 0.1	6.0 ± 0.1	-0.2

TABLE II

SNAP 2 DESIGN OPERATING CONDITIONS TEMPERATURE
AND ENERGY DISTRIBUTION - SNAP 2 DESIGN POINT

Component	Heat Exchange (kw)	Temperature In (°F)	Temperature Out (°F)	Temperature Change (°F)
NaK System				
Reactor (including parasitic load heaters)	54.44*	997	1209	212
Thermoelectric Pump	-2.91	1207	1195	-12
Boiler	-49.18	1195	1004	-191
Mercury System (Primary)				
Mercury Pump	0.07	375	381	6
Bearing Heat Exchanger	0.69	381	441	60
Thermoelectric Pump	2.87	441	692	251
Boiler	48.51	692	1160**	468
CRU	---	1150	628	-522
Radiator-Condenser	-47.09	623	303	-320
Mercury System (Bearings)				
Thrust Bearing	0.33	381	454	73
Alternator Bearing	0.36	381	454	73
Turbine Bearing	0.62	381	511	130
Bearing Heat Exchanger	-0.69	472	427	-45

*Total Reactor Power = 55.4 kw

**Minimum Design Value (Operating value = boiler inlet (NaK) - 10°F)

II. DESIGN POINT ANALYSIS

A. GENERAL

Design point analysis is broken into two phases. The first phase is the establishment of all the system limitations and requirements. This includes the optimization of the system weight as a function of condensing pressure, the selection of NaK flow rate based upon minimum boiler mercury inventory and system performance variations due to sun-shade operation, system tolerances, and reactor outlet temperature variation. In the second phase of the design point analysis, the remaining dependent system temperatures, flows, and pressures are determined.

B. SYSTEM DESIGN POINT SELECTION CRITERIA

The present SNAP 2 system design point is a result of several criteria established either by component performance limitations, minimum system weight, minimum electrical power requirements, variations in manufacturing tolerances or component degradation. The following sections describe these major design point criteria.

1. Minimum Net Electrical Power Delivered

The contract objective for the minimum net power delivered by the system is 3.0 kw.

2. Minimum Gross Electrical Power

The minimum gross electrical power is established at 3.51 kw by summing all the permanent system electrical requirements including:

- a) Parasitic load speed controller (at full load to Agena), 0.138 kw
- b) Short circuit protection capacitors, 0.030 kw
- c) Reactor temperature controller (0.018 kw at conversion efficiency = 87%), 0.021 kw
- d) Follow on instrumentation and signal conditioning (0.015 kw at conversion efficiency = 87%), 0.017 kw
- e) Reserved for system degradation and transient speed control, 0.300 kw
- f) Total control and degradation requirements, 0.51 kw

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g) Minimum net delivered power, 3.00 kw

h) Minimum gross electric power, 3.51 kw

3. Maximum Radiator-Condenser (RC) Exit Temperature

The maximum RC exit temperature has been established at 371°F which was dictated by the minimum NPSH required by the mercury pump based upon a minimum allowable pump inlet pressure of 5.9 psia.

4. Nominal Design Radiator-Condenser (RC) Exit Temperature

The nominal RC exit temperature of 303°F was established as the mean value between:

- a) The maximum of 371°F which occurs while operating in the sun after one year of system operation, with the least favorable combination of manufacturing, and system tolerances
- b) The minimum of 235°F which occurs after startup while operating in the shade with the most favorable combination of manufacturing and system tolerances.

The 136°F difference between the minimum and maximum represents a 40°F variation due to sun-shade operation and 96°F due to specific system manufacturing and performance tolerances. A summary of the parameters considered and their tolerance, and the statistical analysis is given in Table III.

5. Minimum Turbine Inlet Temperature

The minimum turbine inlet temperature was established at 1150°F based upon the maximum value consistent with a nominal 1200°F reactor and a practical reactor control system. During the life of the SNAP Unit, the turbine inlet temperature will probably vary between 1150 and 1200°F. The upper limit will decrease with improvements in the reactor control system.

6. Minimum Boiler Inlet NaK Temperature

The minimum boiler inlet NaK temperature has been established at 1170°F. This allows only 20°F between the minimum NaK inlet and the minimum turbine inlet temperature. Of this, 20°F, one-half is calculated to be required for the mercury vapor temperature drop between the boiler and the turbine. The remaining 10°F represents a reasonable maximum temperature difference between the exiting mercury vapor and the inlet NaK. Analytical estimates of this latter

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TABLE III.

SUMMARY OF THE EFFECT OF PARAMETER VARIANCE
ON THE RC EXIT TEMPERATURE

Parameter	Nominal Value	3σ Tolerance	σ_T Exit	σ_T^2 Exit
Inlet vapor quality (X)	0.982	± 0.004	1.8	3.24
Emissivity (ϵ)	0.91	$\pm 2.5\%$	11.4	130.2
Fin thermal conductivity (k)	17.6 Btu/hr in. $^{\circ}$ F	$\pm 10\%$	8.22	67.5
Mercury flow rate (ω)	19.9 lb/min	± 0.2 lb/min	5.20	27.15
Fin thickness at base (t_b)	14 mil	$\pm 1/2$ mil	2.72	7.4
Fin thickness at centerline between tubes (t_{CL})	6.5 mil	$\pm 1/2$ mil	0.1	0.01
Internal + external heat load (Q)	4.77 kw	± 0.5 kw	4.14	17.17
Analytical error			1.0	1.0
Summation			30.96	253.67
σ_T Exit				15.93*

* $3\sigma_T$ Exit = 47.79 $^{\circ}$ F

temperature difference indicate values less than one degree while experimental measurements to date result in temperature differences from 30 to 40 $^{\circ}$ F. It is felt that the large, experimental values are a result of the difficulty in properly insulating the end of the boiler in an air environment, and in measuring small temperature differences at these temperature levels without special care being taken. Boiler tests are planned with improved insulation and special temperature measuring techniques to verify the design point temperature difference of 10 $^{\circ}$ F.

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7. Nominal Turbine Inlet Pressure

The nominal turbine inlet pressure was established at 115 psia after the 1150°F minimum turbine inlet temperature was established, to provide the necessary degrees of superheat to minimize the moisture in the turbine. This was necessary to obtain high turbine efficiency and to minimize the turbine blade erosion in order to meet the one-year design life requirement.

8. Minimum Boiler Outlet NaK Temperature

The minimum boiler outlet NaK temperature just after system startup is complete, is governed by the requirement to minimize the boiler mercury inventory thus minimizing system weight. Low boiler NaK exit temperatures result in suppressed boiling. Suppressed boiling is defined as the region between where the mercury has been preheated to the NaK temperature and where boiling begins. In this region the NaK temperature is below the boiling point of the mercury. Boiling therefore is delayed or suppressed until the mercury liquid pressure drops to the saturation value corresponding to the NaK temperature. Since the boiler exit pressure is fixed by the turbine, this must be accomplished by reducing the boiler pressure drop (superheat length) (see Figure 3). The net result of suppressed boiling would be several feet of boiler tubing filled with liquid, adding nothing to the boiler performance, but adding considerably to the system weight. Since the pressure drop in the mercury preheat section is negligible, the minimum NaK temperature should be equal to the saturation temperature of the inlet mercury less the NaK ΔT required to preheat the mercury, plus some design pinch temperature difference. For a turbine inlet pressure of 115 psia and a 4 psi pressure drop between the boiler and turbine, and a design boiler pressure drop of 60 psia, the inlet mercury saturation temperature is 996°F. With a NaK preheat temperature drop of 15°F and a design pinch temperature difference of 10°F, the minimum boiler outlet temperature becomes 991°F.

9. Minimum NaK Flow Rate Exclusive of Degradation

Having selected the minimum boiler outlet temperature, the NaK flow rate which insures that this condition be met can be determined from the minimum boiler inlet temperature and the maximum mercury thermal load. This flow rate corresponds to about 75 lb/min. The actual minimum design point flow rate selected was 70 lb/min which corresponds to a NaK outlet temperature

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Figure 3. Typical Boiler Performance

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of 976°F and to 3 ft of additional boiler mercury inventory due to suppressed boiling. This condition will only occur if the boiler inlet sensor is on the lower end of the uncertainty band; if the required pinch temperature for boiling is 10°F, if the NaK pump does not meet performance by 23%, and if the NaK system pressure drop analysis is high by 24%. It is highly improbable that each of these conditions will exist simultaneously. A statistical analysis of these effects indicates that the probable limit of boiler outlet temperature will be 14°F above the minimum value of 976°F for 70 lb/min of NaK, or 990°F which is close enough to the minimum allowable (991°F) to be acceptable.

10. Maximum Steady-State Boiler Inlet NaK Temperature

The boiler inlet NaK temperature will decrease with time as the reactor reactivity decreases. Upon reaching the lower limit, the control drums will be stepped in resulting in an increase in boiler inlet temperature. In addition, the boiler inlet will vary due to NaK flow variations resulting from sun-shade operation effects on mercury cold junction temperature. Furthermore, to reduce high temperatures which may occur after mercury injection is complete, an upper temperature sensor is included in the system to step out the drums to reduce the reactor outlet temperature. To prevent limit cycling, it is necessary to set the upper temperature limit so that the boiler inlet temperature increase will not exceed the temperature sensor setting. A detailed analysis of the control system tolerances is presented.¹ The following is a summary of the results:

a) Reactor outlet temperature increase per step	9.3°F
b) Sun-shade allowance	4.0
c) Motor uncertainty	± 0.93
d) Gear dead space uncertainty	± 2.79
e) Drum worth uncertainty	± 0.28
f) Temperature coefficient uncertainty	± 1.19
g) Temperature sensor uncertainty	± 10.0
h) Model uncertainty	± 5.0
i) Dead band	29
j) Maximum boiler inlet temperature variation (due to sensor drift)	49

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- k) Maximum boiler inlet temperature ($1170 + 49$) 1219
- l) Maximum boiler inlet temperature increase per control drum step, including sun-shade allowance (but not including temperature sensor uncertainty). 20

11. Maximum Turbine Discharge Pressure and Nominal Mercury Flow Rate

The maximum turbine discharge pressure and corresponding nominal mercury flow were established by optimizing the system weight. In this study, the electrical power output was held constant and the independent variables were:

- a) Radiator-Condenser area
- b) Turbine discharge pressure
- c) Number of condensing tubes.

The dependent variables were:

- a) Dry RC weight
- b) Mercury startup inventory
- c) Radiating fin weight
- d) Condensing tube weight
- e) Meteoroid armor
- f) Mercury flow rate
- g) NaK piping weight
- h) Mercury tank weight
- i) Boiler weight (wet)
- j) RC pressure drop.

The method of optimizing was as follows:

- a) A set of curves of the variable weight versus number of condensing tubes were generated for several turbine discharge pressures and a single RC area. Figure 4 is typical of these curves, and shows a minimum system weight occurs at 40 condensing tubes.

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Figure 4. System Optimization for 120 sq ft Radiator-Condenser and 9 psia Turbine Discharge Pressure

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- b) Each of the minimum system weights were plotted on a separate curve to determine the optimum turbine discharge pressure for an RC area of 120 ft^2 . This process was repeated for RC areas of 110, 115, and 125 ft^2 . Figure 5 is typical of the resultant curves and shows that the minimum system weight occurs at a turbine discharge pressure of 9 psia. In fact, 9 psia was the optimum for each of the RC areas considered. The mercury flow rate corresponding to the 9 psia turbine discharge pressure was simultaneously found to be 20 lb/min.
- c) Having determined the minimum system weight for each RC area, these were then plotted as shown in Figure 6 to determine the optimum RC area (120 ft^2) for minimum system weight. The net result of this weight optimization study was the following system design criteria:
- 1) Turbine discharge pressure equals 9 psia
 - 2) Mercury flow rate equals 20 lb/min
 - 3) Radiator-condenser area equals 120 ft^2
 - 4) Number of condensing tubes equal 40.

Since this study was performed, the system components have undergone significant design changes and corresponding weight changes and, therefore, the actual weights shown in Figure 4, 5, and 6 are no longer applicable. However, the shape of the curves and the resultant optimum conditions have not deviated significantly.

12. Established Criteria

Many requirements and design conditions within the CRU have been established by the power conversion system subcontractor, by analysis and/or experiment, and subsequently negotiated and approved by AI. These include:

a) Turbine efficiency	54.3%
b) Turbine shaft power	5610 watts
c) Alternator efficiency (not including capacitor losses)	88%
d) Mercury pump discharge pressure	$525 \pm 20 \text{ psia}$

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Figure 5. Effect of RC Pressure Drop on System Weight (120 ft²) RC

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Figure 6. System Optimization for 40 Tube Radiator-Condenser and 9.0 psia Turbine Discharge Pressure

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e) Mercury pump efficiency	33%
f) Mercury pump maximum inlet temperature	443°F

Bearing power consumption (three pad)

a) Alternator	210 watt
b) Turbine	210 watt
c) Thrust	100 watt

Bearing flow rate (three pad)

a) Alternator	8.7 lb/min
b) Turbine	8.35 lb/min
c) Thrust	8 lb/min

Bearing drain outlet temperature

a) Thrust and alternator	522°F
b) Turbine bearing	579°F

Bearing drain flow

a) Alternator and thrust bearing drain	18.8 lb/min
b) Alternator and thrust bearing	16.7 lb/min
c) Mercury pump leakage	2.0 lb/min
d) Turbine exhaust vapor leakage	0.1 lb/min
e) Turbine bearing drain	8.45 lb/min
f) Turbine bearing	8.35 lb/min
g) Turbine exhaust vapor leakage	0.1 lb/min

C. SYSTEM DESIGN POINT ANALYSIS

Having established the design point criteria and limitations, it is a relatively simple matter to establish the temperatures and pressure throughout the system.

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1. Mercury System Analysis

Since the bearing drain temperatures, pump inlet temperature, and the RC exit temperatures have been previously determined (see Figure 2), the primary result of the mercury thermal analysis is the required bearing heat exchanger effectiveness, the bearing heat exchanger high pressure-mercury outlet temperature, and the enthalpy and quality of the vapor entering the radiator-condenser.

a. Bearing Drain Heat Exchanger Analysis

The following set of equations result in determining the required bearing heat exchanger effectiveness, when it is remembered that the pump inlet temperature must not exceed 443°F when the RC exit temperature is 371°F.

$$\eta_{BE} = \frac{W_{HgBD} (T_{HgBDO} - T_{HgBDHO})}{W_{HG} (T_{HgBDO} - T_{HgPO})} \quad \dots (1)$$

where

η_{BE} = bearing drain heat exchanger effectiveness

W_{Hg} = boiler mercury flow rate - 20 lb/min

W_{HgBD} = bearing drain flow - 27.25 lb/min

W_{HgBD} = alternator bearing flow (8 lb/min)
 + thrust bearing flow (8.7 lb/min)
 + turbine bearing flow (8.35 lb/min)
 + pump leakage flow (2 lb/min)
 + turbine exhaust vapor leakage (0.2 lb/min)

T_{HgBDO} = is the mixed mean bearing drain temperature

T_{HgBDHO} = the bearing drain heat exchange lower pressure-mercury outlet temperature

T_{HgPO} = mercury pump outlet temperature ($T_{HgPO} = T_{HgPi} + 6$)

T_{HgPi} = mercury pump inlet temperature

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The maximum value for $T_{\text{Hg BDHO}}$ is determined from the following equation evaluating the RC exit temperature and pump inlet temperature at their maximum values of 371°F and 443°F, respectively. This equation simply expresses the mixing between the bearing drain heat exchanger outlet and the RC exit stream prior to entering the pump.

$$T_{\text{Hg BDHO}} = \frac{(W_{\text{HgRC}} + W_{\text{HgBD}}) T_{\text{HgPi}} - T_{\text{RCD}} W_{\text{HgRC}}}{W_{\text{HgBD}}} \quad \dots (2)$$

The high pressure mercury bearing drain heat exchanger outlet temperature can then be easily determined from a heat balance:

$$T_{\text{HgBHO}} = T_{\text{HgPO}} + \frac{W_{\text{HgBD}}(T_{\text{HgBDO}} - T_{\text{HgBDHO}})}{W_{\text{Hg}}} \quad \dots (3)$$

b. CRU Thermal Balance

The remaining significant analysis in the mercury liquid system is the thermal balance on the CRU, which determines the enthalpy and quality entering the radiator-condenser.

The entering energy (Q_{in}) is given by (Note: The bearing supply is assumed integral with the CRU):

$$Q_{\text{IN}} = W_{\text{HgPi}} h_{\text{HgPi}} + W_{\text{Hg}} h_{\text{HgTi}} \quad \dots (4)$$

where

W_{HgPi} = the mercury pump inlet flow - 47.05 lb/min

h_{HgPi} = pump inlet enthalpy

h_{HgTi} = turbine inlet mercury enthalpy - 159.27 Btu/lb

W_{Hg} = boiler mercury flow - 20 lb/min

The energy leaving the CRU (Q_{out}) is simply:

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$$Q_{out} = W_{Hg} h_{HgPO} + W_{HgBD} h_{HgBDO} + \frac{Q_{CRU}}{60} + \frac{(PAO)3.413}{60} + W_{HgR/C} h_{HgR/C} \quad \dots(5)$$

where:

h_{HgDO} = mercury pump outlet enthalpy

h_{HgBDO} = bearing drain mixed mean enthalpy

$h_{HgR/C}$ = RC entering vapor enthalpy (assumed equal to CRU vapor exhaust)

Q_{CRU} = net energy radiated and conducted from CRU -34Btu/hr

PAO = electrical energy leaving the CRU

$$PAO = \eta_{alt} \left[\frac{60 W_{Hg} (\Delta h_{TiD}) \eta_T}{3.413} - (P_{Ba} + P_{BTH} + P_{BT}) + P_{HgP} \right]$$

where:

$$P_{HgP} = \frac{\Delta P_{Hg} W_{HgPO}}{\eta_P \rho_{Hg}} \left[\frac{(60)(144)}{778(3.413)} \right]$$

η_{alt} = alternator efficiency - 88% not including losses due to short circuit capacitors

η_P = mercury pump efficiency -33%

Δh_{TiD} = ideal enthalpy drop across turbine -29.2 Btu/lb

η_T = turbine efficiency based on a 9 psia turbine exhaust pressure -54.3%

P_{BA} = alternator bearing power consumption 210 watts

P_{BTH} = thrust bearing power consumption 100 watts

P_{BT} = turbine bearing power consumption 210 watts

P_{HgP} = turbine pump shaft power - watts

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ρ_{Hg} = mercury density at pump inlet (lb/ft³)

ΔP_{Hg} = mercury pump pressure rise (psi)

Equating Equations 4 and 5, the enthalpy of the exiting vapor is determined and therefore, the RC inlet conditions. These results are summarized in Figure 7 and Table IV.

2. NaK System Analysis

In the analysis of the NaK system, the parasitic load heater shall be considered part of the reactor. Therefore, the reactor inlet and outlet temperatures shall actually be the temperatures downstream and upstream of the parasitic load heaters and the temperatures downstream and upstream of the reactor, respectively. This is convenient, since it makes the design point independent of the electrical load. At the design point the temperature rise across a single heater will vary between 1 and 7°F.

A heat exchanger type analysis of the TE pump provides an equation for the TE mercury pump outlet temperature as follows:

$$T_{\text{HgTEO}} = T_{\text{HgBHO}} + \eta_{\text{TE}} (T_{\text{NaKTEi}} - T_{\text{HgBHO}}) \quad \dots (6)$$

where:

T_{HgTEO} = TE pump outlet mercury temperature

T_{HgBHO} = bearing heat exchanger outlet high pressure-mercury temperature

T_{NaKTEi} = TE pump inlet NaK temperature

η_{TE} = TE pump heat exchanger effectiveness -0.332

The mercury inlet temperature to the boiler is equal to the TE pump outlet temperature less that due to the line heat loss.

$$T_{\text{HBi}} = T_{\text{HgTBO}} - \frac{Q_{\text{TEBR}}}{60 W_{\text{Hg}} C_{\text{Hg}}} \quad \dots (7)$$

* THIS TURBINE EFFICIENCY BASED ON AN EXHAUST PRESSURE OF 9.0 psia

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Figure 7. SNAP 2 System Flow and Energy Schematic

TABLE IV
DESIGN POINT SYSTEM, ENERGY, AND POWER BALANCE

I. ELECTRICAL POWER REQUIREMENTS

Payload (kw)	3.00
Parasitic load controller, discriminator, and preamplifier (at full load to Agena) (kw)	0.138
Short circuit protection capacitors (kw)	0.030
Reactor temperature controller (0.018 kw at conversion efficiency = 87%) (kw)	0.021
Follow on instrumentation and signal conditioning (0.015 kw at conversion efficiency - 87% (kw)	0.017
Reserved for system degradation and transient speed control (kw)	<u>0.300</u>
Total control and degradation requirements (kw)	<u>0.51</u>
Minimum required alternator power (kw)	3.51
Excess design point power (kw)	<u>0.72</u>
Estimated gross alternator output (kw)	4.23

II. TURBINE POWER REQUIRED

A. Alternator

Electrical output (kw)	4.23
Efficiency (%)	88
Shaft power required (kw)	4.806

B. Mercury Pump

Useful flow (lb/min)	45
Developed head (psi)	519
Efficiency (%)	33
Shaft power required (kw)	0.284

C. Bearings and Seals

Journal bearings (kw)	0.420
Thrust bearings (kw)	0.100
Shaft power required (kw)	<u>0.520</u>
Total turbine shaft power required (kw)	5.610

TABLE IV (Continued)
DESIGN POINT SYSTEM, ENERGY, AND POWER BALANCE

III. REACTOR AND PRIMARY LOOP HEAT BALANCE

Total reactor power generated (kw)	55.37
Reactor heat loss to space (kw)	-1.80
Primary loop heat losses (kw)	-3.17
Hot leg (0.120 kw to space) (kw)	-0.562
Boiler shell (kw)	-0.627
Cold leg (0.040 kw to space) (kw)	-1.724
Thermoelectric pump (kw)	-0.043
Expansion compensator (kw)	-0.005
Parasitic load heaters (to space) (kw)	-0.110
Hot leg brackets (kw)	-0.024
Cold leg brackets (kw)	-0.025
Boiler brackets (kw)	-0.050
Primary loop heat input (parasitic load) (kw)	<u>1.02</u>
Heat to mercury (kw)	51.42
Heat to mercury in TE pump (kw)	2.91
Heat to mercury in boiler (kw)	48.51

IV. BOILER HEAT BALANCE

Net heat input to mercury (includes 0.054 kw line loss from TE pump to boiler (kw)	51.37
Heat input to thermoelectric pump (kw)	2.909
Heat radiated from line to boiler (kw)	-0.054
Heat input to boiler (kw)	49.188
Heat radiated and conducted from boiler to APU (kw)	-0.677
Heat content of mercury entering boiler* (kw)	4.72
Heat content leaving RC (kw)	3.060
Heat from pressure regulator (kw)	0.080
Heat from bearing drains (kw)	0.824
Heat from mercury pump (kw)	0.069
Heat from bearing drain heat exchanger (kw)	0.689
Heat lost by radiation from turbine supply line (kw)	<u>-0.09</u>
Heat content entering turbine (kw)	56.00

*Referenced to 32°F

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TABLE IV (Continued)

DESIGN POINT SYSTEM, ENERGY, AND POWER BALANCE

V. CRU HEAT BALANCE	
Heat content entering turbine (20 ppm) (kw)	56.00
Heat content leaving through liquid mercury (kw)	-1.61
Heat content entering mercury pump (47.05 ppm) (kw)	9.326
Heat content leaving mercury pump (20 ppm) (kw)	-4.033
Heat content leaving alternator and thrust bearing drain (18.8 ppm) (kw)	-4.573
Heat content leaving turbine bearing drain (8.45 ppm) (kw)	-2.330
Heat radiated from CRU (kw)	-0.01
Gross alternator power out (88% efficiency) (kw)	<u>-4.23</u>
CRU exhaust heat content (19.8 ppm) (kw)	50.15
VI. RADIATOR-CONDENSER HEAT BALANCE	
CRU exhaust heat content (kw)	50.15
Primary loop heat loss to RC (kw)	2.90
CRU mercury casing and line losses (kw)	0.10
Conduction, radiation to cold compartment (kw)	-0.30
Heat content of exiting liquid mercury (kw)	<u>-3.06</u>
Net heat rejection rate to space (kw)	49.79
VII. INSTRUMENT COMPARTMENT HEAT BALANCE	
Radiation from thermal barrier (kw)	0.01
Conduction from radiator-condenser (kw)	0.03
Conduction along electrical leads (kw)	0.26
Conduction and radiation to Agena* (kw)	0.00
Instrumentation (diagnostic and follow-on) (kw)	0.090
Controllers (full load to parasitic load heaters) (kw)	0.156
Short circuit capacitors (kw)	<u>0.030</u>
Net heat rejection rate to space (kw)	0.58

*Present design conditions

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TABLE IV (Continued)

DESIGN POINT SYSTEM, ENERGY, AND POWER BALANCE

VIII. OVERALL PACKAGE HEAT BALANCE

A. Heat Generated in Package

Reactor (kw)	55.37
Reflector (kw)	0.70
Shield (kw)	<u>0.50</u>
	56.57

B. Heat Rejected From Package

From radiator-condenser to space (kw)	49.79
From NaK lines and parasitic load heater (kw)	0.27
From reactor to space (kw)	1.80
From reflector to space (kw)	0.70
From shield to space (kw)	0.50
From instrument compartment to space (kw)	0.58
To payload as electricity (kw)	3.04
To instrument compartment from payload	<u>-0.11</u>
	56.57

Where:

T_{HgBi} = boiler inlet mercury temperature

Q_{TEBR} = net heat radiated from mercury line to the boiler from the
TE pump (186 Btu/hr)

W_{Hg} = mercury flow rate (20 lb/min)

C_{Hg} = mercury specific heat (0.03238 Btu/lb-°F)

The boiler heat load (Q_B) is determined by the energy required to preheat, boil, and superheat the mercury, as given by:

$$Q_B = 60 W_{\text{Hg}} (h_{\text{HgBO}} - h_{\text{HgBi}}) \quad \dots (8)$$

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Where:

h_{HgBO} = the boiler outlet mercury vapor enthalpy — 159.52 Btu/lb at 1160°F and 119 psia

h_{HgBi} = boiler inlet mercury liquid enthalpy (Btu/lb)

and from Reference 4:

$$h_{\text{HgBi}} = 1.070 + 3.352 \times 10^{-2}T - 2.069 \times 10^{-6}T^2 + 1.245 \times 10^{-9}T^3$$

where:

T = saturated mercury temperature in °F and $32^\circ\text{F} \leq T \leq 900^\circ\text{F}$.

The boiler inlet NaK temperature ($T_{\text{NaK Bi}}$) becomes simply:

$$T_{\text{NaK Bi}} = T_{\text{NaK BO}} + \frac{Q_B + Q_{\text{BR}}}{60 W_{\text{NaK}} C_{\text{NaK}}} \quad \dots (9)$$

where:

$T_{\text{NaK BO}}$ = boiler outlet NaK temperature (°F)

$T_{\text{NaK Bi}}$ = boiler inlet NaK temperature (°F)

Q_{BR} = net boiler heat loss by radiation and conduction (2310 Btu/hr)

W_{NaK} = NaK flow rate (70 lb/min)

C_{NaK} = NaK specific heat — 0.209 Btu/hr-°F C 1100°F .

The temperature change in the NaK line from the TE pump to the boiler is given by:

$$T_{\text{NaK TEO}} = T_{\text{NaK Bi}} + \frac{Q_{\text{TEBR}}}{60 W_{\text{NaK}} C_{\text{NaK}}} \quad \dots (10)$$

Where:

$T_{\text{NaK TEO}}$ = TE pump NaK outlet temperature

Q_{TEBR} = net heat radiated from NaK line from TE pump to boiler (368 Btu/hr).

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The NaK inlet temperature to the TE pump is given by a heat balance on the TE pump as follows:

$$T_{\text{NaK TEi}} = T_{\text{NaK TEO}} + \frac{(h_{\text{Hg TEO}} - h_{\text{Hg BHo}}) W_{\text{Hg}} + Q_{\text{TER}}}{W_{\text{NaK}} C_{\text{NaK}}} \quad \dots(11)$$

Where:

$T_{\text{NaK TEi}}$ = TE pump NaK inlet temperature (°F)

$h_{\text{Hg TEO}}$ = TE pump mercury inlet enthalpy (Btu/lb)

$h_{\text{Hg BHo}}$ = TE pump mercury outlet enthalpy (Btu/lb)

Q_{TER} = net heat radiated and conducted from TE pump .

Equations 6 through 11 can be solved by iteration. Once these equations are solved, the reactor inlet and outlet temperatures can be determined by:

$$T_{\text{NaK RO}} = T_{\text{NaK TEi}} + \frac{Q_{\text{RTER}}}{60 W_{\text{NaK}} C_{\text{NaK}}} \quad \dots(12)$$

and

$$T_{\text{NaK Ri}} = T_{\text{NaK BO}} - \frac{Q_{\text{BRR}}}{60 W_{\text{NaK}} C_{\text{NaK}}} \quad \dots(13)$$

Where:

$T_{\text{NaK RO}}$ = reactor outlet temperature (°F)

$T_{\text{NaK Ri}}$ = reactor inlet temperature (°F)

Q_{RTE} = net heat radiated and conducted from the line to the TE pump from the reactor (1633 Btu/hr)

Q_{BRR} = net heat radiated and conducted from the line to the reactor from the boiler (5997 Btu/hr).

The total reactor and parasitic load heater power required is given simply as:

$$Q_{\text{React}} = 60 W_{\text{NaK}} C_{\text{NaK}} (T_{\text{NaK RO}} - T_{\text{NaK Ri}}) + Q_{\text{RR}} \quad .$$

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Where:

Q_{React} = reactor and parasitic load heater power (Btu/hr)

Q_{RR} = heat conducted and radiated from reactor (1706 Btu/hr)

The temperature results of this analysis for the nominal operating condition of an RC exit temperature of 303°F and boiler inlet temperature of 1195°F are given in Figure 7. The design point energy and power balance is shown in Table IV. More complete results are presented.²

D. OFF DESIGN PERFORMANCE

As previously discussed, the design point has been established at the following primary conditions:

1) Turbine mercury flow (lb/min)	20
2) NaK flow (lb/min)	70
3) Turbine inlet temperature (°F)	1150
4) Turbine exhaust pressure (psia)	9.0
5) Boiler inlet temperature (°F)	1195
6) RC exit temperature (°F)	303

However, during actual orbital operation, the operating condition will be continuously varying and even the nominal condition may be different from the design point due to sun-shade operation, reactor control variation, boiler mercury flow regulator tolerance, RC pressure drop uncertainties, pressure regulator tolerances, and NaK system and pump uncertainties. A continuous effort is being made to minimize those conditions which, at present, result in the following possible variations in operating conditions:

1) Turbine mercury flow (lb/min)	19 ± 0.2
2) NaK flow (lb/min)	83 ± 13
3) Turbine inlet temperature (°F)	1175 ± 25
4) Turbine exhaust pressure (psia)	8.65 ± 0.35
5) Boiler inlet temperature (°F)	1195 ± 25
6) RC exit temperature (°F)	303 ± 68

The primary components affected by these variations are the RC, boiler, turbine, alternator, parasitic load heater, and parasitic load controller. The effect on the boiler and RC operation has already been discussed and, therefore,

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the primary concern here is variation in turbine power and, consequently, the maximum power requirements of the alternator and the parasitic load control and heater. The calculation of the turbine off-design performance was made using the following approximate analysis which was felt to be adequate for initial estimates of the small variations involved. A much more detailed analysis which includes variations in turbine efficiency is in preparation.

1. Effect of Turbine Inlet Flow and Temperature Variations

For a choked first stage nozzle, the continuity equation becomes:

$$\frac{W \sqrt{T_T R}}{P_T A} = C \quad \dots (14)$$

where:

W = the weight flow (lb/sec)

T_T = total temperature ($^{\circ}R$)

P_T = total pressure (psia)

R = gas constant (ft/ $^{\circ}R$)

A = flow area times an appropriate flow coefficient (in.²)

C = a constant which for a perfect gas with $k = 1.663$, $C = 4.12$.

Therefore, the relation to determine the deviations in inlet pressure due to variations in flow and/or temperature is simply:

$$P_{T2} = P_{T1} \frac{W_2}{W_1} \sqrt{\frac{T_2}{T_1}} \quad \dots (15)$$

Using the new inlet pressure and temperatures, the inlet enthalpy can be determined and the change in turbine power can be determined from the relation.

$$\Delta P_W = \frac{60 \eta_T (h_i - h_{eiD}) W - P_{WQ}}{3.1413} \quad \dots (16)$$

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where

η_T = turbine efficiency = 54.3% (assumed constant for small perturbations)

η_i = turbine inlet enthalpy

h_{eiD} = ideal turbine exhaust enthalpy

P_{W0} = nominal turbine power at 1150°F and 20 lb/min = 5610 watts

The results of this analysis modified by the alternator efficiency of 88% are shown in Figure 8 and the tabulated variations corresponding to the system operating condition in Table V.

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Figure 8. CRU-V Off-Design
Performance Inlet Temperature
and Flow Variation

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TABLE V
OFF-DESIGN POWER VARIATIONS

Parameter	Mean Value	Variation	Variation in Alternator Power (watts)
Turbine inlet temperature (°F)	1175	±25	90
Turbine flow (lb/min)	19.9	± 0.2	130
CRU exhaust pressure (psia)	8.65	± 0.35	120
Total Variation			340

2. Effect of Turbine Discharge Pressure

This analysis was performed with the following assumptions:

- a) First stage power is constant
- b) Second stage theoretical efficiency is constant at $\eta = 61\%$
- c) Overall turbine theoretical efficiency at design point condition is 60%
- d) First stage discharge conditions are:

$$h = 151.3 \text{ Btu/lb}$$

$$P = 51 \text{ psia}$$

$$X = 0$$

- e) First stage power is 2300 watts.

The analysis consisted of (a) determining the ideal enthalpy drop for each discharge pressure and (b) multiplying by the stage efficiency and adding the first stage power. To obtain the change in performance with discharge pressure for the expected overall turbine efficiency of 54.3% (based on previous experimental results which showed a significant decrease in experimental turbine compared to the analytically determined efficiency) the following relation was used:

$$\Delta P_W = \frac{\eta_T}{\eta_{TD}} (P_W - P_{WOD}) \quad \dots (17)$$

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where

η_T = estimated experimental turbine efficiency (54.3%)

η_{TD} = design turbine efficiency (60%)

P_W = turbine power as calculated above

P_{WOD} = turbine power at the design point as calculated above = 6130 watts.

This analysis will be improved in the near future and finally replaced by experimental data as it becomes available. The results modified by the alternator efficiency of 88% are shown in Figure 9 and Table V.

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Figure 9. CRU-V Off-Design
Performance Exhaust
Pressure Variation

3. Effect of System Parameter Variations on Electrical Component Performance Requirements

As can be seen from Table V, the maximum power variation from the alternator under the worst possible combinations of conditions is 340 watts. From Table IV, this results in an overall maximum electrical load of 4.57 kw.

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While this is considerably over the present design requirements of 3.51 kw, no problems are anticipated since recent tests indicate that the present alternator can handle at least 6.0 kw. If the payload does not have a steady minimum load to absorb the excess power it may be necessary to increase the capability of the parasitic load and load control. In any event the present system design point appears capable of 4 kw minimum net power initially with 3.7 kw available after a year of operation due to system degradation. As will be discussed later, the system power can be upgraded to 5 kw net electrical power with a minor variation in design point.

4. Effect of Ground Operation

The primary differences between ground and orbital operation are the effects of gravity and the vacuum tank wall. As far as the vacuum tank wall is concerned, steady-state operation on the ground could be adequately simulated with a large vacuum tank and a wall temperature around -50°F. Practically, it is not economically desirable. The two primary effects which have to be considered are (a) the increased RC exit temperature and (b) the increased turbine discharge pressure.

The variation in RC exit temperature was calculated.³ In system tests the subcooler height will be approximately 6 in. with an exit temperature of about 450°F. To account for this, an auxiliary ground subcooler is provided between the RC exit and the mercury pump inlet. During ground operation, kerosene at 60°F will be circulated to assure that the RC exit stream is cooled to at least 370°F.

The increase in turbine back pressure is a result of the following influences:

- a) The effect of gravity fixes the subcooler height at approximately the level in the pressure regulator.
- b) The high sink temperature (vacuum tank wall temperature) requires higher radiator temperature and/or increased condensing heat rejection area.

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To obtain the approximate variation in CRU back pressure, the following approximate analysis was made.

The net heat rejection from the condensing section in space is given by

$$Q_{NET} = \eta_f A_C \sigma \epsilon T_{RC}^4 - Q_S \quad \dots (18)$$

where

η_f = average condensing fin effectiveness (0.87)

A_C = condensing area (99 ft²)

σ = Stephan-Boltzmann Constant (0.173×10^{-8} Btu/hr-°F⁴-ft²)

T_{RC} = mean condensing temperature (600°F)

ϵ = RC surface emissivity (0.90)

Q_S = energy received from space including direct solar, albedo, and earth radiation - 6170 Btu/hr for entire RC when 50% of RC is exposed to direct solar radiation.

To obtain the same condensing length in a vacuum chamber, the tank wall temperature must satisfy the equation.

$$Q_{NET} = \eta_f A_C \sigma \mathcal{F} (T_{RC}^4 - T_{W_o}^4) \quad \dots (19)$$

where:

$\mathcal{F}$ = the radiation interchange factor between the RC and the vacuum chamber, 0.88 for PSM-1 and 0.86 for FS-1

T_{W_o} = vacuum tank wall temperature required for solar simulation.

Equating Equations 18 and 19 and solving for T_{W_o} gives:

$$T_{W_o} = \left[\frac{Q_S}{\eta_f A_C \sigma} - \left(\frac{\epsilon}{\mathcal{F}} - 1 \right) T_{RC}^4 \right]^{1/4} \quad \dots (20)$$

For the developmental system test configuration the equivalent temperature is -88°F.

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For the flight system test configuration, cooling to absolute zero temperature would not be adequate. These low results are primarily due to the second term in Equation 20 which represents the heat originating from the RC which is reflected back and absorbed by the RC. Since adequate cooling on the ground cannot be easily obtained, all system tests will incorporate some kind of auxiliary subcooler.

The difference in wall heat rejection, per square foot of RC surface, (ΔQ_W) between the space simulating condition and the actual vacuum tank wall temperature is simply:

$$\Delta Q_W = \sigma \mathcal{F} (T_W^4 - T_{W_0}^4) \quad \dots (21)$$

or from Equation 20

$$\Delta Q_W = \sigma \mathcal{F} \left[T_W^4 - \frac{Q_S}{\eta_f A_C \sigma \mathcal{F}} + \left(\frac{\epsilon}{\mathcal{F}} - 1 \right) T_{RC}^4 \right] \quad \dots (22)$$

The increase in wall heat flux which corresponds to an increase in condensing length and temperature is given by:

$$\Delta Q_W = \left(\frac{\partial Q}{\partial L} \right) \Delta L + \left(\frac{\partial Q}{\partial T} \right) \Delta T \quad \dots (23)$$

where

Q = the heat flux from the RC Btu/hr-ft²

L = condensing length (inches)

T = condensing temperature (°R).

The increase in inlet RC pressure (ΔP_{CRU}) which corresponds to the change in CRU back pressure is given by:

$$\Delta P_{CRU} = \left[\frac{\partial(\Delta P_{RC})}{\partial L} \Delta L + \rho_{Hg} (\Delta L + L_{PR} - L_{SC}) \right] \quad \dots (24)$$

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where

ΔP_{RC} = condensing pressure drop (psi)

ρ_{Hg} = mercury density (lb/in.³)

L_{PR} = mercury level in pressure regulator (in.)

L_{SC} = subcooler nominal level = 14 in.

The first term on the right of Equation 20 represents simply the change in condensing pressure drop due to a change in condensing length (the effect of temperature has been neglected), and the second term represents the change in pressure due to the effects of gravity on the displaced liquid subcooler.

The final relation gives the mean condensing temperature rise based upon one-half of the rise in pressure drop plus the rise due to the displaced head of liquid mercury.

$$\Delta T = \left(\frac{\partial T}{\partial P} \right) \left[\frac{\Delta L}{2} \left(\frac{\partial \Delta P_{RC}}{\partial L} \right) + (\Delta L + L_{PR} - L_{SC}) \rho_{Hg} \right] \quad \dots (25)$$

where $(\partial T / \partial P)$ is evaluated along the saturated vapor line. Equations 22 through 25 can be solved directly for the change in CRU back pressure by eliminating the change in condensing length resulting in:

$$\Delta P_{CRU} = \frac{\left(\frac{\partial \Delta P_{RC}}{\partial L} + \rho_{Hg} \right) \left\{ \sigma \mathcal{F} \left[T_W^4 - \frac{Q_S}{\eta_f A_C \sigma \mathcal{F}} + \left(\frac{\epsilon}{\mathcal{F}} - 1 \right) T_{RC}^4 \right] + \left(\frac{\partial Q}{\partial T} \right) \left(\frac{\partial T}{\partial P} \right) (L_{SC} - L_{PR}) \rho_{Hg} \right\}}{\frac{\partial Q}{\partial L} + \left(\frac{\partial Q}{\partial T} \right) \left(\frac{\partial T}{\partial P} \right) \left[\frac{1}{2} \frac{\partial \Delta P_{RC}}{\partial L} + \rho_{Hg} \right]} - \rho_{Hg} (L_{SC} - L_{PR}) \quad \dots (26)$$

The constants in Equation 22 were evaluated for the normal condensing conditions and were found to be:

$$\frac{\partial \Delta P_{RC}}{\partial L} = 0.137 \text{ psi/in.}$$

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$$\frac{\partial Q}{\partial L} = 21.0 \text{ Btu/hr-in.-ft}^2$$

$$\frac{\partial T}{\partial P} = 11.0^\circ\text{F/psi}$$

$$\frac{\partial Q}{\partial T} = 8.0 \text{ Btu/hr-}^\circ\text{F-ft}^2$$

$$\text{Hg} = 0.465 \text{ lb/in.}^3 \quad C = 590^\circ\text{F.}$$

Substitution into Equation 22 results in:

For the developmental system test

$$\Delta P_{\text{CRU}} = 1.35 \times 10^{-11} T_W^4 - 0.101 (14 - L_{\text{PR}}) - 0.148$$

And for the flight system test

$$\Delta P_{\text{CRU}} = 1.32 \times 10^{-11} T_W^4 - 0.101 (14 - L_{\text{PR}}) - 0.237.$$

Figures 10 and 11 present the results of this analysis for several pressure regulator levels for the developmental and flight systems, respectively. It is apparent that unless temperatures below 0°F are employed for the flight system test, the CRU back pressure will be above design. Present physical limitations prevent the CRU from being located below station 6 and correspondingly the normal level in the pressure regulator is 8.2 in. Present indications are that the CRU can be satisfactorily operated at a back pressure up to 10 psia and, therefore, a vacuum tank wall temperature less than 100°F will be acceptable for ground system tests.

Operation at a back pressure of 9.7 psia will result in a decrease in power output of 120 watts (see Section II-D-2) which is less than the degradation allowance. Therefore, the high ground operating back pressure will not prevent demonstration of the 3-kw system capability.

5. Growth Potential

The present system can supply 4.2 kw gross or 3.7 kw net electrical power. A slight modification of the speed controls would only be required if the payload didn't have a constant demand of about 500 watts. These results are based upon a turbine efficiency of 54.3% and an alternator efficiency of 88%. Present test results indicate that turbine efficiencies of 60% and alternator efficiencies of 90% are better design values for future units. On this basis, 21.4 lb/min of Mercury with turbine inlet and exhaust pressures of 125 psia and

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Figure 10. Development System Turbine
Ground Operating Conditions

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Figure 11. Flight System Turbine
Ground Operating Conditions

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9 psia respectively will provide 5.5 kw gross or 5 kw net electrical power. Only small changes in the boiler, thermoelectric pump, and radiator condenser would be required to accommodate these conditions. The CRU would remain virtually unchanged except for slight changes in the turbine blading since the alternator has already demonstrated a 6 kw capability. Larger power requirements can be handled either by increasing the size of the combined rotating unit, boiler, etc. (PCS) or by combining multiple units of the 5 kw PCS. The former has the advantage of providing higher cycle efficiencies, therefore, a smaller radiator condenser, and lower system weight. A comparison of the reactor power requirements is shown in Figure 12. The use of multiple 5 kw units has the advantages of using developed components, requiring qualification of only one set of components, and greater reliability through the use of redundancy. For example, a 15 kw system using one redundant unit has a reliability of .865 as compared with .840 for a single 15 kw system (see Figure 12).

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Figure 12. SNAP 2 Objectives and Redundent

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III. RADIATOR-CONDENSER ANALYSIS

A. THE TWO-DIMENSIONAL STEADY-STATE RADIATOR CODE

1. Outline of the Analysis

a. The Radiator-Condenser Model

The radiator-condenser consists of a truncated cone (Figure 13), to which, at the top and the bottom, respectively, the vapor and liquid manifolds are attached. The manifolds are joined by tubes, in which the required condensation and subcooling of the mercury takes place. The tubes are brazed to the inner surface of the cone; to the outside surface of the tubes may be brazed fins of a material of high thermal conductivity, such as copper or silver.

The analysis (a digital code) neglects the manifolds, the tubes, and the basic cone, and in the direct calculations considers only the radiating fin. The presence of the manifolds is acknowledged in the boundary conditions, and the effect of geometric factors such as tube shape, the presence or absence of internal tube armor, and the existence of the basic cone, is accounted for as a temperature drop between the fluid and the radiating surface.

Figure 14 represents a typical cross-section of the fin-tube assembly. The code computes results for a single fin, which is defined as the radiating surface of the cone from the centerline of any tube to the midplane between that tube and the one adjacent to it, from the top to the bottom of the RC. Such a fin is shown in Figure 15, and in cross-section as the area BDFC shown in Figure 14; it may be rectangular in cross-section, or have the one-dimensional linear taper shown.

Some assumptions implicit in this choice of model are listed below:

- 1) All of the fins are exposed to identical conditions of mercury flow, mercury quality, and external heat loads from the sun, the earth, and the hot NaK system within the RC.
- 2) The rate of heat absorption from the NaK system is kept essentially constant throughout the length of a given tube by the nearly constant temperature of the condensing portion, coupled with the fact that the surface of the subcooler is coated with a layer of gold foil ($\alpha = 0.05$), which prevents the absorption of excessive heat in this cooler area.
- 3) Preliminary analysis, to be verified by a detailed analysis (see Figure 16) indicates that the position of the liquid-vapor interface

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Figure 13. Honeycomb Radiator-Condenser
With Emissivity Coating

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Figure 14. Typical Cross Section of a Fin and Half-Tube

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Figure 15. Typical Fin

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Figure 16. CRT Plot 1

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does not change greatly with the circumferential variation of the external heat load: a tube on the hot side would require a greater condensing length to condense the same amount of mercury as a tube on the cold side. This larger condensing length would cause a greater condensing pressure drop; but the pressure in each manifold is constant around the RC circumference, so that the pressure drop in each tube must be the same. Consequently, the flow through each tube is automatically adjusted resulting in nearly equal condensing lengths.

For radiator-condenser thermal calculations, the average values of flow and quality are used, and the total heatload imposed upon the RC from sources other than the mercury is calculated for a given orbital position and assumed to be distributed evenly over the surface of the RC. This approach is justified by the fact that the mercury outlet temperature which the RC is designed to provide is the temperature of the combined condensate from all the tubes, and is thus, itself, an average value.

NOTE

The conditions described in (a)(2) result in uneven distribution of the net heat rejected by the RC. The heat rejected by any two adjacent tubes, however, will be approximately equal, so that the assumption of an adiabatic midplane between tubes, used in the code as one of the boundary conditions, is approximately correct even under space conditions.

- 4) No heat enters a fin from beneath, i. e., along the surface EG as shown in Figure 14; all heat enters along the surface BD, at the temperature of the midpoint A.
- 5) The heat from the NaK system, which in fact enters the fin along the line EG shown in Figure 14, has already been accounted for in the space load.
- 6) In the case of the copper fin brazed to the steel face sheet of the honeycomb RC, the thermal resistance of the face sheet ($k = 0.72 \text{ Btu/hr-}^\circ\text{F-in.}$; $t = 0.005 \text{ in.}$) is so great that the heat flow along the face sheet from the mercury in the tube is negligible compared with that along the fin.

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- 7) The heat from the tubes actually enters the fin along the path EE_R (see Figure 14), which is not a negligible portion of the fin length BC. The point at which the heat enters the fin is, for computational purposes, effectively relocated at point A by an approximate analysis.

b. The Basic Equation

The premise upon which this code is based is:

$$\Sigma(\text{heat gained by conduction}) = \text{net heat lost by radiation} \quad \dots (27)$$

The left-hand side of this equation is obtained by summing the heats entering a fin element by conduction across each of the four boundaries: r , $(r + \Delta r)$, θ , and $(\theta + \Delta\theta)$, (see Figure 17), and taking the limit of this sum as Δr and $\Delta\theta$ both approach zero.

The right-hand side is obtained by taking the difference between the heat radiated by an element of area, and the heat absorbed by that element from its environment:

$$Q_{\text{net}} = \sigma\epsilon\Delta A [T^4 - T_s^4] - Q_{\text{external}} \quad \dots (28)$$

In practice, the Q_{external} may be set equal to zero, and an equivalent space temperature T_s can be calculated; or the space temperature may be set equal to absolute zero, while the heat load from space is added to the heat load from the NaK piping, etc., to give Q_{external} ; or both a T_s and a Q_{external} may be used.

The final equations in two dimensions are as follows:

Rectangular Fin:

$$\nabla^2 T = \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T}{\partial \theta^2} = \frac{\sigma\epsilon}{kt} [T^4 - T_s^4] - \frac{Q_{\text{ext}}}{kt} \quad \dots (29)$$

Taper Fin:

$$\begin{aligned} \nabla \cdot t \nabla T &= t(\theta) \left\{ \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{1}{r^2} \left[\frac{\partial^2 T}{\partial \theta^2} + \frac{1}{t(\theta)} \cdot \frac{\partial T}{\partial \theta} \cdot \frac{\partial t}{\partial \theta} \right] \right\} \\ &= \frac{\sigma\epsilon}{k} [T^4 - T_s^4] - \frac{Q_{\text{ext}}}{k} \end{aligned} \quad \dots (30)$$

a.

b.

c.

$$\Delta_1 = y_{m+1} - y_m = \left(\frac{L}{M}\right), \text{ WHERE } L = \text{TOTAL LENGTH OF FIN}$$

$M = \text{NUMBER OF LENGTHWISE NODES}$

$$\Delta_2 = \xi_{n+1} - \xi_n = \frac{1}{N}, \text{ WHERE } N = \text{NUMBER OF CROSSWISE NODES}$$

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Figure 17. Definition Sketch of Fin for Use in Deriving Basic Differential Equation

c. The Method of Solution

The fin is divided into M nodes of width Δ_1 , in the r direction, and N nodes of width Δ_2 in the θ direction. Equations relating the temperature at the node (m, n) to those at the adjacent nodes are derived from Equations 31 and 32 as described below. (M times N) linear equations result, and are solved simultaneously by the code.

(1) The Differential Equation

The following change of variable is effected in Equations 29 and 30:

$$r = y_o + y$$

$$\xi = \theta / \theta_{\max}$$

which thereupon become Equations 31 and 32 (note that $b_o = y_o \theta_{\max}$):

Rectangular Fin:

$$\nabla^2 T = \frac{\partial^2 T}{\partial y^2} + \frac{1}{y_o + y} \frac{\partial T}{\partial y} + \frac{y_o^2}{b_o^2 (y_o + y)^2} \frac{\partial^2 T}{\partial \xi^2} \quad \dots (31)$$

Tapered Fin:

$$\nabla \cdot t \nabla T = t(\theta) \left\{ \frac{\partial^2 T}{\partial x^2} + \frac{1}{y_o + y} \frac{\partial T}{\partial y} + \frac{y_o^2}{b_o^2 (y_o + y)^2} \left[\frac{\partial^2 T}{\partial \xi^2} + \frac{1}{t(\theta)} \frac{\partial t(\theta)}{\partial \xi} \cdot \frac{\partial T}{\partial \xi} \right] \right\} \quad \dots (32)$$

The differential portions of Equations 31 and 32 are put into manageable form by means of the following finite-difference approximations, which are obtained from Taylor's-series expansions of $T(y, \xi)$.

$$\frac{\partial T}{\partial y} = \frac{T(m+1, n) - T(m, n)}{\Delta_1} = \frac{T(m, n) - T(m-1, n)}{\Delta_1}$$

$$\frac{\partial T}{\partial \xi} = \frac{T(m, n+1) - T(m, n)}{\Delta_2} = \frac{T(m, n) - T(m, n-1)}{\Delta_2}$$

$$\frac{\partial t}{\partial \xi} = \frac{t(m, n+1) - t(m, n)}{\Delta_2} = \frac{t(m, n) - t(m, n-1)}{\Delta_2}$$

$$\frac{\partial^2 T}{\partial y^2} = \frac{T(m+1, n) - 2T(m, n) + T(m-1, n)}{\Delta^2}$$

$$\frac{\partial^2 T}{\partial \xi^2} = \frac{T(m, n+1) - 2T(m, n) + T(m, n-1)}{\Delta_2^2}$$

(2) The Radiation Portion

The expression involving T^4 is linearized as follows:

let $T_k = T_{k-1} + \delta T$, where k refers to the k^{th} iteration and
 δT = the change in T from the $(k-1)^{\text{st}}$
to the k^{th} iteration.

Then

$$[T_k^4 - T_{k-1}^4] = [T_{k-1} + \delta T]^4 - T_{k-1}^4.$$

Expanding and throwing away higher order terms gives:

$$T_k^4 \cong T_{k-1}^4 + 4T_{k-1}^3 [T_k - T_{k-1}].$$

And,

Rectangular Fin:

$$\nabla^2 T = \frac{\sigma \epsilon}{kt} [T^4 - T_s^4] - \frac{Q_{\text{ext}}}{\sigma \epsilon}$$

becomes:

$$\nabla^2 T_k - \frac{\sigma \epsilon}{kt} S T_k = -\frac{\sigma \epsilon}{kt} \left[3T_{k-1}^4 + S^4 + \frac{Q_{\text{ext}}}{\sigma \epsilon} \right] \quad \dots (33)$$

where

$$S = 4T_{k-1}^3$$

Tapered Fin:

$$\nabla \cdot t \nabla T - \frac{\sigma \epsilon}{k} \left[T^4 - T_s^4 - \frac{Q_{\text{ext}}}{\sigma \epsilon} \right]$$

becomes:

$$\nabla \cdot t \nabla T_k - \frac{\sigma \epsilon}{k} S T_k = -\frac{\sigma \epsilon}{k} \left[3T_{k-1}^4 - T_s^4 + \frac{Q_{\text{ext}}}{\sigma \epsilon} \right] \quad \dots (34)$$

where

$$S = 4T_{k-1}^3$$

(3) The Boundaries

The finite-difference equations were modified in the neighborhood of each boundary to satisfy the following boundary conditions:

- a) The base fin temperature is equal to the mercury temperature less the temperature drop through the mercury film, tube wall, braze, and meteoroid armor.
- b) An adiabatic midplane exists between tubes.
 $[(\partial T / \partial \xi)_{\theta_{\text{max}}} = 0]$ This assumption is consistent with the earlier one that all the fins are under identical thermal conditions.
- c) There is no vertical heat flow at the top rim of the fin.
 $[(\partial T / \partial y)_{y_0} = 0]$ This is a conservative assumption since condensing is assumed to begin where the condensing tubes are brazed to the outer skin, which is generally between 1 to 2 in. below the top of the Radiator Condenser (R/C). In other words the heat conducted to the upper two inches of R/C exceeds the amount conducted along the tubes and brackets into the top of the condensing section.
- d) The temperature along the bottom rim of the fin is equal to the fin base temperature at the bottom of the tube. This is a good approximation, since the bottom of the fin is in contact with the liquid manifold along its full length.

d. The Result

After the difference equations are inserted into the differential equation, the result set equal to the linearized radiation equation, and due allowances made for the boundary conditions, the equations used by the code are reached.

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e. Future Work

(1) Noncondensable Layer

The code is in the process of being modified to allow for the existence of a layer of noncondensibles between the end of the condensing region and the beginning of the subcooler. The approach being used will not affect the method used in calculating the fin temperatures, but will employ a third version of the equation to calculate the wall ΔT . In this equation, the heat transfer coefficient will be set equal to (k/t) of the noncondensable material.

(2) Effect of Gold Foil

The present assumption that the heat from the hot NaK system is absorbed at a constant rate down the length of a tube is valid as long as the inner surface of the subcooler is coated with a layer of gold foil ($\epsilon = 0.05$). However, there is some indication that, because of the gold foil, the mercury may freeze in the subcooler during startup, and that insufficient heat will be conducted down the fin from the condensing region above to melt this mercury, by the time the pressure in the vapor manifold has been raised to the desired startup value. Modifications to the code to accommodate the analysis of the RC performance with the gold foil removed, is in process.

f. The Coding of the Equation

(1) Outline of the Flow Plan

(a) The Setting Up of the Initial Conditions

- 1) Interior points: all interior points are initially set at the temperature at which, if the entire RC were actually at this temperature, it would radiate the required amount of heat.

The equation involved

$$Q_1 = A_{RC} \{ \sigma \epsilon [T_T^4 - T_s^4] - Q_{ext} \} , \quad \dots (35)$$

is solved for T_T ,

where

T_T = the uniform initial temperature

Q_1 = the desired heat removed from the mercury.

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- 2) Boundary points: along boundary 1 (y, o), the temperature profile in the condensing region is given $(T_o - \Delta T_w)_y$ where the T_o 's are the saturation temperatures corresponding to the computed condensing pressure drop in the tube, and the ΔT_w 's are the corresponding wall temperature drops. In the subcooling region, the initial fin-base temperature profile is based upon a mercury temperature profile linear between the interface temperature and the mercury outlet temperature. The fin base temperatures are obtained by subtracting from these mercury temperatures the computed wall ΔT .

Along the other boundaries, the temperatures are given by the method described in (c)(3) above.

g. The Calculations

(1) First Convergence (E1)

The code first sets up a matrix of (M x N) simultaneous equations, let us say those in the ξ -sweep, using the initial temperatures of (1)(a) above, provides a solution, thus obtaining a temperature distribution throughout the fin. It then sets up a similar matrix, using the equation for the y-sweep and the temperature distribution just obtained for the ξ -sweep, and obtains a new temperature distribution. It repeats this procedure until:

$$\left| \frac{T_k - T_{k-1}}{T_k} \right|_{\max} \leq EI, \text{ the required convergence criterion.}$$

(2) Second Convergence (E2)

The temperature pattern arrived at in (1)(a) enables the code to calculate the heat flow into the fin base, and from this, the new location of the interface. This changes the temperature profile on boundary 1, and the code computes a new temperature distribution in the fin by the method described above. This procedure is repeated until:

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$$\left| \frac{Q \text{ entering fin base} - Q \text{ removed from Hg}}{Q \text{ entering fin base}} \right| < E2, \text{ the required convergence criterion.}$$

Note: The Q removed from the Hg is the same as the net heat radiated.

(3) Third Convergence (E3)

Having obtained an E2 convergence, the code then determines whether the heat removed from the mercury is equal to the desired heat removed. It rarely is; therefore, the code changes the fin thickness by a pre-determined increment, and begins again at the beginning, with (1)(a) above, and continues until the greater of

$$\left| \frac{Q \text{ removed from Hg} - Q_1}{Q_1} \right| \quad \left| \frac{Q \text{ into fin base} - Q_1}{Q_1} \right| < E3, \text{ the desired convergence criterion.}$$

where

Q_1 = the desired amount of heat removed from the mercury.

If necessary, the code will interpolate to find the thickness increment required to obtain an E3 convergence.

h. Options and Special Features

(1) Plots

The following CRT plots can be obtained for each run, if desired. Examples of these plots shown in Figures 16, 18, 19, and 20 apply to the current RC design.

- a) "Heat Flow Into Base of Fin, " and "Wall ΔT " vs "Distance From Tube Inlet" (CRT Plot 1)
- b) "Coolant Temperature, " and "Wall Temperature" vs "Distance From Tube Inlet" (CRT and Plot 2 - Figure 18)
- c) Temperature profiles across the fin, up to 3 levels (m - values) (CRT Plot 3 - Figure 19)
- d) Isotherm map of the fin, showing up to 15 isotherms (CRT Plot 4 - Figure 20)

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Figure 18. CRT Plot 2

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Figure 19. CRT Plot 3

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Figure 20. CRT Plot 4

(2) Doubler

If necessary, the effect of a bracket support doubler around the top of the RC can be included.

(3) Initial Fin Thickness

If a series of runs are made, in which only the number of tubes varies, the code can be made to calculate the initial fin thickness values for all cases after the first, using the equation $N^2 t = \text{constant}$, where N = the number of tubes, and t = the initial fin thickness of the first case.

B. OPTIMIZING A RADIATOR-CONDENSER SYSTEM FOR
FOR MINIMUM WEIGHT1. Fixed Points

Several features of the design are fixed. These normally include:

L = the slant height of the RC

D_T, D_B = the top and bottom diameters of the RC

$\dot{w}$ = the total mercury flow rate

ΔP_{max} = the maximum allowable mercury pressure drop in the RC

$T_{\text{exit}}(\text{max})$ = the maximum allowable mercury exit temperature

$P(o)_{\text{min}}$ = the minimum probability that the RC tubes will suffer no meteoroid punctures during 1 yr in orbit.

Other restrictions sometimes given are:

a) $N = 4n$: The number of tubes should be a multiple of 4, for convenience in manufacture since the RC is made in 4 sections.

b) Restrictions on a tube dimension:

1) In the honeycomb RC, the tubes are required to fill the 0.30 in. space between the honeycomb face sheets.

2) The minimum width of a tube may be set by manufacturing restrictions

3) Fin centerline thickness — manufacturing techniques may set a minimum practical value upon the fin tip thickness.

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2. Calculation Procedure

a. Tubes

Several values of N, the tube number, are chosen, all other variables being held constant, and the condensing pressure drop and temperature distribution corresponding to each value of N is calculated by means of the pressure-drop code.

b. Fin

The tube number and geometry, and the temperature profile calculated in (2)(a) above, with the fixed data of (1), are used by the Steady-State Radiator Code to compute the fin base thickness, for a tapered or nontapered fin, which will permit the mercury to be cooled to the required outlet temperature.

c. Armor

The total armor thickness needed to give the required nonpuncture probability is calculated as outlined.² From this, when due allowance has been made for the protection afforded by the tube wall, face sheet, and fins, the required net armor thickness is calculated.

d. Weight

The physical data obtained from the preceding codes is fed into the Mercury Inventory Code, a transient code which describes the preliminary heating of the total RC by the introduction of hot mercury vapor. This code calculates among other things the weight of the RC, with and without the mercury inventory. A plot of system weights vs N will show a minimum weight, from which the optimum number of tubes may be selected. Clearly, the effect upon system weight of different tube geometries, or of fin geometry, materials of construction, etc., can be determined in the same way.

3. Effect of Variables

Some of the ways in which changes in the components can affect the total system weight are mentioned below.

a. Number of Tubes

If a greater number of tubes is placed around the same cone, the tubes will have to be closer together. Moreover, the heat flow through each tube will be less, to give the required total heat flow. These two factors are

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equivalent to shorter fins, and a lower heat flux per fin; hence the fins can be thinner. Therefore, the armor will have to be thicker, and more of it will be required, to cover the increased vulnerable area.

If the number of tubes is increased and the same pressure drop is to be maintained, the tube size must be decreased and the greater number of smaller tubes will weigh more than the smaller number of larger tubes.

b. Taper of Tubes

More steeply tapered tubes reduce the amount of mercury required for startup, though the steeper taper may require an increase in N (the tube number), if the specified pressure drop is to be maintained. If more tubes are required, the remarks of (a) above, will apply. If no additional tubes are required, the top tube width will have to be made greater; also, in this case a thinner fin may be adequate, with corresponding thicker armor.

c. Tube Geometry

Whether the tube is round in cross-section or rectangular has a considerable effect upon the width of the contact path shown as HD in Figure 21. The importance of a thick braze fillet is obvious from the figure. If, for reasons of tube geometry, the path HD becomes narrower, the wall ΔT becomes greater; and the fin base temperature, the basis for the radiating ability of the fin surface, is diminished. To compensate for this, the fins must be made thicker, and the system weight increases.

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Figure 21. Model for Calculation of Wall ΔT

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4. Some Results of Typical Studies

a. Alternative Tube Geometries

The procedure discussed above is illustrated⁴ in which was considered the effect upon total system weight of a number of changes in the tube geometry of a radiator-condenser, and one change in the fin geometry. The changes investigated are listed in Table VI, and the resulting component weights in Table VII. For convenience in making comparisons, the weight differences are given in Table VIII. A brief discussion of the calculation follows:

(1) Conditions

The tubes considered were alternatives suggested for the RC, and were required to fit within the 0.30-in. space between the face-sheets of the honeycomb shell. The dimensions of the shell were those of the current RC design, and the operating conditions those of the current design point of the radiator-condenser. (See Table IX.)

(2) Results

Reference 8 is a survey work rather than a complete study, and does not give the complete optimization procedure for each change; however, it does illustrate very well the changes in system weight which can be expected from given changes in geometry, and the distribution of the changes among the components.

5. Radiator-Condenser Parameter Study

Reference 3 illustrates a variation of the calculation procedure. It starts with an optimized radiator-condenser, and investigates the interaction of the parameters given in Table X. The report is based upon the assumption that, within $\pm 5\%$ of the design point value, changes in each variable affect the RC performance independently. The mercury outlet temperature was used as the measure of performance, and the results were expressed in the following equations:

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TABLE VI
TUBING ALTERNATIVES CONSIDERED

Tube	Change	No. of Tubes	Tube Width at Bottom (in.)	Tube Width at Top (in.)	Tube Wall Thickness (in.)
1	Basis (present design) (rect. tapered)	40	0.125	0.254	0.015
2	Tube 1 with rectangular fin (rect. tapered)	40	0.125	0.254	0.015
3a	Tube 1 with decreased bonded area (rect. tapered)	40	0.125	0.254	0.015
3b	Tube 1 with decreased bonded area (rect. tapered)	40	0.125	0.254	0.015
4	Tube 1, with increased corner radius (rect. tapered)	40	0.125	0.254	0.015
5	Increase wall thickness (rect. tapered)	40	0.125	0.379	0.052
6	Nontapered rectangular tube	40	0.2185	0.2185	0.015
7a	Round, tapered tube	44	0.22(a)	0.27(a)	0.015
7b	Tube 7a, with increased taper	48	0.19(a)	0.27(a)	0.015
8	Round, nontapered tube (a) Diameter	36	0.27(a)	0.27(a)	0.015

Tube	Corner Radius	Bonded Area (%)	Tapered Fin	Thickness (in.)	Armor Thickness (in.)
1	2t	90	yes	0.01639	0.0247
2	2t	90	no	0.01461	0.0265
3a	2t	75	yes	0.01766	0.0234
3b	2t	50	yes	0.02100	0.0201
4	4t	90	yes	0.01800	0.0230
5	1t	90	yes	0.01800	0
6	2t	90	yes	0.01600	0.0251
7a		90	yes	0.01400	0.0260
7b		90	yes	0.01250	0.0286

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TABLE VII
WEIGHTS OF TOTAL SYSTEM AND OF IMPORTANT COMPONENTS

Tube	Weight of Tube (lb)	Weight of Fin (lb)	Weight of Armor (lb)	Dry Weight (lb)	Weight of Hg (lb)	Total Weight (lb)
1	18.78	63.26	7.31	244.58	76.52	321.11
2	18.78	80.76	7.84	262.61	78.99	341.60
3a	18.78	66.77	6.85	247.63	77.01	324.64
3b	18.78	76.00	5.94	255.96	78.14	334.10
4	17.71	67.71	6.83	247.53	74.95	322.48
5	69.53	67.71	0.00	292.05	78.99	371.04
6	20.02	62.18	8.60	245.93	89.20	335.13
7a	18.49	56.66	10.92	241.60*	79.97	321.57*
7b	18.68	52.51	12.32	239.21*	75.15	314.35*
8	16.61	85.76	6.30	263.84	86.35	350.19

*Includes weight of shim to fill in space between the tapered tube and the bottom face sheet

TABLE VIII
WEIGHT DIFFERENCES*

Tube	Tube	Fin	Armor	Mercury	Net Difference ^{††}
1	0.00	0.00	0.00	0.00	0.00
2	0.00	17.50	0.53	2.47	20.49
3a	0.00	3.51	-0.46	0.49	3.53 [†]
3b	0.00	12.74	-1.37	1.62	12.99 [†]
4	-1.07 [§]	4.45	-0.48	-1.47	1.37
5	50.75	4.45	-7.31	2.47	49.93
6	1.24	-1.08	1.29	12.68	14.02
7a	-0.29	-6.60	3.61	3.45	0.46**
7b	-0.10	-10.75	5.01	-1.37	-6.76**
8	-2.17	22.50	-1.01	9.83	29.08

*Weight (tube n) - Weight (tube 1).

All weights are in pounds

[†]See discussion of results

[§]A minus sign indicates a weight saving

**Includes weight of shim

^{††}This column records the differences between the total weights of Column 7, Table VII

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TABLE IX
FIXED VALUES USED IN REFERENCE 4 FOR HONEYCOMB RC

A. Dimensions:

Top diameter (in.)	$D_T = 30.15$
Bottom diameter (in.)	$D_B = 60.00$
Fin length (in.)	$L = 120.85$
Number of tubes	$= 40$

B. Operating Conditions:

Mercury flow rate (total) (lb/min)	$W = 19.8$
Inlet mercury quality	$\chi = 0.975$
Pressure in vapor manifold	$P = 8.45$
Condensing pressure drop (psi)	$P = 2.25$
Heat lost from mercury (Btu/lb)	$H = 133.556$
Outlet Temperature ($^{\circ}\text{F}$)	$T_{\text{exit}} = 317$

TABLE X
PARAMETERS VARIED IN THE RADIATOR-CONDENSER
PARAMETER STUDY

Parameter		Definition
1	T_W	$(T_{\text{Hg}} - T_{\text{fin base}}) (^{\circ}\text{F})^*$
2	x	Inlet vapor quality
3	ϵ	Emissivity of the RC
4	k	Thermal conductivity of the fin (Btu/hr- $^{\circ}\text{F}$ -in.)
5	Q_{ext}	Heat load from sources other than the mercury
6	ω	Total mercury flow rate (lb/min)
7	t_b	Fin thickness at base (mils)
8	t_{ξ}	Fin thickness at centerline (between tubes) (mils)
9	T_o	Temperature of RC at injection ($^{\circ}\text{F}$)
10	T_{vt}	Temperature of vacuum tank ($^{\circ}\text{F}$)
11	T_{exit}	Temperature of mercury leaving the RC ($^{\circ}\text{F}$)

*In this study, ΔT_{ω} was held constant down the length of the tube

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Honeycomb RC (RC-12)

$$\begin{aligned}(T_{\text{exit}} - 371) = & 304 [(1.0145)^{\Delta T} - 1.242] + 8.0 [(50.5)^X - 43.5] \\ & + 1.63 \times 10^4 [(0.015)^\epsilon - 0.0227] - 14.0 [k - 17.7] \\ & + 254 [(1.0806)^Q - 1.415] + 5.9 [(1.232)^\omega - 62.1] \\ & + [1154 - 192.06 t_b + 10.806 t_b^2 - 0.2142 t_b^3] \\ & + 380 [(0.9821)^{t_G} - 0.9645] \\ l_{sc} = & 21.4 \ln \left[\frac{590}{T_{\text{exit}}} \right]\end{aligned}$$

Aluminum Steel RC (RC-14)

$$\begin{aligned}(T_{\text{exit}} - 371) = & 5.31 (\Delta T - 15) + 9.0 [(42.2)^X - 36.8] \\ & + 1.89 \times 10^4 [(0.0117)^\epsilon - 0.0200] \\ & + 570 [(0.9562)^k - 0.608] + 27.3 [Q - 4.48] \\ & + 7.3 [(1.2245)^\omega - 55.2] + 600 [(0.985)^{t_b} - 0.628] \\ l_{sc} = & 23.1 \ln \left[\frac{590}{T_{\text{exit}}} \right]\end{aligned}$$

where

$\Delta T = T_{\text{Hg}} - T_{\text{fin base}}, ^\circ\text{F}$

$x = \text{Mercury inlet quality}$

$\epsilon = \text{Emissivity of outer RC surface}$

$k = \text{Thermal conductivity of fin, Btu/hr-}^\circ\text{F-in.}$

$Q = \text{Heat load from sources other than the mercury, kw}$

$\omega = \text{Total mercury flow rate, lb/min}$

$t_b = \text{Fin thickness at base, mils}$

$t_G = \text{Fin thickness at top, mils}$

$l_{sc} = \text{Subcooler, length, in.}$

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Note — any parameters which remain at their design point values (see Table XI) vanish from the equations. The equations can be used

- a) To determine how much the exit temperature will be affected by a change in a given variable.
- b) To determine the change in one variable, such as t_b , which would be required to compensate for a change in another, such as ϵ , if the mercury exit temperature were fixed.

TABLE XI
DESIGN POINT VALUES USED IN THE
RC PARAMETER STUDY*

Parameter	RC-12	RC-14
$T_{\text{exit}}, ^\circ\text{F}$	371	371
$\Delta T_{\omega}, ^\circ\text{F}$	15	15
x	0.963	0.963
ϵ	0.90	0.88
$k, \text{Btu/hr} - ^\circ\text{F-in.}$	17.7	11.1
$Q_{\text{exit}}, \text{kw}$	4.47	4.48
$W, \text{lb/min}$	19.8	19.8
t_b, mils	13.75	30.8
$t_{\text{CL}}, \text{mils}$	2.00	-

*Several of the design point values have changed since Reference 3 was written

C. CONDENSER PRESSURE DROP

1. General

The accurate prediction of the pressure drop in the SNAP 2 condenser tubes is essential in meeting the design point requirements. If the pressure drop is too large, the manifold pressure must rise and a less efficient system results. Very low pressure drops; although acceptable by design point considerations, are not welcomed since they contribute to making the stable range of condenser operation smaller and increase mercury startup inventory.

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Pressure drop in the condenser is manifested in several ways. Entrance losses occur from the manifold to the tube. This is due to the acceleration of the vapor into the tubes and also the turbulent losses that accompany the acceleration drop. Frictional losses occur throughout the rest of the condenser in the transition section between the manifold and the straight portion of the condenser tube and in the straight portion itself. Upon condensing, part of the inlet vapor momentum is recovered and this rise in pressure reduces the net pressure drop somewhat.

Saturated or slightly wet vapor enters the vapor manifold from the vapor risers. The flow splits and is channeled into 4 quarter sections of the manifold which feed 10 condenser tubes each. A small amount of condensation may take place in the manifold due to heat exchange between it and the top of the radiator. The amount of condensation is small; this reduces the quality of the vapor by a very small amount. The velocity of the wet vapor in the manifold is fairly low, ranging from about 50 ft/sec to zero from one end of the quarter section to the other, for the reference design shown in Figure 22. Because of the variation in velocity, the static pressure of the manifold varies along its length. However, the dynamic head is small and an average static pressure can be assumed over the length of the manifold. The wet flow is accelerated into the mouth of the

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Figure 22. Manifold and Condenser Tubes

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condenser tubes and upon doing so, the static pressure drops. Also, some turbulent losses occur during acceleration. Like the manifold, little or no condensation takes place in the transition section between the manifold and the straight portion of the tube. In the transition, the bends contribute to the static pressure drop. Condensation starts at the mouth of the straight section of the tube. The condensation rate, however, is not constant along the tube, but increases with distance down the tube since the heat rejection area increases with distance down the tube. The quality profile for the vapor then is not a linear one but is obtained by integrating the heat rejection rate. Mathematically:

$$\psi(x) = \psi_1 - \frac{\int_0^x q(x) dx}{W\lambda} \quad \dots (36)$$

where ψ_1 is the quality at the mouth of the straight section. A typical heat rejection curve and a quality curve are shown in Figures 23 and 24. Condensation in the manifold and in the transition section is small usually so that ψ_1 can be taken as the inlet quality to the manifold. The vapor flow at any cross-section along the tube is equal to

$$W(x) = W\psi(x) \quad \dots (37)$$

Throughout the tube, it is assumed that equilibrium prevails and saturated conditions exist. The vapor state follows the saturation curve with pressure drop in the tubes and density decreases with decrease in pressure. Besides the change in cross-sectional areas of the tube, the vapor velocity within the tubes are complicated by the nonlinear quality profile and change in density along the tube. From continuity the velocity is

$$V(x) = \frac{W\psi(x)}{A(x)\rho(P)} \quad \dots (38)$$

A typical velocity distribution is shown in Figure 25 for a tapered tube.

Frictional pressure drop in a condenser tube is appropriately analyzed by considering the results of pressure drop experiments with two-phase two-component flow in straight tubes where no condensation takes place. In this way, pressure recovery due to condensation and the acceleration terms are ruled out and the pressure gradient thus obtained can be used for the frictional portion of the pressure drop calculations in our system.

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Figure 23. Heat Rejection Profile

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Figure 24. Quality Profile

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Figure 25. Velocity Profile

Two-phase pressure drop is defined in the same way as the pressure drop for a single vapor phase, but is modified by a factor ϕ^2 .

$$\left. \frac{dP}{dx} \right|_{\text{Two Phase}} = \phi^2 \left. \frac{dP}{dx} \right|_{\text{Vapor}} = \phi^2 \frac{2f\rho V^2}{g De} \quad \dots (39)$$

f is the friction factor based on vapor flowing alone in the tube. The correction factor ϕ^2 is obtained from the improved Lockhart-Martinelli correlations for two-phase flow and is a function of two parameters, χ^2 , the two-phase modulus, and the vapor Reynolds Number. The two-phase modulus is the ratio of the

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pressure gradient that would result when only the liquid flow is considered to the pressure gradient that would result when only the vapor flow is considered.

$$X^2 = \frac{dP/dx/\text{liquid}}{dP/dx/\text{vapor}} \quad \dots (40)$$

where

$$\frac{dP}{dx} = \frac{2fG^2}{gD_e\rho} \quad \dots (41)$$

and G is the mass velocity for each phase.

The friction factor f is

$$f = \frac{16}{Re} \text{ for laminar flow} \quad \dots (42)$$

and

$$f = \frac{0.079}{Re^{0.25}} \text{ for turbulent flow} \quad \dots (43)$$

Depending on the Reynolds number for each phase flowing separately, the modulus can represent four modes.

Xtt - turbulent liquid, turbulent vapor
Xvt - viscous liquid, turbulent vapor
Xvv - viscous liquid, viscous vapor
Xtv - turbulent liquid, viscous vapor

In general, only viscous-turbulent and viscous-viscous modes occur. A plot of ϕ^2 against X with vapor Reynolds number as a parameter is shown in Figure 1 of reference 5.

Since changes in conditions, pressure, temperature, density, and flow rate change significantly throughout the condenser, and the applicable equations are complex, it is only correct to evaluate the pressure drop by step by step calculations. The following paragraphs discuss the phenomena that occur with condensing pressure drop in more detail, the derivation of the governing equations, and explain their use in the computer program.

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2. Entrance Losses

Figure 26 below shows the entrance to a tube from the manifold. At the entrance of the tube the static pressure of the vapor drops since it must be accelerated into the tube and also upon doing so, turbulent losses occur. It is assumed the turbulent loss is equal to a certain fraction (K) of the acceleration drop and this is included in the total drop. Since the system is compressible, a direct subtraction of the turbulent loss after subtracting the acceleration drop cannot be made. The density change due to the turbulent part of the entrance drop may be very large so that entrance velocities increase and the acceleration drop calculated originally becomes smaller than the actual acceleration drop.

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Figure 26. RC Manifold and
Condensing Tube Configuration

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To circumvent this problem, the turbulent loss is applied first and the acceleration drop later. The acceleration drop is based then upon the new total pressure at the mouth of the tube and a higher and more accurate acceleration drop results. The only problem is that initially an estimate of the acceleration drop is needed to determine the entrance loss. An estimate based on the total pressure of the manifold is sufficient for this purpose and any deviations may be taken into account by modifying the percentage factor (K). Shown in the sketch are numbers that signify the conditions, total pressure, static pressure, and temperature in the system. Point 0 is for conditions in the manifold. Point 1 is for conditions after the estimated acceleration drop. Here the total pressure is taken equal to the manifold static pressure and the static pressure is less by an amount equal to the estimated acceleration drop. Point 2, for conditions after the fraction of the estimated acceleration drop (turbulent loss), is applied to the total pressure at 1 (or manifold pressure). Here the static pressure is the difference between the actual acceleration drop and the new total pressure. The temperature at each of these conditions is related to the pressure by the pressure-temperature relationship for saturated conditions.

The relationship between the total pressure and the static pressure in a compressible perfect gas is given by:

$$\frac{P_T}{P_S} = \left[1 + \frac{k-1}{2} M^2 \right]^{\frac{k}{k-1}} \quad \dots (44)$$

where M is the Mach number and k the ratio of the specific heats. The ratio of the specific heats for mercury vapor is known and is approximately constant at 1.663.

At point 1, the total pressure is equal to the known manifold pressure, but the Mach number is not known, and an estimate must be sought. Mach number for a perfect gas is given by:

$$M = \frac{W\sqrt{RT_T}}{P_TA\sqrt{kg}} \left(\sqrt{\frac{T_S}{T_T}} \right) \left(\frac{P_T}{P_S} \right) \quad \dots (45)$$

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or

$$M = \frac{W\sqrt{RT_T}}{P_T A \sqrt{kg}} \left[1 + \frac{k-1}{2} M^2 \right]^{\frac{k+1}{2(k-1)}} \quad \dots (46)$$

when

$$\frac{P_T}{P_S} = \left[1 + \frac{k-1}{2} M^2 \right]^{\frac{k}{k-1}} \quad \dots (47)$$

and

$$\frac{T_T}{T_S} = 1 + \frac{k-1}{2} M^2 \quad \dots (48)$$

are introduced into Equation 2. For the first estimate of Mach number, the ratios of the pressures and the temperature are assumed to be unity. The total pressure at 1 is known and the total temperature can be obtained from the pressure-temperature relationship for saturated conditions.

Upon obtaining the estimated Mach number, it is then introduced into Equation 3 to obtain a second estimate of Mach number. This process is continued with new estimates until the final Mach number does not differ from the last estimate by more than 0.1%. The final value for Mach number is introduced into Equation 1 to obtain the static pressure at 1.

The turbulent loss was assumed to be a certain fraction of the estimated acceleration drop and is given by:

$$\Delta P_{\text{turbulence}} = K(P_{T1} - P_{S1}) \quad \dots (49)$$

The new total pressure at 2 is given by:

$$P_{T2} = P_{T1} - \Delta P_{\text{turbulence}} = P_{T1} - K(P_{T1} - P_{S1}) \quad \dots (50)$$

With the new total pressure known, the new static pressure can be calculated with Equation 1. Estimates of the new Mach number again have to be

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made and Equation 3 again is iterated upon. The new total pressure and its corresponding temperature are used in this calculation.

3. Condensing Pressure Drop

With the static pressure known at the entrance we can commence with the condensing pressure drop in the tubes. A discussion of the pressure drop in the straight section follows.

Pressure drop in condensing tubes can be appropriately analyzed by the following momentum equation.⁶

$$PA + PdA - (P + dP)(A + dA) - \tau_W dA_W - dB = \frac{1}{g} (W + dW_g + dW_L)(V + dV) - V_g \frac{dW_g}{g} - V_L \frac{dW_L}{g} - \frac{VW}{g} \quad \dots(51)$$

It relates the pressure, shear, and body forces on an element of condensing, boiling, or injection flow to the change in momentum of the constituents of the element. The differential element is shown in Figure 27.

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Figure 27. Differential Model for Condensing Pressure Drop

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It shows a control volume surrounding the vapor but excluding the liquid. Here evaporation from the liquid and injection of vapor is shown. In our case, the condensing process is chosen to be negative injection (rejection) of vapor. It is assumed that the rejected vapor leaves with an axial component of velocity equal to V_g . Since vapor rejection was considered as the condensing process, the other mechanism shown need not be considered. Exclusion of the liquid from the control volume is a valid assumption since its volume and momentum are small.

The first three terms of Equation 51 are the pressure forces from which the pressure gradient will ultimately be obtained.

$\tau_W dA_W$ is the shear force at the wall. τ_W is given by $2f' \rho V^2 A / g D_e P_W$ where f' is the friction factor for two-phase flow and is equal to $\phi^2 f$ discussed before. dB is the body force term and is negligible for orbital operation of the SNAP 2 system. The terms on the right side of the equation represent the net change in momentum of the element. Since the liquid has been excluded dW_L is zero and $dW_g = dW$.

Upon expansion, dropping second order terms and introducing values for the aforementioned, the equation reduces to:

$$-AdP - \frac{2f' \rho V^2 A dA_W}{g D_e P_W} = \frac{W dV}{g} + \frac{1}{g}(V - V_g)dW \quad \dots(52)$$

Letting $y = V_g/V$, and rearranging and dividing by dx , the equation becomes:

$$A \frac{dP}{dx} + \frac{2\phi^2 f \rho V^2 A}{g D_e P_W} \frac{dA_W}{dx} + \frac{W}{g} \frac{dV}{dx} + \frac{V}{g} (1 - y) \frac{dW}{dx} = 0 \quad \dots(53)$$

Noting that

$$V = \frac{W}{\rho A}, \quad \frac{dV}{dx} = \frac{1}{\rho A} \frac{dW}{dx} - \frac{W}{A \rho^2} \frac{d\rho}{dx} - \frac{W}{\rho A^2} \frac{dA}{dx} \quad \dots(54)$$

and applying the chain rule:

$$\frac{d\rho}{dx} = \frac{d\rho}{dP} \frac{dP}{dx} \quad \dots(55)$$

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$$\frac{dV}{dx} = \frac{1}{\rho A} \frac{dW}{dx} - \frac{W}{A\rho^2} \frac{d\rho}{dP} \frac{dP}{dx} - \frac{W}{\rho A^2} \frac{dA}{dx} \quad \dots(56)$$

Also,

$$\frac{dAw}{dx} = Pw(x) \text{ the wetted perimeter} \quad \dots(57)$$

Introducing Equations 56 and 57, Equation 53 becomes

$$A \frac{dP}{dx} + \frac{2\phi_f^2 \rho V^2 A}{D_e} + W \left[\frac{V}{\rho A} \frac{dW}{dx} - \frac{W}{A\rho^2} \frac{d\rho}{dP} \frac{dP}{dx} - \frac{W}{\rho A^2} \frac{dA}{dx} \right] + V(1-y) \frac{dW}{dx} = 0 \quad \dots(58)$$

$d\rho/dp$ along the saturation line can be obtained by the Clausius Clapyron relationship and the derivative of the perfect gas relationship.

$$\frac{dp}{dt} = \frac{\lambda}{T\Delta v} \quad \text{Clausius Clapyron} \quad \dots(59)$$

$$1 = \rho R \frac{dT}{dP} + RT \frac{d\rho}{dP} \quad \text{Perfect gas} \quad \dots(60)$$

Introducing the Clausius Clapyron relationship into derivative of the perfect gas equation and noting that Δv is approximately equal to the reciprocal density, $d\rho/dP$ becomes

$$\frac{d\rho}{dP} = \frac{1}{RT} - \frac{1}{\lambda} \quad \dots(61)$$

The use of the perfect gas relationship and the Clausius Clapyron equation is reasonable since the mercury vapor closely approximates a perfect gas in this region. Upon introducing Equation 61 and the continuity equation into Equation 58 and rearranging, the pressure gradient becomes

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$$\frac{dp}{dx} = \frac{\frac{-2\phi_f^2 \rho V^2}{gD_e} - \frac{V}{gA} (2-y) \frac{dW}{dx} + \frac{\rho V^2}{gA} \frac{dA}{dx}}{\left(1 - \frac{V^2}{g} \left[\frac{1}{RT} - \frac{1}{\lambda}\right]\right)} \quad \dots (62)$$

Knowledge of the state of the liquid and vapor in the tubes, their flow rates, the change of vapor flow along the length of the tube, the cross-section area of the tube and its change along the tube length are necessary before this equation can be evaluated at any point within the tube. Assume for now that the pressure at a particular point is known. Because the state of a saturated vapor is known when only one variable is fixed, the temperature and density of both the liquid and the gas are known. Also, the latent heat of vaporization is known.

$$T_{\circ F} = 461.87 + 128.34 \log P + 46.67 (\log P)^2 \quad \dots (63)$$

$$\rho_{\text{vapor}} = 18.70 \frac{P}{(T + 460)} \frac{\text{lbm}}{\text{ft}^3} \quad \dots (64)$$

$$\rho_{\text{liquid}} = \frac{10,680 - (T + 460)}{12} \frac{\text{lbm}}{\text{ft}^3} \quad \dots (65)$$

$$\mu_{\text{vapor}} = 1.9 \times 10^{-16} T^6 - 6.58 \times 10^{-13} T^5 + 9.395 \times 10^{-10} T^4 \quad \dots (66)$$

$$-7.075 \times 10^{-7} T^3 + 2.96 \times 10^{-4} T^2 - 6.54 \times 10^{-2} T + 5.939 \frac{\text{lbm}}{\text{hr-ft}}$$

$$\mu_{\text{liquid}} = -1.91 \times 10^{-5} T^6 + 6.53 \times 10^{-12} T^5 - 9.265 \times 10^{-9} T^4 \quad \dots (67)$$

$$+ 6.96 \times 10^{-6} T^3 - 2.92 \times 10^{-3} T^2 + 0.644 T - 55.9 \frac{\text{lbm}}{\text{hr-ft}}$$

$$\lambda = 127.4 \frac{\text{Btu}}{\text{lbm}} \quad \dots (68)$$

Since the total flow rate in each tube and the quality profile are established by design point and heat transfer consideration the flow rates of the liquid and gas are known along the tube.

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$$W(x) \text{ vapor} = W\psi(x) \quad \dots (69)$$

$$W(x) \text{ liquid} = W[1 - \psi(x)] \quad \dots (70)$$

The geometry of the tube is already established, so the cross-sectional area and perimeter are known along the length of the tube.

$$A = f(x) \quad \dots (71)$$

$$P_w = f(x) \quad \dots (72)$$

From the cross-sectional area, the densities, and the flow rates, the velocities of both the liquid and vapor can be calculated. The velocity of the vapor is very close to the actual velocity in the tube since the volume of liquid is small. The calculated velocity of the liquid, however, is very much smaller than the actual. Although the calculated liquid velocity is physically not true, this velocity is used in the correlating parameter for two-phase pressure drop. With the velocities, the densities, the viscosities, and the hydraulic diameter known, the Reynolds number for each phase is obtained and subsequently the single phase friction factors and pressure drops can be evaluated for each phase.

With the virtual pressure drops and the Reynolds number for the vapor known, the two-phase correction factor and the two-phase friction factor can be obtained

$$X^2 = \frac{dp/dx/\text{liquid}}{dp/dx/\text{vapor}} \quad \dots (73)$$

$$\phi^2 = f(X, \text{Re vapor}) \quad \dots (74)$$

$$f' = \phi^2 f \quad \dots (75)$$

The first term in Equation 62 is thus completed. The vapor flow rate gradient is known from the flow rate profile and a suitable value for y can be chosen, so that the second term becomes complete. y is a measure of the loss in momentum of the condensing vapor. If the condensing vapor leaves the

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control volume at the stream velocity, none of its momentum can be recovered. The only recovery is in the effective diffusion that takes place in the rest of the vapor after the condensed vapor leaves the system (control volume). It is, however, assumed that some of its momentum is transferred to the vapor. This parameter can be determined by comparing calculated pressure drop results to measured results. Typical values are around 0.5. The area gradient is also known so the third term is complete. All the terms in the denominator are also known. The pressure gradient at this point now becomes known.

The pressure profile can be determined by nodewise calculation of the gradients. If the pressure at a node is known, the gradient can be calculated as shown above. Projection of the profile to the next node establishes the pressure at the next node. The gradient at the first node, however, is only an estimate of the gradient between the nodes. A much better estimate of the gradient between the nodes can be obtained by use of the Runge-Kutta Method.⁷ This method is extensively discussed in most books on numerical analysis. The method employs an averaging type technique to estimates of the gradient at and between the two nodes.

Mathematically,

$$P_{i+1} = P_i + \frac{1}{6} \Delta x \left(\left. \frac{dP}{dx} \right|_1 + 2 \left. \frac{dP}{dx} \right|_2 + 2 \left. \frac{dP}{dx} \right|_3 + \left. \frac{dP}{dx} \right|_4 \right) \quad \dots (76)$$

where Δx is the node length and

$$\begin{aligned} \left. \frac{dP}{dx} \right|_1 &= f(x_i, P_i) \\ \left. \frac{dP}{dx} \right|_2 &= f\left(x_i + \frac{1}{2} \Delta x, P_i + \frac{1}{2} \Delta x \left. \frac{dP}{dx} \right|_1\right) \\ \left. \frac{dP}{dx} \right|_3 &= f\left(x_i + \frac{1}{2} \Delta x, P_i + \frac{1}{2} \Delta x \left. \frac{dP}{dx} \right|_2\right) \\ \left. \frac{dP}{dx} \right|_4 &= f\left(x_i + \Delta x, P_i + \Delta x \left. \frac{dP}{dx} \right|_3\right) \end{aligned} \quad \dots (77)$$

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The gradient is first evaluated at the initial node. The second estimate is made at the midpoint where the pressure is estimated by the first gradient projected from the initial node. The third estimate is made again at the midpoint but where the pressure is estimated by the second gradient projected from the initial node. The last estimate is made at the next node where the pressure is estimated by the third gradient projected from the initial node. The four estimates of the gradient are then averaged by the Simpson Barrel rule. The pressure at the next node is the average gradient projected from the initial node. This calculation is continued from node to node throughout the condensing portion of the condenser tubes.

4. Transition Section

Equation 62 and the Runge-Kutta method are applicable to the transition section also. The friction factor, however, must be modified to take into account the turbulent effects of the bends. An appropriate modifying coefficient need only be introduced into the first term of Equation 62.

A typical pressure profile is shown in Figure 28.

For a more detailed discussion of the computer program developed to solve the foregoing equation see Reference 8.

5. Manifold Pressure Drop

The small change in static pressure along the length of the manifold can be calculated in much the same way as the pressure drop in the condenser tubes. The manifold section between the tubes can be considered as a condenser tube itself and the condenser pressure drop program can be applied.

It is usually assumed that no condensation takes place along the manifold. The quality of the vapor, however, changes since it is assumed that only vapor enters the tubes and what little liquid that enters the manifold flows to the last tube. So after each tube, the quality of the vapor decreases, unless of course, the inlet quality is 100%. It is also assumed that the flow to each tube is the same. Hence, the flow rate and the quality profile in each manifold section between tubes is known. It is assumed that no turbulent losses occur at the mouth of the manifold and after each tube where diffusion of the vapor occurs. Losses at the mouth are charged to the vapor risers.

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Figure 28. Pressure Profile

With the total pressure at the mouth of the manifold known, the static pressure is calculated by the entrance drop routine. Here turbulent losses are assumed to be zero. With the static pressure known, pressure drop is calculated in the same way as the pressure drop in the condenser tubes. The short manifold section between the entrance and the first tube is broken up similarly into 11 nodes and the same nodewise calculations are made.

The total pressure of the vapor before and after the first tube is constant since no turbulent losses upon expansion were assumed. Since the pressure drop along this very short section is extremely small, the difference between the total pressure drop and the static pressure drop is negligible. There is very little density change along the section at the pressure levels investigated to warrant considering the difference between the two. The total pressure at the first tube is taken to be the total pressure at the mouth of the manifold less the static pressure drop.

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The second manifold section between the first and second tubes is treated in the same way. With the total pressure at the first tube known, the static pressure can be calculated. In this calculation the flow for the second section is used. Pressure drop along the second section is also calculated nodewise. Here a lower quality, reduced by the amount of pure vapor that entered the first tube, is used.

The foregoing calculations are made for each section until the last tube is reached. In the case of the honeycomb RC, 10 sets of calculations are made for the manifold section in front of the ten tubes of each quarter section of the whole manifold. The pressure drop in the RC manifold is 0.007 psi.

D. SINGLE TUBE MERCURY CONDENSING TEST

1. General

This section describes the mercury, two-phase flow, condensing experiment. The purpose of this test is to furnish data that will aid in the design and evaluation of the SNAP 2 flight RC. Basically, this test will measure differential pressure and simulated flight system stability.

2. Program Objectives

The present test program is tailored towards providing as much information as possible with regard to present uncertainties in the zero "g" condensing process. The test will provide information in the following important areas:

- a) Pressure drop tests will be performed under a wide range of flow and pressure conditions in order to provide more accurate correlations to use in the flight design RC. The test loop is well instrumented to provide accurate temperature and pressure profile data.
- b) Tests will be performed to determine the amount of momentum recovery from the vapor. The momentum effects can be separated from the friction drop by varying the condensing length at constant flow. In this manner, a complete pressure drop analytical model, including momentum recovery and compressible flow effects can be verified and used to predict system pressure drop.

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c) Tube geometries to be tested:

- 1) Straight circular tubes
- 2) Rectangular tapered tubes of RC dimensions
- 3) Bank of multiple tubes

In this manner, it can be determined whether the pressure drop correlation used is universally applicable to all geometries. Vapor distribution to the tubes may be observed under normal operations and under possible flight perturbations with the multiple arrangements. An accurate determination of the actual RC tube pressure drop will also be obtained.

d) Fluoroscopic methods will be employed to study the following:

- 1) The shape of the liquid-vapor interface in a one "g" (horizontal) environment. This will be useful in predicting the performance of present and future flat RC tests. It will enhance our understanding of interface stability and will shed some light on the potential problem of gas and vapor trapping in the interface area.
 - 2) The flow phenomenon occurring in the various portions of the RC tube. It will determine if there are any flow instabilities, such as plug flow, or similar unstable regions which may lead to a gas or vapor trapping problem in the flight design.
- e) The effects of mercury wetting on interface shape and flow phenomenon will be investigated. It is expected that wetting will occur after an extended operating time during the test and that wetting changes will be observable during a given run. If not, wetting will have to be induced artificially (e.g., hot mercury flush, chemical additions, electro polishing tube, etc.). The flight system must operate satisfactorily under both wetting and nonwetting conditions; during ground and orbital operations.
- f) The effects of noncondensibles on radiator-condenser performance will be studied. The present flight design RC has a design margin included due to the need to operate with noncondensibles. These

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arise from several sources: (1) hydrogen leakage, (2) residual pump down gas in Hg system, (3) outgassing from mercury system components at operating temperatures. The single tube condensing test will use nitrogen and hydrogen to study the separate effects to be expected in the flight system. Deliberate introductions of fixed volumes of nitrogen will take place and the temperatures, pressures, and flows will be observed. Fluoroscopic examination will also be utilized during this perturbation. Hydrogen will be introduced from a capsule at a constant rate, rather than by discrete volumes, since it diffuses out of the RC tube itself. The rate will attempt to simulate the flight mercury system in-leakage to verify whether our flight design RC has an adequate design margin for noncondensibles.

- g) The effects of tilting the tube both down and up (so that gravity aids and hinders the flow) will be studied with respect to temperature and pressure distributions, interface shape, gas entrainment, flow stability, etc. It will be possible to determine several important, still undetermined pieces of data from this experiment. (1) It will help to determine both interface shape and flow stability under a negative "g" condition. The SNAP 2 Flight System may encounter such negative "g" conditions in case of vehicle attitude control system failure and resultant tumbling. (2) It will determine how strongly the condensing pressure drop and flow regime is affected by gravity. This will aid in interpreting what type of behavior is expected of vertical system and RC test of the flight configuration.
- h) Vibration testing of the tube to determine the effects of external body forces on the interface. This would simulate the effects of the CRU vibration on the RC interface and flow stability.
- i) The use of a glass condensing tube may be feasible if fluoroscopic examination proves unsuccessful in the study of the particular flow regimes.

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3. Equipment Description

a. Phase 1 — System Checkout-Single-Non-taper-Circular-Horizontal Steel Tube

During early testing of the tube, two modifications were deemed necessary in order to obtain more confidence in the measurements of the required parameters.

The loop at present consists essentially of a mercury boiler, 8-ft test section, mercury receiver and condenser cooling which is provided by a damper controlled, centrifugal blower and air distribution plenum (Figure 29).

Figure 29. Test System Sketch

Instrumentation includes thermocouples at 2-in. intervals along the test section, pressure monitoring monometers at 12-in. intervals and a calibrated collection vessel for flow rate measurement. An immersion thermocouple and absolute pressure transducer are located at the inlet to the condenser for obtaining initial mercury conditions.

A vacuum pump is provided for loop evacuation and may be used during operation for noncondensable gas removal.

The entire apparatus has double containment to ensure maximum safety to the operating personnel.

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b. Phase 2 — Single and Multiple, Tapered Steel Tubes

The equipment for this phase of testing will be similar to that used in Phase 1 except the test section will consist of a rectangular tapered tube, simulating the type of tubes used in the SNAP 2 flight system. This tube will have provisions for tilting (longitudinally) and vibrating. Provisions will be made for inserting multiple tubes into the inlet and outlet manifolds.

4. Test Description

The mercury vapor flow rates will be varied from approximately 0.25 to 1.0 lb/min. Inlet pressure to the test section will cover the range of about 2 to 15 psia.

The liquid-vapor interface may be obtained in the tube for a given flow rate, saturation pressure and cooling air flow rate. Altering the cooling flow rate causes an increase or decrease in the required condensing surface. A change in the condensing surface moves the interface along the tube.

The interface is located by the thermocouples located on the tube in the absence of noncondensable gases. In the presence of these gases, the accurate location of the interface will be made with the aid of the fluoroscope. That the interface can be established in this way is clearly shown in Figure 30 which is a photograph of the fluoroscope during a condensing run. In general, the tube

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Figure 30. Fluoroscope Showing Liquid Vapor Interface in Condensing Tube

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runs full of mercury in the condensing section; however, periodically the region near the interface does not run full as was the case when this picture was taken. The subcooling at the interface produces a large temperature drop; see Figure 31 for a typical temperature profile for the tube. The absolute pressures in the condensing section are then obtained from the manometers.

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Figure 31. Temperature Profile, Temperature vs Tube Length, Flow Rate = 0.2 lb/min

The tapered tubes will be tested as above. Complete testing will also be performed during tilting of the tube.

Fluoroscopic studies will be made during vibration of the tube in order to evaluate the stability of the system under such circumstances.

Noncondensable gas injection (N_2 , H_2) will be included in all the above tests. Nitrogen gas will be injected in known volumes and hydrogen gas will be diffused into the tube at rates similar to those predicted for loss of gas from the reactor fuel.

5. Preliminary Results

Initial tests on the apparatus have allowed an evaluation of the required parameter measuring devices and are noted below.

a. Temperature

The expected temperature distribution for the condensing and subcooling regimes was obtained in all the tests so far performed. A typical temperature profile for the tube appears in Figure 31. It is apparent that the temperature instrumentation is adequate.

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b. Pressure

The pressure measurements thus far have not been satisfactory. The connecting lines from the tube to the manometers have been redesigned and are being installed for the next set of tests.

c. Flow Rate

Flow measurements were attempted with a sharp-edge orifice plate, but at the required small flow rates, it is probable that sufficient sensitivity had not been obtained. Therefore, future testing will utilize a calibrated, glass vessel for volumetric flow measurement.

d. Fluoroscopy

A fluoroscope has been adapted to the test assembly and results from the initial checkout are excellent. The liquid-vapor interface is easily seen as well as the behavior of the mercury in the condensing and subcooling areas. Movies of the fluoroscope screen will be attempted upon resumption of operations.

E. FLOW DISTRIBUTION ANALYSIS

An IBM-7090 program is being developed to study the overall thermodynamic and hydrodynamic performance of the SNAP 2 radiator-condenser when it is placed in an earth orbit.

The system model used for the program consists of vapor risers, a split vapor manifold, an even number of condensing tubes, and a liquid manifold. (See Figure 2.) There are two identical systems of vapor risers, each feeding one-half of the vapor manifold. The condensing tubes are tapered as are the attached fins, which are of the trapezoidal type. The liquid manifold is one piece and circular.

Mercury vapor enters the system at the CRU outlet with a known flow rate, pressure, and quality. It flows through the vapor risers and enters the vapor manifold at two locations. It then passes through the vapor manifold, into the condensing tubes where it is condensed. The subcooled liquid is collected in the liquid manifold where it is further cooled before finally leaving at the manifold exit. In the program this exit is given an arbitrary location.

The purpose of the program is to examine the effect of the unsymmetric external and internal heat loads (due to the presence of the sun and asymmetry

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of the boiler) on the overall radiator condenser performance. To do this, the system pressure drop, the condensing length in each tube, the temperature of the liquid mercury leaving the system and other quantities of interest will be calculated when the SNAP unit is in an arbitrary orientation with respect to the sun.

The necessary calculations are carried out as follows:

A flow distribution is assumed in the vapor manifold and for the first iteration the total weight flow of mercury is assumed to be equally distributed among all the condensing tubes. The pressure in the risers and in the vapor manifold can then be calculated using the existing 2-phase pressure drop program (III-C). The flow distribution in the vapor manifold is adjusted until the pressure drop for the two paths are equal. Using the same program, the pressure drop in each condenser tube is calculated for the condensing length. In the remainder of the tube, the subcooled region, the pressure drop is calculated by a direct integration of the one-dimensional momentum equation. Thus, the pressure is obtained at the exit of each condensing tube.

At this point, a flow distribution is assumed in the liquid manifold. Here two effects must be accounted for; the frictional loss and the mixing loss at the junction of condenser tubes and liquid manifold. The mixing pressure drops are calculated using standard correlations of the form

$$\Delta P_m = K \frac{\rho V^2}{2g}$$

where K is an empirical number depending on geometry and Reynolds number. The frictional loss is calculated by the standard formula:

Where

$$f = 64/N_{Re} \quad \text{if laminar}$$

$$f = 0.046/(N_{Re})^{0.2} \quad \text{if turbulent}$$

Again the assumed flow distribution in the liquid manifold is corrected until the pressure drop by each of the two possible paths to the exit are equal.

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The overall pressure drop for each possible path through the whole system is now calculated. There is one possible path for each condenser tube. These are summed, the mean formed, and the flow through each condenser tube is corrected according to the η^{th} root of the deviation of the pressure drop for that path from the mean. η is perhaps 2.0 but can be adjusted. The entire calculation is now repeated. When all of the overall pressure drops deviate from the mean by less than a predetermined amount, the calculations are terminated.

The pressure drop code requires as input the quality at given points (nodal points). These are calculated pointwise from

$$\Delta X = \frac{Q \cdot \Delta l}{W hfg}$$

where Δl is the node length, W the flow in the tube, hfg , the latent heat. Q is the heat flux in Btu/ft-min with the proper sign attached.

For the piping interior to the RC (vapor risers) Q is assumed to be known. For the vapor manifold, condensing tubes, and liquid manifold Q is calculated from the Stephen-Boltzmann law assuming the radiation node dominates. In the case of the vapor manifold and condenser, advantage is taken of the nearly constant saturation temperature to calculate all the Q 's beforehand. This is done in the usual way:

$$Q = \sigma \epsilon \eta T^4 \times (\text{fin width}) = \frac{\text{Btu}}{\text{length-time}}$$

The fin efficiency η is found from curves by a simple fit which works well over the range of values of interest here.

In the subcooled region of the condenser tubes the Q calculated in this way must be modified because of the very rapid change in mercury temperature with length. This is done by multiplying the A as calculated above by $(T/T_s)^{4.0}$ where T is the temperature at the preceding node end. This method is not generally valid because the efficiency is not temperature independent, but a direct examination of curves⁹ shows it to be quite accurate for the range of parameters under study here.

Under the assumption of constant properties no temperature calculation need be made in the liquid manifold until the flow pattern is finally established. Therefore, after the final pressure calculation the temperature drop in the liquid manifold is calculated. Here

$$\Delta T = \frac{Q \cdot \Delta l}{W C_p} .$$

Q is calculated by assuming that the metal exterior to the liquid manifold forms a rectangular fin. The fin efficiency is calculated as before. Here mixing must be considered at every tube-manifold junction. This is done by an energy balance in the usual way. The final calculation gives the temperature of the liquid leaving the liquid manifold. This flow distribution analysis is presently being programmed and is expected to be checked out in 4 to 6 weeks.

F. INTERFACE STABILITY IN RECTANGULAR TUBES

It is known that stable interfaces can be maintained under any orientation within a gravity field, provided that the diameter is under a certain critical dimension.^{45,46} It has been questioned as to whether or not a stable interface can be maintained in a rectangular tube where only one of its dimensions is equal to or under the critical dimension. Tests with mercury in tapered rectangular glass ducts were made to answer this question. The tests showed that stable interfaces can be maintained in rectangular tubes when: (1) in the inverted position, both dimensions are equal to or less than the critical dimension, and (2) in the horizontal position, the vertical dimension only is equal to or less than the critical dimension.

The glass ducts were made by sandwiching two plates of glass in a "V" configuration between two other plates of glass, so that a tapered duct with constant height equal to the thickness of the sandwiched plates resulted. Two ducts were made, with heights equal to 0.12 in. and 0.25 in. under and over critical size respectively.

In the inverted position, a stable interface was formed near the apex of the "V" in the 0.12-in. duct where the width was approximately 0.175 in. 0.175 in. appeared to be the critical dimension. Here both dimensions satisfied

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the stability criterion. In the 0.25-in. duct, no interface could be maintained anywhere within the "V," the height being above the critical dimension.

In the horizontal position where the height is the vertical dimension, stable interfaces can be maintained anywhere within the "V" in the 0.12-in. duct. No stable interface can be maintained in the 0.25-in. duct. Here the vertical dimension is the controlling dimension.

In the horizontal position where the width is in the vertical dimension, stable interfaces can be maintained in both ducts near the apex of the "V" where the width is equal to or less than the critical dimension.

Stable interfaces in rectangular ducts where only one dimension meets the critical size can be maintained in any orientation within the gravity field only when the interface boundary along the larger dimension is fixed. This occurs at the mouth of the duct where the boundary can attach itself to the edge of the glass. Within the "V," the longer boundary is free to move along the surface of the glass and this boundary must be considered as a one-dimensional interface that is capable of instability. These results are described in more detail.¹⁰

G. STABILITY OF RC

1. Stability of RC Under Zero "G" Condition

a. Discussion

Because some or all of the inlet momentum to the condenser tubes is recovered in the condensing process, it is possible that negative pressure drops can occur within the tube. These generally occur at short condensing lengths where the frictional portion of the pressure drop is smaller than the momentum recovery and when the entrance losses are very small. In this negative pressure drop region, there exists a condensing length below which the gradient of pressure drop with condensing length is negative, when flow rate is a positive function of condensing length. Negative gradients with condensing length are undesirable, in that disturbances of the interface in this region are magnified and the liquid leg becomes unstable. The criterion for this stability is that

$$\frac{d\Delta P_c}{dL} > 0$$

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where ΔP_c is the condensing pressure drop between the manifold and the interface. This criterion was applied to the RC whose tube geometry is described in III G2. Condensing pressure drop was calculated as in IIIc.

b. Results

Pressure drop vs condensing lengths were determined for startup conditions where the condensing length is as short as 20 to 40 in. and for steady state conditions where the condensing length is 110 in. In all cases the condensing pressure drop is always positive; there is no region of negative pressure drop, hence no region of negative gradient with condensing length. This is attributed to the entrance loss which nullifies any momentum recovery. An unstable region does exist when the entrance loss is omitted from the calculations, however the normal condensing length is well above this unstable region. Under zero "G" conditions one is confident that the RC liquid leg will be stable.⁴⁷

c. Interface Stability

Under true zero "G" interfaces of any size are theoretically stable (see equation 82).⁴⁵

2. Stability of RC With Tumbling

a. It is possible that tumbling of the SNAP 2-Agena vehicle will occur during orbital operation of the SNAP 2 system. The effects of tumbling on the radiator-condenser could be adverse and therefore an evaluation was performed.¹¹

Because the center of mass of the SNAP 2-Agena combination lies well below the subcooler section of the RC within the Agena, tumbling causes negative acceleration within the RC. This condition would be similar to ground operation if the RC were to be inverted. Depending on the magnitude of the acceleration, the pressure of the vapor may not be capable of supporting the weight of the subcooled liquid with the result that liquid fills the tubes and enters the vapor manifold. Or if the vapor is capable of supporting the hydrostatic head of the liquid, the liquid leg may be unstable to small disturbances. In the first case, there exists nonequilibrium and in the second case unstable equilibrium. Unstable equilibrium is bilateral in that not only can liquid proceed to the vapor manifold after a small disturbance in that direction, but the vapor leg (condensing section) can proceed to the liquid manifold when the disturbance is in the other direction.

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Another form of instability that may occur is that of the liquid-vapor interface. This instability can be considered independently of liquid leg equilibrium and stability, for it may occur during stable conditions of the liquid leg. When the dimensions of the tube are larger than some critical value associated with the local acceleration, the interface will break and liquid will run into the condensing portion of the tube. At the same time, vapor spills into the sub-cooler portion of the tubes and into the liquid manifold with results just as drastic as those of liquid leg instability.

A study was initiated to determine the equilibrium conditions of the RC during tumbling and to determine its stable and unstable regions. Interface instability was also investigated. This was done for steady-state operation only.

b. Discussion

Figure 32 shows all the dimensions of the SNAP 2-Agena combination that are of importance. Shown are the two cylindrical isobars for the pressure regulator that will occur during tumbling about two perpendicular axes. It is seen that in both cases the isobars intersect the projection of the outer tube well below the liquid manifold. The intersection of these isobars for the inner tube are well above the liquid manifold in both cases. With such a distribution of pressure, one would expect fairly complicated distributions of flow and condensing length during tumbling. Each tube and its relationship with the others must be considered under this situation. A brief look at this problem indicated a detailed treatment would require excess complexity. Since the importance of the problem at this time does not warrant this treatment, the following simplified axial model was developed.

In the axial model, only axial dimensions are considered, all the components of the system are assumed to lie on the centerline. In this way, the pressure distributions are uniform from tube to tube and condensing lengths are constant from tube to tube. Condensing length then becomes a convenient independent variable for the analysis.

At a particular tumbling rate, the condensing length must decrease so that the pressure of the condensing vapor at the interface increases sufficiently to support the hydrostatic head of the liquid leg. Decreasing condensing length increases the vapor interface pressure in three ways. First, with reduced

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Figure 32. Schematic of Tumbling System

condensing length, the temperature of the radiator must increase in order to have the same heat rejection rate. The vapor manifold pressure will rise with temperature and this is reflected in an increase in interface pressure. Secondly, the condenser pressure drop decreases with decrease in condensing length; and thirdly, the pressure drop decreases with increased density.

It is convenient to use condensing length as the independent variable in order to generate an equilibrium curve. With the condensing length known, the required manifold temperature (pressure) can be calculated. And with the manifold conditions known for a given condensing length, the condenser pressure drop can be calculated and the interface pressure becomes known. The difference between this interface pressure and that at the design condensing length with no tumbling, can be attributed to the hydrostatic head during tumbling. The length of this head is known and the tumbling rate can be calculated. In this way, an equilibrium curve is generated.

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Due to the simplicity of this model, one can only seek approximate answers or an expected range of answers. This axial model will give optimistic results unless a good deal of pessimism is built into it. The optimism lies in the description of the hydrostatic head. The hydrostatic head is the integral of $\rho w^2 r$ from the regulator isobar to the interface isobar. When only axial dimensions are used, the radius of the interface isobar is shorter than the actual and the hydrostatic length is shorter, hence higher rotation rates are necessary to make up the hydrostatic head. Also, there is some possible optimism in the evaluation of manifold pressure, which is obtained by heat transfer considerations.

For these reasons, three cases were run. The first and most optimistic used the axial dimensions in the description of the hydrostatic head and a variable manifold pressure. The second case used the axial dimensions, but assumed the manifold pressure to be constant at the design value. The increase in vapor manifold pressure is a major contributor in balancing the hydrostatic head so higher tumbling rates can be experienced. Without this increase, a lower equilibrium curve is generated and conservative model results. The third case, like the second, assumes constant manifold pressure, but uses the actual radius of the interface in the outer tube with tumbling about the $\phi = 90^\circ$ axis (Figure 32). Here an abundance of conservatism is built into the model so that it is felt that the equilibrium curve thus generated will be the lowest and onset of stability on this curve will represent the lower limit of the range.

In the first case, manifold pressure is related to manifold temperature via the following saturation curve.

$$T_{vm} = 461.87 + 128.33 (\log p) + 46.67 (\log p)^2 \quad \dots (78)$$

The temperature is related to the condensing length by:

$$T_{vm} = \left[\frac{ld + ld^2/240}{1 + l^2/240} \right]^{0.25} T_{vmd}$$

where d stands for design point. This equation was obtained by simplified considerations of the heat rejection capabilities of the RC. The pressure drop between the manifold and the interface was calculated with the condensing pressure

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drop computer program for the following characteristics and dimensions of the condenser tubes. The pressure drop program is described in Section III-C.

Tube height (in.)	0.270
Bottom tube width (in.)	0.125
Top tube width (in.)	0.249
Tube length (in.)	120
Dog leg length	1.9
Dog leg area change (in. ²)	0.0156
Dog leg entrance loss coefficient	0.5
Liquid momentum dissipation factor	0.57
Tube friction factor	Blasius smooth pipe
Flow rate (lbm/min)	19.8 — to condenser- variable in disturbed tube
Condensing length (l)	variable
Manifold pressure	variable

The interface pressure is output from the program (Figure 33).

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Figure 33. RC Pressures and Temperatures
for Variable Manifold Pressure

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The difference between the interface pressure and the regulator pressure is attributed to the hydrostatic head. The regulator pressure was equated to the interface pressure for the design condensing length of 108 in. with no tumbling and in this way frictional pressure drops were taken into account. The hydrostatic head is the integral of $\rho\omega^2 r$ between the regulator and the interface, and is given by

$$\Delta p_{hs} = 0.607 \times 10^{-3} \omega^2 [r_{\text{interface}}^2 - 5080] \text{ psi} \quad \dots (80)$$

where r represents the axial distance from the center of gravity and is a linear function of condensing length. The equilibrium tumbling rate ω is readily calculated knowing Δp_{hs} and r .

The second case is similarly calculated except that manifold pressure is assumed constant. The third case is similar to the second except that the actual radius (R) is used in Equation 80.

Tumbling rates for all three cases were calculated for condensing lengths between 100 and 108 in. These are shown in Figure 34.

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Figure 34. Equilibrium Tumbling Rate

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With the equilibrium curves established for the entire condenser, attention is focused on the stability of the liquid leg in one tube. An unstable condition exists when the rate of change in vapor interface pressure with change in condensing length is larger than the rate of change in liquid interface pressure during small disturbances that increase or decrease the condensing length. An unbalanced pressure force exists at the interface which continues to drive the interface in the direction of the initial disturbance. Correctly, it is the unbalanced force over the length of the liquid leg that accelerates the liquid leg in that direction; however, since the dynamics are not presently in consideration, this explanation suffices. Stated mathematically the criterion for stability is:

$$\left. \frac{dp}{dl} \right|_{\text{liquid interface}} \geq \left. \frac{dp}{dl} \right|_{\text{vapor interface}} \quad \dots (81)$$

dp/dl (liquid) is simply the derivative of Equation 80 and dp/dl (vapor) can be approximated by the difference in interface pressure evaluated at two points about the condensing length in question. Flow rate variation with condensing length is included in this evaluation.

Both liquid and vapor pressure gradients are plotted in Figure 35 for the first case and in Figure 36 for second and third cases versus condensing length. The intersection of the two is the limit of stable operation in each case. Equation 80 was modified in case three to take into account geometrical considerations (the change in the actual radius R with change in condensing length l). The limit of stable operation in the first case occurs at a condensing length of 106.3 in. In the second and third case, projection of the gradient curves indicates the limit of stable operation to be below 80 and 76 in. respectively. At 106.3 in. the maximum tumbling rate allowable for the first case is 0.86 radians per sec. For the second and third cases, projection of each equilibrium curve to 80 and 76 in. respectively indicates the onset of stability at 0.4 and 0.5 radians per sec respectively.

c. Interface Stability

Although the liquid leg may be stable, the interface itself may not, and the onset of its instability will dictate the maximum tumbling rate. The standard method of determining interfacial stability of fluids in tubes found in many hydrodynamic books is used here.⁴⁶ The criterion for interfacial stability is stated simply by the following equation.

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Figure 35. Liquid and Vapor Pressure Gradient vs Condensing Length—Case 1, Variable Manifold Pressure

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Figure 36. Liquid and Vapor Pressure Gradients vs Condensing Length for Constant Manifold Pressure — Cases 2 and 3

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$$D \leq 1.835 \sqrt{\frac{\sigma}{a(\rho_f - \rho_v)}} \quad \dots (82)$$

where

D = tube diameter

σ = surface tension between the two fluids

a = local acceleration

ρ_f = liquid density

ρ_v = vapor density

Although the equation is set up for round tubes, it can be successfully used for rectangular tubes where D represents the larger dimension. Rearranging Equation 82 one obtains the equation for the maximum local acceleration that can be sustained by an interface in a given tube.

$$a \leq \left[\frac{1.835}{D} \right]^2 \frac{\sigma}{(\rho_f - \rho_v)}$$

Using the tube height, 0.270 in., which is the larger dimension of the tube, for D , 380 dynes/cm for surface tension and 810 lbm/ft³ for the liquid density, the maximum acceleration is 82.5 in./sec². For tumbling:

$$a = \omega^2 r = 82.5 \text{ in./sec}^2$$

Inserting $r = 85.3$ in. for steady-state operation, the maximum tumbling rate is 0.98 radians/sec. This value is above that set by the liquid leg considerations, therefore, the liquid leg value must be taken as the tumbling rate limit.

d. Conclusions and Other Considerations

During steady-state operation, the tumbling rate of the SNAP 2-Agena combination must be kept below 0.4 radians/sec (approximately 4 rpm) to ensure stable conditions in the radiator condenser. This value is set by hydrodynamic stability considerations of the liquid leg (subcooler section). Interface stability is also maintained at this value and may be maintained up to 0.98 radians/sec.

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In the real case where hydrostatic lengths are different in each tube due to the cylindrical isobars and the variation in flow rate and condensing length, one would expect a different equilibrium curve and different stability range. But a good deal of conservatism was figured into the horizontal isobar cases, so that the 4 rpm figure can be accepted as the limit. The actual limit for the real case is expected to lie somewhere between 4 and 8 rpm.

The question of stability under natural rotation in earth orbit has been considered. It, however, is dwarfed in comparison to the tumbling rates dealt with in this study. Natural rotation is in the order of 90 min per orbit or approximately 0.01 rpm.

H. THERMAL CONDUCTANCE TESTS OF SNAP 2 RADIATING FIN

1. General

The effectiveness of the SNAP 2 radiator condenser is dependent partly on the overall lateral conductivity of the copper fin and steel honeycomb sandwich combination. Due to the lack of this overall conductivity data, past analyses have had to be conservative. Conduction heat transfer through the braze and steel sections were neglected and all heat transfer was charged to the copper fin alone. This analysis, however, may not be conservative. If the value of the conductivity in the copper fin is significantly degraded due to diffusion of impurities (braze alloy elements) into the copper fin during the brazing operation, the analyses will be optimistic.

Early tests indicated that the effective thermal conductivity of a brazed composite section (12 mil copper, 2 mil 731 braze, 6 mil steel) was between 75 and 85% of the conductivity of pure copper, based on the area of copper alone. These results were obtained by relating thermal conductivity to electrical resistivity, and could only be considered a rough estimate of the degradation in thermal conductivity. These results indeed supported the concern for degradation that had been previously expressed.

Because of this concern for degradation and the lack of confidence in the relationship between electrical resistivity and the thermal conductivity, thermal conductivity tests on composite specimens were proposed. To facilitate obtaining thermal conductivity data and to remain conservative in radiator analyses, it was proposed to test composite sections containing the copper fin, braze,

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and top steel face sheet only. The bottom face sheet and honeycomb would be neglected from tests and future analyses, and conservative results would be expected. The amount of conservatism would be small, of course, since the conductance in this amount of steel is small in comparison to the conductance available in the copper fin.

Not only was confidence lacking in the relationship of electrical resistivity to thermal conductivity, but also there was the question of the temperature dependence of thermal conductivity. If the relationship between electrical and thermal were good at room temperature, the temperature at which the electrical measurements were made, there was no guarantee that thermal conductivity would be the same percentage of pure copper at the higher radiator temperatures. The determination of a possible temperature dependence became a secondary purpose for the proposed tests.

Thermal conductivity is much more difficult to obtain than electrical resistivity, therefore it would be expedient to use electrical measurements if the relationship between the two holds and if there is little or no dependence on temperature. It was proposed also to test these thermal conductivity specimens electrically, to determine if the relationship is good, in the event that there is little or no temperature dependence.

Thermal conductance, the sum of all the conductivity-thickness products of the laminate, was expected to be linear with copper fin thickness. This was based on the assumption that the diffusion depth of the impurities into the pure copper would be constant and that the reduction in conductivity in this zone would be constant. By varying the copper thickness above this value of diffusion depth, one would expect a linear variation in conductance, with a slope equal to the conductivity of pure copper. If the diffusion depth or the concentration of impurities within the whole copper section were a function of copper thickness, a nonlinear conductance curve might result. To check this assumption, at least three copper thicknesses had to be tested for each braze alloy used.

Heat transfer analysis on various composite fin specimen configurations indicated that the nontapered rectangular radiating fin was the best test configuration for the accurate results that were needed. This method would, of course, be predicated on knowing the radiation temperature level and the emissivity accurately. This could be done within accepted tolerances so the fin configuration was adopted for the tests.

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Two types of braze alloy have been proposed for joining the copper fin to the steel face sheet. They are a copper-manganese braze (75% copper-25% manganese) and 731 braze (approximately 70% copper, 30% silver, and 1% lithium from whence its name 731 comes). These two brazes were subsequently tested.

2. Discussion

a. Conductance, Theory, and Application

Figure 14 shows the cross section of the SNAP 2 RC tapered fin. The thermal conductance through any lateral cross section of it may be given by:

$$K = k_{st}t_{st} + k_{br}t_{br} + k_{dif}t_{dif} + k_{Cu}t_{Cu} \quad \dots(83)$$

where

k = thermal conductivity

t = thickness

st = steel

br = braze

dif = diffusion zone

Cu = pure copper

If the conductivities and the diffusion depth are constant, the equation reduces to the following linear equation.

$$K = K_1 + k_{Cu}t_{Cu} \quad \dots(84)$$

where

$$K_1 = k_{st}t_{st} + k_{br}t_{br} + k_{dif}t_{dif} \quad \dots(85)$$

since t_{Cu} is the only variable.

Since the pure copper thickness is the original copper thickness less the diffusion depth ($t_{Cu} = t_{Cu_o} - t_{dif}$) the equation may be expressed as:

$$K = K_2 + k_{Cu}t_{Cu_o} \quad \dots(86)$$

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Although K_1 is always positive, K_2 will be negative when the diffusion depth is very large. When negative, this linear curve passes through zero conductance at a small positive value of original copper thickness. This curve is shown in Figure 37. This is physically not true in this range; there is always positive conductance. Actually, in this neighborhood the slope of the curve is not that of pure copper, but approaches that of the braze and steel combination. At t_{Cu_0} equal to zero, the conductance should be that of the braze and steel. This is shown by the dashed line in Figure 37.

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Figure 37. Expected Curve for Conductance

For analytical purposes, however, the linear relationship can be used when t_{Cu_0} is much larger than the intercept value and when conductivities and diffusion depth are constant. The minimum thickness to be used for the RC will be 6 mils, whereas the intercept value is expected to be between 0 and 4 mils, based on electrical resistivity measurements. Design of the tapered fin can first be based on the thicknesses of pure copper required to remove the heat from the condenser. Then an additional thickness of copper equal to the intercept value can be added to it for the manufacturing specifications.

3. Test

a. Test System Configuration

A photograph of the system utilized for thermal conductivity determinations is presented in Figure 38. It shows a copper capsule enclosed in a glass bell jar. The fin specimen, not shown, is suspended within the capsule. Brass plates were fitted to the bell jar at both the top and bottom. Specimen thermocouples were led through the bottom plate to a thermocouple selector switch, and subsequently to the potentiometer. The vacuum line, a thermocouple vacuum gage, and an ionization vacuum gage were mounted on the top plate. In order to achieve the vacuum required to sufficiently reduce convection losses from the surface of the specimen, an oil diffusion vacuum system was installed to supplement the mechanical vacuum pump.

b. Capsule Configuration

A half shell section of the copper capsule used as the heat sink is shown in Figure 39. It is 5-1/4 in. in diameter and 11 in. long. Shown also is a fin specimen (1 in. by 8 in.) suspended within the capsule. The ratio of capsule area to specimen area (28.2 to 1) is sufficient to reduce significantly the absorption of reflected radiation from the capsule. The exchange factor closely approximates the emissivity of the fin.

Copper tubing was coiled around the capsule and bonded to the capsule with silver solder. Water was used to cool the capsule. The interior of the capsule was coated with PT-404, a high emissivity coating ($\epsilon = 0.90$, at room temperature) to absorb heat emanating from the surface of the specimen. The interior surface of the capsule was first grooved to enhance the absorption characteristics of the capsule. Chromel-alumel thermocouples were attached to the exterior of the capsule (top, side, and bottom) so that the ambient temperature could be known

c. Specimen Configuration

Uniform thickness fin specimens were fabricated. Each specimen consisted of an 8-mil nominal PH15-7 Mo steel sheet brazed on both sides to OFHC copper sheets. This was done to ensure against bimetallic warping of the specimen during test. Two braze alloys were investigated, alloy 731 (70 Cu - 29.5 Ag - 0.5 Li) and a Cu-Mn alloy (75 Cu - 25 Mn). The 731 braze

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Figure 38. Test Apparatus

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Figure 39. Test Capsule

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alloy joint width was 1.5 mils thick nominal in each specimen, while the Cu-Mn braze alloy joint width was 2 mils nominal. Copper sheet thicknesses of 7, 10, 15, and 20 mil nominal were employed. Micrometer measurements indicated that the specimens were of essentially constant thickness (i. e. ± 0.2 mil) over the length of the specimen. Brazing cycles for both alloys were: 1750°F for 20 min, cooled to 100°F for 8 hr, and held at 1050°F for 1 hr. The procedure was adopted to heat treat the PH15-7 Mo to the RH1050 condition while brazing. All specimen dimensions were 8-in. long and 1-in. wide. One-half of each specimen was coated with the SNAP 2 high emissivity ceramic coating to serve as a high emissivity surface. The remainder of the specimen was polished to affect a low emissivity surface for maximum heat retention. The edges of the specimen were also polished.

Five-mil-diameter iron-constantan thermocouples were used to measure fin temperatures. The thermocouples were placed in the specimens by drilling holes (approximately 12-mil diameter) through the specimens and inserting the tips. In most specimens, three thermocouples were located within 1/10 in. of the tip, one thermocouple approximately 2 in. from the tip and three approximately 4 in. from the tip. See Figure 39.

Two nichrome heaters enclosed in a copper block served as the heat source.

To minimize convection heat transfer from the fins, the maximum tank pressure was established at 1μ .

4. Results

a. Thermal Conductance Tests

The temperature differences between the tip thermocouples were small (less than 1°F in most cases), so these were averaged and assigned to the average thermocouple placement distance from the tip of the fin. The difference in temperature due to placement in the neighborhood of the tip is negligible since the temperature gradient is practically zero there. This is plainly shown by the temperature profiles in Figure 40. The middle thermocouples and the top thermocouples were treated individually, however, since differences in their placement result in large temperature differences. This can be seen in Figure 40 also.

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Figure 40. Typical Temperature Profile for Two Fins

The three capsule thermocouples were also averaged. In most runs, the differences between these thermocouples were about 1°F , but as high as 3°F .

Figure 41 shows the spread in the calculated value of conductance versus the original copper thickness. The data as expected are fairly linear with thickness over the range investigated. It does show the expected trend at small values of thickness, C7 and D7 are slightly high indicating a decrease in overall conductivity (lower slope). The intercept of these lines are at copper thickness of 0.6 mil and 4.4 mils and the slopes are 18.8 and 18.4 Btu/hr-in. - $^\circ\text{F}$ (225 and 221 Btu/hr-ft- $^\circ\text{F}$) for the 731 braze and the copper manganese braze respectively.

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Figure 41. Thermal Conductance vs Original Copper Thickness

Figure 41 shows the superiority of the 731 braze over the copper manganese. Its intercept value is approximately 3.8 mils smaller than that of the copper manganese. This means that the SNAP 2 tapered fin can be 3.8 mils thinner when 731 braze is used instead of copper manganese and represents a weight saving of 30 lb.

To determine the dependence of conductance on temperature, the conductance calculated for each thermocouple region was compared at different average temperature levels. In this way, error between thermocouples is minimized and placement errors are ruled out.

Within the limits of the experimental error, it was not possible to establish any definite temperature dependence. This may be due to temperature dependent errors in individual thermocouples and possibly readout error. In most individual cases, the dependence, whether positive or negative, is small.¹²

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An error analysis was made¹² to determine the accuracy of the calculated conductance, thermocouple error, potentiometer error, readout error, thermocouple placement error, and error in the measurement of emissivity. The largest, and most probable error for the 4-in. fin length region is $\pm 4\%$ and the largest, most probable error for the 2-in. fin length region is between $\pm 4\%$ and $\pm 10\%$. The longer the measured region the more accurate the conductance results.

5. Comparison of Electrical Conductivity to Thermal Conductivity

a. Theory

Both thermal and electrical transfer are analogs in that thermal transfer is electrical in nature. Their rate equations shown below are similar for one dimensional transfer in composite metals.

$$q = -b \sum k_i t_i \left(\frac{dT}{dx} \right) \quad (\text{thermal}) \quad \dots (87)$$

$$I = \frac{b}{\ell} \sum c_i t_i \left(\frac{dV}{dx} \right) \ell \quad (\text{electrical}) \quad \dots (88)$$

where

$$\sum k_i t_i = K \quad \dots (89)$$

and

$$\frac{b}{\ell} \sum c_i t_i = \frac{1}{R} = c \quad \dots (90)$$

The term $\sum k_i t_i$ in Equation 89 has been called conductance although it differs in form from the electrical conductance (reciprocal resistance) in Equation 90. Upon expanding both terms for conductance for the SNAP 2 fin, we get

$$k_{st} t_{st} + k_{bi} t_{bi} + k_{dif} t_{dif} + k_{Cu} t_{Cu} \quad \dots (91)$$

$$\frac{b}{\ell} (c_{st} t_{st} + c_{br} t_{br} + c_{dif} t_{dif} + c_{Cu} t_{Cu}) \quad \dots (92)$$

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Assuming constant conductivities in a tapered fin, these conductances may be expressed as:

$$K = K_1 + k_{Cu} t_{Cu} \quad \dots(93)$$

$$C = C_1 + c_{Cu} t_{Cu} \quad \dots(94)$$

And since the copper thickness is the original copper thickness less the diffusion depth the equations may be rewritten as follows:

$$K = K_2 + k_{Cu} t_{Cu_o} \quad \dots(95)$$

$$C = C_2 + c_{Cu} t_{Cu_o} \quad \dots(96)$$

Equation 96 describes a curve similar to that discussed in Figure 41. Assuming the effective electrical and thermal diffusion depths (intercepts) are the same, the ratio of thermal conductance of a composite to the thermal conductance of the original copper thickness is equal to the ratio of the electrical conductance of the composite to the electrical conductance of the original copper thickness. Since the slope of the curves are equal to their conductivities (k_{Cu} and c_{Cu}) the first terms of the equations can be expressed by

$$K_2 = -k_{Cu} \Delta t \quad \dots(97)$$

and

$$C_2 = -c_{Cu} \Delta t \quad \dots(98)$$

where Δt is the effective diffusion depth (intercept value). The ratios stated above can be written as follows:

$$\frac{K}{k_{Cu} t_{Cu_o}} = \frac{t_{Cu_o} - \Delta t}{t_{Cu_o}} \quad \dots(99)$$

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$$\frac{C}{c_{Cu} t_{CuO}} = \frac{t_{CuO} - \Delta t}{t_{CuO}} \quad \dots (100)$$

and

$$\frac{K}{k_{Cu} t_{CuO}} = \frac{C}{c_{Cu} t_{CuO}} \quad \dots (101)$$

b. Electrical Resistivity Tests

Specimens similar to the thermal conductivity specimens were tested electrically at room temperature. They differed only in that they were one-half of the "back to back" type tested thermally and their thicknesses varied slightly. Four specimens (7, 10, 15, and 20 mil copper) were brazed with the 731 braze and one (10 mil copper) with the copper manganese braze. Their conductances are plotted in Figure 42 against original copper thickness. Electrical conductances are normalized to a specimen of unit width and unit length.

The electrical conductance curves in Figure 42 are very similar to the thermal conductance curves of Figure 41. The intercept values are, however, slightly different, about 1.1 mils compared to 0.6 mil for the 731 braze and 3.8 mils compared to 4.4 mils for the copper manganese braze. The value of conductance for the OHFC specimen is close to the expected value shown by the upper dashed line. The line through D10 was made parallel to the line through the 731 data so that a value for the copper manganese intercept could be found. It is interesting to note that both conductances have the same pattern about the linear curve for the 731 braze.

A comparison of electrical results to thermal results are made in Figure 43. The base of the figure is the same as Figure 41. The electrical results are plotted proportionately from the dashed line of Figure 42 against the dashed line of Figure 41. The agreement between the two is fair. The 731 electrical results are lower since its thickness intercept value is larger by 0.5 mil and the opposite is true with the copper manganese since its intercept value was lower by 0.6 mil. At 10 mils, the 731 electrical results are 6% lower than the 731 thermal results.

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Figure 42. Electrical Conductance vs Original Copper Thickness

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Figure 43. Comparison of Electrical Conductance to Thermal Conductance

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c. Conclusions

Thermal conductance in composite fins is a fairly linear function of original copper thickness. Design of the tapered copper fin can first be based on the thicknesses of pure copper that are required to remove the heat from the condenser, then an additional thickness of copper equal to the intercept value of 0.6 mils must be added for the design specifications. The bulk of the calculated results, the 4-in. fin length data which are more reliable, lie in the upper section of the spreads shown in Figure 41, so the curves are somewhat conservative. To allow a little more conservatism for the negative part of the error, 1 mil could be accepted as the intercept value for design. The conductivity of pure copper to be used in the design analysis is the 223 Btu/hr ft°F, the average of the slopes of the curves in Figure 41.

Electrical resistivity is a fair method in estimating the conductance in composite fins. It is not as accurate as the thermal measurement, but is a useful method in comparison of conductances providing a standardization is employed. It is recommended that it be used in quality control of the brazed radiator panels. Specimens can be scattered throughout the panel retort during the brazing operation. Comparison of their electrical conductances to those in Figure 43 will indicate whether or not the thermal conductance of the panel meets the design specifications.

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IV. NaK SYSTEM PRESSURE DROP

A. GENERAL

Because of the TE pump design limitations, the primary loop pressure loss is very critical. A large pressure drop means a greater weight penalty because of the necessary increase in TE pump size (See Figure 44). An initial pressure loss of 1.05 psi had been calculated for the NaK system. To provide some contingency for uncertainties in the calculations and pump performance, the NaK tubing size was increased from 1-1/4 in. to 1-3/8 in. and the design pump pressure rise held at 1.05 psi. For the TE pump mounted on the cold leg this resulted in a 0.2 psi contingency.

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Figure 44. SNAP 2 Chromel
Constantan TE Pump Weight
Variation With Pressure Output

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Because of the questionable capability of the cold line pump to provide stable flow during orbital startup, the reference design was chosen as the chromel-constantan pump (See Figure 45) operating between the hot boiler NaK inlet line and the cold boiler mercury inlet line.

At the time, thermoelectric pump concepts for either the hot (chromel-constantan) or the cold leg (lead-telluride) were being considered. The Pb-Te pump cannot be mounted on the hot leg due to temperature limits of the materials.

A weight optimization analysis was performed, allowing the TE pump size to decrease with increasing NaK piping diameters (See Figures 46 and 47). While saving weight by decreasing the TE pump size, extra weight due to the increase in NaK tubing and the expansion compensator almost cancelled this, resulting in a weight saving of less than a pound (See Figure 48).

1. Primary Loop Pressure Drop Analysis

In the present analysis, the system is divided into four sections: the hot NaK leg, cold NaK leg, reactor, and the boiler. The analysis has been coded for the IBM 7090 computer to allow quick evaluation of system changes and new viscosity data over the entire operating temperature range. The equations used in this analysis are given in Tables XII and XIII. Pressure losses were initially determined for the three proposed designs for locating the TE pump in the system. The designs considered are defined as:

hot leg design	TE pump is mounted on the hot leg of the primary loop
cold leg design	TE pump is mounted on the cold leg of the primary loop
parallel design	Two TE pumps mounted on the hot leg of the primary loop in parallel.

The results of this analysis are presented in Reference 16 and in Tables XIV and XV.

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a.

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b.

Figure 45. Thermoelectric Pump

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Figure 46. Weight Penalty and Weight Savings Variations With System Pressure Drop

After the hot leg design had been chosen as the flight design and after the piping layout for the primary loop had been finalized, a new calculation was made. The results of this calculation are shown in Table XVI. As shown, the system and pump contingency has been increased to 0.34 psi for a design pressure drop of 1.05 psi. Since higher NaK flows result in increased boiler operating margin this larger contingency increases the overall system reliability.

a. Primary Loop Pressure Drop Code

Due to the time involved in making the hand calculation, and because of the many changes occurring in the piping layout during the development of the

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Figure 47. Expected NaK Expansion Compensator Weight Increase
With System Pressure Drop

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Figure 48. Total Weight Penalty
Or Savings Variation With
System Pressure Drop

system, a computer program was written to perform the analysis. Input into the program consists of operating temperatures, bend angles, radii, tubing length, inside tubing diameter, inside weld diameters, the number of welds per section, and the NaK flow rate. The loop is divided into four sections at present, although it can be divided into more. Output consists of the loss per section, total loss due to welds, total loss for orbital operation, and total loss for ground operation.

For ground testing, there will be a loss due to buoyant forces induced by the changes in the density resulting from a temperature variation around the loop. This loss and the method of calculation is discussed.¹⁷ The buoyant loss is determined by summing the increment losses around the loop.

For an increment of constant temperature the buoyant loss is given by:

$$\Delta P_{\text{buoyant}} = \sum_{n=0}^N \left(\bar{\rho}_n \Delta Z_n \right)_{\text{cold leg}} - \sum_{n=0}^N \left(\bar{\rho}_n \Delta Z_n \right)_{\text{hot leg}}$$

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TABLE XII
PRESSURE DROP EQUATIONS

Equation Used	For	Reference
$\Delta P = f \frac{L}{D} \frac{\rho v^2}{2g_c}$	Straight tubing	13
$\Delta P_{90} = f \left(\frac{L}{D} \right)_e \frac{\rho v^2}{2g_c}$	90° bend	14
$\Delta P_\theta = C_\theta \Delta P_{90}$	Bends	15
$\Delta P = K \frac{\rho v^2}{2g_c}$	Expansions contractions	14
$\Delta P = 4f' \frac{NG_{\max}^2}{2g_c}$	Shell side of the boiler	13
$f' = \frac{(0.004 + 0.08X)}{(X_T - 1)^n} \frac{(D_o G_{\max})^{-0.15}}{\mu}$		13
for $2000 \leq \frac{D_o G_{\max}}{\mu} \leq 40,000$		

TABLE XIII
PRESSURE DROP COEFFICIENTS

Coefficient	Name	Reference
f	Friction factor	14
C_θ	Correction coefficient for Bend θ	15
K	Resistance coefficient	14
f'	Friction factor for flow over tubes defined in Table XII	13

TABLE XIV

NaK SYSTEM PRESSURE LOSS

Condition	Ground	Orbit
5% flow, 40°F isothermal		
a. hot leg design	0.01 psi	0.01 psi
b. cold leg design	0.01 psi	0.01 psi
c. parallel design	0.01 psi	0.01 psi
5% flow, 1015°F to 1215°F		
a. hot leg design	0.093 psi	0.004 psi
b. cold leg design	0.093 psi	0.004 psi
c. parallel design	0.093 psi	0.004 psi
100% flow, 1015°F to 1215°F		
a. hot leg design	1.04 psi	0.91 psi
b. cold leg design	0.98 psi	0.85 psi
c. parallel design	1.09 psi	0.96 psi

TABLE XV

NaK SYSTEM PRESSURE LOSS FOR THE
SINGLE HOT LEG TE PUMP DESIGN

	PSIA	Condition
Reactor	0.2	
Hot Leg	0.21	In orbit, 100% flow, 1015°F to 1215°F
Boiler	0.25	
Cold Leg	0.25	
Total	0.91	

TABLE XVI
NaK SYSTEM PRESSURE LOSS
PER SECTION

Section	Loss psi
Reactor	0.20
Hot Leg	0.16
Boiler	0.25
Cold Leg	0.10
Best Estimate	0.71
System Contingency	0.17
Pump Contingency	0.17
System Pressure Drop	1.05

where

$\bar{\rho}_n$ the average density over the increment Z_n to Z_{n+1}
 ΔZ_n the increment length in elevation from n to $n+1$
 N the total number of increments

Table XVII represents the pressure drop of the present flight design, including all weld losses, the latest modification in the boiler, and the TE pump in the hot leg. The increase in system and pump contingency to 0.47 psi for a design pressure drop of 1.05 psi is quite desirable as previously discussed. No further significant changes in NaK system pressure drop are anticipated. Figure 49 is a plot of the calculated system pressure loss as a function of flow rate for the constant temperatures shown.

2. Boiler Hydraulic Test

The pressure drop across the boiler (See Figure 50) was initially calculated, using a conservative approach, to be 0.4 psi. Water tests have now been run on two boilers, and at the design point the pressure loss calculated from the measured loss coefficients was 0.25 and 0.24 psi.

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TABLE XVII

NaK SYSTEM PRESSURE LOSS PER
SECTION SHORT BOILER DESIGN

Section	Loss ~ psi
Reactor	0.20
Hot Leg	0.11
Boiler	0.19
Cold Leg	0.08
Best Estimate	0.58
System Contingency	0.24
Pump Contingency	0.23
System Pressure Drop	1.05

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Figure 49. NaK System Pressure Loss

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Figure 50. Model 5C Boiler

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The tests were conducted over a wide range of flow rates. From this information, a total loss coefficient was determined as a function of Reynolds number (See Figure 51). The Reynolds number range ran from approximately 50% below that of the design point flow rate (70 lb/min) to 50% above it. Reference 18 summarizes the results of this test.

The pressure loss coefficient for the boilers is defined by the following equation:

$$\Delta P_B = K_B \frac{v_{NaK}^2 \rho_{NaK}}{2g_c}$$

Where

K_B = The loss coefficient taken from Figure 51 and based upon a Reynolds number at the NaK flow rate in a 1.0 in. ID tube.

v_{NaK} = The velocity of the NaK in 1.0 in. ID tube for the NaK flow rate being used.

ρ_{NaK} = The density of the NaK at the mean boiler temperature.

Figure 52 shows the calculated pressure drop across the boiler as a function of NaK flow rate for the empirical loss coefficient obtained.

3. Pipe, Weld, and Bend Hydraulic Test

The NaK system will have a minimum of twenty welds which will contribute to the total system pressure loss. Tests have revealed that at flow rates in the SNAP 2 design point range, the pressure loss across a weld is significant. The following equation is used for the weld loss coefficient. It is the sum of the coefficient for a sudden expansion and a sudden contraction.

$$K_{Total} = (1/\beta - 1)^2 + 0.23 (1 - \beta^2)$$

Where

β = The flow area with the weld divided by the flow area without the weld.

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Figure 51. Comparison of Total Loss Coefficients For the 5C-4 And 5C-6 Boilers

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Figure 52. Total Pressure Loss For the NaK Side of the 5C-4 Boiler

Thus

$$\Delta P_w = \frac{K_{\text{Total}} v_{\text{max}}^2 \rho_{\text{NaK}}}{2g_c (144)}$$

Where

v_{Max} = The velocity at the weld
flow area.

At present, the total weld loss is calculated to be only 4.1% of the system loss. However, the test mentioned previously indicates that the loss determined by the above equation is too small. In the equation, the weld is assumed to decrease the inside tube diameter by 50 mils, which was determined by actual measurement of typical welds. Due to the uncertainty in the pressure loss, a test will be made in the near future to determine the loss for eleven welds in series, each typical of those expected in the flight system.

Because of the importance of pressure loss in the NaK system, the pressure drop across bends of 30, 45, 60, 90, and 180° will be determined in the same test loop as the welds. With these results, a total loss coefficient as a function of Reynolds number will be determined and plotted similar to the boiler results. These results will be incorporated into the primary loop pressure drop code as have the results of the boiler test.

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V. PRIMARY SYSTEM BRACKET THERMAL LOSSES

A. GENERAL

It is necessary that the NaK be maintained and circulated above 40°F prior to startup. The power for this heating must be supplied by batteries, which results in a system weight penalty. After startup, heat loss through the brackets to the radiator-condenser must be compensated for by increased radiator-condenser performance at an additional cost in system weight. While some NaK system heat loss is desirable to preheat the mercury components prior to injection, radiation is more effective than local conduction through the brackets. Therefore, the heat loss through the brackets should be minimal both before and after startup.

1. Initial Analysis

In the initial analysis³⁴ the primary concern was the nonuniform heat load to the RC in the vicinity of the boiler. To minimize the local RC heating, it was found that gold foil over eight tubes on the RC in the vicinity of the boiler was adequate, and that a radiation shield between the boiler and RC was not required. Later thermal analysis of startup requirements indicated that the portion of the RC that was gold coated would be below the freezing temperature of mercury and these tubes might plug during startup. A more detailed, but preliminary review of the problem, including mercury vapor flow redistribution due to large local heat loads, indicated that even the gold foil could be removed without danger of mercury vapor entering the liquid manifold. A detailed analysis of the flow redistribution resulting from nonuniform heat loads is being programmed on the IBM computer. (See Section III E).

2. Present Analysis

The present analysis is primarily concerned with determining the bracket heat losses and indicating how they can be minimized. Several types of brackets have been considered, including those with vaporizing inserts to eliminate conduction, with low conductivity material inserts, and with coiled spring or ball bearing inserts to minimize contact area. The most promising technique to date appears to be the use of coiled springs or perhaps ball bearings between the clamping device and the part being held (Figure 53). A small test program has been initiated to evaluate the thermal resistance of this type of bracket.

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Figure 53. Bracket Model With Temperature Nodes

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The actual analysis to date has been confined to the bracket designed for the developmental structural system test. Both radiation to and from, and conduction through the brackets were taken into account. Direct view factors were determined by computer programs written for problems of this type.²⁰ The model was broken into five nodes, and a computer program was used to solve the nodal equations. Figure 53 shows the model used, and Figure 54, the thermal circuit.

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Figure 54. Thermal Circuit Used
for the Analysis

Table XVIII gives the total heat loss conducted through the brackets from the primary loop. The anchor bracket losses are estimated. Complete details are presented in Reference 35.

For the ground test losses, it was assumed the tank vacuum (10^{-2} mm) was low enough to eliminate thermal convection but not low enough to eliminate gaseous conduction in the joints between bracket surfaces. The temperature of

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TABLE XVIII

HEAT LOSS BY CONDUCTION THROUGH THE BRACKETS

Bracket Station Number (watts)	Hot Sink	Cold Sink	In a Vacuum		For Ground Test	
			Before Startup (watts)	After Startup (watts)	Before Startup (watts)	After Startup (watts)
132.7	Hot leg	RC	0.40	2.77	0.69	4.97
	Cold leg	RC	0.40	1.87	0.69	3.35
117.0	Hot leg	RC	0.40	2.77	0.69	4.97
	Cold leg	RC	0.40	1.87	0.69	3.35
101.5	Hot leg	RC	0.40	2.77	0.69	4.97
	Cold leg	RC	0.40	1.87	0.69	3.35
83.125	Boiler	RC	0.99	6.95	1.51	11.10
55.2	Boiler	RC	0.99	5.81	1.51	9.27
28.5	Boiler	RC	0.99	4.68	1.51	7.44
Total			5.37	31.36	8.67	52.77
Estimate of Anchor Bracket 2 PSM2A			8.00	37.50	12.00	54.50
Total			13.37	68.86	20.67	107.27

the RC for both environments was 600°F after startup and -160°F before startup. Temperatures of 1215, 1115, and 1015°F were used for the NaK after startup depending on where the bracket fastened to the primary loop.

a. Interface Conductance Loss

In space or a vacuum, there is no conducting fluid between metal surfaces in contact. The absence of this fluid means one less heat transfer path. Tests have revealed the importance of this fluid path in conduction across a joint. The results shown in Figures 55 through 58 indicate that the thermal conductance is from 5 to 30% of the thermal conductance in air under the same conditions. In the analysis, 15% of the value of the thermal conductance in air was used. The contact resistance reduces the conduction heat load by slightly over 50%.

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Figure 55. Thermal Conductance for Aluminum/Aluminum Joint

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Figure 56. Percent Conductance in a Vacuum for Aluminum/Aluminum Joint

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Figure 57. Thermal Conductance of Aluminum/Steel Joint

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Figure 58. Percent Conductance in a Vacuum
for Aluminum Steel Joint

Thermal contact conductance increases with contact pressure. However, once in space, there will be no load except that caused by thermal expansion. The amount of area in contact at a joint is very important to the magnitude of the heat conducted across the interface. At the boiler, or tube to bracket joint, the area is quite small, because of the sliding spring design. This is not true for other joints in the bracket design.

b. Radiation Losses

The heat transferred from the primary loop to the brackets by radiation was found to be very significant at high temperatures. At low temperatures, it is almost negligible. The following illustrates this fact for the boiler brackets.

	Radiation (watts)	Conduction (watts)	Total (watts)
After Startup	15.5	13.9	29.4
Before Startup	0.16	2.97	3.13

Therefore, at low temperatures radiation contributes about 5% to the total heat load from the boiler to the boiler brackets. This compares with the figure of greater than 50% for the higher temperatures.

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c. Proposed Thermal Contact Test

Due to the importance of the thermal contact resistance in the present analysis, a test has been proposed to determine this important quantity. The test will be designed to evaluate the spring type contact used in the brackets as a function of temperature across the joint, contact pressure, and environmental pressure.

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VI. INSTRUMENT COMPARTMENT THERMAL ANALYSIS

A. GENERAL

The SNAP 2 instrument compartment is composed of eleven modules mounted on the inside of an aluminum ring which is 5 ft in diameter. The modules control and monitor all operations of the power unit. The instrument compartment is eight in. high and is located between station 8.0 in. and station 0.0 in. This study was made to determine:

- 1) The maximum and minimum component temperatures at both prestart and full power phases.
- 2) The type of emissivity coatings to be applied to the inner panel, module surfaces, outer panel and load ring surfaces.
- 3) The effects of two types of thermal barriers: (a) Inverted truncated cone (b) Convex Dome shaped
- 4) The conduction and radiation from electrical wires.
- 5) The conduction from the mercury subcooler.
- 6) The heat leak through the thermal barrier due to fasteners, etc.

1. Prestart Analysis

The steady-state and transient temperature fluctuations of the SNAP 2 instrument compartment at prestart power phase were computed by summing the external and internal heat loads and equating them to radiation losses to space.

Internal conduction between panels, radiation interchange between modules, and an assumed adiabatic SNAP 2 Agena interface were included in this analysis.

a. Method

(1) External Heat Inputs

There were two orbits considered, they are:

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- a) Sun-shade transient orbit where the plane of the orbit is parallel to the sun's rays, (see Figure 59).
- b) Constant-sun, constant-shade orbit where the plane of the orbit is perpendicular to the sun's rays.

Figure 59. Constant Sun,
Constant Shade Orbit

In both cases, the orbit is assumed circular and the projection of the vehicle axis passes through the center of the earth. Another assumption is that the sun is infinitely far away, such that the rays striking the earth are parallel, (the actual maximum angle is less than 32 minutes).

For any orbit there are three major sources of external heat input: (a) direct sunlight (b) thermal radiation from the earth (c) sunlight reflected off the earth and the earth's atmosphere. For the calculation of external heat inputs, the instrument compartment has the shape of a cylinder.

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(a) Calculation of External Heat Inputs for Sun-Shade
Transient Orbit

(1) Direct Solar

The heat input to one panel of the instrument compartment due to direct sunlight is given by:

$$Q_{DS} = q_S \alpha_S A(\phi) \quad \dots (102)$$

where:

$$q_S = \text{solar constant} = 443 \text{ Btu/hr-ft}^2$$

$$\alpha_S = \text{absorbtivity to solar radiation (based on using Tin Oxide as the emissivity coating; } \alpha_S = 0.3, \epsilon_T = 0.9) = 0.3$$

$A(\phi)$ = projected area of one panel on a plane perpendicular to the sun's rays.

$A(\phi)$ depends on the orbital position angle ϕ and can be closely represented by the function:

$$\begin{aligned} A(\phi) &= A_o \sin \phi, \quad 0 < \phi < (90 + \theta) \\ &= 0, \quad (90 + \theta) < \phi < 360 \end{aligned}$$

where:

$$A = \text{projected panel area } 0 \geq A \geq 1 \text{ sq ft}$$

$$\theta = \cos^{-1} \left(\frac{R}{R+h} \right) = 31^\circ 47'$$

$$R = \text{Radius of earth} = 3960 \text{ miles}$$

$$h = \text{Height of orbit} = 700 \text{ miles}$$

We can rewrite equation (1) as:

$$Q_{DS} = q_S \alpha_S A_o f(\phi)$$

Where:

$f(\phi)$ is the trig part of $A(\phi)$;

$$0 \leq f(\phi) \leq 1$$

Then:

$$Q_{DS} = (443) (0.3) (1) f(\phi) = 133 f(\phi) \text{ Btu/hr.}$$

This analysis is based on the one panel which has the maximum projected area.

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(2) Thermal Earth

The heat input to one panel of the instrument compartment due to thermal radiation from the earth is given by:

$$Q_{TE} = F_{S \rightarrow E} \frac{A_{IC}}{16} \alpha_T \epsilon_E q_{bE} \quad \dots (103)$$

where:

$F_{S \rightarrow E}$ = view factor from satellite to earth

A_{IC} = Total instrument compartment surface area = 16 ft^2

α_T = absorbtivity to thermal radiation = 0.9

(Again based on using Tin Oxide as emissivity coating)

q_{bE} = black body radiation of earth = σT_E^4

T_E = earth's average temperature = 450°R

σ = Stefan-Boltzman Constant = $0.1713 \times 10^{-8} \text{ Btu/hr-ft}^2\text{-}^\circ \text{R}^4$

The view factor for a cylindrical object above the earth whose axis passes through the earth's center is calculated as follows:

$$F_{S \rightarrow E} = \frac{2B - \sin 2B}{2}$$

where:

$$B = \sin^{-1} \frac{R}{R+h} = 58^\circ 13'$$

then

$$F_{S \rightarrow E} = 0.18$$

from Equation 2 we have

$$Q_{TE} = 11.4 \text{ Btu/hr}$$

The heat input from the earth's thermal radiation is thus a constant and independent of orbit position.

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(3) Reflected Solar

The heat input due to sunlight reflected off the earth is given by:

$$Q_{RS} = q_S \alpha_S \tau_{ES} F_{S \rightarrow E} \frac{A_{IC}}{16} \overline{\cos w(\varphi)} \quad \dots (104)$$

where:

τ_{ES} = reflectivity of earth to solar radiation = 0.4

w = angle of incidence of sun's rays on the earth.¹⁹

$\cos w(\varphi)$ must be averaged over that portion of the earth's surface which is seen by the one panel of the instrument compartment. Neglecting the azimuthal variation of $\cos w(\varphi)$,

then:

$$\begin{aligned} \overline{\cos w(\varphi)} &= \frac{1}{\theta} \int_{(\varphi-\theta)}^{\varphi} \cos w \, dw \\ &= \frac{1}{\theta} \left[\sin \varphi - \sin (\varphi - \theta) \right] \quad \dots (105) \end{aligned}$$

Therefore, the heat input due to reflected sunlight will vary with orbital position because the angle of incidence of the sun's rays varies over the earth's surface.

It is assumed that as the vehicle moves behind the earth, the reflected solar input drops linearly to zero as the vehicle enters the shade.

Equation 104 is not exactly correct, since an overall view factor is only valid for thermal radiation and not reflected radiation. However, the contribution of reflected sunlight to the total heat input is so small as to make the error negligible. Equation 104 can therefore be rewritten as:

$$Q_{RS} = q_S \alpha_S \tau_{ES} F_{S \rightarrow E} \frac{A_{IC}}{16} \left[\frac{1}{\theta} g(\varphi) \right] \quad \dots (106)$$

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where:

$g(\varphi)$ is the trig part of $\cos w(\varphi)$

$$0 \geq g(\varphi) \geq 1$$

$$g(\varphi) = \begin{cases} \sin \varphi - \sin (\varphi - \theta) & , & 0 < \varphi < 90 \\ (90 + \theta - \varphi) \cos w(90) & , & 90 < \varphi < 90 + \theta \\ 0 & , & 90 + \theta < \varphi < 270 \\ (\varphi - 270) \cos w(300) & , & 270 < \varphi < 270 + \theta \\ \sin \varphi - \sin (\varphi - \theta) & , & 270 + \theta < \varphi < 360 \end{cases}$$

From equation 105

$$Q_{RS} = 17.3 g(\varphi) \text{ Btu/hr} .$$

For plots of the various heat inputs, see Figure 60.

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Figure 60. SNAP 2 External Heat Inputs

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The previous calculations have been for one panel of the instrument compartment. Figure 61 shows the two types of thermal barriers,

- 1) inverted truncated cone,
- 2) convex dome shaped

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Figure 61. Two Shapes of the Thermal Barrier

(b) Constant Sun, Constant Shade Orbit

The external heat inputs for the constant sun - constant shade orbit may be obtained directly from the foregoing results. Since the vehicle in this case will always be at an angle of 90° with the line $\phi = 0^\circ$ in Figure 59, the external heat inputs are just those constant values obtained by setting $\phi = 90^\circ$ in the previously listed expressions of the sun-shade transient orbit. Figure 61 shows the constant inputs for the hot and cold sides. They may be compared with the variable input of the sun-shade transient orbit.

(2) Summary of External Heat Inputs

1) Sun-Shade Transient Orbit:

$$Q_{(\phi)} = 133 f(\phi) + 11.4 + 17.3 g(\phi) \text{ Btu/hr}$$

2) Constant Sun-Constant Shade Orbit:

$$Q_{(\phi)} = 147 \text{ Btu/hr}$$

(3) Internal Heat Inputs

The internal heat inputs during the prestart power phase are: (1) electronic equipment and (2) radiation interchange between modules.

(4) Thermal Interchange Between Panels

Figure 62 shows the layout of the instrument compartment while Table XIX lists the estimated power dissipation of the individual modules

VIEW LOOKING FORWARD
STATION 0.0
SCALE: 1 cm = 6 in.

INSTRUMENT MODULES ARE 6" HIGH, 9" WIDE, AND 6" DEEP

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Figure 62. SNAP 2 Instrumentation Compartment

TABLE XIX
JOULEAN DISSIPATION OF MODULES

Panel Number (see Figure 63)		I^2R -PS (Prestartup) (watts)	I^2R -SS (Steady State) (watts)	Estimated Weight (lb)
1	Temperature Controller	15	15	5
2	Position Indicator	12	12	8
3	Load Controller	0	12	3
4	Hg Pressure and Flow No. 1	11	11	5
5	Nuclear Instruments	15	15	4
6	Hg Pressure and Flow No. 2	11	11	5
7	Temperature Bridge	7.2	7.2	6
8-9	Startup Programmer	4.5	13	16
10	Void	-	-	-
11-12-13	Parasitic Load Controller	0	144	18
14-15	Short Circuit Capacitors	0	30	28
16	Load Switch	5	5	7

during the prestart power phase along with the estimated component weight. To determine the effects of radiation interchange between modules, it was necessary to determine the view factor F . The view factor, F , was computed²⁰ while the overall interchange factor, $\mathcal{F}$, was computed using the results of the view factor code (Reference 17 with Reference 21). It was assumed while computing F (the overall interchange factor) that the modules are coated with PT 404, a black paint having an emissivity of 0.90 and the thermal insulation barrier is coated with gold leaf having an emissivity of 0.05.

The present SNAP 2 reference design includes the use of these coatings, and experimental tests validate these values.

While there is only one layout of the instrument compartment which is shown in Figure 62, two sets of view factors and interchange factors were

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computed. One set is for an inverted truncated cone shaped thermal barrier which limits the radiation interchange between modules while the other set is for a convex, or dome shaped, thermal barrier which in turn aids radiation interchange due to the high reflectivity of the gold emissivity coating applied to its underside.

The internal conduction between panels was determined by knowing the thermal conductivity of the 6061 T6 Aluminum Panels and considering joint conductance between panels. In adding up the sum of the internal heat inputs for this analysis, the SNAP 2-Agena Interface was assumed to be adiabatic (neither a source nor a sink).

(5) Radiation Losses to Space

To determine the radiation losses to space it was necessary to compute the fin efficiencies of both the aluminum panels and the mag-thoria load ring located between station 0.0 in. and station 4.0 in. The maximum temperature drop (ΔT) across the mag-thoria considering joint conductance between the aluminum panel and the mag-thoria load ring was 10°F and occurs on the cold side of a constant sun-constant shade orbit. This calculation was made with Reference 22 and checked with Reference 9. The calculation for determining the fin effectiveness of the aluminum panel using Reference 9 gave a fin efficiency of 95%.

This report includes the results in curve form of the aerodynamic heating study done earlier. These curves are included to enable one to choose an emissivity coating for the external surface of the aluminum and mag-thoria panel. Tin-oxide is chosen on this basis (it is stable at the maximum "aerodynamic heating" temperature) and other coatings already tested have shown very significant degradation due to ultraviolet wavelengths. Figure 63 is a plot of the heat input rate vs flight time curves due to aerodynamic heating for various wall temperatures, considering radiation losses to space. These curves were generated by using Reference 23. Figure 64 is a plot of the maximum aerodynamic heating temperature vs flight time for the instrument compartment panel surface, considering radiation losses to space and was generated by using Reference 23.

By adding the external and internal heat loads and equating them to radiation losses to space, using Reference 22 the panel temperatures were

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Figure 63. Heat Input Rate vs Flight Time

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Figure 64. Maximum Temperature vs Flight Time
Due to Aerodynamic Heating

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determined while the instruments were operating at their prestart power phase.

b. Results

The minimum panel temperature occurred when panels 8 through 15 were subjected to full shade in a constant sun, constant shade orbit. Figure 65 gives two temperature distributions of this case (panels 8 through 15 subjected to full shade).

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Figure 65. Prestart Temperature Distribution for Constant Sun, Constant Shade Orbit

One temperature distribution (the underlined numbers) corresponds to using the dome shaped thermal barrier while the other set of numbers corresponds to using the inverted truncated cone shaped thermal barrier.

One can see from Figure 65 that the effects of using a convex dome as the shape of the thermal barrier greatly help in raising the temperatures of the "cold" panels. With a dome shaped thermal barrier and the minimum allowable temperature being -65°F , no electrical heaters will be needed.

Figure 66 gives two temperature distributions when panels 8 through 15 are in direct sun; the instrument compartment has been rotated 180° with respect to Figure 65.

Table XX shows the transient temperature fluctuations for the prestart power phase based on using the dome shaped thermal barrier.

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Figure 66. Prestart Temperature Distribution for Constant Sun, Constant Shade Orbit

2. Post Startup Analysis

The Steady-State and transient temperature fluctuations of the SNAP 2 Instrument Compartment at full power phase were computed by summing the external and internal heat loads and equating them to radiation losses to space. As in the prestart analysis, internal conduction between panels, radiation interchange between modules, and an assumed adiabatic SNAP 2-Agena interface were included in this analysis.

TABLE XX

TRANSIENT FLUCTUATIONS FOR PRESTART POWER PHASE
DOME SHAPED BARRIER

Module Number	Minimum Temperature (°F)	Maximum Temperature (°F)
1	-19	91
2	-13	98
3	1	111
4	11	118
5	-2	107
6	-13	98
7	-28	86
8-9	-41	71
11-12-13	-50	62
14-15	-42	69
16	-29	79

a. Method

(1) External Heat Inputs

Same as Prestart Analysis.

(2) Internal Heat Inputs

The internal heat inputs during full power phase are as follows:

- a) I^2R loss at full power phase (Jouleian Dissipation)
- b) Conduction and radiation from electrical wires.
- c) Conduction from mercury subcooler through 91 LD Phenolic Resin.
- d) Radiation from the thermal barrier proper as well as that due to bolts and other fasteners.
- e) Radiation interchange between modules
- f) Internal conduction between panels.

Examining each internal heat input individually, - (a) I^2R loss at full power phase - Table XIX lists the estimated power dissipation of the individual modules during the full power phase along with component estimated weight, (b) Conduction and radiation from electrical wires - an approximate, analytical method was employed to calculate the SNAP 2 instrument compartment wire temperature gradients and the heat conducted and radiated into the instrument compartment from the same wires.

Figure 67 shows a typical insulation covered wire connected at one end to an instrument module and with the other end connected to equipment located above the thermal barrier.

T_0	- TEMPERATURE AT CONNECTION ABOVE THERMAL BARRIER
T_e	- TEMPERATURE AT THERMAL BARRIER
T_L	- TEMPERATURE AT CONNECTION TO INSTRUMENT MODULE (BELOW THERMAL BARRIER)
Q_e	- HEAT FLUX AT OR ACROSS THERMAL BARRIER
Q_L	- HEAT FLUX CONDUCTED INTO INSTRUMENT MODULE
e	- LENGTH OF INSULATION-COVERED WIRE ABOVE THERMAL BARRIER
L	- TOTAL LENGTH OF INSULATION-COVERED WIRE
REGION 1	- REFERS TO THAT PART LOCATED ABOVE THERMAL BARRIER
REGION 2	- REFERS TO THAT PART LOCATED BELOW THERMAL BARRIER
T_S	- AMBIENT TEMPERATURE

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Figure 67. Sketch of Insulation Covered Wire

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The generalized equation for either region one or region two is as follows:

$$\frac{d^2T}{dx^2} - \frac{UC}{KA} (T - T_s) = 0$$

where T_s is the ambient temperature corresponding to either region one or region two.

In making the preceding equation applicable for radiation, the UC term must contain an equivalent convection coefficient for radiation, h_r as follows:

$$\frac{1}{UC} = \frac{\ln(R_2/R_1)}{2\pi KI} + \frac{1}{2\pi R_2 h_r}$$

To determine the value of the temperature (T_r) used to calculate h_r , a conservative temperature distribution was assumed, the average temperature was then determined by integrating the fourth power of this distribution, thereby giving a conservative value of h_r .

Solution for region (1): ($0 \leq x \leq e$)

$$\frac{d^2T}{dx^2} - \left(\frac{UC}{KA} \right) (T - T_{s1}) = 0$$

Therefore

$$T_{1(x)} - T_{s1} = A_1 \sinh \alpha_1 x + B_1 \cosh \alpha_1 x$$

where

$$\alpha_1 = \sqrt{UC/KA}$$

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Boundary conditions:

$$(1) \text{ At } x = 0, \quad T = T_o$$

$$(2) \text{ At } x = e, \quad Q_{e(1)} = Q_{e(2)}, \quad T_{e(1)} = T_{e(2)}$$

From boundary conditions above:

$$B_1 = T_o - T_{s_1} \quad \dots (107)$$

$$T_e - T_{s_1} = A_1 \sinh \alpha_1 e + B_1 \cosh \alpha_1 e \quad \dots (108)$$

$$Q_e = KA \frac{dt_1}{dx} = KA \alpha_1 \left(A_1 \cosh \alpha_1 e + B_1 \sinh \alpha_1 e \right) \quad \dots (109)$$

Solution for region (2): ($e \geq x \geq 1$)

$$\frac{d^2 t}{dx^2} - \frac{U_2 C}{KA} (T_2 - T_{s_2}) = 0$$

Therefore

$$T_{2(x)} - T_{s_2} = A_s \sinh \alpha_2 x + B_2 \cosh \alpha_2 x$$

where

$$\alpha_2 = \sqrt{U_2 C / KA}$$

Boundary conditions

$$(1) \text{ At } x = e, \quad Q_{e_1} = Q_{e_2}, \quad T_{e_1} = T_{e_2}$$

$$(2) \text{ At } x = 1, \quad T = T_1$$

From boundary conditions above:

$$T_e - T_{s_2} = A_2 \sinh \alpha_2 e + B_2 \cosh \alpha_2 e \quad \dots (110)$$

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$$T_1 - T_{s_2} = A_2 \sinh \alpha_2 L + B_2 \cosh \alpha_2 L \quad \dots(111)$$

$$Q_e = KA \frac{dt_2}{dx} = KA \alpha_2 A_2 \cosh \alpha_2 e + B_2 \sinh \alpha_2 e \quad \dots(112)$$

Equation 107 gives the solution for constant B_1 , Equations 108 and 109 when simultaneously solved give constant A_1 in terms of T_e and Q_e . The constants A_2 and B_2 can be evaluated by solving simultaneously Equations 108 and 110 and 109 and 112. Equation 111 (with the above values of A_2 and B_2) gives the solution for T_L as a function of L .

To determine the length of wire below the thermal barrier at which a transition to low temperature wire can occur, $T_L = 250^\circ\text{F}$ is substituted in Equation 110 and L is solved. The specific wire type used in this calculation was 16 gauge 4 conductor TW, which has the lowest surface area to copper cross-sectional area ratio. After performing the calculation, it was found that the transition from high temperature wire to low temperature wire can occur at a minimum distance of 8 in. below the thermal barrier. Similarly, upon substitution of $T_L = 150^\circ\text{F}$ in Equation 111 and solving for L , the minimum length of wire, at which the heat flux conducted into the instrument module is negligible is sixteen in. As before, the specific type of wire used in this calculation was 16 gauge 4 conductor TW, which has the lowest surface area to copper cross-sectional area ratio.

In the solutions previously mentioned, the $\alpha_2 L$ product was greater than 3.0 and for a αL product of greater than 3.0, the following equation is applicable:

$$Q_e = \alpha_2 KA \left(T_e - T_{s_2} \right)$$

Solving the above, Q_e for one wire (16 gauge 4 conductor TW) equals 23.0 Btu/hr.

The total heat conducted and radiated from the electrical wires is 260 watts. Since nearly all the wires below the thermal barrier are greater

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than 16 in. in length, the heat radiated from these wires can be equally divided by 16 to get the internal heat load per panel. Expressed in Btu per panel per hr this heat load is 55.4 Btu/hr panel

(3) Conduction from Mercury Subcooler

The heat conducted from the mercury subcooler will be entering the instrument compartment through the 91 LD Phenolic Resin. The Phenolic Resin in question has a cross sectional area of 0.354 sq ft and a thermal conductivity of 0.2 Btu/hr ft °F. Expressed in Btu per panel per hr this heat load is 6.4.

(4) Radiation by the Thermal Barrier Including Bolts, Fasteners, etc.

The present design calls for anodized aluminum fasteners to be used in holding the thermal barrier together and to its supporting structure.

Due to the high thermal conductivity and high emissivity of anodized aluminum, the heat leak into the instrument compartment will be 153.0 Btu/hr. By replacing the anodized aluminum fasteners with 91 LD Phenolic Resin fasteners and coating the phenolic resin fasteners with gold leaf ($\epsilon = 0.05$), the heat leak is reduced to 8.2 Btu/hr or approximately 0.5 Btu/hr panel. This calculation is based on using a dome shaped thermal barrier supported by a sheet of mag-thoria. In the case of the truncated cone shaped thermal barrier, the heat leak (using gold leaf coated 91 LD Phenolic Resin) is further reduced to 6.4 Btu/hr or 0.4 Btu/hr panel.

Radiation interchange between modules was calculated in the same manner as the Prestart Analysis VI(1). Internal Conduction between panels was calculated in the same manner as the Prestart Analysis VI(1). As in the Prestart Analysis, the SNAP 2 Agena Interface was assumed to be adiabatic.

Radiation losses to space VI(3) were calculated in the same manner as the Prestart Analysis VI(1) with the following exceptions:

- a) I^2R is evaluated at full power
- b) The additional internal heat inputs are included.

b. Results

The maximum panel temperature occurred when panels 8 through 15 were subjected to full sun in a constant sun, constant shade orbit. Figure 68

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Figure 68. Full Power Phase
Temperature Distribution
for Constant Sun,
Constant Shade
Orbit

gives two temperature distributions of this case. One temperature distribution (the underlined numbers) corresponds to using the dome shaped thermal barrier while the other set of numbers corresponds to using the inverted truncated cone shaped thermal barrier. One can see from Figure 68 that using a convex dome as the shape of the thermal barrier helps in lowering the temperature of the "hot" panels.

Figure 69 gives two temperature distributions when panels 8 through 15 are in full shade; the instrument compartment has been rotated 180° with respect to Figure 68.

Table XX shows the transient temperature fluctuations for the full power phase based on using the dome shaped thermal barrier. The present SNAP 2 unit reference design is the dome shaped thermal barrier. Since

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UNDERLINED NUMBERS ARE IN °F AND CORRESPOND TO
DOME SHAPED THERMAL BARRIER
NUMBERS BELOW UNDERLINED NUMBERS ARE IN °F
AND CORRESPOND TO TRUNCATED CONE SHAPED
THERMAL BARRIER

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Figure 69. Full Power Phase
Temperature Distribution
for Constant Sun,
Constant Shade
Orbit

TABLE XXI

TRANSIENT FLUCTUATIONS FOR FULL POWER PHASE,
DOME SHAPED THERMAL BARRIER

Panel	Temperature (°F)	Temperature (°F)
1	78	176
2	70	174
3	57	172
4	40	166
5	59	170
6	71	172
7	84	176
8-9	95	179
11-12-13	108	183
14-15	100	182
16	89	179

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sufficient testing has not been completed regarding degradation of the Tin Oxide emissivity coating due to ultraviolet radiation at low temperatures, a calculation was made using an absorptivity of 0.5 to determine the influence of possible degradation. This temperature distribution is shown in Figure 70. Comparing Figure 70 with Figure 68 the increase in absorptivity raised the temperature of module 11-12-13 (Parasitic Load Controller) from 183 to 225°F.

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Figure 70. Full Power Phase Temperature Distribution for Constant Sun, Constant Shade Orbit - ($\alpha_s = 0.5$)

The internal heat load will be minimized by using 91 LD Phenolic Resin fasteners (low thermal conductivity) emissivity coated with gold leaf ($\epsilon = .05$) to secure the thermal barrier. The heat leak using the above fasteners will be 8.2 Btu/hr as opposed to 153.0 Btu/hr Anodized Aluminum type fasteners.

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VII. HYDROGEN DIFFUSION

A. GENERAL

Operation of the SNAP 2 unit will result in the addition of hydrogen to the reactor coolant. The source of the hydrogen is the zirconium hydride moderator within the fuel elements. At the temperatures existing within the fuel during reactor operation, the hydride substantially dissociates into zirconium and molecular hydrogen. A significant hydrogen pressure develops within the fuel and the result is diffusion of hydrogen through the fuel element cladding into the coolant. A diffusion barrier applied between the fuel and the cladding holds the hydrogen loss rate to a level which is acceptable from the standpoint of reactivity loss over a one-year period. The hydrogen entering the NaK is subsequently lost to space either by diffusing directly through the primary system pipe walls or by first diffusing into the mercury through the boiler then to space through the secondary system pipe walls.

In the following sections, an analysis is developed to determine for the presently designed SNAP 2 system:

- 1) The steady-state concentration distribution of hydrogen both in the NaK and the mercury.
- 2) The effect of hydrogen on the system performance together with an indication of potential problem areas.
- 3) The hydrogen concentration versus time relationship in both the NaK and the mercury during the approach to steady-state.

The analysis is later extended to evaluate the effect of a number of possible system modifications. The modifications have the objective of either minimizing the quantity of hydrogen entering the mercury power conversion system or facilitating the rejection to space of the hydrogen that does enter the PCS. Finally, a program is outlined that will, by the application of analysis and experiment, ensure that the effects of hydrogen on the SNAP 2 unit performance will be reduced to an acceptable level within the present SNAP 2 schedule.

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B. ANALYSIS

1. General

The migration of the hydrogen from the reactor coolant to space and to the mercury system is dependent upon the relations governing the diffusion of a gas through a porous medium. A diatomic gas such as hydrogen is believed to diffuse through metallic media in the atomic state. Consequently, the rate of diffusion is proportional to the difference in the square roots of the hydrogen pressures on either side of the barrier. The appropriate relationship in differential form is:

$$dq = \frac{S}{d} \left[P_1^{1/2} - P_2^{1/2} \right] dA \quad \dots (113)$$

where

dq = differential diffusion rate, $\text{cm}^3(\text{STP})/\text{hr}$

S = permeability of the barrier, $\text{cm}^3(\text{STP})\text{mm}/\text{hr}\text{-cm}^2\text{-atm}^{1/2}$

dA = differential superficial diffusion area, cm^2

d = wall thickness, mm

P_1, P_2 = inner and outer hydrogen pressures, respectively, atm

The permeability S is a function of wall material and temperature and involves dissociation and absorption of the gas at the wall, the diffusion of the atomic species, and the recombination of the atoms and desorption at the outside surface. Investigators have determined values for S for different metals. Figure 71 displays the available data for the materials mentioned in this section.

Before proceeding further with the analysis, it should be mentioned that the above relation and associated hydrogen permeability data are to be used under conditions where their applicability may be questionable. As discussed,²⁴ the square root of pressure relation may not hold at the very low hydrogen pressures which will be found to exist in the SNAP 2 system. Furthermore, permeability data is normally obtained in experiments at one atmosphere hydrogen pressure and their application at very low hydrogen pressures represents a substantial extrapolation of this data. The validity of hydrogen permeation

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Figure 71. Comparative Permeabilities of Hydrogen Through Various Metals

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obtained with a pure free gas when applied to a situation where the gas is in a solution with a liquid metal, may also be suspect. Finally, the corrosive effect of liquid metals on the system walls may influence the effective permeability of the piping for hydrogen.

2. NaK System and Boiler

Hydrogen enters the NaK coolant from the reactor at a steady-state rate of $Q \text{ cm}^3(\text{STP})/\text{hr}$ and leaves the system by diffusing directly to space through the system piping at $q_{ns} \text{ cm}^3(\text{STP})/\text{hr}$ or by diffusing into the mercury PCS through the boiler tube walls at $q_{nm} \text{ cm}^3(\text{STP})/\text{hr}$. At steady-state

$$q_{ns} + q_{nm} = Q , \quad \dots(114)$$

but

$$q_{ns} = \int dq_{ns} = \int \frac{S}{d} \left[P_n^{1/2} - P_s^{1/2} \right] dA , \quad \dots(115)$$

and

$$q_{nm} = \int dq_{nm} = \int \frac{S}{d} \left[P_n^{1/2} - P_m^{1/2} \right] dA , \quad \dots(116)$$

where P_n refers to the hydrogen pressure in the NaK, P_s refers to the hydrogen pressure in the surroundings (space), and P_m refers to the hydrogen pressure in the mercury. Equation 115 includes the entire NaK system except the boiler tubes and Equation 116 applies to the boiler tubes alone.

The hydrogen in the NaK is in solution and for unsaturated solutions exerts a dissociation pressure given by the relation

$$P_n^{1/2} = k w_n , \quad \dots(117)$$

where P_n is the hydrogen pressure, w_n is the hydrogen weight fraction and k is the equilibrium constant. Figure 72, taken from Reference 24 shows the dependence of k on temperature. The low rate of hydrogen input coupled with the relatively high NaK flow rate leads to the assumption that the weight fraction of hydrogen in the NaK will be essentially uniform throughout the NaK system. Using the maximum specified reactor hydrogen leak rate² and the design NaK flow rate, the incremental w_n/cycle is in the part-per-billion range.

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Figure 72. Equilibrium Constant For
Hydrogen-NaK 78 System

Since it is later shown that the steady state w_n is in the ppm range, the preceding assumption is justified. If it is further assumed that the hydrogen pressure in the surroundings is zero, Equation 115 becomes

$$q_{ns} = w_n \sum \left(S \frac{A}{d} k \right)_{ns} , \quad \dots (118)$$

where a summation of the product of permeability, diffusion area, equilibrium constant and reciprocal wall thickness is taken over regions of uniform surface area and wall thickness. Although S and k are functions of temperature, appropriate average values may be used for each incremental A/d .

A further simplification of the analysis results if it is assumed that point for point in the boiler P_m is sufficiently smaller than P_n that P_m may conveniently be taken as zero in the calculation of q_{nm} . This assumption is shown to be valid in Section VII(C) Equation 114 provides a conservative estimate of the amount of hydrogen diffusing into the mercury. Under these conditions, Equation 116 reduces to

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$$q_{nm} = w_n \sum \left[S \frac{A}{d} k \right]_{nm} . \quad \dots (119)$$

The combination of Equation 118 and 119 with Equation 114 leads directly to the following solution for w_n

$$w_n = \frac{Q}{\sum \left[S \frac{A}{d} k \right]_{nm} + \sum \left[S \frac{A}{d} k \right]_{ns}} . \quad \dots (120)$$

Having obtained w_n , q_{ns} and q_{nm} are directly calculable as are the hydrogen pressures at any point in the NaK system given the temperature. Assuming ideal gas behavior and no accumulation of hydrogen in the boiler, the hydrogen weight fraction in the mercury and the partial pressure at the boiler exit can be calculated from q_{nm} , the mercury flow rate M_m , and the total pressure at the boiler exit P_t .

$$w_m = a \frac{q_{nm}}{M_m} , \quad \dots (121)$$

where M_m is in gm/hr and "a" is a conversion factor equal to $8.93 \times 10^{-5} \text{ gH}_2/\text{cm}^3 (\text{STP})\text{H}_2$.

$$P_m = w_m \frac{MW_{\text{Hg}}}{MW_{\text{H}_2}} P_t , \quad \dots (122)$$

where MW_{Hg} and MW_{H_2} are molecular weights of mercury and hydrogen, respectively.

3. Mercury System

The steady state analysis of the distribution of the hydrogen in the mercury vapor system has been treated in considerable detail.²⁵ The analysis assumes the hydrogen is confined to the vapor region of the mercury system, i. e., no hydrogen leaves the radiator-condenser with the liquid mercury. Consequently, at steady-state, all the hydrogen entering through the boiler is forced to diffuse to space between the boiler outlet and the RC liquid interface. In addition, the hydrogen is assumed to move through the system at the same velocity as the mercury vapor.

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For hydrogen flowing in mercury vapor through a uniform conduit of internal diameter D_i cm, wall thickness d mm and permeability S cm³(STP)mm/hr-cm²-atm^{1/2} the hydrogen pressure P_m at any point x cm from the inlet is given by

$$P_m^{1/2} = P_{mo}^{1/2} - \frac{a\pi MW_{Hg} D_i S P_t}{2MW_{H_2} M_m d} x \quad \dots (123)$$

where "a," MW_{Hg} , MW_{H_2} , M_m and P_t are as previously defined, and P_{mo} is the hydrogen pressure at $x = 0$. Equation 123 is valid for small changes in P_m with constant P_t and negligible resistance for hydrogen migration to the pipe wall. The case of a highly permeable membrane with mass transfer of hydrogen from the bulk flow to the interface (wall) considered is treated in Section VII B-5.

Using Equation 123 and continuity considerations, the hydrogen pressure at the inlet to the RC was determined. It was found that hydrogen loss from the boiler to the RC was negligible. Since, the mercury vapor flow down the RC tubes goes to zero at the liquid mercury interface, it was anticipated that hydrogen would tend to build up in this region. Consequently, back diffusion of hydrogen in the RC was considered. To determine the hydrogen pressure along an RC tube, the tube was divided into 10 nodes as shown

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where

$$g_i = \frac{P_i A_i V_i}{R_{H_2} T} , \quad \dots (124)$$

and

$$Q_i = \left(\frac{S A}{d} \right)_i P_i^{1/2} . \quad \dots (125)$$

In Equation 124

g = mass flow of hydrogen, g/hr

A = cross-sectional area for bulk flow, cm^2

V = mercury vapor velocity, cm/hr

T = absolute temperature, $^{\circ}\text{K}$

R_{H_2} = specific gas constant for hydrogen, R/MW_{H_2}
equal to $41.03 \text{ cm}^3 \text{ atm/g } ^{\circ}\text{K}$

P = hydrogen pressure, atm

In Equation 125, it should be remembered that A refers to the area for diffusion through the tube wall. The back diffusion term is given by³

$$N_A = \frac{D_{AB} A P_t}{R T P_B} \left(\frac{\Delta P_A}{\Delta X} \right) , \quad \dots (126)$$

for steady state diffusion of hydrogen through non-diffusing mercury vapor. In Equation 126

A = molar transfer rate of A (hydrogen) q moles/hr

D_{AB} = binary diffusivity for system A-B, cm^2/hr

R = universal gas constant, $82.06 \text{ cm}^3 \text{ atm/g mole } ^{\circ}\text{K}$

P_B = log mean pressure of B (mercury vapor), atm

P_t = system pressure, atm

$\Delta P_A / \Delta X$ = hydrogen pressure gradient per node, atm/cm

A = cross-sectional area for back diffusion, cm^2

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The diffusivity of hydrogen in mercury vapor may be estimated using an equation taken from Reference 4

$$D_{AB} = 6.69 \frac{\left[T^3 \left(\frac{1}{MW_A} + \frac{1}{MW_B} \right) \right]^{1/2}}{P_t \sigma_{AB}^2 \Omega_{D,AB}} \text{ cm}^2/\text{hr} \quad \dots(127)$$

in which

σ_{AB} = collision diameter, Angström units

$\Omega_{D, AB}$ = a dimensionless function of the temperature and the intermolecular potential field for one molecule of A and one of B.

and T , MW_A , MW_B , and P_t have been previously defined.

Calculations were made using the above ten node model by starting with node 1 and the known value for P_o . Performing a mass flow balance on node i in conjunction with P_{i-1} enables P_i to be determined. However, when the last node is reached and the mass flow into the node is set equal to the diffusion through the walls plus the back diffusion the hydrogen pressure required at the interface becomes very large. The net result of these calculations was to show that the bulk of the hydrogen does collect at the interface. The limiting value for the hydrogen pressure is the 6.1 psia at the RC outlet.

Further calculations were made starting at the interface end of the tube and subdividing node 10 into ten additional nodes. In this calculation, the hydrogen pressure at the interface was estimated at 5.1 psia, corresponding to a 1 psi Hg partial pressure at an interface temperature of 460°F. These calculations were made knowing P_i and solving for P_{i-1} until the values obtained intersected those found in the previous calculations. A plot of hydrogen pressure versus distance down an RC tube found in this manner is shown in Figure 73. The peak hydrogen pressure occurs where the velocity goes to zero. The assumption of hydrogen velocity equal to the velocity of the mercury vapor forces the point of zero hydrogen velocity to be at the liquid mercury interface. Under these conditions, only approximately 15% of the hydrogen entering the

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Figure 73. Hydrogen Pressure in SNAP 2 Radiator-Condenser

RC escapes through the wall. Thus in order for the total hydrogen inflow to escape, there must be a region, approximately 7.3 in. in length ahead of the interface in which the hydrogen pressure is in the vicinity of 5 psia.

The preceding analysis assumes a temperature drop of approximately 130°F through the 7.3 in. of hydrogen-filled tube. This is approximately the same drop that would have been obtained through about 6 in. of liquid mercury subcooler. Since subcooling in the hydrogen is not as effective as that in a mercury-filled tube, the temperature at the liquid mercury interface should be greater than previously assumed corresponding to a greater mercury partial pressure and, consequently, a slightly lower hydrogen pressure.

For this reason, a thermal analysis was performed to determine how much temperature drop could be expected for various lengths of hydrogen-filled tube. It was assumed that condensing has stopped before the mercury enters the hydrogen region. Thus the mercury temperature at the inlet to the hydrogen region was fixed at 590°F. The tube wall temperature was found by matching

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heat flux from the wall by radiation. Using this temperature in conjunction with the temperature profile in the subcooler, the wall temperature drop over various lengths of hydrogen-filled tube was calculated.

Next the length of the tube required to allow the hydrogen input rate to escape was calculated for various hydrogen pressures. The mercury pressure at the liquid interface is known for each hydrogen pressure since the total pressure must equal 6.1 psia. Knowing the mercury vapor pressure, the temperature may be found and the Δt across a given length of hydrogen-filled tube was determined in this manner.

A plot of Δt versus Δx for each of these methods was made and is shown in Figure 74. The point of intersection of the two curves gives the solution for Δt and Δx . Knowing the required diffusion rate of hydrogen and the length, the hydrogen pressure at the interface was calculated to be 4.8 psia. Thus 7.5 in. of condenser tube ahead of the interface contain hydrogen at 4.8 psia.

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Figure 74. Determination of Hydrogen Lengths Required to Satisfy Operating Conditions

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4. Approach to Equilibrium

It is of interest to know how rapidly the NaK system approaches steady-state. A hydrogen balance in differential form consistent with the assumptions in Section VII B-2 yields

$$\frac{dw_n}{d\theta} = \frac{a}{G} [Q - w_n (e + f)] \quad \dots(128)$$

where

θ = time, hr

G = NaK inventory, grams

$e = \Sigma \left(S \frac{A}{d} k \right)_{ns}$, $\text{cm}^3(\text{STP})/\text{hr}$

$f = \Sigma \left(S \frac{A}{d} k \right)_{nm}$, $\text{cm}^3(\text{STP})/\text{hr}$

Integration of this equation with a boundary condition of $w_n = 0$ at $\theta = 0$ leads to

$$w_n = \frac{Q}{e + f} \left[1 - e^{-\frac{a}{G} (e + f) \theta} \right] \quad \dots(129)$$

or, in terms of the ratio of w_n to the steady-state hydrogen weight fraction $w_{n\infty}$

$$\frac{w_n}{w_{n\infty}} = 1 - e^{-\frac{a}{G} (e + f) \theta} \quad \dots(130)$$

The analysis of the rate of hydrogen buildup in the mercury system is not so straightforward as that for the NaK system. However, with certain simplifying assumptions a reasonable estimate of the time behavior of the hydrogen concentration in the mercury vapor can be obtained. While recognizing that the hydrogen pressure within the mercury vapor is in fact a variable, it is assumed that some effective average value may be used in Equation 113. With $P_s = 0$, one may then write for the diffusion of hydrogen from the mercury system

$$q_{ms} = \Sigma (S \frac{A}{d})_{ms} P_m^{1/2} \quad \dots(131)$$

However, for a vapor mixture of hydrogen in mercury one may also write

$$P_m = w_m \frac{MW_{Hg}}{MW_{H_2}} P_t \quad \dots(132)$$

where w_m is the weight fraction of hydrogen in the mercury, P_t is the total pressure and MW_{Hg} and MW_{H_2} refer to the molecular weights of mercury and hydrogen, respectively. Here it is implied that a suitable average system pressure is known. From the two equations above, a differential mass balance for hydrogen takes the form

$$\frac{dw_m}{d\theta} = \frac{a}{H} \left[q_{nm} - \Sigma (S \frac{A}{d})_{ms} \left(\frac{MW_{Hg}}{MW_{H_2}} P_t \right)^{1/2} w_m^{1/2} \right] \quad \dots(133)$$

where H is the mass of mercury vapor and "a" is the previously defined conversion factor. The integration of this equation is considerably simplified if q_{nm} is taken as constant at, for convenience, the steady-state value. Under these conditions, the time-concentration relation will actually only apply for the case of an initially hydrogen-free mercury system instantaneously exposed to a constant hydrogen source. The result will be of value in that the time constant obtained under these conditions may be added to that obtained in the transient analysis of the NaK system to obtain a conservative estimate of the behavior of the mercury system. Actually, the time constant for the mercury system will be found to be small compared with that for the NaK system and little error is introduced. Taking q_{nm} constant and $w_m = 0$ at $\theta = 0$, integration of Equation 133 yields

$$w_m^{1/2} = w_{m\infty} \left\{ 1 - e^{-w_m^{1/2} \left[\frac{a q_{nm\infty}}{2 H w_{m\infty}^{1/2}} \theta - w_m^{1/2} \right]} \right\} \quad \dots(134)$$

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where

$$w_{m\infty}^{1/2} = \frac{q_{nm\infty}}{\Sigma \left(S \frac{A}{d} \right)_{ms} \left(\frac{MW_{Hg}}{MW_{H_2}} P_t \right)^{1/2}} \quad \dots(135)$$

In terms of the ratio of w_m to $w_{m\infty}$

$$\left(\frac{w_m}{w_{m\infty}} \right)^{1/2} = 1 - e^{- \left[\frac{aq_{nm\infty}}{2H w_{m\infty}} \theta - \left(\frac{w_m}{w_{m\infty}} \right)^{1/2} \right]} \quad \dots(136)$$

The evaluation of Equation 130 and Equation 136 is considered in Section C.

5. Hydrogen Diffusion Through a Highly Permeable Membrane

The use of hydrogen "windows" in the NaK and mercury systems to facilitate the diffusion of the hydrogen to space has been proposed. Factoring the presence of a highly permeable material into the steady-state calculations for the NaK system is quite simple. Since the hydrogen in the NaK continuously recirculates, all that is required is to modify the $\Sigma (S \frac{A}{d} k)$ term to reflect the proportion of the system having the large permeability. In the case of the mercury vapor system where once-through hydrogen flow is postulated, it is necessary to include mass transfer of the hydrogen from the mercury vapor to the pipe wall to determine the overall diffusion rate. The analysis is complicated by the fact that within the mercury, the transport of the hydrogen to the wall is directly proportional to the hydrogen pressure difference whereas a square root pressure difference governs flow through the wall. The treatment of hydrogen diffusion from a conduit where both mass transfer to the wall and diffusion through the wall are important has been developed³⁰ and leads to the following equation for the distance downstream from the entrance to reach any ratio of m to m_0

$$x = g \left\{ \left(1 + h m_o \right)^{1/2} - \left[1 + h m_o \left(\frac{m}{m_o} \right) \right]^{1/2} \right. \\ \left. + \ln \frac{1 - (1 + h m_o)^{1/2}}{1 - \left[1 + h m_o \left(\frac{m}{m_o} \right) \right]^{1/2}} \right\} \quad \dots(137)$$

where

x = distance from entrance, cm

m = molar flow rate of hydrogen, g mole/hr at x

m_o = initial molar flow rate of hydrogen, g mole/hr

$$g = \frac{Mn}{\pi D_i k_x MW_{Hg}} \quad \text{cm} \quad \dots(138)$$

$$h = \frac{MW_{Hg}}{MmP_t} \left(\frac{2k_x b d}{S} \right)^2, \quad \text{dimensionless} \quad \dots(139)$$

b = conversion factor, $22.4 \times 10^3 \text{ cm}^3 \text{ (STP) } H_2 / \text{gmole } H_2$

k_x = average mass transfer coefficient, $\text{gmole}/\text{cm}^2 \text{ hr}$

and the other quantities have been previously defined. The average mass transfer coefficient k_x is estimated through a conventional heat-mass transfer analogy for turbulent flow in a circular conduit³¹

$$Nu_{AB} = 0.023 Re^{0.8} Sc^{0.3} \quad \dots(140)$$

where

$$Nu_{AB} = \frac{k_x Di}{c D_{AB}} \quad \dots(141)$$

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$$\begin{aligned}
 c &= \text{total molar concentration, moles/cm}^3 \\
 D_{AB} &= \text{binary diffusivity for system A-B cm}^2/\text{hr} \\
 \text{Re} &= \frac{\rho D_i V}{\mu} \quad \dots(142)
 \end{aligned}$$

ρ = bulk density, gm/cm³

V = bulk velocity, cm/sec

μ = bulk viscosity, g/cm sec

$$\text{Sc} = \frac{\mu}{\rho D_{AB}} \quad \dots(143)$$

The diffusivity of hydrogen in mercury vapor can be estimated as described in VII (B-2) using Equation 127.

C. RESULTS FOR THE PRESENT SYSTEM

1. Steady-State

Using the NaK system and boiler parameters shown in Figure 75 together with the maximum specified hydrogen leak rate from the reactor, the steady-state hydrogen weight concentration in the NaK system is found to be 9.2 ppm. Under the postulated model 69% of the hydrogen diffuses directly to space from the NaK and 31% enters the mercury system through the boiler. The hydrogen concentration in the mercury vapor at the boiler exit is 1.3 ppb corresponding to a partial pressure of 1.0×10^{-6} atm. The hydrogen pressure in the NaK at the boiler NaK inlet is 8.4×10^{-5} atm using an equilibrium constant of $1000 \text{ atm}^{1/2}$ at 1195°F. Thus the hydrogen pressure in the NaK is a factor of 84 greater than that in the mercury at the corresponding point, substantiating the assumption made earlier to neglect P_m in calculating q_{nm} .

By summing the pressure drops due to diffusion, the CRU turbine, and the various pipe configuration changes, the hydrogen pressure at the inlet to an RC tube was found to be 8.9×10^{-8} atm. As shown in VII(B-2) the pressure rises gradually down the tube to a value of about 7×10^{-6} atm at a distance of 7.5 in. ahead of the liquid mercury interface. At this point, the pressure jumps abruptly to approximately 0.33 atm (4.8 psia) and remains at this level. A plot of hydrogen pressure versus position in the RC appears in Figure 73. In the analysis by

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NaK PIPING, REACTOR HOUSING AND BOILER SHELL FABRICATED FROM TYPE 316 STAINLESS STEEL.

BOILER TUBES FABRICATED FROM HAYNES 25 ALLOY

$$\sum \left(s \frac{A}{d} k \right)_{NS} = 1.939 \times 10^6 \text{ cm}^3 (\text{STP}) / \text{hr}$$

$$\sum \left(s \frac{A}{d} k \right)_{NM} = 8.55 \times 10^5 \text{ cm}^3 (\text{STP}) / \text{hr}$$

NaK INVENTORY = 1.65×10^4 grams (36.3 lb)

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Figure 75. NaK System and Boiler Parameters

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which the temperature drop through the 4.8 psia hydrogen region was determined, it was assumed that the temperature at this region was the same as the interface temperature with zero hydrogen present. This means that the interface is displaced an amount equal to the length of the high pressure hydrogen region. Taking into account the temperature drop through the hydrogen region and the loss in subcooler length, a net increase in RC exit temperature of approximately 50° F should result from the pressure distribution found above. The preceding results are very conservative in view of the recent experimental results which indicate that the hydrogen leak rates are only about 50% of those assumed in this analysis.

It should be noted, that in the above analysis the back-diffusion of hydrogen was evaluated using a diffusion equation for one component diffusing through a second stagnant component. In this approach, the back-diffusion utilizes the entire cross-section of the tube equally. It seems likely that in reality a large part of the back-diffusion will occur in the low velocity region of the mercury vapor boundary layer. The effect of an increase in back-diffusion along the tube walls would be to more evenly distribute the hydrogen along the length of the tube and to place a layer of hydrogen adjacent to the wall.

The distribution of the hydrogen into the condensing region will bring about a net reduction in the overall heat transfer coefficient. This reduction is brought about by the requirement that the condensing vapor diffuse through the layer of noncondensibles on the condensing heat transfer surface. The effect of noncondensibles on condensing heat transfer coefficients in air-steam and nitrogen-steam systems is illustrated in Figure 76 taken from Reference 25. Since the diffusion coefficients of mercury through hydrogen is at least 10 times as large as the coefficients for the systems mentioned above, it is expected that the effect of hydrogen on the mercury condensing heat transfer coefficient will be significantly less. The presence of hydrogen in the condensing section and the consequent reduction in overall heat transfer coefficient will manifest itself ultimately in the form of additional condensing length required or increased copper fin efficiency through increased fin thickness. The percent increase in the affected condensing length required due to hydrogen is plotted in Figure 77 versus the reduction in heat transfer coefficient. Completion of a two-dimensional diffusion analysis in the boundary region should allow an estimate to be made of the effect on condensing length.

The presence of a layer of hydrogen directly above a liquid mercury interface continually being disturbed by impinging droplets of mercury may lead

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Figure 76. Effect of Non-Condensibles on Heat-Transfer Coefficient

to gas entrainment or even slugging of gas pockets in the condensate outlet. The jet pump on the mercury pump suction line should aid in the breaking up of large bubbles of gas. Small bubbles may collect in the eye of the pump and lead to cavitation and vapor binding of the unit. Work is presently underway to determine the hydrogen swallowing capability of a small high-speed mercury pump.²⁶ The presence of small hydrogen bubbles in the high pressure bearing stream is not believed detrimental to bearing performance.

An additional effect of the recycle of hydrogen from the RC to the boiler would be an increase in the hydrogen pressure within the boiler leading to a reduction in the rate of hydrogen entering the mercury. If q_{nm} is reduced, q_{ns} will increase due to a slight rise in the hydrogen concentration in the NaK. In an extreme case where the hydrogen is assumed to be uniformly distributed throughout the mercury system with diffusion to space occurring only in the mercury vapor region the steady-state rate of hydrogen flow into the mercury is calculated to be about 3% of the specified reactor hydrogen leak rate with a

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Figure 77. Effect of Reduction
in Heat-Transfer Coefficient
on Condensing Length

resultant steady-state hydrogen concentration in the NaK of 12.8 ppm and a hydrogen pressure in the mercury at the boiler outlet of 7.4×10^{-5} atm. The above condition represents the minimum effect consistent with the assumed mechanism of hydrogen diffusion. The main question, as yet unanswered; is just what the rate of hydrogen removal from the RC will be due to entrainment. The answer to this question is not analytically evident.

2. Approach to Equilibrium

Consistent with the system model originally proposed in VII B-1 and 2 the time required for the hydrogen in the NaK systems to reach 95% of the steady-state concentration is obtained from Equation 130. Using parameters from Figure 75 it appears that 198 hr will be required. Similarly, for the mercury system, Equation 136 in conjunction with parameters from Figure 78 yields a value of 8.3 hr to approach within 95% of the steady-state hydrogen concentration. It is apparent that the mercury system approach to steady-state will be limited by the

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Figure 78. Mercury Vapor System and Parameters

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NaK system and that very little time lag exists between the NaK and mercury systems from the standpoint of hydrogen diffusion.

D. POSSIBLE SYSTEM MODIFICATIONS

There exists a number of areas in which system modifications may profitably be employed to either reduce the amount of hydrogen entering the mercury or facilitate its rejection to space; thereby, minimizing the effect of the hydrogen on system performance. The modifications fall into two general categories. These are (a) "windows" of highly permeable material either in the NaK or mercury systems and (b) hydrogen barriers on the boiler tubes.

1. Windows

The application of a chrome-molybdenum steel (Croloy 9M) and columbium (niobium) as hydrogen windows has been examined. Croloy 9M is a 9% chrome, 1% moly low carbon steel alloy having a permeability to hydrogen approximately 4 to 5 times that of 316 SS in the 1200 to 1000°F region. The use of this material has been proposed as an alternate boiler material and its application in this regard is discussed in References 27, 28, and 29.

According to Reference 28, a boiler with Croloy shells and the inner shell vented to space would reduce the steady-state rate of hydrogen flow into the mercury by 57%. While this reduction is significant, it by no means eliminates the problem since the hydrogen length in the RC would still be about 3.2 in. long. Furthermore, fabrication of the entire NaK piping system of Croloy as well provides only a small additional reduction of 0.4 in. of hydrogen in the RC. The use of Croloy as a boiler tube material due to its mercury corrosion resistance is highly detrimental to the hydrogen situation resulting in increases of 135 to 160% in the amount of hydrogen entering the mercury depending upon whether or not Croloy shells are used as well. Croloy 9M is about 3 times as permeable as the Haynes 25 tube material. Due to the very low pressure of hydrogen in the mercury the use of Croloy in the mercury vapor lines will not significantly improve the diffusion of the hydrogen from the mercury.

Columbium has a hydrogen permeability about three orders of magnitude above that of the 316 SS and the Haynes 25 in the 1000 to 1200°F range. Its use as discussed in VII B-5 either in the NaK or mercury systems should reduce the

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amount of hydrogen reaching the RC to a negligible value. As shown in reference (5) a columbium mercury vapor line between the boiler and turbine would reduce the hydrogen concentration in the vapor to 1% of the boiler exit value in 55% of the available tube length. The problems involved in joining columbium to dissimilar metals is significant and it could not be used in this area without considerable development work. If, on the other hand, the inner boiler shell were fabricated of columbium and a suitable getter sealed within the shell, it appears that the hydrogen diffusion rate into the mercury could be reduced by as much as 99%, without the necessity of welding the columbium to any other material since the inner shell would not be vented to space. A potential barrier to the use of columbium as a window may be the degree of hydriding encountered at the temperatures and hydrogen pressures expected in the system. As described in Reference 26, experimental evaluation of columbium windows as a means of removing hydrogen from a mercury vapor system is in progress.

The use of hydrogen windows in the RC above the liquid interface is impractical due to the complex honeycomb construction of the radiator and the present use of several different materials in its fabrication.

The use of a gas separator between the RC and the mercury pump may be feasible but has not been examined in detail. Such a device might utilize a centrifugal force field to separate the gas and a nonwetted porous surface to allow the hydrogen to diffuse to space. Problems inherent in such a device would be structural strength of a porous membrane and excessive pressure drop in the pump suction line.

2. Hydrogen Barriers

The development and testing of hydrogen diffusion barriers has been underway for an extended period in the nuclear industry. In Reference 30, a tungsten-silicon coating applied in 2 steps to one side of a Haynes 25 membrane provided a marked reduction in hydrogen permeability. Such a coating if applied to the boiler tubes has the potential of reducing the hydrogen input to the mercury by 98%. Unfortunately, the behavior of this and other coatings in NaK or mercury is not known. While it appears that hydrogen barriers have much to offer, considerable testing is yet required before they may be used with confidence in the SNAP 2 boiler.

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E. REMAINING PROGRAM

Both analytical and experimental efforts are being continued in order to resolve the current uncertainties associated with the effects and amount of hydrogen within the SNAP 2 system.³¹ Analytical models are being continually updated to correspond to the latest flight designs. The models are presently being improved to allow for more refined effects (e.g., two dimensional diffusion in the radiator-condenser). The analytical model will also be updated as new experimental permeability information becomes available. A number of experimental activities are presently planned or in progress to resolve some of the uncertainties and correlate the analytical models. These are:

- 1) Consideration is being given to experimentally inserting hydrogen into the NaK system, and possibly into the mercury system of PSM-1 (nonnuclear, thermal-hydraulic performance test of the NPU flight configuration), PSM-3 (nonnuclear, startup test system with a horizontal flat RC, and RC-11. PH 15-7 Mo (honeycomb thermal performance test, flight design). System performance can then be observed as a function of time following insertion. Decisions on the necessity for these tests will be made at a later date, pending further analytical and experimental information.
- 2) More consistent hydrogen permeation measurements are presently being made on PH 15-7 Mo material. The permeation measurements will be extended to a brazed honeycomb panel with rectangular tube and copper fin.
- 3) Chemical reactions and solubility of hydrogen in mercury are being investigated.
- 4) Hydrogen and other noncondensibles will be inserted into the single tube condensing experiment in order to determine the effects on heat transfer, pressure drop, gas entrainment at the interface, condensing coefficient, and shape of the liquid interface.
- 5) A program has been initiated at the University of Michigan to determine the effects of hydrogen on the cavitation of mercury.
- 6) The Thompson Ramo Wooldridge test program being performed on the Sunflower program will be followed closely. This program

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includes the introduction of hydrogen directly into the mercury stream entering the mercury pump. The program may be extended to CRU V testing if found desirable.

- 7) A program is being formulated at Thompson Ramo Wooldridge to determine the gas swallowing capabilities of the CRU V jet-centrifugal pump and the journal and thrust bearings behavior with noncondensibles present. In addition this program will study possible methods of eliminating the effects of entrained gases (e.g., accumulators).

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REFERENCES

1. Specification for SNAP 2 Reactor Temperature Control Subsystem NR7550-22, (September 6, 1963)
2. Specifications for SNAP 2 (Thermoelectric Pump) Design Point, Atomics International Specification NR7550-42 (June 18, 1963)
3. P.M. Gresho, "Radiator-Condenser Parameter Study," NAA-SR-TDR-7981 (November 30, 1962)
4. E.A. Carruthers, "Alternative Tube Geometries for the SNAP 2 Radiator-Condenser," NAA-SR-TDR-8895 (September 6, 1963)
5. C.J. Baroczy and V.D. Sanders, "Flow Vapors Condensing in a Straight Horizontal Tube," NAA-SR-6333 (June 1, 1961)
6. A.H. Shapiro, "Compressible Fluid Flow," Vol. I (Roland Press Company, 1953)
7. J.B. Scarborough, "Numerical Mathematical Analysis," 3rd Ed. (John Hopkins Press, 1955)
8. R.P. Forslund, "Pressure Drop Program for Condensing Mercury in the SNAP 2 Condenser," NAA-SR-TDR-9064 (September 17, 1963)
9. Seymour Lieblein, "Analysis of Temperature Distribution and Radiant Heat Transfer Along a Rectangular Fin of Constant Thickness," NASA-TND-196 (November, 1959)
10. R.P. Forslund, "Interface Stability in Rectangular Tubes within Gravity Field," NAA-SR-TDR-8960 (September 4, 1963)
11. R.P. Forslund, "Stability of the SNAP 2 Radiator-Condenser with Vehicle Tumbling During Steady-State Operation," NAA-SR-TDR-8843 (August 5, 1963)
12. R.P. Forslund and J.D. Stearns, "Determination of the Lateral Thermal Conductance in the SNAP 2 Radiator-Condenser Fin-Test," NAA-SR-TDR-9859 (September 4, 1963)
13. W.H. McAdams, "Heat Transmission," 3rd Ed. (McGraw-Hill Book Co. Inc., New York, 1954)
14. Technical Paper No. 410, "Flow of Fluids Through Valves, Fittings, and Pipe," (Engineering Divisions of Crane Co., Chicago, Ill., 1954)
15. M. Adler, "Flow in Curved Channels, Z AMM," Vol. 14, pp 257-275 (October 1934)
16. J. Stultz, "Primary Loop Pressure Drop," NAA-SR-MEMO-TDR-8372 (April 4, 1963)

NAA-SR-9087

DECLASSIFIED¹⁹⁸

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REFERENCES

17. J. Stultz, "Primary Loop Pressure Drop, " NAA-SR-TDR-8045 (December 19, 1963)
18. J. Stultz, "Pressure Drop Across the NaK Side of the SNAP 2 5C-4 Boiler, " NAA-SR-MEMO-TDR-8345 (March 23, 1963)
19. D. Berman, "SNAP 2 Prestartup System Temperature Analysis, " NAA-SR-TDR-6747 (September 20, 1961)
20. K.A. Toups, "A General Computer Program for the Determination of Radiant-Interchange Configuration Factors, " NAA-SID-62-393
21. P. Gresho, M.P. Berhold, "Radiant Interchange Code" Deck No. 9W-021 (September 1962)
22. R.R. Liquori, and J.W. Stephenson, Astra Inc., Raleigh N.C., "Astra Heating Program, " Astra 417-5.0, (January 1, 1961)
23. Deck Number 3K021, "Flat Plate, Turbulent Flow, " program obtained from SID
24. E.M. Mouradian, "Hydrogen Distribution by Diffusion in SNAP 8 Loop System, " NAA-SR-MEMO-8630 (June 17, 1963) Confidential
25. D.J. Ritchie, "Effect of Hydrogen Diffusion on Mercury Condensation, " NAA-SR-MEMO-8114 (June 3, 1963) Secret
26. "Sunflower Power Conversion Quarterly Report, " Thompson Ramo Wooldridge, Inc., Tapco Div. ER-5386 (March 1963 - May 1963)
27. S.L. Shulman, "Croloy Boiler Evaluation, " NAA-SR-MEMO-8708 (July 10, 1963) Confidential
28. P. Barbour, "SNAP 2 Boiler Modifications - Effect on Hydrogen Diffusion into the PCS, NAA-SR-MEMO-8763 (July 1963) Confidential
29. M.G. Weisman, "The Structural Use of Croloy in the Model 5C Boiler, " NAA-SR-MEMO-8669 (August 1, 1963)
30. "Sunflower Power Conversion System Quarterly Report, " Thompson Ramo Wooldridge, Inc., Tapco Division, ER-5264 (December 1962 - February 1963)
31. R.T. Lancet, "Proposed Hydrogen Diffusion Program, " NAA-SR-MEMO-8682 (June 28, 1963) Secret
32. M. Weisman and D. Burgess, "SNAP 2 Meteoroid Protection, " NAA-SR-TDR-8098 (January 1, 1963)

NAA-SR-9087

199

0317020030

REFERENCES

33. J. Stultz, "Pressure Drop Across the NaK Side of the SNAP 2 5C-6 Boiler," NAA-SR-TDR-8472 (April 9, 1963)
34. G.F. Burdi, "Boiler Shield Analysis," NAA-SR-TDR-8042 (December 17, 1962)
35. "Heat Transfer Analysis of the Primary Loop," NAA-SR-TDR-8900 (October 28, 1963)
36. E. Fried, "Thermal Joint Conductance in a Vacuum," ASME Publication, Paper Number 63-AHGT-18 (December 1962)
37. G.F.C. Rogers, "Heat Transfer at the Interface of Dissimilar Metals," International Journal of Heat and Mass Transfer Volume 2, New York (1961)
38. Deck Number 9K050, "Isos Program for Analogous Circuit Network for Heat Flow," obtained from SID
39. E. C. Bauer Jr., SNAP 2 Instrument Compartment Wire Temperature Gradients," NAA-SR-TDR-8574 (June 3, 1963)
40. E.C. Bauer Jr., "Thermal Analysis of SNAP 2 Instrument Compartment," TDR-8551 (June 26, 1963)
41. R.B. Bird, W.E. Stewart, E.N. Lightfoot, "Transport Phenomena," J. Wiley and Sons, Inc., New York (1960)
42. P. Barbour, "Effectiveness of Columbium Windows in Reducing Hydrogen Leakage to the Mercury System," To be published (Confidential)
43. E.A. Steigerwald, "The Permeation of Hydrogen Through Construction Materials," Thompson Ramo Wooldridge, Inc., Tapco Division, ER-4776 (November 30, 1961)
44. P.S. Flint, "The Diffusion of Hydrogen Through Materials of Construction," KAPL-659 (December 14, 1951)
45. TRW "The SNAP 2 Power Conversion System Topical Report No. 13 OFFBASE Testing" TRW Report No. ER-4670
46. TRW "The SNAP 2 Power Conversion System Topical Report No. 8 Condenser Development and Design Study" TRW Report No. ER-4670
47. R. P. Forslund "SNAP 2 RC Stability Under Zero "G" Conditions" NAA-SR-TDR 9126 (October 24, 1963)