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AUXILIARY POWER SYSTEMS FOR A LUNAR ROVING VEHICLE

by E. P. Erlanson

Prepared by
GENERAL ELECTRIC
Philadelphia, Pa.

for Lewis Research Center

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GENERAL ELECTRIC
Spacecraft Department
Philadelphia, Pa.

for Lewis Research Center

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

FOREWORD

The research described herein, which was conducted by the General Electric Spacecraft Department, was performed under NASA Contract NAS 3-7627 with Mr. Lloyd W. Ream, Space-Power Systems Division, NASA Lewis Research Center as Technical Manager. The report was originally issued as General Electric Document No. 66SD4395, June 1966.

ABSTRACT

Five types of power systems are considered for use with a two man lunar roving vehicle for a 45-Day Mission. Life support and cabin environmental control system conceptual designs are investigated. The benefits of thermal integration of the power and life support system are determined. Comparisons of nonintegrated and integrated designs are made to determine the combination with the most merit. An integrated isotope Brayton cycle - life support system is selected as the one with the most potential for use with the roving vehicle.

SUMMARY

The main objective of this study was to determine from several potential power systems the best candidate for a lunar roving vehicle, taking into account closed life support systems and the possibilities for thermal integration. The power systems considered were hypergolic engine, hydrogen-oxygen engine, fuel cell, isotope Brayton cycle and isotope Rankine cycle. The closed loop life support systems recycled water and oxygen as principal functions. A two-man crew was used for equipment sizing.

The roving vehicle mission duration was taken as 45 days. The assumed vehicle configuration was a 12-foot cabin (extended MOLAB) on four wheels attached to one or more two-wheeled trailer modules carrying the power system and fuel. The available electrical power required is approximately 3 kw for full vehicle operation, including locomotion. Each complete vehicle is to be transported to the lunar surface by an unmanned Saturn V-LLV with a payload capability of 28,000 pounds. The trailer modules were constrained to have a maximum weight of 3000 pounds. A 100-watt electrical output RTG is located on the side of the cabin for the 30-day dormant period power.

The lunar thermal environment was considered for heat rejection constraints. The chosen terrain placed the vehicle in a bowl with a 9-degree upward slope in all directions. Life support analyses included environment control of the cabin, electronic equipment, thermal control, and extravehicular suit water, oxygen, and thermal balances.

Thermal integration of the power and life support systems, or the use of power equipment waste heat to operate life support processes, was studied for each power system to determine the advantages and shortcomings. A formal evaluation of both integrated and non-integrated vehicles was conducted to select that combination with the most potential for use in the lunar roving vehicle.

The major results were:

a. An integrated isotope Brayton cycle has the greatest potential for use in the 45-day mission lunar roving vehicle. The isotope requirement for this system is 14 KWe; the single trailer module weight slightly less than 3000 pounds.

b. Thermal integration reduces isotope systems electrical power requirements from 2.85 KWe to 1.85 KWe. The replacement of electrical heat with power system heat is feasible for the majority of the roving vehicle life support processes entry requirements.

c. The cabin and power system trailer module are capable of heat rejection and control during the lunar environment extremes without recourse to use of expendables or reduction in operations.

d. Shorter mission times improve the merit of the fuel cell, longer mission times (and successive missions with one vehicle) improve the merit of the isotopic power systems. The point of equal merit for the evaluation conducted was 34 days.

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SECTION 1

INTRODUCTION

1.1 BACKGROUND

Various methods are being considered for lunar surface exploration following the initial Apollo LEM landings around the end of this decade. A possible method of approach, which has been studied in some depth, is a two man lunar roving vehicle with the acronym MOLAB (Reference 1-1 and 1-2). As part of the basic Apollo-Saturn V system the MOLAB is envisioned for use with mission durations not exceeding 14 days due to booster weight limitations. Fuel cells have been the selected power source in the MOLAB studies because they are relatively light weight and will be flight proven in the Gemini and Apollo programs. MOLAB could be the complete expeditionary force or it could function as part of a lunar base-vehicle system that would have a total crew complement of six or more (Reference 1-3 and 1-4).

The prospects for long duration single vehicle expeditions and base/roving vehicle complexes have been examined in anticipation of post Apollo and requirements beyond the 14-day MOLAB (Reference 5). The vehicle of particular interest is the two-man intermediate exploration vehicle (IEV) which has a 42 day mission duration, as studied in Reference 1-5. The vehicle configuration is similar to a MOLAB, with a one foot cabin extension and the addition of a second trailer module to carry power system fuel (Figure 1-1). This vehicle was adopted as the basic configuration for the 45 day mission lunar roving vehicle considered in this study. It is assumed that a single Saturn V-Lunar Landing Vehicle would deliver the roving vehicle to the lunar surface. The two-man crew would arrive by another flight system at some later time. The study emphasis was placed by direction on a single 45 day mission. However, it is recognized, as investigated by others, that a more economic long-range plan would be to reuse the vehicle on multiple missions, as with vehicle/base complexes. An extension of manned vehicle active operation time could have an impact on methods used in some of the different vehicle systems, such as power, life support, enviromental control, locomotion, and the scientific experiments. Closed loop systems, or methods which reprocess the same material, may offer certain advantages when long term missions are considered. For the life support system, oxygen and water can be reused and recovered with existing design concepts. The closed system concept is also descriptive of power systems using radioisotopic energy sources with long half-lives.

Two previous studies (References 6 and 7) have explored the design of life support and power systems for a six man earth orbiting vehicle and a two man lunar shelter. The primary purpose of this work was to examine thermal integration of the two systems, using waste thermal energy in place of electrical energy from the power system to operate life support process equipment. It was found that this integration was feasible and resulted in reduced power system size and weight. Thermal integration may provide advantages for a lunar roving vehicle.

1.2 PURPOSE

The objectives of this study were to examine different possible power systems and associated life support systems, and determine the extent and practicality of thermal integration when used on a 45 day two man lunar roving vehicle. Evaluations of integrated roving vehicles with each power system were also required to determine the benefits of thermal integration. Results of past studies, including the work on two previous thermal integration contracts, were to be re-applied where applicable.

1.3 STUDY TOPICS

The electrical power systems considered in this study were:

a. Radioisotope dynamic systems

   1. Brayton cycle power system
   2. Rankine cycle power system

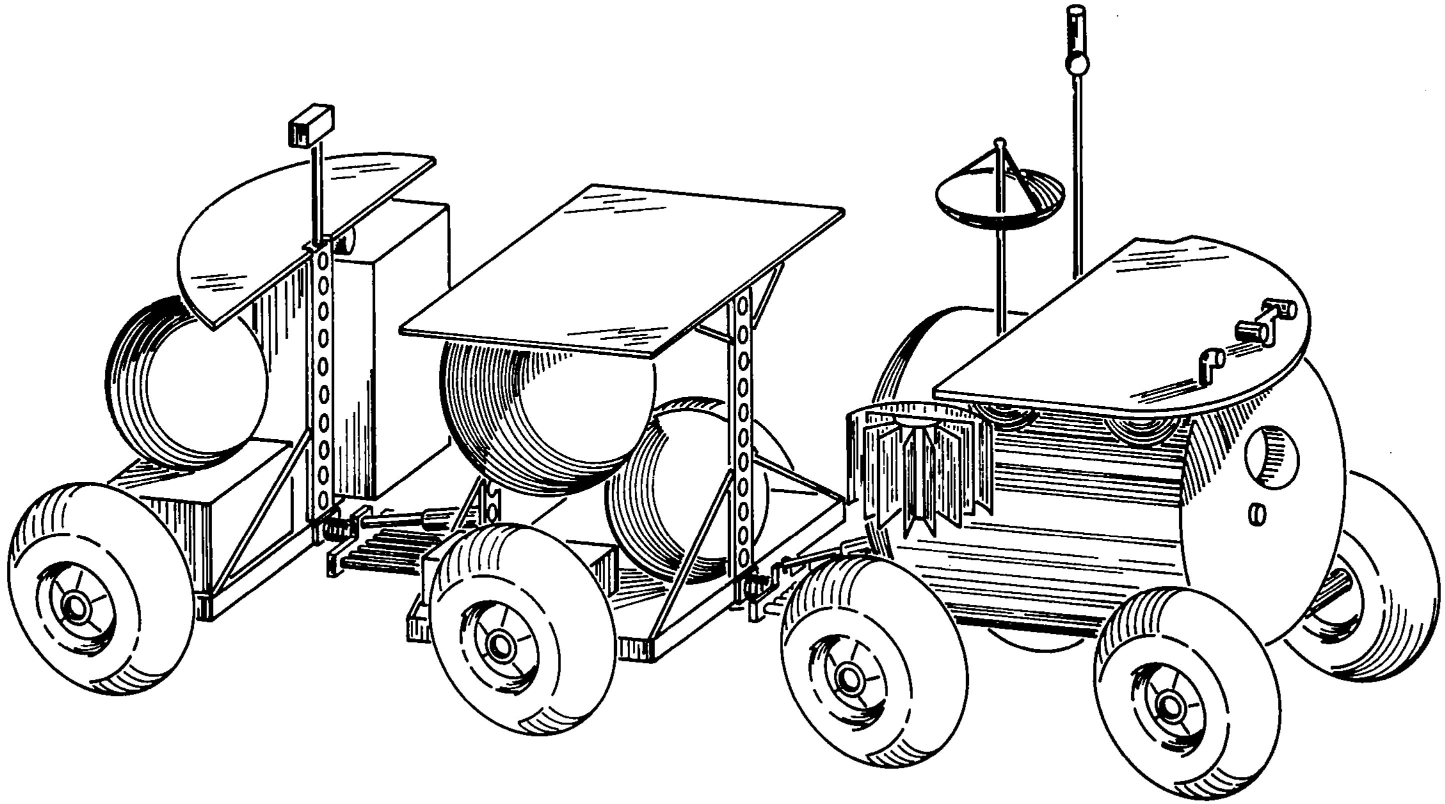


Figure 1-1. Basic 45 Day Lunar Roving Vehicle

b. Chemical power systems.

1. High temperature fuel cell
2. H-O internal combustion engine system
3. Hypergolic combustion engine system

The life support subsystems examined were as follows:

1. CO_2 regeneration by the sabatier method
2. Urine treatment by distillation and pyrolysis
3. Waste water distillation
4. Food preparation
5. Cabin atmosphere cooling

Cabin environment control, including consideration of an absorption refrigeration system was analyzed.

Study analyses included the following areas:

a. Electrical power requirement for each configuration

b. Power system energy requirements or heat source size

c. Size and weight of heat rejection equipment

d. Isotope source configuration and weight

e. Effects of thermal integration on power and life support systems

f. Temperature level and energy rate requirements for life support processes

g. Thermal integration equipment design

h. Power system component design to the extent of determining size and weight

i. Life support equipment design to the extent of size, weight, and determination of their thermodynamic characteristics

j. Lunar environment

k. Vehicle mission requirements

l. Systems evaluation

1.4 APPROACH

The initial requirement in the study was to establish a philosophy such that each system was designed on a consistent basis. The requirement was met by analyzing and extrapolating from previous studies the lunar environment, roving vehicle mission requirements, vehicle equipment electrical requirements, crew consumable requirements, design philosophy, and assembling all of this information into the design guidelines (Section 3). Some important general guidelines are:

a. Two-man crew

b. Unmanned placement on lunar surface followed by 30 day dormant period

c. 45-day manned mission

d. Near lunar equator location with system to be capable of all functions during lunar day and night

e. Vehicle configuration to be a modified MOLAB - Also called intermediate exploration vehicle (IEV) (Figure 1-1).

Following identification of the significant restraints, studies were made of the power and life support systems and nonintegrated designs were reached. The electrical power requirements of the life support system were established, along with the temperature requirements of each process. The waste heat temperatures and quantities from the power systems were identified, and the degree to which electrical power could be replaced by thermal power was determined. The electrical power output requirements were then reduced by the power replaced with heat in the life support systems and the power systems were resized for the new electrical load. Effects on the life support equipment of replacing electrical energy with heat energy were investigated, and design modifications performed where necessary.

Following the systems design phases, a comparison of the resulting combinations were made, according to established criteria. Selected integrated combinations were compared with the present MOLAB standard, a fuel cell with nonintegrated life support system.

1.5 REFERENCES

1-1. Apollo Logistics Support System (ALSS) Payloads, The Boeing Company, Document D2-36072-3, April 1965 (Two Volumes)

1-2. "ALSS Payloads", Bendix Corporation, Final Report for NASA Contract NAS 8-11287, June 1965.

1-3. Initial Concept of Lunar Exploration Systems for Apollo (LESA), Boeing Company, November 15, 1963

1-4. Operation and Logistics Study of Lunar Exploration Systems for Apollo, General Dynamics/Forth Worth, NASA CR-62018, February 1965

1-5. "Study of Human Factors and Environmental Control - Life Support Systems", Lunar Exploration Systems for Apollo, Prepared by Garrett Division, AiResearch Manufacturing Company, for George C. Marshall Space Flight Center, Report Number SS 3243 (Six Volumes)

1-6 Hanson, K. L., Thermal Integration of Electric Power and Life Support Systems for Manned Space Stations, NASA Contractor Report NASA CR-316, General Electric Company, November 1965

1-7. Woods, R. W., and Erlanson, E. P., Study of Thermal Integration of Electric Power and Life Support Systems for Manned Space Stations, General Electric Company, Document Number 66SD4231, Final Report for NASA Contract NAS 3-6478, January 26, 1966

SECTION 2

RESULTS

This section presents a summary of the results of the study and discusses their implications.

2.1 STUDY RESULTS

A. The evaluation of the power systems and their associated life support systems produced the following ranking for a 45 day mission:

		Figure of Merit
a.	Integrated Brayton Cycle	234
b.	Integrated Rankine Cycle	216
c.	Fuel Cell	214
d.	Integrated Hydrogen-Oxygen Engine	127

(The Hypergolic Engine was eliminated from evaluation because of excessive weight)

Figure 2-1 shows the mission time dependence of the relative figure of merit of the power systems. The complete evaluation and supporting evidence can be found in Section 3.

B. The two integrated isotopic systems and a fuel cell system were compared for the three criteria: weight, deployed area and electrical power requirements.

	Fuel Cell + Open Loop LSS	Brayton + Closed Loop LSS	Rankine + Closed Loop LSS
Power System + Life Support System			
Weight (Pounds)	6674	3208	3386
Deployed Area (Square feet)	230	379	267
Electrical Power Requirements (Gross)	3.15kw	2.43kw	2.47kw

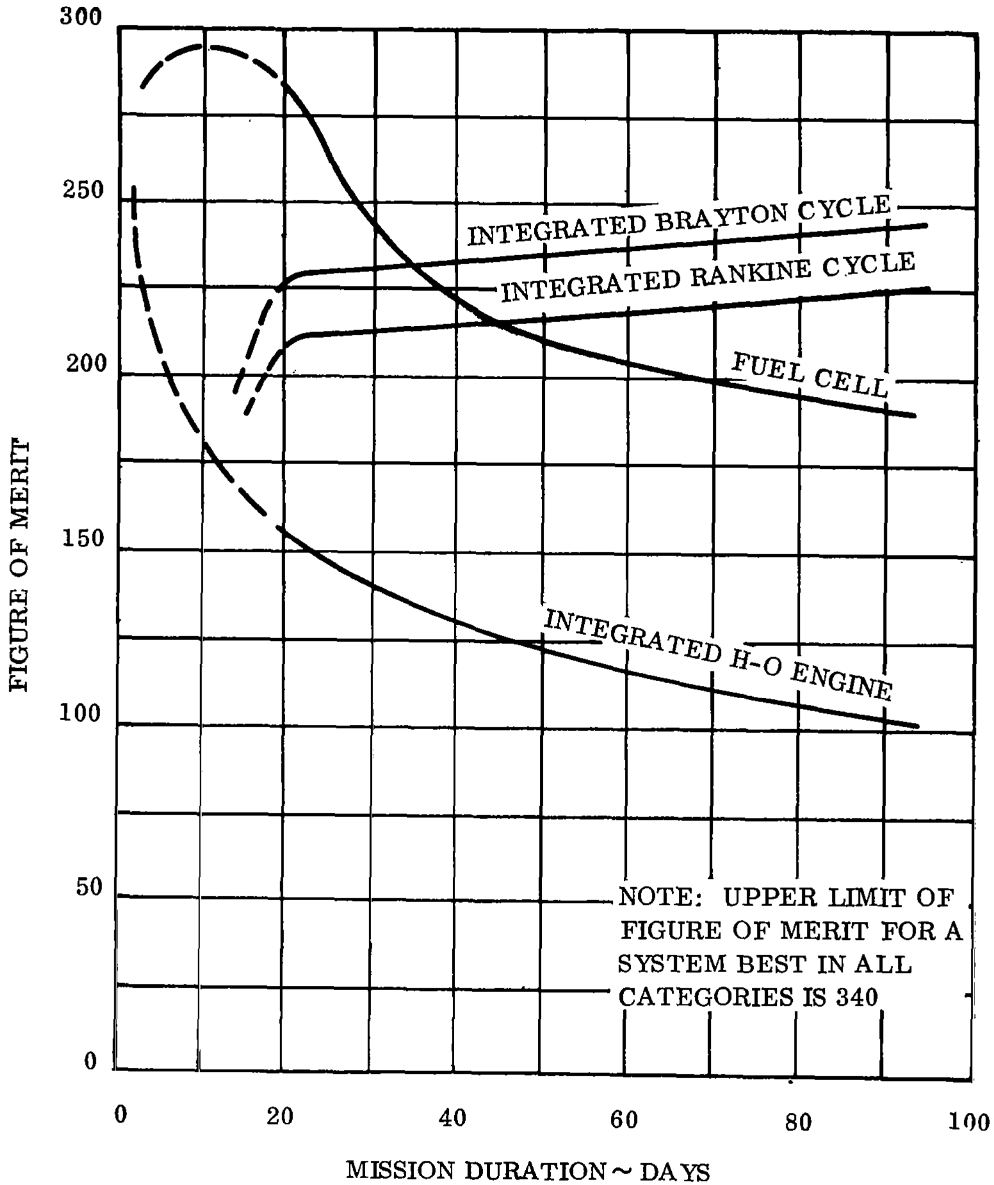


Figure 2-1. Evaluation of Auxiliary Power Systems for a Lunar Roving Vehicle

C. Thermal integration reduces the isotopic systems electrical power requirements by 970 watts. The $H_2 - O_2$ engine power required is reduced 880 watts by thermal integration. The process heat required by life support equipment is essentially all supplied by thermal integration. The remaining life support electrical power consists of pumps, blowers, electrolysis, illumination (germicidal), and pyrolysis which are all definitely unfeasible for operation by thermal power.

D. The reduction in required electrical power changes the $H_2 - O_2$ engine system weight from 12,500 pounds to 8,730 pounds. The thermally integrated Rankine isotope source is reduced from 36 to 29 thermal kilowatts. The integrated Brayton cycle heat source requirements are reduced from 18.5 to 14 thermal kilowatts.

E. The radiator shading by lunar terrain and the elevated lunar surface day temperatures, for the considered model, did not significantly affect roving vehicle design or operation, although the presence of a hot day surface did limit radiator configurations.

F. The isotope power systems require one trailer module for the 45 day mission, the fuel cell power system requires two trailers, and the integrated hydrogen-oxygen system requires three trailers to carry the fuel.

G. The lightest isotopic system designs are composed of constant output alternators with secondary batteries accounting for peak power demand.

H. The best general heat rejection method for the roving vehicle trailer and cabin is to use a direct space oriented radiator. Absorption refrigeration does not provide benefits for electronic equipment heat rejection. Also, as results of the previous study (Reference 2-1) show, a vapor compression system requires additional electrical power, weighs more, reduces reliability and is otherwise unattractive.

I. For the fuel cell and $H_2 - O_2$ engines, product water can be used to reduce the size of the cabin radiator. However, the amount of reduction is less than 20%. Also studies for longer term base roving vehicle complexes (Reference 2-2) indicate that it is advantageous to return the water to a base for reseparation into hydrogen and oxygen.

J. Magnesium oxide can be used in cannisters in the pressurized suits to remove CO_2 from suit-air. This represents a simplification over the molecular seive process and saves weight.

K. Minimum electrical power requirements were found with the integrated hydrogen-oxygen engine, 1.72 kw_e (Net).

L. The dormant period thermal control is accomplished by having low flow pumps and blowers (operating with 13 watts RTG Power) to circulate the cabin oxygen and radiator fluid.

M. The vapor fin radiator provides advantages over the fin-tube design for use on the lunar roving vehicle.

Table 2-1 presents a summary of the significant parameters derived during the study, for all the power systems examined. Figures 2-2 through 2-6 show the overall trailer-cabin configuration for each power system.

2.2 DISCUSSION OF RESULTS

The hypergolic engine was eliminated from the systems evaluation because its weight of 24,000 pounds (integrated) plus the rest of the roving vehicle (estimated at 6000 pounds) exceeds the 28,000 pounds now being considered as the maximum Saturn V - LLV Payload (Reference 2-2).

As can be seen in Figure 2-1, the figure of merit for the chemical systems decreases noticeably with increasing mission duration. The isotope systems are semiweight independent with a per-day increase of 18 pounds while the fuel cell use is 130 pounds per day. Thus the isotope systems not only weigh less, but are more adaptable to extended or

SUMMARY TABLE 2-1. POWER SYSTEMS FOR LUNAR ROVING VEHICLE (45-DAY MISSION)

		System Type	H_2-O_2 Engine	Hypergolic Engine	Isotope Brayton	Isotope Rankine	Fuel Cell
Type		B	Chemical Dynamic Piston	Chemical Dynamic Piston	Nuclear Dynamic Turbine	Nuclear Dynamic Turbine	Chemical Static
Working fluids		B	H_2-Steam	Combustion products	Argon	Mercury	H_2-O_2-steam
Rotation rate (rpm)		B	6000 (max)	6000 (max)	67,500 & 12,000	40,000	---
Electrical power		B	400 Hz	400 Hz	400 Hz	2000 Hz	DC
Fuel		B	H_2, O_2	N_2O_4, UDMH/N_2H_4	Isotope	Isotope	H_2, O_2
Maximum temperature (°R)		B	4000	5800	1950	1710	960
"Cycle efficiency"		N-I	0.28	0.25	0.25	0.15	0.61
		I	0.28	0.25	0.23	0.13	
System efficiency		N-I	0.22	0.20	0.16	0.08	0.44
		I	0.31	0.26	0.20	0.10	
Average net power (kw)	Electrical	N-I	2.60	4.68	2.85	2.85	2.26
	Electrical	I	1.72	3.05	1.88	1.85	
	Thermal	I	0.71	0.97	0.97	1.00	
	Total	I	2.43	4.02	2.85	2.85	
Average gross power (kw)	Electrical	N-I	2.68	4.77	3.51	3.71	3.15
	Electrical	I	1.77	3.13	2.43	2.47	
	Thermal	I	1.13	1.17	1.19	1.16	
	Total	I	2.90	4.30	3.62	3.63	
Net power system weight		N-I	10,070	28,500	1,813	1,944	4,990
		I	6,810	18,700	1,520	1,730	
Gross power system weight (pounds)		N-I	12,820	36,450	3,160	3,290	6,490
		I	8,960	24,250	2,870	3,080	
Number of trailer		N-I	4	12	1	1	2
		I	3	8	1	1	
Weight per trailer (pounds)		N-I	3,200	3,040	3,160	3,290	3,245
		I	3,000	3,030	2,870	3,070	
Volume (cu ft)		N-I	1,100	1,300	150	135	260
		I	800	850	135	120	
Net power system weight/lb / Net useful power kw		N-I	3,850	6,100	640	680	2,200
		I	2,800	4,650	530	610	
Gross power system weight/lb / Net useful power kw		N-I	4,900	7,800	1,100	1,200	2,900
		I	3,700	6,000	1,000	1,100	
Waste heat (kw)		N-I	7.1	14.1	13.5	31.9	3.3
		I	3.0	8.7	10.5	24.9	
Radiator inlet/outlet temperature (°R)		N-I	960-950	960-950	860-520	1065-855	670-585
		I	950-940	950-940	830-520	1065-840	
Radiator area (sq ft)		N-I	190	95	285	130	120
		I	150	62	205	100	

Note 1: Code: N-I Nonintegrated system
I Integrated system
B Both systems

Figure 2-2. Lunar Roving Vehicle with Hypergolic Engine Power System

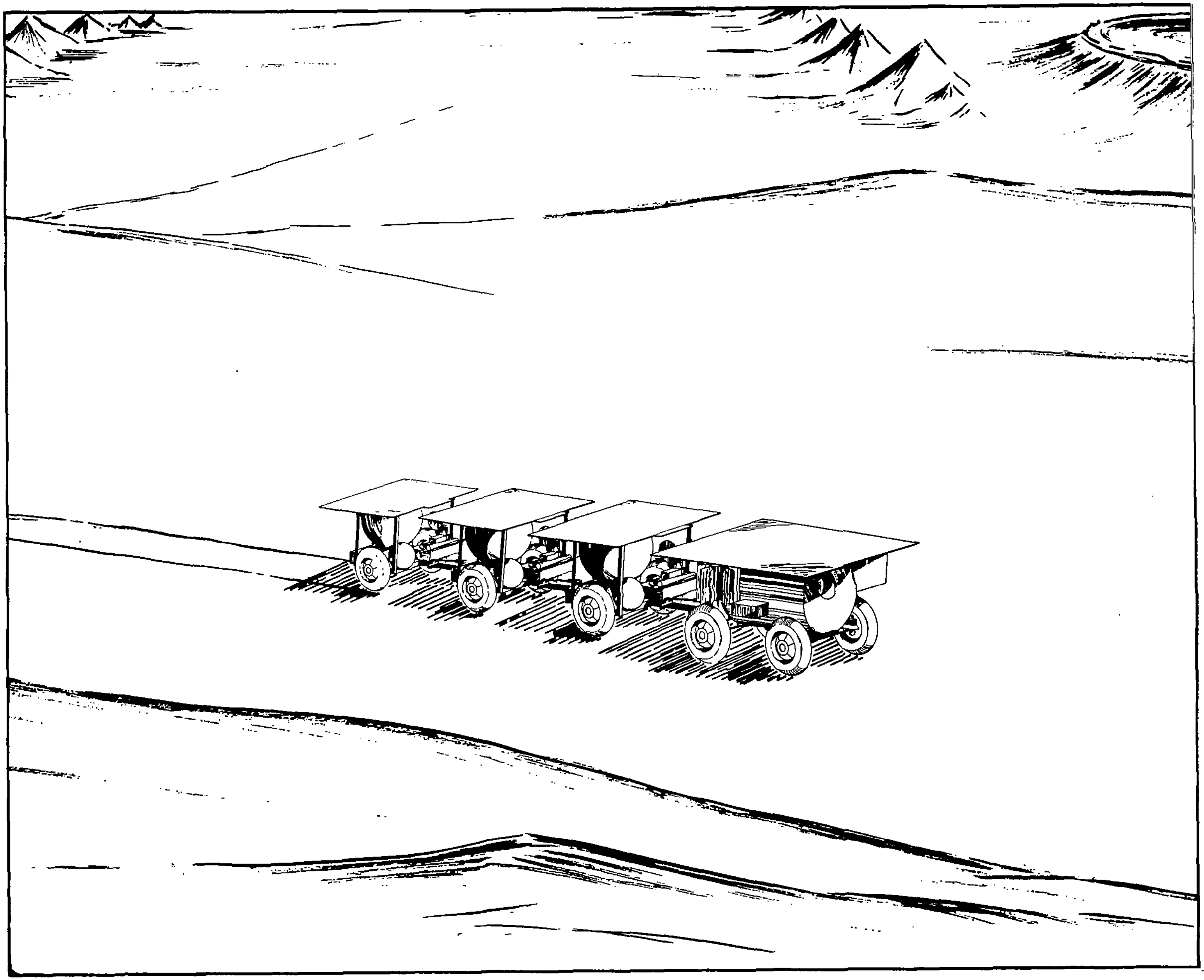


Figure 2-3. Lunar Roving Vehicle with Hydrogen-Oxygen Engine Power System

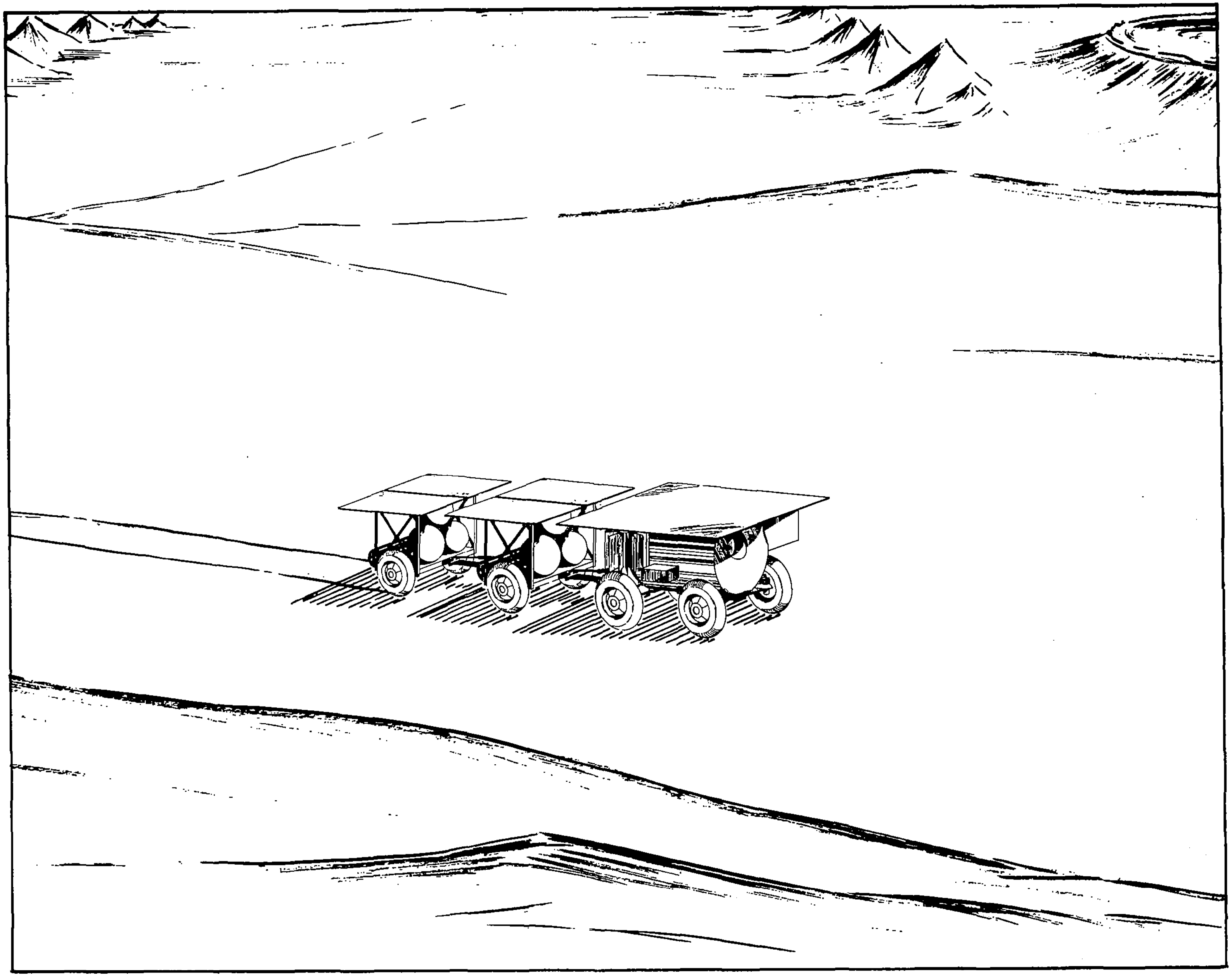


Figure 2-4. Lunar Roving Vehicle with Fuel Cell Power System

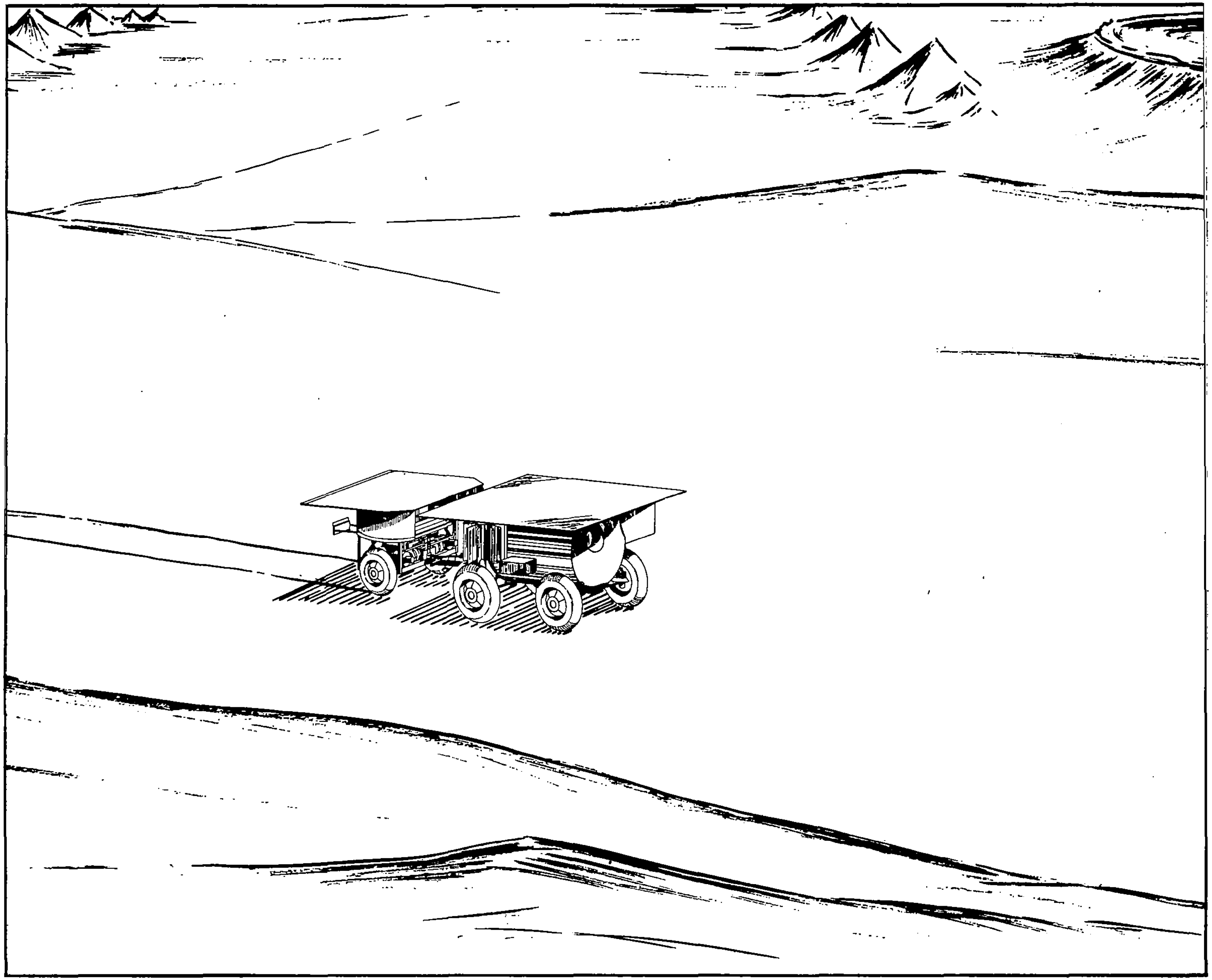


Figure 2-5. Lunar Roving Vehicle with Brayton Cycle Power System

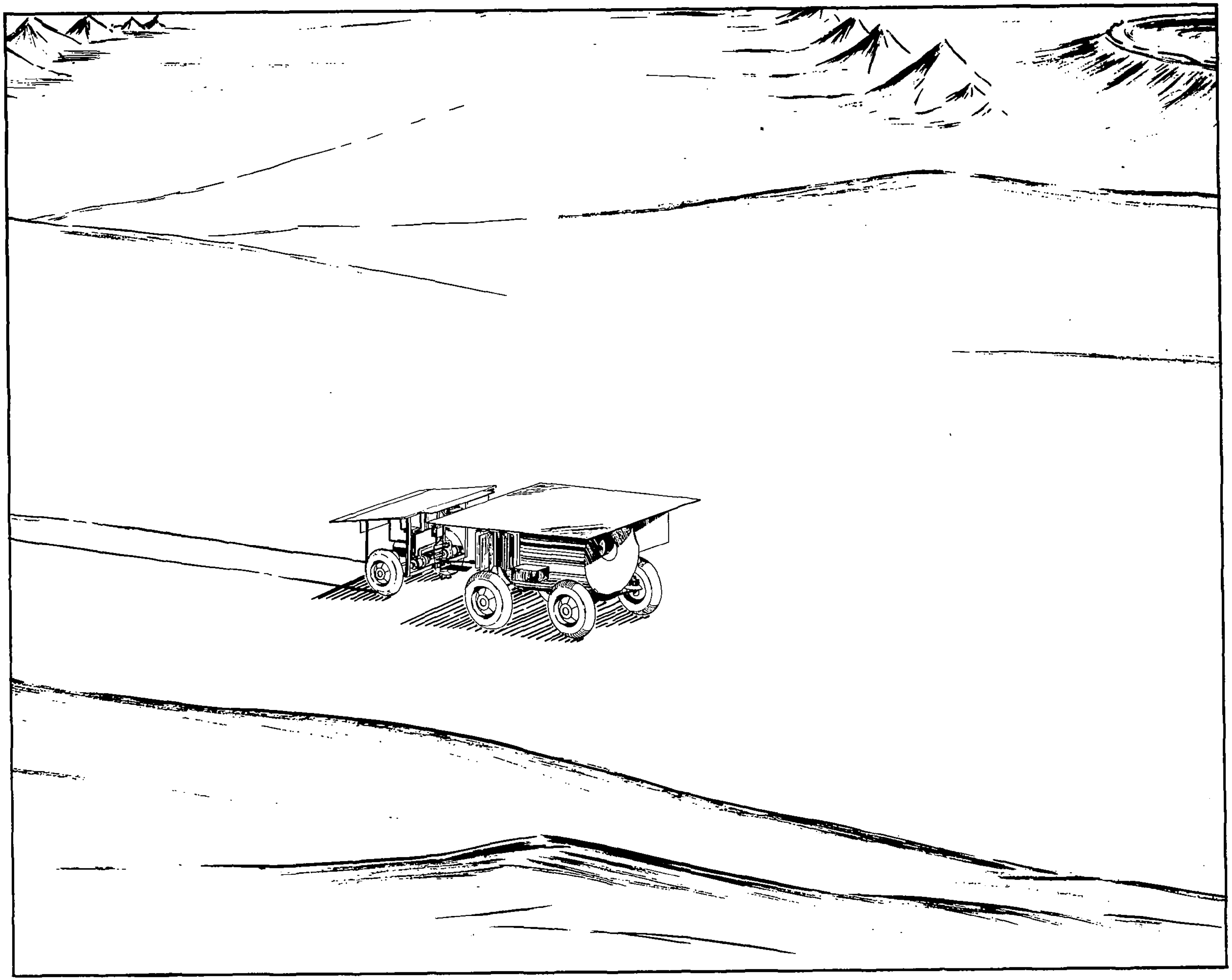


Figure 2-6. Lunar Roving Vehicle with Mercury Rankine Cycle Power System

multiple missions. Two significant items in the fuel cell's favor were the lack of large isotope requirement, and its development status. The matching of the isotope systems shows that the smaller amount of isotope fuel required by the Brayton system outweighs the Rankine cycle's advantages in having a smaller radiator, its present more extensive development, and less dependence on proper life support heat removal when thermally integrated.

The 880 watts of electrical power reduction for the hydrogen-oxygen engine includes approximately 180 watts for trailer locomotion, since thermal integration reduces the number of trailers from four to three. The CO_2 is not processed to recover oxygen when the H_2 - O_2 engine is used, for it is simpler and lighter to transport extra oxygen and use it directly. The difference in energy requirements for the isotopic and chemical systems is then due to CO_2 processing, principally in water electrolysis.

The change in isotope requirements is considered significant because available references indicate that likely fuel candidates will be both expensive and scarce for the model program schedule. Further reduction in isotope thermal source sizes beyond thermal integration can be made by: a) decreasing the rejection temperature, or b) using a life support system in which oxygen is not recovered. Decreasing the rejection temperature would impose a weight and complexity penalty on the system, but would reduce source requirements, since it increases thermal efficiency. Use of an open loop for the oxygen portion of the Life Support System would decrease weight slightly (for a 45 day mission), simplify the system, and reduce source requirements to 10. 5 kw in the Brayton system and 22. 5 kw in the Rankine system. If the vehicle were to be reused for several 45 day missions, then the 8 pounds per day of oxygen weight penalty would reach a point where the closed cycle with oxygen recovery would be desirable.

Prior to the study, direct supply of electrical power from an alternator for all profile demands was felt to be feasible. As work progressed it became apparent that the initial equipment weights for the fuel cell, Brayton cycle and Rankine cycle were quite dependent on maximum output. To minimize weight, the choice was then made to smooth the power profile with a secondary battery, resulting in a more level output. Although this results in additional battery weight, the total system is lighter and additional power is available to assist in startup, emergency or repair periods.

Thermal integration of the fuel cell power system with life support was examined briefly. In terms of the evaluation method used, thermal integration would reduce total weight by perhaps 1000 pounds, at the expense of the "Thermal Integration" penalties. The fuel cell system would still weigh 5500 pounds (and require two trailer modules) versus the 3000 pounds for the isotopic systems and its figure of merit for the criteria used would improve only slightly.

2.3 REFERENCES

2-1. Woods, R. W. and Erlanson, E. P., "Study of Thermal Integration of Electric Power and Life Support Systems for Manned Space Stations", Final Report for NASA Contract NAS 3-6478, General Electric, Missile and Space Division Document No. 66SD4231, January 26, 1966.

2-2. "Study of Human Factors and Environmental Control - Life Support Systems", prepared by Garrett Division, AiResearch Manufacturing Company, George C. Marshall Space Flight Center, Report Number S. S. 3243-3, April 1965.

SECTION 3

DESIGN REQUIREMENTS AND SYSTEMS COMPARISONS

3.1 DESIGN GUIDELINES

3.1.1 INTRODUCTION

The design guidelines which were established to define the design requirements, constraints and philosophy were based on the contract work statement and References 3-1 through 3-5. The guidelines include:

a. Study scope
b. System requirements
c. Lunar environment model
d. Mission profile
e. Design requirements and restraints

3.1.2 STUDY SCOPE

This is a study of power and life support systems and their thermal integration, as applied to a two man lunar roving vehicle with a 45 day active mission.

The power systems considered were:

a. Radioisotope Brayton Cycle
b. Radioisotope Mercury Rankine Cycle
c. High Temperature Fuel Cell
d. Hydrogen-Oxygen Combustion Engine
e. Hypergolic Combustion Engine

The life support and environmental control processes considered were:

a. CO_2 regeneration by the Sabatier method*
b. Urine treatment by distillation and pyrolysis
c. Waste water distillation
d. Food preparation
e. Cabin atmosphere cooling
f. Electronic equipment cooling
g. life support equipment cooling
h. Absorption refrigeration

System designs of the roving vehicle were made for each of the power systems on both an integrated and a nonintegrated basis. Components were designed only to the extent of determining size and weight when data was not available from previous studies. Equipment arrangements were made to show compatibility with available volume.

3.1.3 SYSTEM REQUIREMENTS

The model for the lunar roving vehicle is the six wheel semiarticulated concept described in Reference 3-1. Some modifications were required because of the 45 day mission and other study requirements.

3.1.3.1 System Specifications

a. Nonintegrated Power Requirements. The maximum power to the power bus is 6.0 kw. The load profile is shown in Table 3-1.

b. Dormant Period. There is a 30 day dormant period on the lunar surface prior to crew arrival.

*Cabin air is not regenerated to the extent of removing O_2 from CO_2 for vehicles powered by a fuel cell or H-O engine. For these systems oxygen is provided from a power system-life support stored supply.

c. Launch Environment. The vehicle is considered isolated from aerodynamic and solar fluxes. The isotope power systems will have exterior access for heat rejection.

d. Lunar Environment. The lunar environment is based on the description in References 3-1 and 3-2. The vehicle shall be active for 45 days. The location will be near lunar equator.

e. Total Roving Vehicle Launch Weight. 28,000 pounds.

3.1.3.2 Cabin Description

Figure 3-1 depicts the IEV Cabin and the general equipment arrangement.

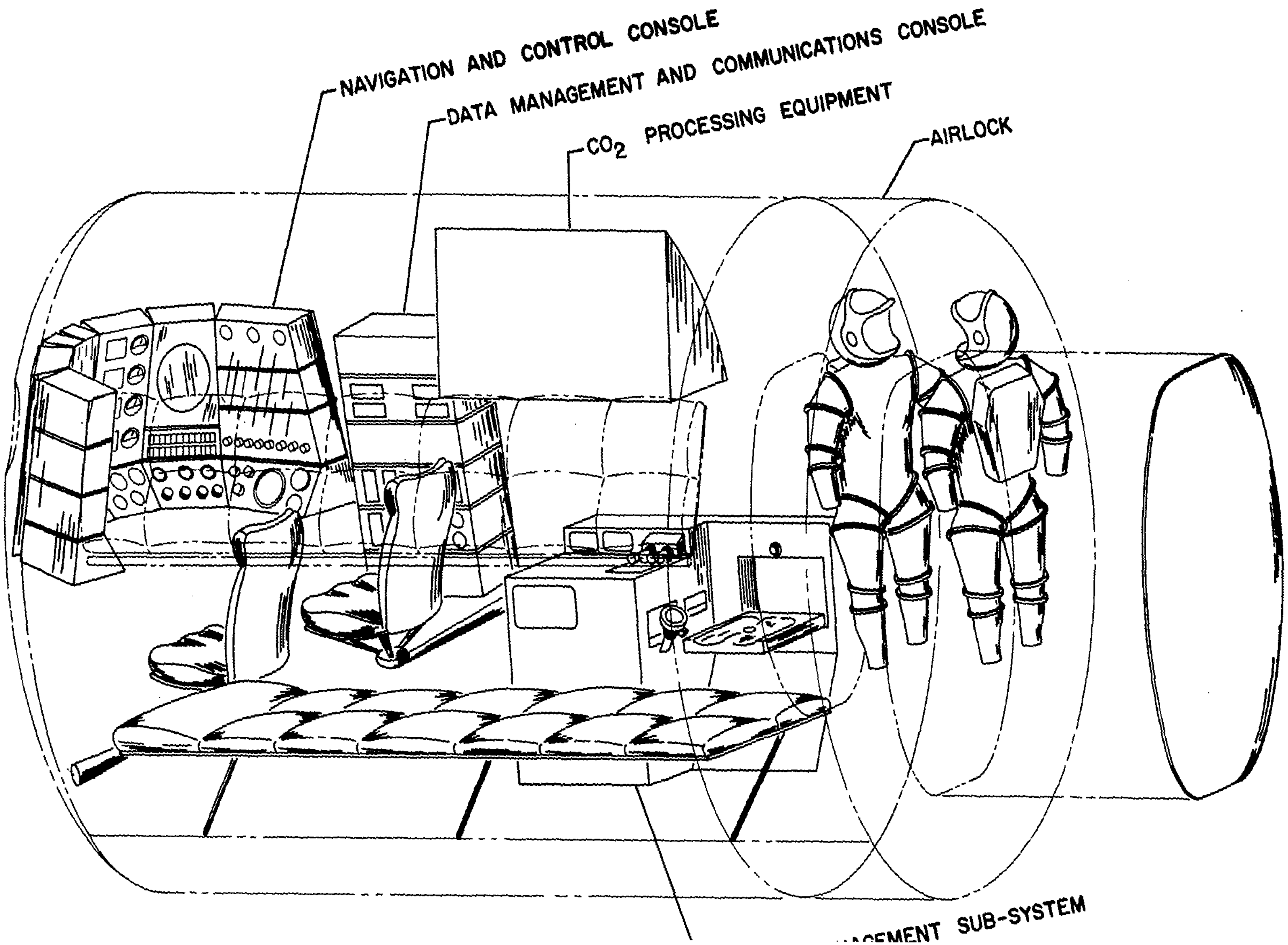
a. Cabin volume - 245 ft^3 from Reference 3-3.

b. Cabin atmosphere (occupied)

O_2 partial pressure	180 mm Hg
N_2 partial pressure	180 mm Hg
CO_2 partial pressure	3.8 mm Hg
H_2O content	50% relative humidity
Temperature	$72^{o}F$
Leak Rate	1 lb of atmosphere/24 hr period

c. Dormant period cabin atmosphere

O_2	180 mm Hg
(no N_2, CO_2, or H_2O)	
Temperature	minimum $60^{o}F$
	maximum $100^{o}F$
Leak Rate	0.5 lb/24 hr period


NAVIGATION AND CONTROL CONSOLE
DATA MANAGEMENT AND COMMUNICATIONS CONSOLE
CO_2 PROCESSING EQUIPMENT
AIRLOCK
SUB-SYSTEM

d. Airlock

Volume	80 ft^3
Atmosphere	same as cabin when cabin is occupied
Number of cycles	120 during 45 day mission
Atmosphere Recovery	Down to 1 psia

e. Thermal Properties

Cabin heat flux - 76 Btu/hr. in for lunar day maximum

428 Btu/hr. out for lunar night minimum (111% of Figure 11-4 of Reference 3-2)

Cabin ECS radiator - $\alpha_s/\epsilon = 0.05/0.84$

specific coating weight (0.15 lb/ft^2)
maximum area - as required, maximum width = 12 ft

Internal cabin dissipation (from Reference 3-2 and power users)

From LSS to liquid loop	3240	
From LSS to cabin air		3416
From equipment to liquid loop	1636	
From equipment to cabin air		409
	4876 Btu/hr	3825 Btu/hr

Dormant cabin electric heat load - 100w (from RTG)

3.1.3.3 Lunar Suit

Design	Litton/Apollo "hard" type
Atmosphere	Same as cabin
Power requirements	50w, continuous
Heat rejection	Uses 1.8 pounds H_2O/hour, supplied from cabin
CO_2 removal	a) Regenerable cannisters processed by the cabin CO_2 closed loop system power for hypergolic and isotope systems.
	b) Open loop sieve exhausting to vacuum for the H-O engine and fuel cell power systems.

3.1.3.4 Roving Vehicle Electrical Power

a. Dormant vehicle RTG

Power Output - 100 watts electrical (all dc)

Location - On side of cabin (Reference 3-1)

Radiation criteria (25 rem total, 50% contribution by radiation sources)

b. Vehicle equipment power requirements

Experiments (from Reference 3-1, 3-3)	200 to 3400 watts
Life support (from Reference 3-2) and environmental control	1749 watts (closed loop CO_2) 965 watts (open loop CO_2)
Lights	20 watts internal 80-180 watts/external
Locomotion drive (6 motors)	300-2300 watts (7kw momentary)
Navigation	140 watts
Communications and data management	140 to 340 watts (Reference 3-4)
Suit battery charging	50 watts (2 suits)

c. Power profile

Figure 3-2 and 3-3 depict the rover power requirements for a six day period. Note that the profile repeats every three days. Table 3-1 is a listing of the power requirements and the energy for the three day cycle. Section 3.1.5 describes the power users.

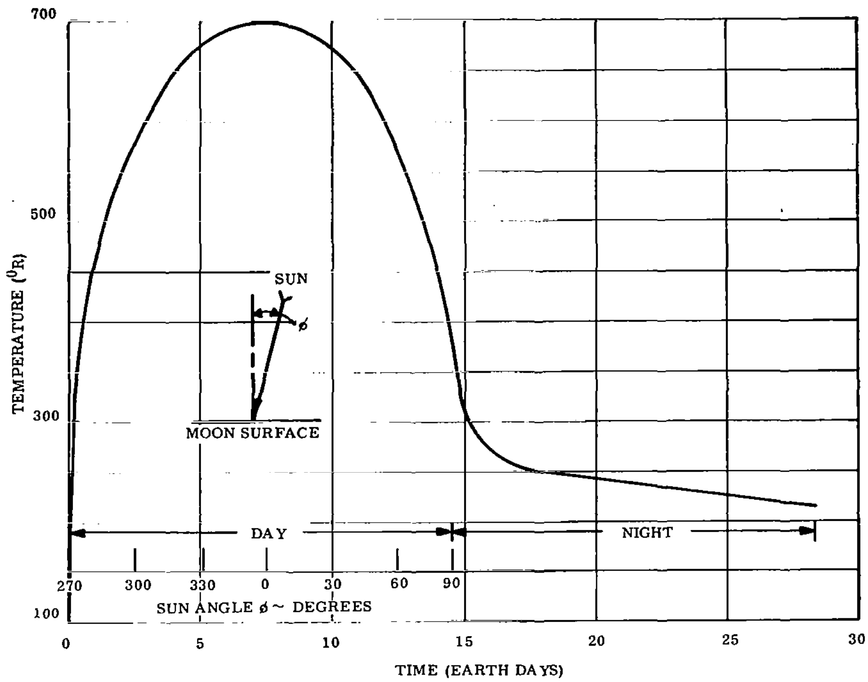


Figure 3-2. Estimated Moon Surface Temperature History for an Area Near the Equator

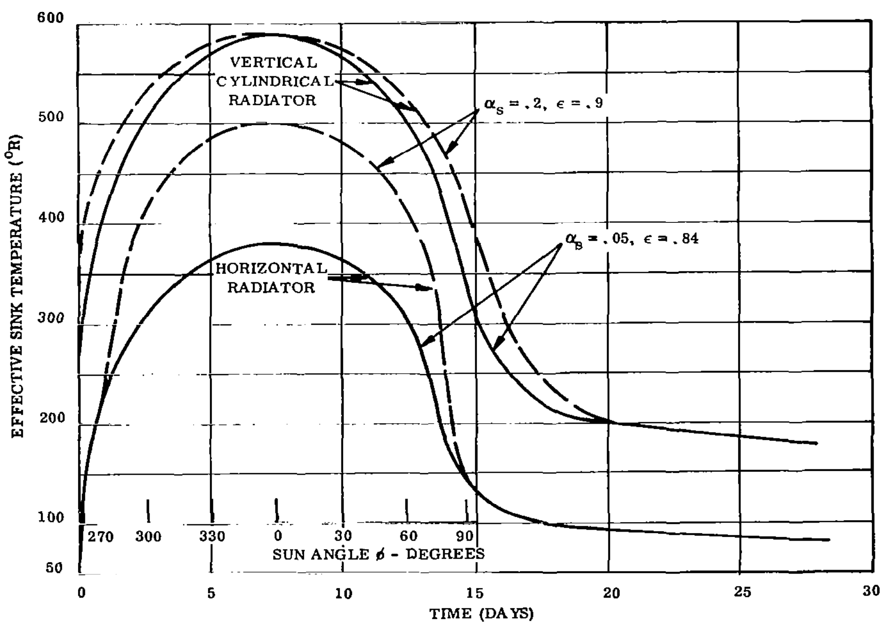


Figure 3-3. Estimated Effective Lunar Sink Temperature

d. Power division

1/3 28 volts 2/3 400 Hertz	dynamic systems	50% 28 volts 50% 400 Hertz	Fuel cell

e. Power Conversion Efficiencies

Efficiency	92%	
Efficiency	87%	
Frequency change efficiency	91%	From Reference 3-2
Voltage regulation	90%	
Battery charge/discharge efficiency	70%	

3.1.4 LUNAR ENVIRONMENT MODEL

The major elements of the lunar environment model are described in Reference 3-2. For the purposes of this study it has been assumed that the lunar roving vehicle is located in a dish shaped crater with the rim at a large distance (relative to the dimension of the vehicle) and 9 degrees above the radiator horizontal. In the aforementioned reference, mention was made of the fact that the lunar surface interacts with the vehicle and in particular the presence of the vehicle on the lunar surface tends to raise the local lunar surface temperature. For the purposes of this study, it is assumed that the moon's surface is unaffected by the presence of the roving vehicle and that the moon's surface acts as an infinite temperature sink. This premise has been made in view of the fact that the lunar rover is in motion part of the time. The assumed lunar surface temperature behavior is shown in Figure 3-2. The effective sink temperature is shown in Figure 3-3. The derivation for this effective sink temperature can be found in Appendix A.

TABLE 3-1. ROVER POWER PROFILE (CLOSED LOOP CO_2)

Day		Time (Hr)	Level	Kw-Hr	Accum Kwh
1	a	2	2.25	2.25	2.25
	b	2	2.45	2.45	4.70
	c	2	4.65	9.3	14.00
	d	4	2.45	9.80	23.80
	e	3	4.65	13.95	37.75
	f	5	2.45	12.25	50.00
	g	8	2.15	17.2	67.2 [49.2 Open Loop]
2	a	1	2.25	2.25	2.25
	b	2	5.25	10.5	12.75
	c	3	2.45	7.35	20.10
	d	2	4.35	8.70	28.00
	e	3	2.45	7.35	36.15
	f	1	4.35	4.35	40.50
	g	4	2.45	9.80	50.30
	h	8	2.15	17.2	67.50 [49.5 Open Loop]
3	a	1	2.25	2.25	2.25
	b	3	5.85	17.55	19.80
	c	2	2.45	4.90	24.70
	d	e	4.65	13.95	38.65
	e	3	2.45	7.35	46.00
	f	2	4.65	9.30	55.30
	g	2	2.45	4.90	60.20
	h	8	2.15	17.2	77.40 [59.4 Open Loop]
			Average		[2.95 kw Closed Loop] [2.20 kw Open Loop]

3.1.5 MISSION PROFILE

The various power users on the lunar roving vehicle have been arranged in the power profile, Figures 3-4 and 3-5, from a guideline mission standpoint. These users, their intervals and durations are as follows:

Function	Power Level	Duty Cycle
Life Support Processes	1740w (closed loop) 965w (open loop)	Continuous Continuous
Locomotion	7000w 2300w 2460w (Night) 1400w	Momentary at locomotion start 5 hrs/24 hr period 2 hr } every third 3 hr } 24 hr period
Extravehicular Activities		5 hr/man/24 hr period
Experiments	200w 3200w	Continuous For 3 hours, once every 72 hour period
Lights	50w 200w 100w 400w	Day locomotion and sleep period Extravehicular activity during day Day stationary Maximum night locomotion
Navigation-TV-Command	140w 50w	Continuous during active periods During sleep periods
Suit Pack Charging	50w	19 Hours/24 hr period (When not in use)

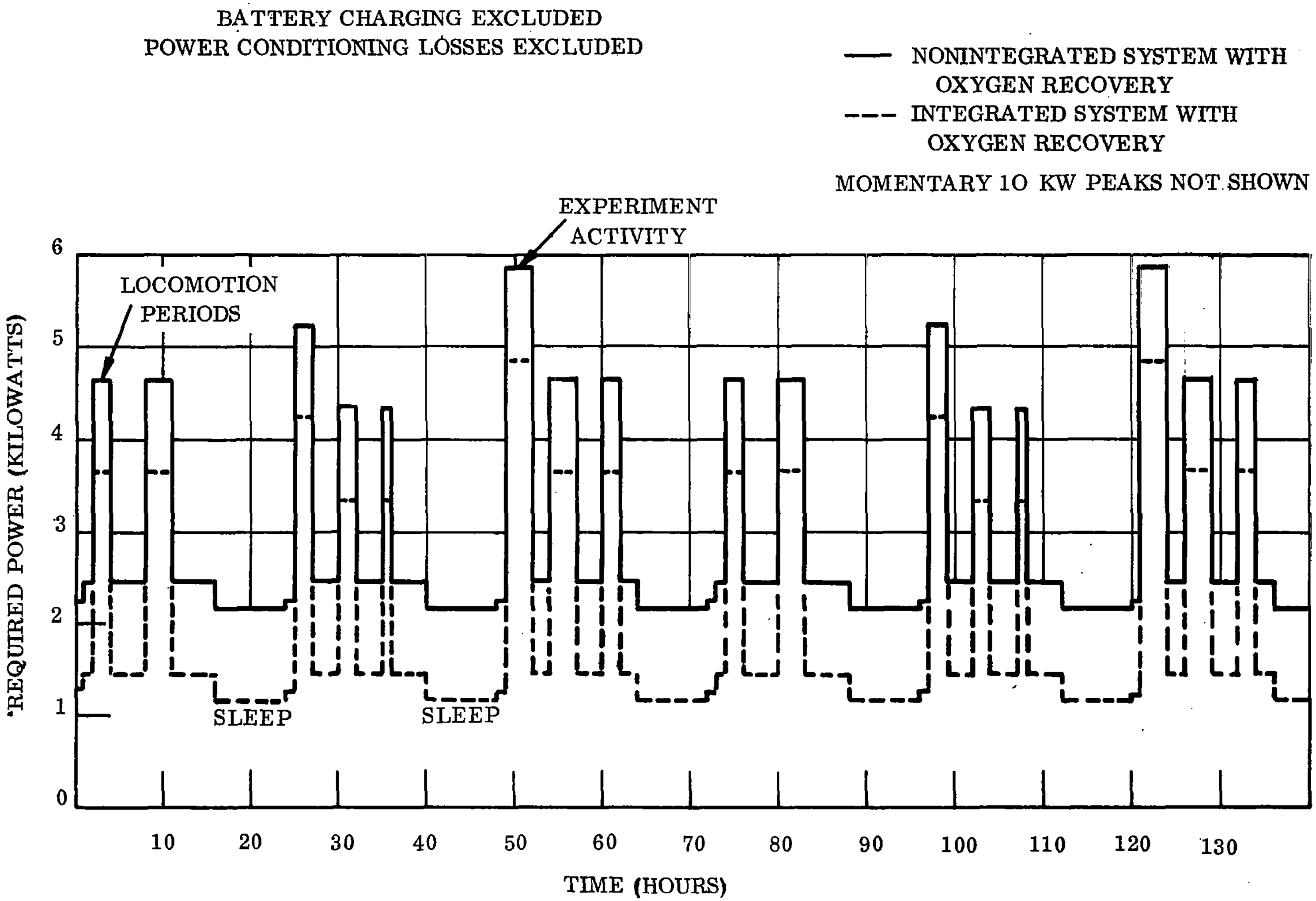


Figure 3-4. Roving Lunar Vehicle Power Profile-Closed Loop

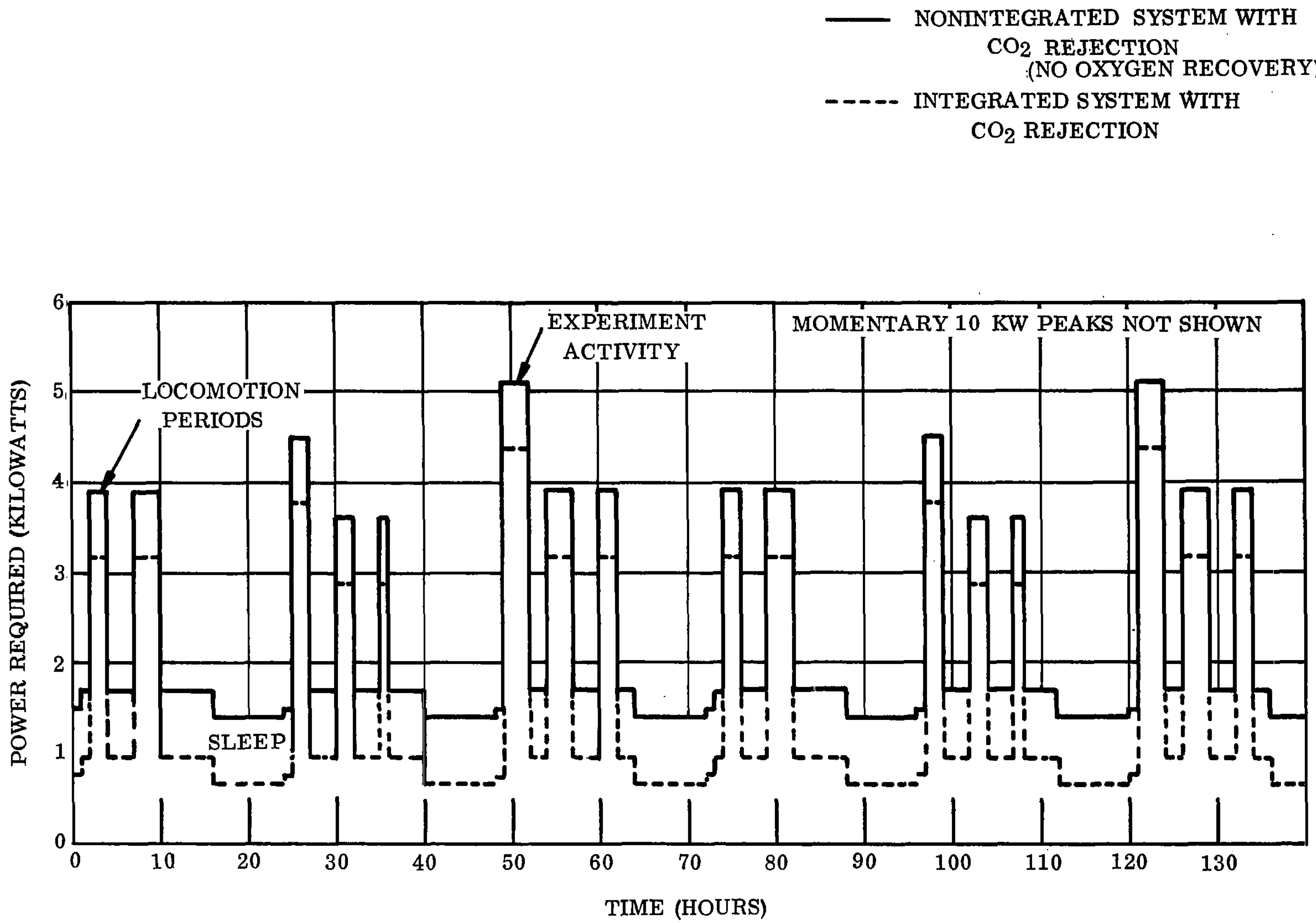


Figure 3-5. Roving Lunar Vehicle Power Profile-Open Loop

Function	Power Level	Duty Cycle
Communications, Telemetry, Data Management	140w	Continuous
	240w	During earth transmissions or computer operations
	340w	Both transmit and compute
Sleep Time		8 hr/24 hr period

3.1.6 DESIGN REQUIREMENTS AND RESTRAINTS

3.1.6.1 Vehicle Development and Launch Schedule

The anticipated initial lunar use of the lunar rover is 1972. The equipment at the component and subsystem level is assumed to be manufactured in 1970. The development philosophy is to utilize current concepts for equipment design.

3.1.6.2 Redundancy Requirements

Critical components shall be duplicated for increased success probability. Alternatively, where two or more identical "critical" components are required by design, then $3\frac{n}{2}$ components shall be incorporated. A critical component is defined as one that is necessary for astronaut survival. The necessary power to operate the life support equipment is of course also necessary for survival.

3.1.6.3 Isotope Source

The primary power system isotope heat source shall be Pu_{238}. Isotope fuel availability and cost are not investigated in this study.

The source, heat exchanger, and shielding are analyzed to the extent of determining weights and configuration. System considerations should be made for prelaunch and flight heat rejection. A preliminary design of thermal control of the source during dormant and active lunar periods are made to determine size, weight and location requirements. The shielding requirements are based on the dose defined in Section 3.1.3.4 as a maximum for a mission.

The RTG secondary power source shall be analyzed and designed within the same constraints. The combined radiation for the RTG and isotope sources shall be considered for shield design purposes for the cases when both appear on the roving lunar vehicle.

3.1.6.4 Life Support System Equipment

The life support equipment design is based on the two-man lunar shelter studied under NAS3-6478 (Reference 3-2).

An extravehicular suit (one for each man, 2 total) and its life support functions is incorporated into the life support system. The carbon dioxide produced by the astronaut is processed to recover oxygen for the Brayton, Rankine and hypergolic power systems, and is removed from the air and exhausted from the vehicle for the hydrogen-oxygen and fuel cell power systems. The environmental control of the cabin air removes portions of the heat dissipated in both the life support equipment and the electronic equipment (Section 3-2). The average metabolic data for a 24-hour period (suit-5 hours, and cabin -19 hours) are as follows:

Oxygen Consumption	2.78 lb	Input
Water Allowance	9.32 lb	Input
Food (dry)	2.00 lb	Input
Carbon Dioxide	3.39 lb	Output
Urine Water	3.30 lb	Output
Fecal Water	0.37 lb	Output

Urine and fecal solids	0.32 lb	Output
Respiration and Perspiration Water	6.70 lb	Output
Metabolic Water	1.05 lb	(Produced)
Latent and Sensible Heat	16,800 Btu	(8,000Btu rejected in suit)

3.1.6.5 Power System Design

The hydrogen-oxygen, hypergolic and fuel cell power systems design are based on information supplied by NASA. Equipment size and weight estimates are based on linear scaling for the two engines, and module additions for the fuel cell. In consonance with References 3-1 and 3-3 concepts, the power system will be carried on a two-wheeled trailer module. Also from Reference 3-3, the basic trailer module is limited to 3000 pounds, of which 600 pounds is basic mobility equipment and supports. For power systems weighing appreciably over 2400 pounds, an additional trailer module (s) will be required. The power required for each trailer module is 800 watts during locomotion and the clearance between the wheels will be 8 feet.

The isotope power systems designs are to be based on the work conducted on NAS3-6678 (Reference 3-2) where applicable. Scaling can be used for certain components to arrive at weights and sizes for the Rover application. The location and weight requirements for the chemical systems are also applicable to the isotope systems.

Analyses are made to determine the peak power provisioning requirements, considering the desirability of power system equipment versus use of batteries.

3.1.6.6 Heat Rejection

Rejection of thermal energy from the cabin and power systems is analyzed. The chief method of rejection will be radiation. Suit cooling is by evaporation of water, while the water produced by the fuel cell and hydrogen-oxygen engines can be used for some peak load cabin cooling.

Radiator configurations for the lunar shelter studies on contract NAS 3-6478, Reference 3-2, were re-examined in the light of roving vehicle requirements. Concepts with promise are analyzed to determine suitable radiator-heat exchanger shapes, sizes and weight.

The lithium bromide-water absorption refrigeration system studied in Reference 3-2 is to be applied to the lunar rover for rejection of cabin heat and compared with direct rejection methods.

Radiator analysis shall consider the surface condition with 9 degrees shading. The shading is considered equivalent to placing the vehicle in a "dish" with the dish rim at that angle above the horizontal all around the vehicle (Section 3.1.4).

3.1.6.7 Thermal Integration

The life support and power system are examined following their design and life support electrical power replaced with power system rejected heat, wherever possible. The integrated systems weight, shape and power requirements were determined. The philosophy was to use practical designs, emphasizing maximum utilization of the power source energy.

3.2 SYSTEMS EVALUATION AND COMPARISON

3.2.1 METHOD

A goal of the study was to select a power system-life support system combination from the possibilities in the design guidelines with the selected system the one with the best merit for the two man 45 day lunar roving vehicle. Thermal integration of the power and life support system provides both advantages and penalties to the vehicle so the approach was to study both nonintegrated and integrated vehicles. Fuel cells had previously been selected for the 14 day MOLAB mission and were regarded as the state-of-the-art power system for mobile lunar laboratories. Thus the fuel cell provides a standard for comparison for the more advanced systems. Specific evaluation areas were:

a. Gross electrical power requirement
b. Vehicle power and energy utilization
c. Quantity of isotope
d. Contribution to total vehicle weight
e. Thermal integration penalties to power and life support systems.
f. Effects on vehicle configuration and performance
g. Adaptability to evolving mission requirements
h. State of system development

Study and evaluation of cost were specifically excluded from scope of the study though cost is indirectly included in terms of weight, development and isotope quantity.

With the applicable performance areas defined, a method of determining their relative importance and applying them to the different power systems on a common basis was required. Previous thermal integration studies (References 3-2 and 3-5) assigned a weight factor to each performance area and a Degree of Performance (DOP) for each power system in each performance area. The product of the weight factor and DOP was then the Performance Index (PI). The sum of the PI's gave a Figure of Merit for the systems. Comparison of these Figures of Merit determined the best power-life support combination. This method was also used in this study to evaluate the different systems. The evaluation is shown in Table 3-2. Note that the PI had a mandatory zero minimum and a 10 maximum for each performance area.

3.2.2 SYSTEMS EVALUATION

The weight factors and the DOP for each performance area are discussed in the following paragraphs. The hypergolic power system is not included because of excessive weight. The data presented in Table 2-1 is useful in the evaluation, since direct comparison for several of the performance areas can be made.

TABLE 3-2. SYSTEM EVALUATION WORK SHEET FOR LUNAR ROVING VEHICLE (45-DAY MISSION)

NI = Nonintegrated
INT = Integrated
DOP = Degree of Performance
PI = Performance Index

PERFORMANCE AREA	WEIGHT FACTOR	POWER SYSTEMS													
		H_2-O_2				FUEL CELL		BRAYTON				RANKINE			
		NI		INT		NI		NI		INT		NI		INT	
		DOP	PI	DOP	PI	DOP	PI	DOP	PI	DOP	PI	DOP	PI	DOP	PI
a. Electrical Power Required	3	6	18	10	30	3	9	1	3	7	21	0	0	7	21
b. Overall System Utilization (1) Energy	3	0	0	5	15	10	30								
(2) Power	3							6	18	10	30	0	0	2	6
c. Isotope Quantity	5	10	50	10	50	10	50	4	20	6	30	0	0	2	10
d. Power System Weight	7	0	0	3	21	5	35	10	70	10	70	10	70	10	70
e. Thermal Integration Penalties	2	10	20	0	0	10	20	10	20	6	12	10	20	8	16
f. Effects on Vehicle Configuration and Performance	5	0	0	1	5	3	15	6	30	7	35	9	45	10	50
g. Adaptability to Evolving Mission Requirements	3	0	0	2	6	5	15	10	30	8	24	10	30	9	27
h. System Development	4	3	12	0	0	10	40	5	20	3	12	6	24	4	16
FIGURE OF MERIT			100		127		214		211		234		189		216

3.2.2.1 Gross Electrical Power Requirement (Weight Factor = 3)

This factor is a measure of thermal integration in that an integrated system would require less electric power than one which is nonintegrated. It is also a measure of power system parasitic power and the required conditioning of the raw output to make it compatible with the rest of the vehicle systems. This is then a significant item in evaluation, although not the most important.

The integrated hydrogen-oxygen engine with open-loop life support requires the least electrical power, 1.72 kw. Direct 400 Hertz generation of alternating current and variation of engine output to meet load demands directly are principal contributors to this system's low power requirements. The next two systems are nearly equal, with the integrated Brayton system requiring 2.43 kw and the integrated Rankine system 2.47 kw. The life support system requires slightly more power when a Brayton system is used, but the Rankine System must have its ac power conditioned prior to use. Both battery charging and CO_2 processing (electrolysis of water) increase the power requirements of these systems over the H_2-O_2 engine. Each thermally integrated system requires less power than the nonintegrated systems.

The nonintegrated Hydrogen-Oxygen engine supplies power more efficiently than the other nonintegrated systems for reasons identical to those for the integrated system; these are direct 400 Hz ac generation and variable output capability without penalty. The fuel cell, although combined with an open loop life support system (LSS), required (1) dc to ac conditioning to produce ac for locomotion; (2) battery charging to accommodate peak demands; and (3), parasitic power to maintain cell temperature. Its gross power of 3.15 kw ranks the fuel cell second in the nonintegrated category and fifth overall. The nonintegrated Brayton System ranks next at 3.51 kw. The electrolysis of water in the closed loop oxygen recovery system is the major additional contribution over that of the fuel cell. The nonintegrated Mercury Rankine system requires the most power. Its ac to dc conditioning requirement elevates its gross power to 3.71 kw.

3.2.2.2 Vehicle Power and Energy Utilization (Weight Factor = 3)

This performance area is a measure of: (1) power system thermal efficiency; and (2) power system size reduction with thermal integration. The chemical systems are truly energy limited because of fuel consumption, while the isotope systems are powered limited. Therefore, these two types of thermal sources are rated separately within this performance area, each against a zero to 10 scale.

For the chemical systems the best system in this category is the fuel cell. Its high thermal efficiency of 0.44 is much greater than the Hydrogen-Oxygen Engine with an efficiency (including thermal heat to life support) of 0.31. The lowest scoring system in this category is the nonintegrated Hydrogen-Oxygen Engine, with an efficiency of 0.22. (For a comprehensive numerical listing, see Table 2-1.)

The minimum source size for the isotopic power systems is the integrated Brayton cycle with a thermal efficiency (including life support thermal heat) of 0.20. Next in the rankings is the nonintegrated Brayton cycle with an 0.16 efficiency. The Mercury Rankine cycles follows; its integrated system efficiency (including life support thermal heat) is 0.10. This is followed by the nonintegrated Rankine cycle with an efficiency of 0.08. Higher efficiencies could be realized for both isotopic cycles if temperature and pressure constraints were removed. This study is conservatively oriented with stress placed on the use of developed hardware concepts.

3.2.2.3 Isotope Quantity (Weight Factor = 5)

This criterion measures the amount of isotope required for use by the lunar roving vehicle power system. From available data (such as References 3-6 and 3-7) Pu_{238} is both scarce and expensive, making the amount required for a power system an important area for consideration.

The chemical systems all rate best in this category, since their isotope complement is less than 3000 watts, in the RTG. The integrated isotope Brayton system ranks next, with a 14 thermal kilowatt requirement (exclusive of the RTG). The nonintegrated Brayton cycle requires 18.5 thermal kilowatts, following the integrated system. The integrated and nonintegrated Mercury Rankine cycles require 29 and 36 thermal kilowatts respectively, and rank lowest in this category. It should be pointed out that radiation is not a primary consideration with the isotopes for this case. The majority of the 12-1/2 rem total dose for the 45 day mission comes from the RTG. By comparison the RTG dose for the 14-day MOLAB is 30 rem (Reference 3-7).

3.2.2.4 System Weight (Weight Factor = 7)

For launch considerations the significant factors are payload size and weight. Most present studies use $5,000 per pound as a working number for payload delivery to the lunar surface. The full capability of each Saturn V will be used to deliver equipment and supplies to the lunar surface, thus light lunar roving vehicles would be advantageous in providing additional space on the booster.

Another weight-related factor is the bearing stress on the lunar surface. A heavier power system in this case means additional trailer modules to allow sufficient wheel bearing area. Because of the importance of system mass and because of the wide range between systems, this performance area was assigned a weight factor of 7.

The integrated isotopic systems were found to be the lightest, the basic Brayton system weighing 1540 pounds and the Rankine system 1730 pounds. To these weights are added 750 pounds of water for suit cooling and 600 pounds of locomotion equipment and structure to arrive at trailer module weights of 2890 pounds and 3080 pounds respectively. To obtain the contribution to total vehicle weight, 300 pounds of life support equipment in the cabin must be included. Since the number of trailers vary for the different power systems while the LSS weight is relatively constant, the weights hereafter referred to will all be trailer module weights. The next heavier systems were the nonintegrated

Brayton cycle and Rankine cycle, weighing 3190 lb and 3290 lb respectively. Previous studies have shown that the Rankine power system was considerably lighter than a Brayton powered system for the same application. Safety, re-entry integrity and criticality studies that have since been conducted indicate that weight is fairly dependent on heat source size (in thermal kilowatts). The Brayton cycle source is therefore lighter than the Rankine system because of the former's higher thermal efficiency for these designs. Considering the chemical systems, however, the differences among isotopic systems are so small (maximum 400 lb) when compared to the large weight increase of the next system in rank (fuel cell at 6490 lb) that the isotopes can all be scored with a degree of performance of 10. The fuel cell power system weight was found to be 6490 lb, more than twice the Brayton or Rankine cycle, so a degree of performance of 5 was assigned. The integrated Hydrogen-Oxygen engine was next at 8730 lb and was rated with a 3. In last place was the nonintegrated hydrogen-oxygen engine at 12,500 lb.

3.2.2.5 Thermal Integration Penalties to Power and Life Support Systems (Weight Factor = 2)

This performance area is to evaluate the penalties of thermal integration on the power and life support systems. Additional equipment is required to accomplish thermal integration and certain interdependences and modifications of controls are created by combining two systems. However, development of equipment would appear to be straightforward using conventional designs, so that thermal integration does not place a large penalty on any of the power systems and is of small concern to life support equipment. Hence, the relatively small weight factor of 2 was assigned to this performance area. Two methods of thermal integration are possible for the power systems considered. One uses flexible stainless steel woven hoses with an inner layer of Dupont H film to connect the trailer modules and the cabin. The other would use a heat pipe from a boiler on the back surface of the D configuration 100 watt RTG mounted in the cabin. Both of these concepts are examined in Section 4.5. Use of the RTG would make the thermal integration independent of the power system as to the kind of equipment required. It would only serve to reduce power system size. The use of flexible hoses has been the conventional concept of this study and the evaluation will be made on that basis.

The nonintegrated systems receive a degree of performance of 10. The integrated rankine system ranked next because the amount of heat removed by the LSS is a small portion of the total, and the power system is not too dependent on this heat removal for operation since its rejection temperature is relatively high. The integrated Brayton power system ranked lower because the power system is more dependent on the thermal integration heat removed, and the lower available heat temperature is nearer to the lower limit required by the desorption of CO_2 processing equipment. The integrated Hydrogen-Oxygen engine scored lowest in this category because connections were required between the three trailer modules and the cabin instead of just with a single trailer module. It also required life support equipment timed operation because the heat available during the sleep period is not sufficient to operate all processes at once.

3.2.2.6 Effects on Vehicle Configuration and Performance (Weight Factor = 5)

The various power systems, when applied to the lunar roving vehicle, create vehicle systems with different capabilities, despite efforts to provide power system design concepts which are equivalent from a mission standpoint. It is expected that a lunar rover with the greatest capability (range, speed, maneuverability, power, etc.) would be most desirable to use. Two major considerations are the number of trailer modules and radiator size. This performance area was deemed to be important in determining the value of the different power systems and was therefore assigned a Weight Factor of 5.

The system ranked first in this performance area is the integrated Mercury Rankine system. It requires but one trailer module and has the minimum radiator area. The nonintegrated Rankine power system follows with still one trailer, but an additional 30 square feet of radiator area. The integrated and nonintegrated Brayton systems ranked next, with degrees of performance of 7 and 6. Only one trailer module is required, but with radiator areas much larger than the Rankine system is. The next power system is the fuel cell with a degree of performance of 3 because of its two trailer module requirement. The integrated and nonintegrated Hydrogen-Oxygen engine systems are ranked last, because they require 3 and 4 trailer modules respectively.

3.2.2.7 <u>Adaptability to Evolving Mission Requirements (Weight Factor = 3)</u>

This performance area measures the ability of the power system to:

a. Operate in different profile modes
b. Operate for periods longer than those required for a single 45-day mission
c. Shutdown and restart abilities

All of these factors are important for flexibility of the lunar base/roving vehicle complex. This study, however, is limited to a single 45-day mission with a specific estimated power profile, so these considerations were de-emphasized by assigning a Weight Factor of 3.

The nonintegrated isotopic systems are ranked highest in this area because:

a. The power system can operate for a long time. (Most equipment designed is for 10,000 hour minimum life.)

b. Restart after dormancy has been demonstrated many times.

c. The life support system operates with electric power, and the power system is independent of the LSS.

The integrated Mercury Rankine and Brayton power-life support combinations rank next for the same reasons as above, except that thermal power is supplied to the life support systems. The integrated Brayton System ranks slightly behind the other systems because of the dependence on the LSS for removing heat.

The fuel cell was given a degree of performance of 5. The lower score is based on the life limits of the cells themselves (which are not proven for beyond the 45-day mission) and the 100 pound-per-day weight penalty for extending the mission. The cells must also be maintained at some elevated temperature during shut down; or the startup energy requirements are high, and the warmup time will be extended. The fuel cell is somewhat more capable of handling fluctuating loads at the expense of specific fuel consumption. The Hydrogen-Oxygen Engines (integrated and nonintegrated) were ranked last because of their limited life expectancy.

3.2.2.8 State of System Development (Weight Factor = 4)

The power systems studied in the contract are in stages of development ranging from feasibility studies and demonstration to flight qualification and operation. This performance area measures the status of each system in terms of development time (and cost) required. A Weight Factor of 4 is assigned to this area.

The life support equipment is essentially common to all combinations under consideration and consequently can be left out in determination of degree of development. Most of the processes have operated at the laboratory level. Two areas needing further development are methane pyrolization and the magnesium oxide system for CO_2 removal. The integrated systems are arbitrarily graded as less developed than the nonintegrated systems, because the work has not actually been carried out to prove design concepts through testing.

The fuel cell is the power system in the most advanced stage of development. Low temperature cells have operated successfully on manned Gemini flights. Although the high-temperature cells have not been flight proven, they could be replaced (without significant differences) by low temperature units, should the need arise.

The Rankine power system equipment is next most advanced in development. Turbo machinery and auxiliary equipment have been extensively tested by TRW. The direct boiling of mercury in the source is not anticipated to be a design problem. Condensing radiator flow stability is simplified with the presence of lunar gravity; orientation should be consistent enough so that droplet collection would always occur at the condensed end.

The Brayton cycle ranks next; equipment is under test at present. Gas turbine history has been excellent. Based on that experience Brayton cycle problems that may occur in development should be straightforward. Intermediate loop heat removal from the isotope source should provide as reliable a method as the previous direct gas exchange examined on NAS 3-2799 and NAS 3-6478.

The H_2-O_2 engine (ranking last) has operated, but not on long-duration runs. Oxidizer injection is critical and an operating problem. Lubrication and its contamination of exhaust products is another area requiring further work. The device is small, and concurrent development utilizing several operating engines would probably not be unduly expensive.

3.2.3 INTEGRATED SYSTEMS SELECTION

The resulting integrated-systems combination with the highest Figure of Merit is integrated Brayton-closed life support. Its Figure of Merit is 234, compared with the integrated Rankine cycle Figure of Merit of 216. The significant advantages of the Brayton cycle over the Rankine cycle are its higher thermal efficiency (for this application) and its resulting lower isotope requirements for a heat source. The integrated Brayton cycle scores higher than the nonintegrated Brayton cycle because of:

a. Reduced electrical power requirements
b. Increased thermal power utilization
c. Reduced isotope requirements

The Rankine cycle is a major source of electrical power on earth and has high thermal efficiencies in these applications. If future technologies are able to improve the thermal efficiencies of the Rankine system in the 2-3 kw range, then it might prove the equal of a Brayton system in an evaluation similar to the one conducted in this study.

The integrated Hydrogen-Oxygen Engine, although an improvement over the nonintegrated system, ranked lower than the isotopic systems. Weight and adaptability were major shortcomings of this system for the 45-day mission.

3.2.4 INTEGRATED SYSTEM - FUEL CELL COMPARISON

Both the integrated Rankine and Brayton systems ranked above the fuel cell system in the Figure of Merit rating so both were included in this comparison. Performance in the areas of weight, electrical power requirements, and radiator area is as follows:

Power System and Life Support System	Fuel Cell and Open Loop LSS	Integrated Brayton and Closed Loop Life Support System	Integrated Ranking and Closed Loop Life Support System
Figure of Merit for 45-day Mission	214	234	216
Weight (lb)	6674	3208	3386
Total Radiator Area (sq ft)	230	379	267
Electrical Power Requirements (kw)	3.15	3.43	2.47

The most significant difference in the systems is the larger weight of the fuel cell system which results from the fuel storage requirements. Using the above criteria the integrated isotope systems are superior to the fuel cell.

The fuel requirement for the fuel cell is a function of the length of the mission and, as shown in Figure 2-1, the relative merits of the systems is time dependent. Tables 3-3 and 3-4 contain the evaluations made for 22-day and 90-day missions. For the 90-day mission the fuel cell system weight included approximately 11,000 pounds of fuel which makes this system unattractive for this application.

TABLE 3-3. SYSTEM EVALUATION WORK SHEET FOR LUNAR ROVING VEHICLE (22-DAY MISSION)

PERFORMANCE AREA	WEIGHT FACTOR	POWER SYSTEMS													
		H_2-O_2				FUEL CELL		BRAYTON				RANKINE			
		NI		INT		NI		NI		INT		NI		INT	
		DOP	PI	DOP	PI	DOP	PI	DOP	PI	DOP	PI	DOP	PI	DOP	PI
a. Electrical Power Required	3	8	24	10	30	3	9	1	3	7	21	0	0	7	21
b. Overall System Itilization															
1) Energy	3	0	0	5	15	10	30								
2) Power	3							6	18	10	30	0	0	2	6
c. Isotope Quantity	5	10	50	10	50	10	50	4	20	6	30	0	0	2	10
d. Power System Weight	7	0	0	5	35	9	63	9	63	10	70	9	63	10	70
e. Thermal Integration Penalties	2	10	20	0	0	10	20	10	20	6	12	10	20	8	16
f. Effects on Vehicle Configuration and performance	5	0	0	2	10	10	50	5	25	6	30	8	40	9	45
g. Adaptability to Evolving Mission Requirements	3	0	0	3	9	5	15	10	30	8	24	10	30	9	27
h. System Development	4	3	12	0	0	10	40	5	20	3	12	6	24	4	16
FIGURE OF MERIT			106		149		277		199		229		177		211

NI = Nonintegrated
INT = Integrated
DOP= Degree of Performance
PI = Performance Index

TABLE 3-4. SYSTEM EVALUATION WORK SHEET FOR LUNAR ROVING VEHICLE (90-DAY MISSION)

PERFORMANCE AREA	WEIGHT FACTOR	H_2-O_2				FUEL CELL		BRAYTON				RANKINE			
		NI		INT		NI		NI		INT		NI		INT	
		DOP	PI	DOP	PI	DOP	PI	DOP	PI	DOP	PI	DOP	PI	DOP	PI
a. Electrical Power Required	3	4	12	10	30	2	6	2	6	9	27	0	0	9	27
b. Overall System Utilization															
1) Energy	3	0	0	3	9	10	30								
2) Power	3							6	18	10	30	0	0	2	6
c. Isotope Quantity	5	10	50	10	50	10	50	4	20	6	30	0	0	2	10
d. Power System Weight	7	0	0	1	7	4	28	10	70	10	70	10	70	10	70
e. Thermal Integration Penalties	2	10	20	0	0	10	20	10	20	6	12	10	20	8	16
f. Effects on Vehicle Configuration and Performance	5	0	0	1	5	2	10	6	30	7	35	9	45	10	50
g. Adaptability to Evolving Mission Requirements	3	0	0	1	3	3	9	10	30	8	24	10	30	9	27
h. System Development	4	2	8	0	0	10	40	6	24	4	16	7	28	5	20
FIGURE OF MERIT			90		104		193		218		244		193		226

NI = Nonintegrated
INT = Integrated
DOP = Degree of Performance
PI = Performance Index

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SECTION 4
LIFE SUPPORT SYSTEMS

4.1 INTRODUCTION

Closed cycle life support processes, where oxygen and water are reused several times by astronauts, offer advantages for long term missions. This process equipment requires, in general, heat energy to function. For this study, equipment sized for two men and used in a lunar roving vehicle is of particular interest. Initially, analyses are required to develop design concepts and arrangements for the two-man roving vehicle application. Then the endothermic power requirements for the closed cycle equipment are identified as to temperature level and amount of heat. The process equipment is re-analyzed to determine the effects of using available waste heat from the electrical power system. The final result is nonintegrated and integrated system designs.

4.1.1 SUBSYSTEMS AND SYSTEM COMBINATIONS

Life support subsystems considered for the lunar roving vehicle consist of:

a. Cabin atmosphere cooling during dormant and active periods.

b. Carbon dioxide removal from the cabin atmosphere by the Sabatier process. Recovery of the oxygen by the Sabatier method is not required when the vehicle utilizes a hydrogen-oxygen fuel cell or H-O engine power system. The amount of oxygen used to produce process power would exceed the amount of oxygen used directly by the men. The carbon dioxide produced by the suited men, when they are outside on the lunar surface, is collected and returned to the vehicle for processing (when oxygen recovery is utilized).

c. Water recovery from urine by distillation and pyrolysis.

d. Watter recovery from wash water by distillation.

e. Food preparation.

Solid waste management will consist of a bag type apparatus which is assumed to not affect the vehicle power requirements.

The four system categories to be considered are:

a. Nonintegrated system without oxygen recovery.
b. Nonintegrated system with oxygen recovery.
c. Integrated system with oxygen recovery.

This investigation will consider, in addition to active cabin operation, dormant period thermal control and transient periods such as start-up. The detailed calculations for analyses conducted in this section may be found in Appendix B.

4.1.2 GUIDELINE INFORMATION

Crew Size:	Two men
Mission Length:	45 days
Dormant Period:	30 days after Lunar Landing
Transit Period:	3 days
Cabin Pressure:	360 mm Hg (normal)
O_2 Partial Pressure:	180 mm Hg
N_2 Partial Pressure:	180 mm Hg
CO_2 Partial Pressure:	3.8 mm Hg
Relative Humidity:	50%
Normal Cabin Temperature:	72°F dry bulb
Ventilation Rate-Cabin:	35 CFM/Man
Ventialtion Rate-Suit:	50 CFM/Man*
Suit Pressure Drop:	5.0 inches W.G. @ 360 mm Hg
Cabin Volume:	245 cu ft
Man Lock Volume	80 cu ft
Egressions:	120 in 45-day mission. Both men will exit together for 5 hours per day.
Time in Suit:	450 hours total.

*During minimal work while in suit. However when the backpack is utilized, liquid cooled under garments are utilized to remove metabolic heat.

Each crew member will occupy the cabin for 19 hours and the suit 5 hours per day. Therefore 79.2% of the time will be at normal metabolic output and 20.8% of the time will be at a substantially higher metabolic output. The overall average metabolic data per man day is as follows:

Oxygen Consumption	2.78 lb
Water Allowance	9.32 lb
Food (Dry)	2.00 lb
Carbon Dioxide	3.39 lb (1)
Urine Water	3.30 lb
Fecal Water	0.37 lb
Urine and fecal solids	0.32 lb
Respiration and Perspiration water	6.70 lb (2)
Metabolic Water	1.05 lb
Latent and Sensible Heat	16,800 btu (3)

a. Approximately 1.55 lb is produced while man is in the suit. 1.84 lb is collected in the cabin.

b. Assumes that this water is collected in the suit and returned to the cabin system.

c. Approximately 8,000 btu will be dissipated while man is in suit. 8,800 btu is rejected to the cabin environmental control system.

A possible overall man-day water balance is shown in Figure 4-1. Note that water recovery efficiencies are given since the water balance in the lunary roving vehicle is more critical than for the MORL type vehicle or the lunar shelter. This is due to the large amounts of water consumed in the pressure suit back-pack for thermal control. The thermal control evaporates approximately 1.8 pounds of water per hour per man.

Gas Inventory Requirements

Open Loop Systems	Weight
Oxygen	348 lb
Nitrogen	48 lb

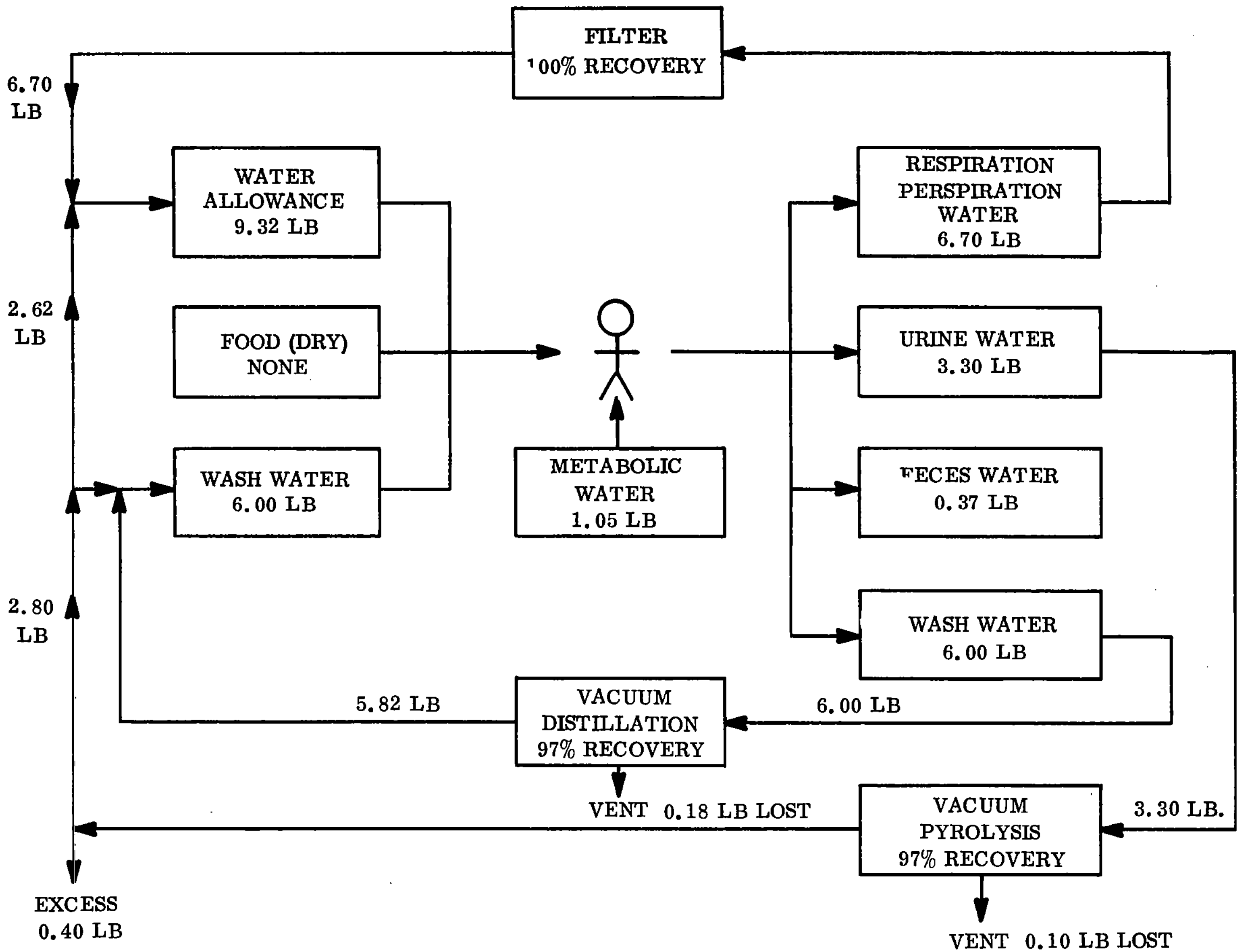


Figure 4-1 Possible Man-Day Water Balance - 19 Hours in Cabin and 5 Hours in Suit

Gas Inventory Requirements (cont'd)

Closed Loop Systems	Weight
Oxygen	80 lb
Nitrogen	48 lb

4.1.3 SUIT AND BACK-PACK DESCRIPTION

When the men exit from the lunary roving vehicle they will encounter probably the harshest environment to which man has yet been exposed. These include:

a. Both extremes of temprature.
b. Ultra low pressure
c. Full intensity of solar flux.
d. Cosmic radiation
e. Meteroid particles
f. Rugged terrain.

Two types of pressure suits are being developed to protect the men from these environments:

a. "Soft" suits which consist of:

1. Undergarment with cooling coils
2. Flexible pressure garment
3. Helmet, boots and gloves
4. Thermal insulation overgarment
5. Micrometeroid protective garment
6. Backpack portable life support system

b. "Hard" suits which consist of:

1. Undergarment with cooling coils
2. Rigid pressure garment with helmet boots and gloves attached
3. Back pack portable life support system.

The "hard" suit was selected for this study because of the inherent safety feature of metal construction. Also the suit requires less physical effort for mobility, has fewer accessories and may be operated at higher internal pressures if desired. Both types utilize a similar back pack.

a. The back pack portable life support system provides:

1. Environmental gas circulation and cooling
2. Oxygen for breathing
3. Liquid coolant circulation thru undergarment and cooling
4. Carbon dioxide control
5. Condensation and collection of metabolic water vapor
6. Emergency oxygen supply

Cooling for both the gas and coolant circuit is provided by an evaporative heat exchanger. Water is utilized as the expendable refrigerant and removes approximately 1000 btu per pound of water evaporated. The 1800 btu hr suit heat load consists of metabolic, electrical, solar flux and heats of absorption or adsorption of carbon dioxide. Water is evaporated at a rate of 1.8 pounds per hour to provide cooling.

The oxygen for breathing may be stored either cryogenically, super critically or at a high pressure. The lightest weight and smalles volume is the high pressure system which meters oxygen into the suit as required via a demand type pressure regulator.

Carbon dioxide may be either collected and stored or collected and jettisoned depending on whether oxygen recovery from the carbon dioxide is required. An absorption type system (molecular sieves) is utilized for merely collection and venting of the carbon dioxide. However when the carbon dioxide is to be collected and stored a more efficient absorption method is utilized.

The metabolic water is condensed in the evaporative heat exchanger and is collected and stored in wicks or sponges. A transpiration type evaporative heat exchanger may possibly utilize the metabolic water for cooling.

Emergency oxygen is provided to supplement the normal supply in emergencies such as component failures or extreme leakage of the suit.

4.2 DORMANT PERIOD ANALYSIS

Following vehicle launch from the earth's surface, cabin environment control must be provided through a 3-day flight and 30 days on the lunar surface, while the main power system is dormant. A 100 watt radioisotope thermoelectric generator supplies 100 watts to the cabin for vehicle maintenance activities. These activities include telemetry monitoring, communications, experiment maintenance, and pumps and fans for environment control. The method of control and its power requirements are described in the following paragraphs:

4.2.1 REQUIREMENTS AND ASSUMPTIONS

The following list represents the factors on which the dormant period analysis was conducted:

a. The worst case dormant period consists of:

    1. 3 days in transit between earth launch and lunar landing.
    2. 15 days exposed to the lunar day.
    3. 15 days exposed to the lunar nite.

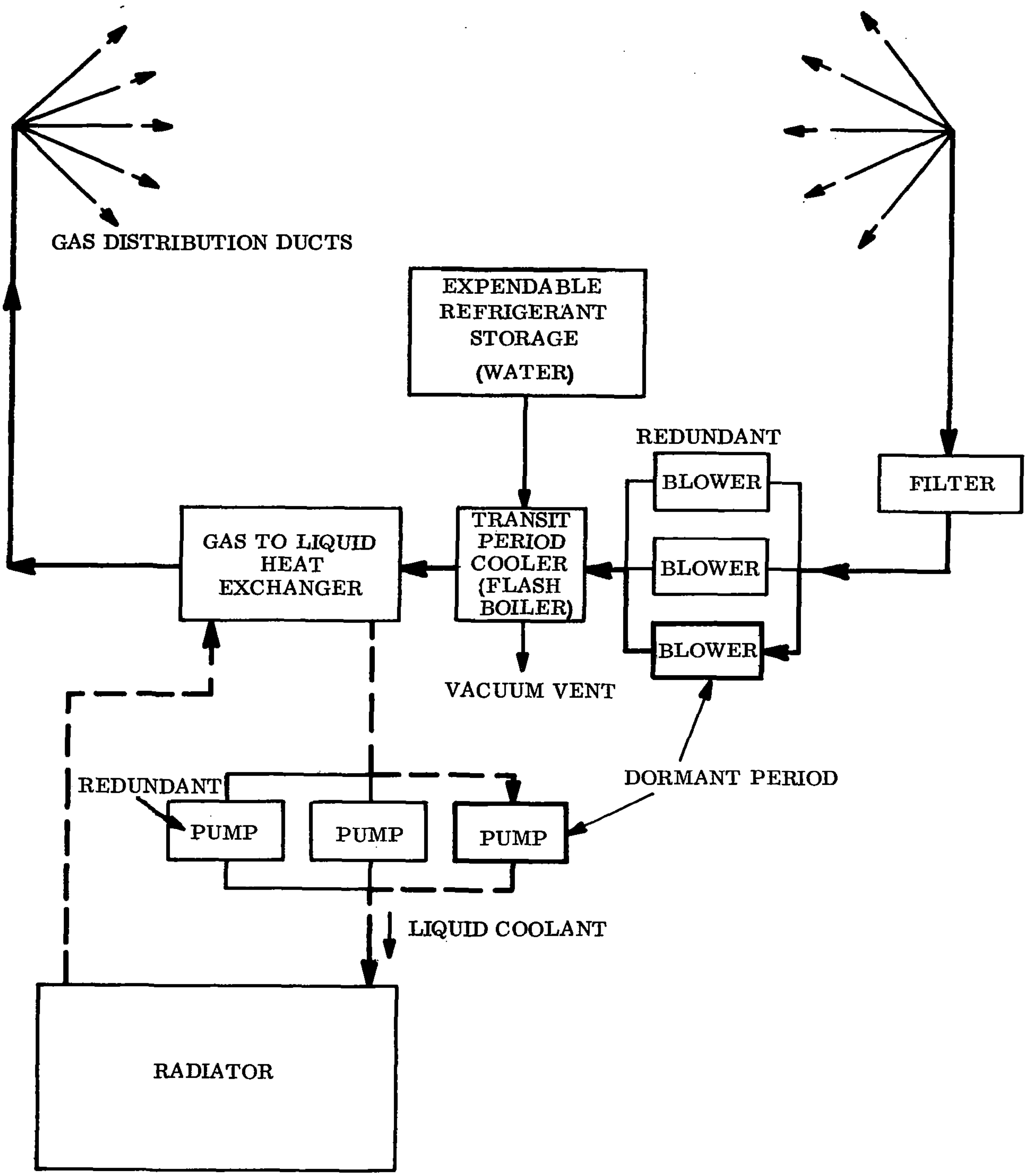


Figure 4-2 Block Diagram for Thermal Control During Dormant Period

b. During the 3-day transit period, there will not be an influx of solar heat.

c. Natural convection in the lunar environment is negligible.

d. The internal cabin temperature is maintained between 60 and 100°F.

e. The internal cabin pressure is maintained at 180 mm Hg pure oxygen. This arrangement permits forced convection thermal control, a lower pressure to reduce leakage and does not require auxiliary pressure controls for the dormant period. The same oxygen partial pressure sensor is thus utilized for both the dormant and active period.

f. All the 100-watt radioisotope thermoelectric generator (RTG) power is eventually dissipated as heat to the cabin.

g. The manlock is closed and has no active temperature or pressure control. Passive thermal control will possibly be provided by conduction through the walls.

h. A blower will circulate the cabin atmosphere to minimize hot spots, e.g., electrical components. See Figure 4-2.

4.2.2 TRANSIT PERIOD

During the three-day (72 hr) transit, the 100-watt RTG will release 7200 watts to the cabin. It is assumed that a shroud is employed during this period, thus the radiators cannot be depended on to reject this heat. The cabin has an assumed heat capacity of 800 btu/$^{\circ}$F. The temperature rise will then be in the order of 30°F for the three days and some local heating problems may occur. Utilization of the dormant period fan will alleviate hot spots but the total energy could still create some problems. It has been postulated that a ground takeoff average temperature of 75°F is probable. To stay beneath the 100°F limitation, approximately 4 pounds of water in a flash boiler heat exchanger will suit the needs (see Figure 4-2.) This boiler is also a possible extreme condition or emergency safeguard when the cabin is occupied. Details can be found in Appendix B.

4.2.3 LUNAR DAY

The maximum lunar day heat load is 100 watts electrical from the RTG and 20 watts heat leakage through the cabin walls. Using 180 mm Hg oxygen as a dormant period pressurant, it is found that 38 cfm flow will suffice to remove this heat energy with a gas Δ t of 90°F. The electrical power required will be 9 watts for the fan and 3 watts to circulate the liquid through the radiator. A more complete treatment of this analysis can be found in Appendix B.

4.2.4 LUNAR NIGHT

The cabin heat loss through the walls exceeds the RTG electrical energy input by about some 20 watts (maximum just prior to lunar dawn). This low level loss can be accounted for within the allowable temperature limits, or some RTG thermal heat could be utilized for temperature maintenance.

4.3 NONINTEGRATED SYSTEM WITH NO OXYGEN RECOVERY

Figure 4-3 illustrates the Life Support System block diagram for a nonintegrated (thermal) system in which oxygen is not recovered from the carbon dioxide.

Required quantities of oxygen are available from the hydrogen-oxygen fuel cell or combustion engine power system. Oxidizer stroage tunks note that the coolant pump and cabin atmosphere blower for the dormant period are included in the diagram.

4.3.1 CABIN ENVIRONMENTAL CONTROL

The cabin atmosphere consists of 180 mm Hg oxygen, 180 mm Hg nitrogen and small amounts of carbon dioxide, water vapor and trace gases. The blower circulates the cabin atmosphere through a particulate matter filter and then a heat exchanger which removes the required

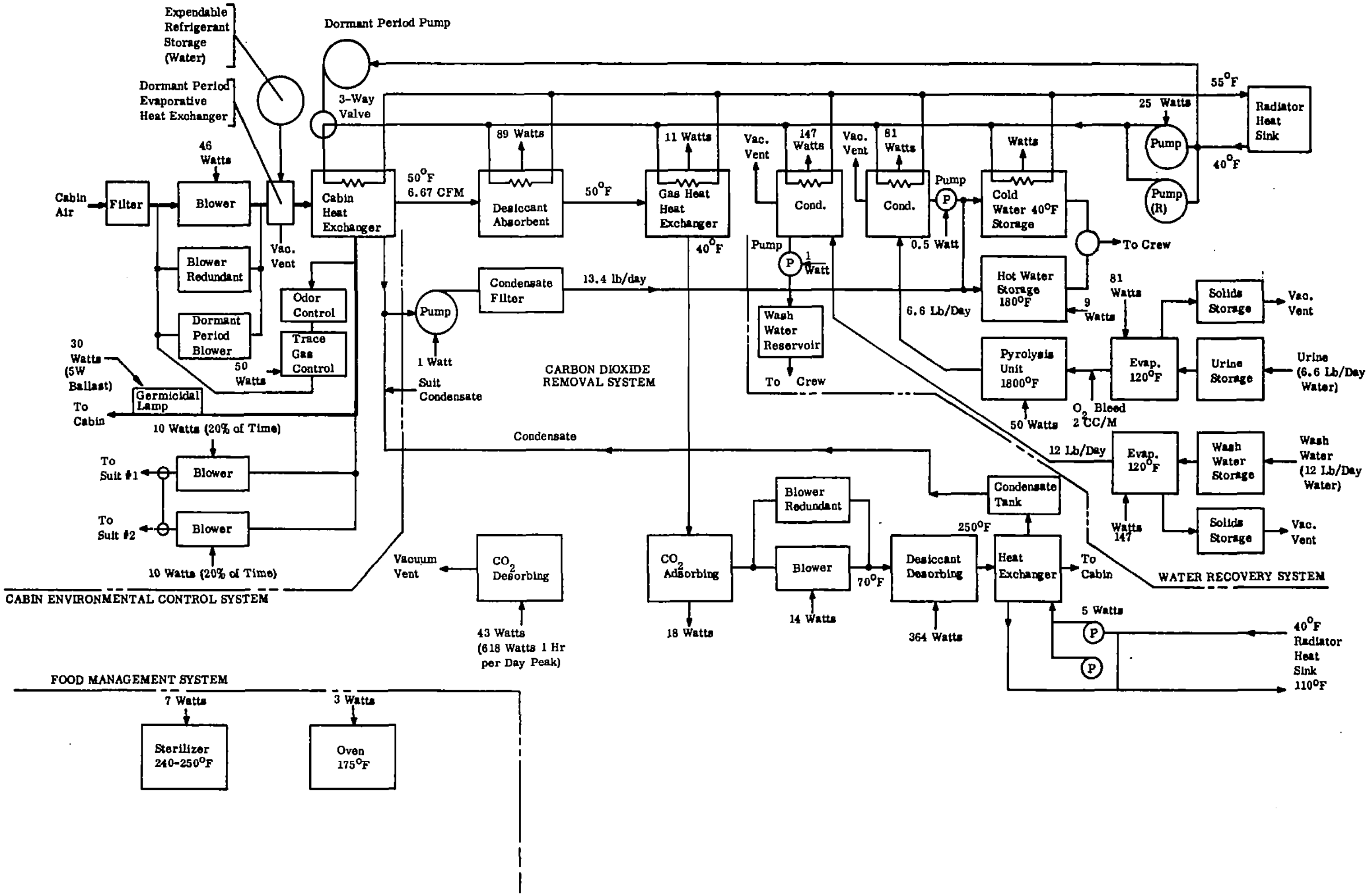


Figure 4-3 Block Diagram of Open Loop Nonintegrated Life Support System

latent and sensible heat. Most of the cooled and dehumidified atmosphere is returned directly to the cabin. The liquid coolant in the heat exchanger is a heat transport medium which is circulated to an external radiator where the heat is rejected to the spatial heat sink. The atmosphere and coolant circulation is critical to the crew survival, consequently a redundant blower and pump are provided. A bypass odor and trace gas control is located on the gas exit line of the heat exchanger. The odor control consists of the bed of activated charcoal which absorbs heavy hydrocarbons, e.g., mercaptans, indole, phenol, body odors, cooking ordors, etc. The trace gas control consists of a catalytic oxidizer (burner) which oxidizes carbon monoxide, hydrogen, ozone and methane into products such as carbon dioxide and water which are easily removed and controlled. The oxidizer is heated electrically and requires approximately 50 watts of power.

Bacteria floating or suspended in the cabin atmosphere are controlled by a germicidal lamp located in the air duct. The ultraviolet energy emitted from the lamp kills a large percentage of the bacteria in the atmosphere flow. The lamp continuously consumes approximately 30 watts of electrical power.

Since the men will don their suits for emergency conditions or may wear them between stops during the locomotion part of a "day" (24-hour period), a flow rate of 20 cfm (at 360 mm Hg) through an air hose is required to cool each suited man. The flow has a pressure drop of 4 inches water gage. These figures are calculated for routine duties in a pressure suit at 300 btu/hr sensible heat load. The latent heat load is easily handled by the flow. For simplicity and ease of flow control, each suit has its own blower. However, for redundancy the two suit circuits may be valved together so that one blower provides flow for both suits. Considering an overall small blower efficiency of 25 percent each blower will consume approximately 10 watts while operating.

The condensate in the cabin heat exchanger collects at a rate of approximately 13.4 lb/day (4.2 ml/min).

The condensate is continuously pumped by a small (1 watt) pump through a filter which contains a bacterial filter, activated charcoal and ion exchange resin to produce potable water.

Table 4-1 delineates the impurities found in a typical heat exchanger condensate from a closed environment manned test. Also the equivalent weight of anions and cations are shown. The cations are the goverining ions and subsequently, they are used to size the filter bed. Amberlite MB-3, a mono-bed exchange resing is ideal for this application. It is a mixture of strongly acidic cation and strongly basic anion exchange resins. The resins insure that the acid formed by initial contact of a salt with a particle of cation exchange is immediately neutralized by the neighboring particles of anion exchanges. This maintains a neutral pH in the effluent water. This resin contains a small amount of indicator on the anion exchanger, thereby gives a visual indication of exhaustion of the resin. As the deionizing capacity of the resin is exhausted, the color of the bed changes from a blue-green to a yellow-brown color. The amount of resin required is calculated below:

TABLE 4-1. TYPICAL CABIN HEAT EXCHANGER CONDENSATE IMPURITIES FOR MANNED SYSTEMS**

		meq Anion*	meq Cation*
Ammonia as N ppm	6.0	---	0.35
Alkalinity $CaCo_3$ ppm	24	0.48	0.48
Sulfate as SO_4 ppm	0.8	0.017	---
Chloride as C1 ppm	0.5	0.028	---
pH	6.7	---	0.0002
Sp Cond.	50	---	---
Phenol, ppm	0.580	---	---
Solids, ppm	4	---	---
		0.525	0.8302

* Equivalent weights based on 1 liter of water.

**Analysis by Betz Laboratories, Inc. for condensate from 2-man 15-day closed environment test at the General Electric Company

Based on Rohm & Hass literature:

$$\frac{0.830 \text{ meq}}{\text{liter}} \quad x \quad \frac{0.69 \text{ gm/ml}}{0.46 \text{ meq/ml}} \quad = \quad \frac{1.25 \text{ gm MB-3}}{\text{liter } H_2O}$$

Assume bed is 80% efficient:

$$\frac{1.26}{0.80} = 1.5 \text{ gm MB-3 per liter of water processed}$$

For the above process rate of 13.4 lbs. per day, approximately 0.91 pounds of resin are required for a 45 day mission.

Activated charcoal is required to remove the Phenol in the water. Based upon 1 liter of H_2O, 0.58×10^{-3} gm phenol will require 1.70×10^{-3} grams charcoal (Barneby-Cheney type PC-5 33% capacity). For the above process rate of 13.4 lb per day, approximately 0.001 pounds of charcoal are required for a 45-day mission.

The 4×10^{-3} gm of solids may be readily removed along with bacteria by filter media such as manufactured by Pall Corporation. The filter is sized by the 0.3 micron filtration requirement for bacteria.

The resin and the charcoal are mixed to reduce packaging problems. The filtered water is pumped directly to the potable water storage containers.

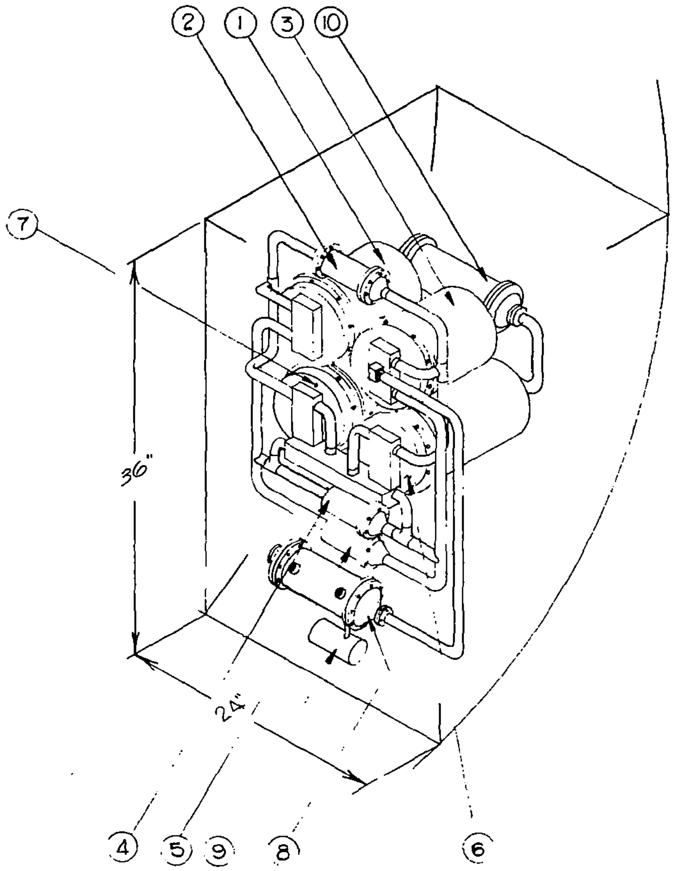
4.3.2 CARBON DIOXIDE REMOVAL

A small amount (6.67 cfm) of the cabin atmosphere flow is diverted from the cabin heat exchanger to the carbon dioxide removal section. The process gas is removed from the heat exchanger since this gas has the smallest amount (weight) of water vapor. The process gas is then passed through a desiccant bed of silica gel which absorbs the water vapor such that the gas has a dew of $-40^{\circ}F$. The absorption process is exothermic, consequently a coolant flow is provided to cool the canister. The cooling also maintains the efficiency of the desiccant. The gas is then cooled to approximately 40 ^{o}F in a gas to liquid heat exchanger and is passed through a carbon dioxide absorbing canister containing molecular sieve material. The rpocess gas is first dried by the desiccant since the molecular sieve has a preferential affinity for water vapor. The sieve will not absorb the carbon dioxide if large quantities of water are contained in the process gas. The carbon dioxide absorption process is also exothermic but the low flow and the cooled gas permits the canister to dissipate sufficient heat to the gas flow to maintain a reasonable collection efficiency. The process gas is circulated through the carbon dioxide removal subsystem by a blower which continuously consumes 14 watts of electrical power. Since this blower is critical to the survival of the crew, a redundant blower is provided.

The subsystem is arranged so that a desiccant and a molecular sieve canister are absorbing while a second canister of each material is being desorbed and readied for its abosroption cycle (see Figure 4-4). The alternate absorbing/desorbing cycle of each canister permits continuous removal of carbon dioxide. The processed and reconditioned gas which exits from the carbon dioxide absorbing canister is then passed through the desorbing desiccant canister. Since the processed gas is very dry, the gas readily removes the water from the desiccant when the canister is heated. The humidified and heated gas is returned to the cabin heat exchange in nearly the same condition as it entered the subsystem except the temperature is higher and the carbon dioxide is removed.

While the desiccant is being rejuvenated, the second molecular sieve canister is being purged of carbon dioxide. This is accomplished by isolating the canister and venting it to spatial vacuum.

The desorption of the sieve is an endothermic process. If no heat is added to the sieve while the sieve is desorbing, the sieve temperature will drop until the desorption process virtually stops. If the process is to be completed efficiently, heat must be added to the sieve to assure desorption. The major thermodynamic problem involved is the transfer of heat to the sieve material since the heat must be conducted through a random arrangement



1 DESICCANT CANISTER
2 GAS HEAT EXCHANGER
3 DESICCANT CANISTER
4 BLOWER
5 REDUNDANT BLOWER
6 MOLECULAR SIEVE CANISTER
7 MOLECULAR SIEVE CANISTER
8 HEAT EXCHANGER
9 CONDENSATE TANK
10 HEAT EXCHANGER (THERMALLY INTEGRATED SYSTEM ONLY)

Figure 4-4 Carbon Dioxide Removal System

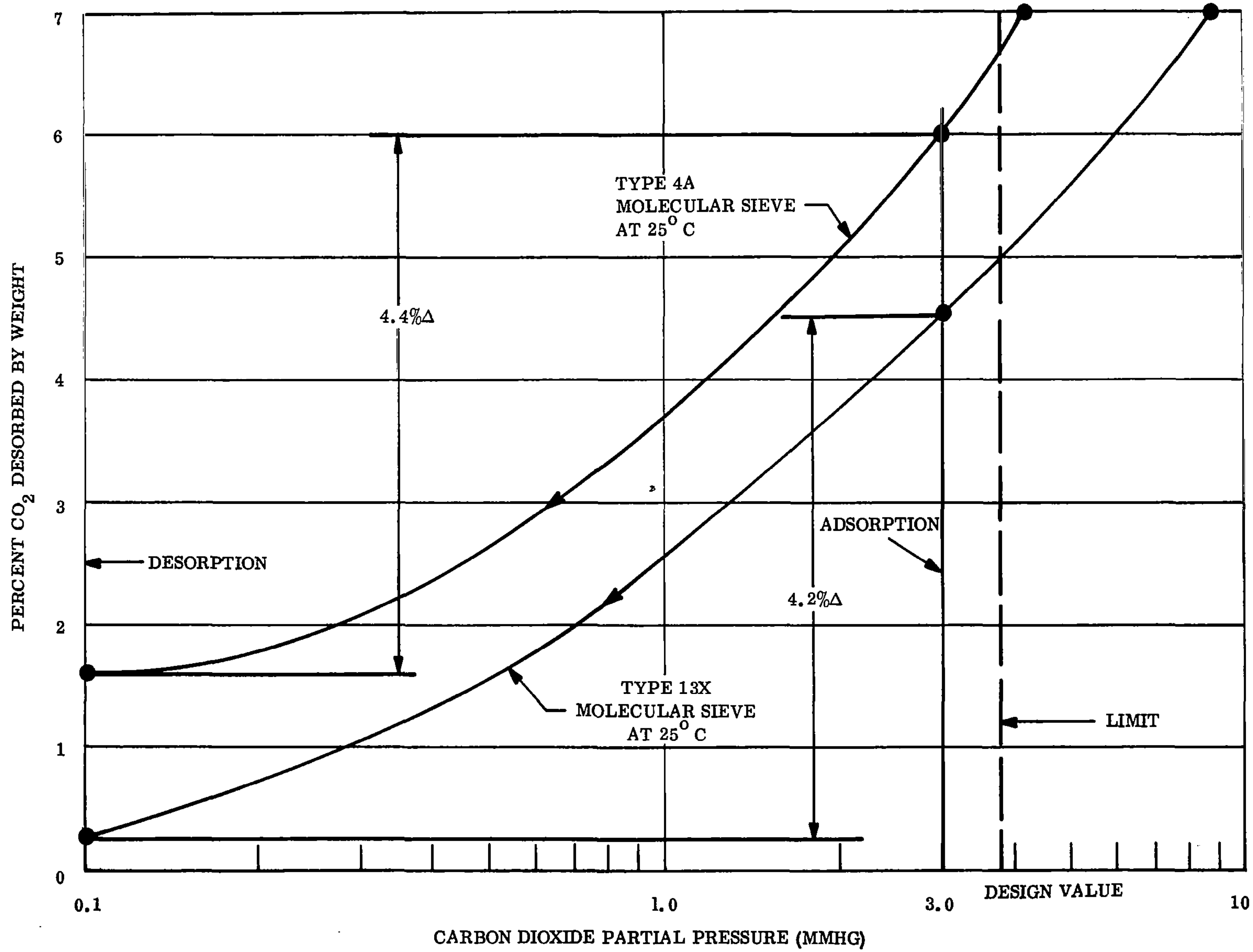


Figure 4-5 Carbon Dioxide Partical Pressure (mm Hg)

of sieve pellets (a poor conductive path). A recent development by General Electric permits rapid heat transfer to the sieve material and thus requires a lower temperature energy source. Also, since the carbon dioxide is not recovered in this system, lower (spatial vacuum) pressure is utilized to desorb the sieves and lower temperatures are required. Figure 4-5 illustrates the $25^{o}C$ ($78^{o}F$) isotherms for type 4A and 13 X Molecular Sieves. Establishing of a 3.0 mm Hg CO_2 partial pressure design valve and a 0.1 mm Hg desorption pressure permits a 4.4 and a 4.2 delta desorption percentage (respectively) by weight. Both sieve materials have approximately the same properties in this operating range, however, the 13 X material approaches complete desorption (zero percentage CO_2) at a higher pressure and the rate of desorption is more rapid at the lower pressures. Consequently the 13 X Molecular Sieve material is selected as the carbon dioxide absorption material. During normal desorption, electrical heaters are utilized to only maintain the sieve at cabin temperature, thus a minimum of electrical power is required. Also once a day the sieve is heated to $400\text{-}600^{o}F$ to expel all sieve contaminants which degrade performance. The carbon dioxide production rate is 1.84 pounds per man for the 19 hours in the cabin, or about 0.2 lb/hr. The adsorption/desorption cycle duration selected is one hour.

With a 50 percent sieve bed absorption efficiency (and a 4.2 percent absorption delta percentage) the canister size required will be 9.5 pounds for one half hour of continuous operation adsorption Two canisters are used, one in operation and one desorbing at all times.

4.3.2.1 Carbon Dioxide Absorption Cycle Cooling

The sieve absorption process is exothermic at a rate of 300 btu/lb of carbon dioxide absorbed. Cooling is provided by the cooled process gas flow at a rate of 60 btu/hr to assure high absorption efficiency.

4.3.2.2 Carbon Dioxide Desorption Cycle Heating

It is assumed that the canister is at $78^{\circ}F$ at the completion of the absorbing cycle. Electrical heat is provided to maintain this temperature during the endothermic desorption cycle. The endothermic power requirement is the same as the exothermic power of 60 btu/hr (18.0 watts). Once a day the canisters are heated to $400^{\circ}F$ to expel sieve contaminants. Assuming a total two canister weight of 25 pounds (19.0 sieve plus 6.0 shells) at an average specific heat of 0.25 btu/lb $^{\circ}F$ the electrical power requirement is 600 watts, the total average power is 18 watts + $\frac{600}{24}$ = 43 watts.

4.3.2.3 Water Vapor Absorption Cycle Cooling

The silica gel desiccant absorption process is exothermic at a rate of approximately 1300 btu/lb of water vapor absorbed. If the process gas temperature is to remain low then the heat of absorption is removed by a glycol cooling coil. For the by-pass flow selected of 6.67 cfm, 0.231 pounds of water will be collected per hour.

The amount of exothermic process heat is then:

$$\frac{1300 \text{ btu}}{\text{lb}} \times \frac{0231 \text{ lb}}{\text{hr}} = 300 \text{ btu/hr (about 89 watts) cooling}$$

An additional 11 watts per hour of heat is removed to achieve a gas temperature of $40^{\circ}F$.

The carbon dioxide absorption process adds 18 watts to the process gas flow and the blower adds 14 watts for a total of 32 watts. This raises the gas temperature 30° and the process gas enters the desorbing desiccant canister at $70^{\circ}F$.

4.3.2.4 Water Vapor Desorption Cycle Heating

The silicon gel regeneration (desorption) cycle is an endothermic process. Figure 4-6 shows the adsorption/desorption isotherms for silica gel. A 25 percent delta desorption (by weight)

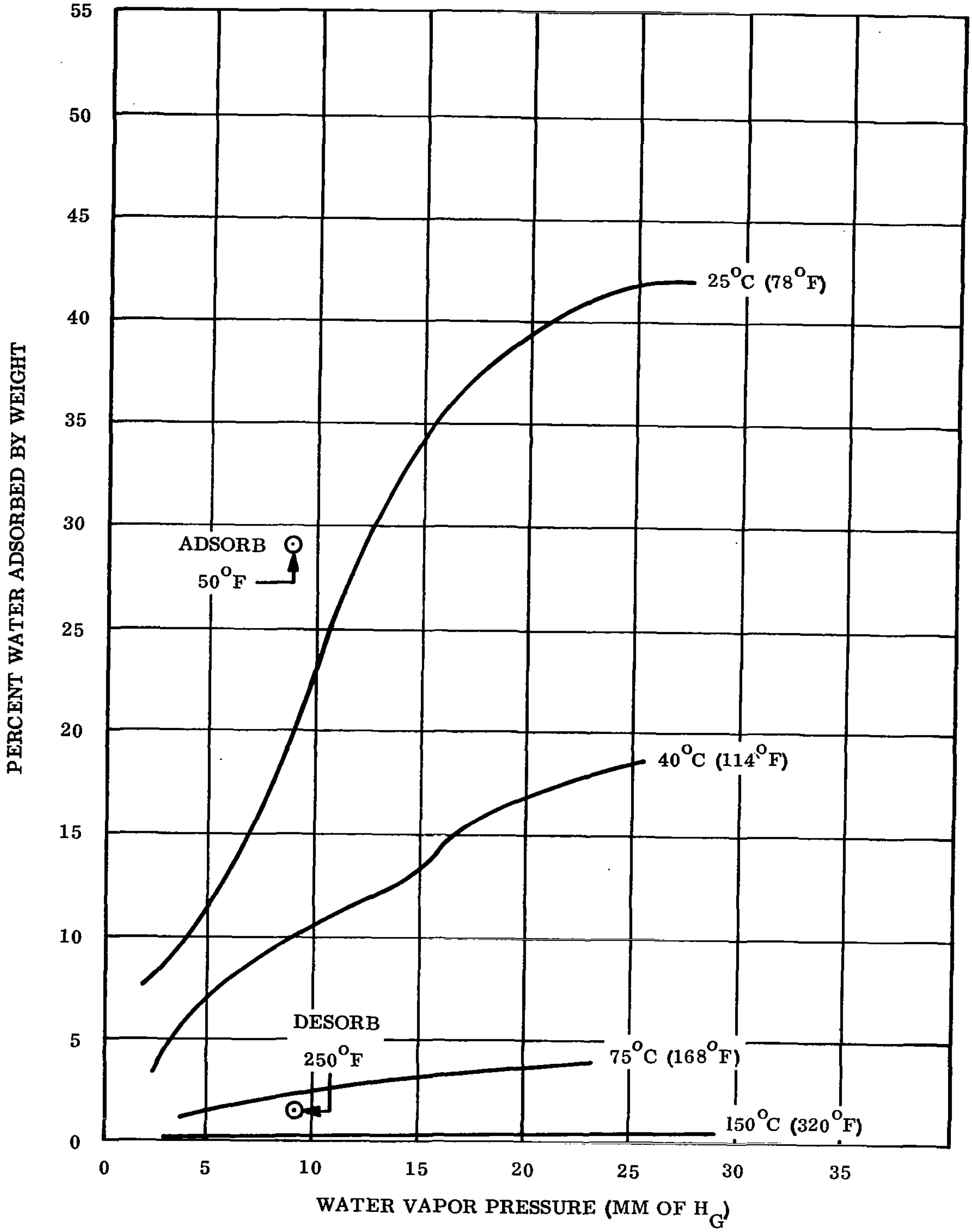


Figure 4-6 Silica Gel Isotherms

is used and an absorption efficiency of 25 percent is assumed. For a one hour cycle, the two silica gel beds weigh 3.7 pounds.

The total weights of the two canisters is assumed to be 6.0 pounds (3.7 lb gel and 2.3 lb shells) and the average specific heat to be 0.25 btu/lb ^{o}F. The temperature rise is 180^{o}F and approximately 80 watts of electric power is required for operation.

The desorption process also required the addition of 89 watts (same as absorbing cycle) to remove the water from the gel. In addition, the process gas flow is heated to the higher temperature; which requires 195 watts.

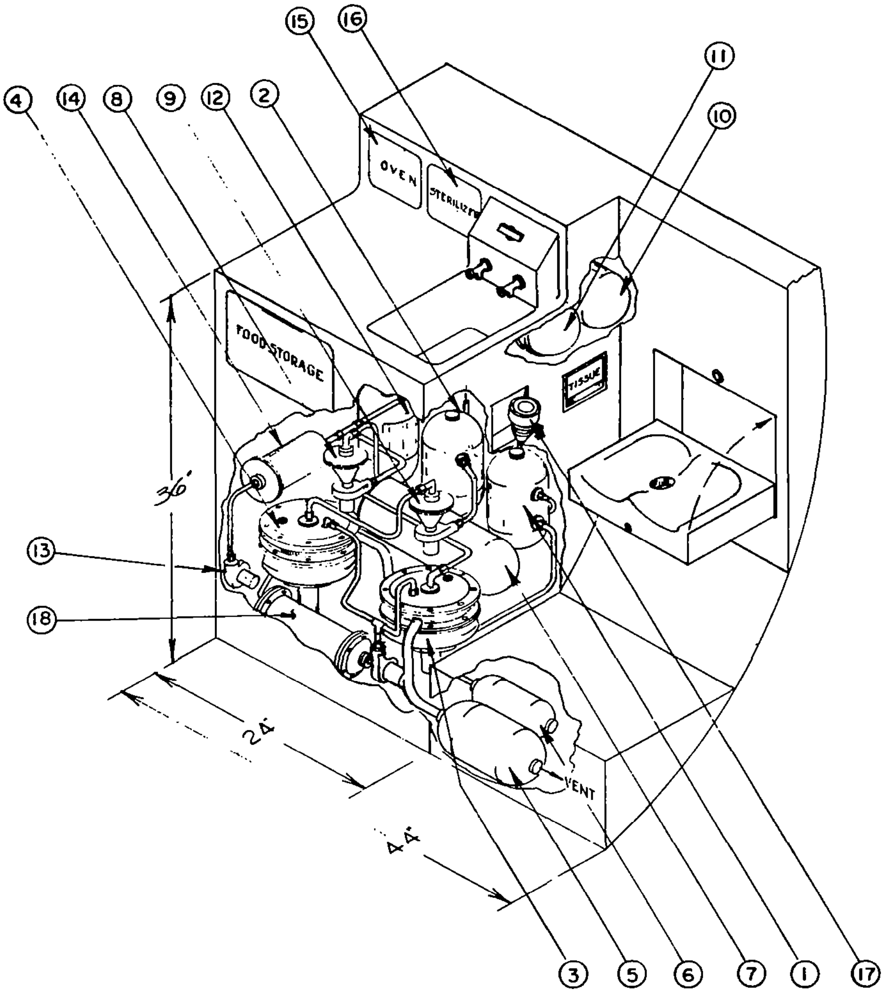
In summary, the total average power requirement to regenerate the disiccant is 364 watts.

Canisters	80 watts
Description	89
Glas Flow	195
Total	364 watts

The hot humid air which exits from the desiccant canister is cooled and dehumidified in a gas to liquid heat exchanger. Approximately 80 percent of the latent and sensible heat (290 watts) contained in the air is transferred to a liquid collant loop. The remaining heat (74 watts) is vented to the cabin atmosphere with the air flow. The liquid coolant is pumped by redundant 5 watt pumps to a radiator heat sink. The coolant enters the radiator at 110^{o}F and exits at 40^{o}F to the heat exchanger. When the humid air enters, the heat exchanger is cooled and water is condensed. This water flows to a condensate tank by the lunar gravitational force and is pumped to the condensate filter for processing.

4.3.3 WATER RECOVERY

The water recovery system consists of all equipment necessary to store and process the urine and wash water. A possible equipment arrangement is shown in Figure 4-7. Each crew member will occupy the cabin for 19 hours and the suit 5 hours per day. 79.2 percent



1 URINE STORAGE RESERVOIR
2 WASH WATER STORAGE RESERVOIR
3 URINE EVAPORATOR
4 WASH WATER EVAPORATOR
5 URINE SOLIDS STORAGE
6 WASH WATER SOLIDS STORAGE
7 PYROLYSIS UNIT
8 POTABLE WATER CONDENSER
9 WASH WATER CONDENSER
10 POTABLE COLD WATER RESERVOIR
11 POTABLE HOT WATER RESERVOIR
12 WASH WATER RESERVOIR
13 PUMP
14 CONDENSATE FILTER
15 OVEN
16 STERILIZER
17 URINAL
18 HEAT EXCHANGER (THERMALLY INTEGRATED SYSTEM ONLY)

Figure 4-7 Water Recovery System

of the time will be at normal metabolic output and 20.8 percent of the time will be at a substantially higher metabolic output (in the suit). The overall average metabolic data per man-day is used to determine a possible overall man-day water balance as shown in Figure 4-1. Note that water recovery efficiencies are given since the water balance in the lunar roving vehicle is more critical than for the MORL type vehicle or the lunar shelter studied previously. This is due to the large amounts of water consumed in the suit back-pack for thermal control. Also it is assumed that all the water produced by the man while in the suit will be collected and returned to the vehicle for processing.

4.3.3.1 WATER RECOVERY FROM URINE

The 7.0 pounds of urine voided by the crew per day contains approximately 6.6 pounds of water and 0.4 pounds of solids (approximately 5 percent). The urine is first processed in a low temperature/pressure evaporator, then subjected to high catalytic temperatures and finally condensed. The low temperature (120°F) and low pressure (1.7 psia) distillation minimizes the amounts of impurities (organics, ammonia, etc.) volitized with the water, thus separating the majority of the impurities from the water. Note that since there is approximately one-sixth normal gravity on the lunar surface, there is no need for centrifugal impellers to induce phase separation during boiling. The electric power requirement for evaporation is 81 watts.

The urine solids (5 percent of the urine) will settle to the bottom of the evaporator due to the lunar gravity. This permits easy removal of the solids (e.g. by a piston type device) or the evaporator may be sized to store all the solids.

The water vapor from the distillation process is far from being potable. The vapors are heated to 1800°F in the presence of a platinum catalyst and a small amount of oxygen (2 cc/min) bled into the system. The water vapor impurities are oxidized to form fairly unsolvable oxides (e.g. CO_2, NO_2, etc.) The water vapors are condensed as potable water and the oxides are vented to vacuum. The pyrolysis process consumes approximately 50 watts of electrical power, however, only a small amount of this power is utilized to heat the vapor.

A gas to gas heat exchange permits the pyrolyzed vapor to preheat the incoming vapor so that the thermal energy is conserved. The majority of the power is required to overcome the radiation and conduction heat losses through the superinsulation jacket.

The condenser is a gas to liquid heat exchanger which removes sufficient energy from the water vapor to permit liquefication. The thermal energy is transferred to the vehicle liquid coolant loop. Since there is approximately one-sixth normal gravity on the lunar surface, there is no need for an active or a passive means of phase separation other than the differences in density between the vapor and the condensate. The condensate is then pumped to the potable water storage reservoirs by a small pump which utilizes an average power of 0.5 watts. The cold water storage reservoir is cooled to approximately $40^{o}F$ by the vehicle liquid coolant while the hot water storage reservoir is heated to approximately $180^{o}F$ by an electrical heater. The hot and cold water is then dispensed to the crew as required. A mixing valve is utilized to obtain water temperatures at any point between the $40^{o}F$ and $180^{o}F$ range.

4.3.3.2 WATER RECOVERY FROM WASH WATER

The crew utilizes approximately 12.0 pounds of wash water per day. This water need not be potable and since it contains only small amounts of solid contaminants (0.25 to 1.0 percent), low temperature/ pressure distillation suffices to recover the wash water for reuse. If contaminants should increase substantially over the long mission, a portion of the distilled vapor may be passed through the pyrolysis section of the urine water recovery system and potable water utilized to replenish the wash water supply. This will essentially dilute the contaminant concentration and maintain the impurities at an acceptable level. The electrical power requirement for distillation is 147 watts.

The washwater solids (0.25 to 1.0 percent) settle to the bottom of the evaporator and are removed or stored in the evaporatore.

The water vapor from the distillation process is then liquefied in a condenser and pumped to

TABLE 4-2. POWER REQUIREMENTS (WATTS) - NON INTEGRATED

Cabin Environmental Control System		Weight (lb)	Average Process Power (Watts)	Average Support Power (Watts)
Consisting of:	Air Filter	0.5	0.0	0.0
	Blower	2.0	0.0	46.0
	Redundant Blower	2.0	0.0	0.0
	Container/ Expendable Refrigerant	17.0	0.0	0.0
	Dormant Period Blower	0.5	0.0	0.0
	Evaporative Heat Exchanger	3.0	0.0	0.0
	Cabin Heat Exchanger	6.0	0.0	0.0
	Odor Control Canister	1.5	0.0	0.0
	Trace Gas Control	2.0	50.0	0.0
	Germicidal Lamp/ Ballast	1.0	25.0	5.0
	Glycol Pump	2.0	0.0	25.0
	Redundant Glycol Pump	2.0	0.0	0.0
	Dormant Period Pump	0.5	0.0	0.0
	Suit Blower #1	1.0	0.0	2.0
	Suit Blower #2	1.0	0.0	2.0
	Ducting	10.0	0.0	0.0
	Misc. Hardware & Plumbing	4.0	0.0	0.0
	Controls	1.0	0.0	2.0
		57.0 lb.	75.0 watts	82.0 watts
			75 watts	

Carbon Dioxide Removal System				
Consisting of:	Desiccant Canister	3.0	364.0	0.0
	Desiccant Canister	3.0	0.0	0.0
	Gas Heat Exchanger	0.5	0.0	0.0
	Mole Sieve Canister	12.5	43.0	0.0
	Mole Sieve Canister	12.5	0.0	0.0
	Blower	1.0	0.0	14.0
	Redundant Blower	1.0	0.0	0.0
	Misc. Hardware & Plumbing	2.5	0.0	0.0
	Controls	2.0	0.0	5.0
	Heat Exchanger	2.0	0.0	0.0
	Condensate Tank	0.5	0.0	0.0
	Pump	0.5	0.0	5.0
	Redundant Pump	0.5	0.0	0.0
	Misc. Hwd	0.5	0.0	0.0
	Heat Leakage	0.0	0.0	20.0
		42.0 lb.	407.0 watts	44.0 watts
			451.0 watts	

TABLE 4-2. POWER REQUIREMENTS (WATTS) - NON INTEGRATED (CONT'D)

Water Recovery System			
Consisting of: Recovery From Urine			
Urine Storage	2.0	0.0	0.0
Solids Storage	1.0	0.0	0.0
Evaporation	5.0	81.0	0.0
Pyrolysis Unit	10.0	50.0	0.0
Condenser	2.0	0.0	0.0
Condensate Pump	0.5	0.0	0.5
H.E. Condensate Pump	0.5	0.0	1.0
H.E. Condensate Filter	1.5	0.0	0.0
Cold Water Storage	2.5	0.0	0.0
Hot Water Storage	2.5	0.0	9.0
Misc. Hardware & Pumping	2.0	0.0	0.0
Controls	1.5	0.0	3.5
Heat Leakage	0.0	0.0	20.0
	31.0 lb.	131.0 watts	34.0 watts
		165.0 watts	
Recovery From Wash Water			
Wash Water Storage	3.0	0.0	0.0
Solids Storage	1.0	0.0	0.0
Evaporator	7.0	147.0	0.0
Condenser	2.5	0.0	0.0
Condensate Pump	0.5	0.0	1.0
Wash Water Reservoir	3.0	0.0	0.0
Misc. Hardware & Pumping	1.5	0.0	0.0
Controls	0.5	0.0	2.0
Heat Leakage	0.0	0.0	20.0
	19.0 lb.	147.0 watts	23.0 watts
		170.0 watts	
Food Management System			
Consisting of: Oven	5.0	3.0	0.0
Sterilizer	7.5	7.0	0.0
Misc. Hardware & Pumbing	2.0	0.0	0.0
Controls	0.5	0.0	2.0
Heat Leakage	0.0	0.0	10.0
	15.0 lb.	10.0 watts	12.0 watts
Misc. Structure & Enclosure		22.0 watts	
TOTALS NONINTEGRATED SYSTEM	184.0 lbs	770.0 watts	195.0 watts
NO OXYGEN RECOVERY		965 watts	

storage by a pump which utilizes an average power of 1.0 watt. The storage reservoir retains the water until it is utilized for washing.

4.3.4 FOOD MANAGEMENT SYSTEM

Past studies have included the water cooling and heating storage reservoirs as part of food management. However, the block diagram (Figure 4-3) shows that these items are more properly a part of the water recovery system. The food management system consists of an oven to reconstitute the rehydrated food and a sterilizer to control bacterial contamination of eating utensils.

The oven is utilized to merely maintain the rehydrated food package at 170°F for approximately 5 minutes to assure proper reconstitution. This requires an average of 3 watts electrical power which is utilized mainly to overcome heat leakage from the oven since the food is put into the oven at 180°F.

The sterilizer is utilized to sterilize utensils used for eating. Also such items as boots or gloves which contact the lunar surface may be sterilized in this unit to minimize the possibility of bacterial contamination. The sterilizer requires approximately 7 watts to produce superheated steam at 240-250°F.

4.3.5 RESULTS

Table 4-2 summarizes the average electrical power requirements for a thermally nonintegrated life support system in which oxygen is not recovered from the carbon dioxide.

4.4 NONINTEGRATED SYSTEM WITH OXYGEN RECOVERY

The nonintegrated life support with recovery contains the same water recovery and food management equipment as discussed in the previous section (4.3). The differences are the area of carbon dioxide collection in the cabin and suit and the process for recovery of the oxygen. The changes are as follows:

a. The desorption process is accomplished at a high temperature and pressure since the carbon dioxide is to be collected. Collection at the lower sieve temperature and pressure would require larger equipment and more electrical power.
b. The silica and molecular sieve method for collection of the carbon dioxide is too large and heavy to be utilized for collection of the carbon dioxide while the men are in their space suits. A regenerable magnesium oxide absorption system is much lighter and therefore used with the suits.
c. A Sabatier reactor, condenser, electrolysis unit, and methane pyrolyzer are added to the system to remove the oxygen from the carbon dioxide.

4.4.1 CABIN ENVIRONMENTAL CONTROL

The basic system is the same as that described in 4.3.1. The inclusion of the oxygen recovery subsystem in the vehicle requires the expenditure of more electrical energy and consequently the liberation of more heat to be the cabin environment. More cabin atmosphere and liquid coolant are then required to be circulated for thermal control. The blower and pump electrical power requirement are increased to 50 and 30 watts respectively, but without a significant weight change.

4.4.2 CARBON DIOXIDE REMOVAL AND OXYGEN RECOVERY

The molecular sieve is heated to 250°F from the normal temperature of 78°F to permit removal of the entrained carbon dioxide. The sieves and canisters weigh approximately 20 pounds and are recycled every hour using 270 watts of electrical power.

The carbon dioxide gas is removed from the sieve by a suction gas pump which stores the carbon dioxide in an accumulator. The pump requires 33 watts for operation and is not back-up by a redundant pump since the whole oxygen recovery system may be shut down while the pump is repaired.

The carbon dioxide is continuously bled into the Sabatier reactor where in the presence of a catalyst, the carbon dioxide is hydrogenated to form water and methane. The process is exothermic and liberates approximately 94 watts of thermal power. The water is separated from the methane in a condenser and is pumped to the electrolysis unit. The water is electrolyzed to form hydrogen and oxygen at the expenditure of 330 watts of electrical power. The oxygen is returned to the cabin for rebreathing and the hydrogen is returned to the Sabatier reactor for reuse.

The methane separated from the water in the condenser flows to a pyrolysis unit which thermally cracks the methane to form carbon and hydrogen. The carbon is discarded while the hydrogen is reused in the Sabatier reactor. A more complete description of the process may be found in References 4-1 and 4-2.

4.4.2.1 CARBON DIOXIDE COLLECTION IN PRESSURE SUITS

The mission profile for the lunar roving vehicle requires the men to spend 20 percent of their time in pressure suits exploring the surface. In obeisance to the philosophy of carrying a minimum amount of expendables, the carbon dioxide produced by the men while in their suits is collected and returned to the vehicle for recovery of the oxygen. The collection of carbon dioxide by the same method as utilized in the vehicle (molecular sieves) can not be utilized in the suit back pack because of the prohibitive size and weight requirement that a man would have to transport. A sieve weight of 55.8 pounds is required to absorb 1.55 pounds of carbon dioxide produced during 5 hours in the suit. The desiccant and cannister will increase the weight to approximately 100 earth pounds. Also the method of absorption of carbon dioxide by lithuim hydroxide can not be utilized in spite of its high capacity due to the difficulties in removal of the carbon dioxide. A temperature of $1360^{o}F$ and a pressure of 0.025 atmosphere would be required to regenerate the lithuim hydroxide.

One of the more promising methods of collection of carbon dioxide for this application is the use of magnesium oxide as an absorbent. Initially the magnesium oxide is preconditioned with steam to convert the oxide to a hydroxide $Mg(OH)_2$. This permits a high absorption capacity when exposed to carbon dioxide.

$$Mg\ (OH)_2\ (solid) + CO_2\ (gas) + H_2O\ (gas) \longrightarrow Mg\ CO_3 \bullet 2H_2O$$

The absorption process is exothermic and produces approximately 1060 btu per pound of carbon dioxide absorbed. The absorption efficiency of the Mg $(OH)_2$ for 40 percent utilization requires 5.14 pounds for a 5-hour period inthe suit. A canister weight of 1.36 pounds is envisioned for a total canister filled weight of 6.5 pounds.

4.4.2.2 SUIT CANISTER REGENERATION

Regeneration of the backpack CO_2 removal canister is complex because the dissociation of the Mg CO_3 must be implemented. This requires a two-step process, first dehydration, and then dissociation at approximately 900°F. A temperature-time profile somewhat as indicated by Figure 4-8 will be realized during this two step process.

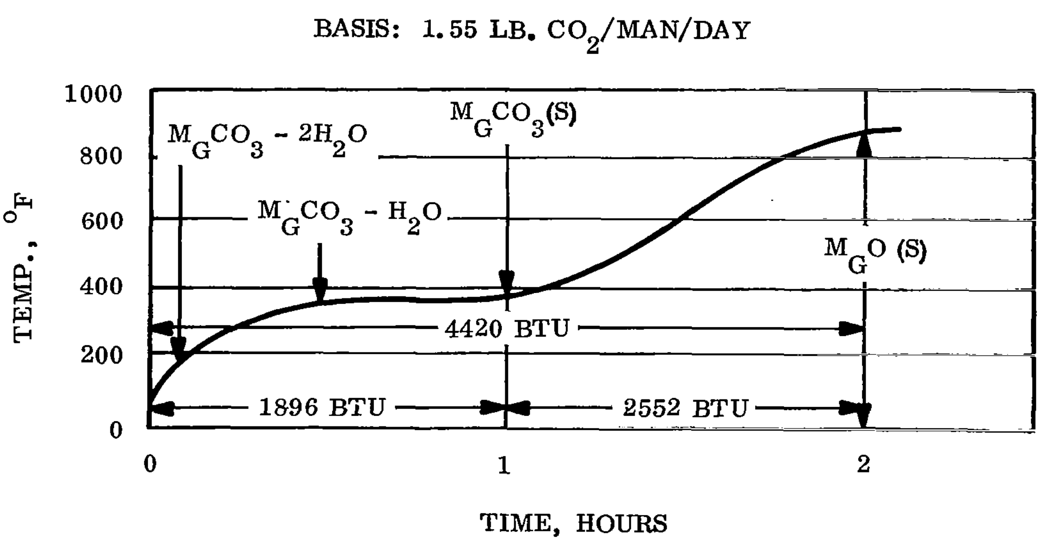


Figure 4-8. MgO Regeneration Thermal Profile

The regeneration of the absorbent is accomplished in three stages when the expended canister is returned to the vehicle.

a. The canister is heated to approximately 350°F and evolved water vapor is rejected to the cabin heat exchanger.

$Mg\,CO_3 \bullet 2H_2O \longrightarrow Mg\,CO_3 + 2H_2O + \Delta H_1$

b. The canister is then heated to approximately 900°F and the carbon dioxide is removed by the gas pump.

$Mg\,CO_3 \longrightarrow Mg\,O + CO_2 + \Delta H_2$

c. Water is then injected into the hot canister to produce the required absorbent hydoxide.

$Mg\,O + H_2O \longrightarrow Mg\,(OH)_2$

The regeneration rate may vary depending on the rate of heat transfer to the bed, and the temperature gradient effects inherent in the canister design.

The sensible heat required to bring the canister to 350°F is calculated to be approximately 626 btu (based on Cp=1.9 btu/°F). The heat of dehydration (ΔH_1) required to remove the H_2O is 1270 btu.

To accomplish regeneration of the carbonate to the solid oxide requires addition of approximately 980 btu per pound of CO_2 removed (ΔH_2), or approximately 1520 btu for the stated capacity of 1.55 pounds of CO_2 per cycle. In addition to the heat energy required to affect decomposition of the carbonate molecule, additional sensible heat is required to raise the canister temperature to the 900°F operating temperature. This amounts to approximately 1050 btu per regeneration. The "latent" and sensible heat totals are noted in Figure 4-8. No allowance is made in that figure for losses by heat transfer to the surroundings. These processes require the following input energy per regeneration:

a.	Heat of dehydration	1270 btu
b.	Sensible heat to attain 350°F	626
c.	Heat of dissociation	1520
d.	Sensible heat to attain 900°F	1032
		4448 btu

If the canisters are regenerated for a 24-hour period, approximately 55 watts are required per canister or 110 watts for two canisters average.

As in Section 4. 3. 2. 4 a heat exchanger has been added to cool and dehumidify the air which exits from the desiccant canister as it is being regenerated. Also the condenser for the oxygen recovery unit is attached to this coolant line. The coolant pump power requirement is increased to 7 watts.

4. 4. 2. 3 WATER RECOVERY

Water is utilized to reactivate the MgO after regeneration to produce Mg $(OH)_2$ which is a more efficient absorbent for carbon dioxide. The water is expelled from the absorbent during regeneration and passes to the cabin heat exchanger where it is condensed and returned to the potable water storage tank. The effect on the environmental control and condensate pump and filter is insignificant.

The canister is configured as a cylinder with radial flow of gas from the center to the outside surface. This minimizes pressure drop through the bed and provides maximum retention time. The canister must be designed to withstand the 900°F regeneration temperature. Adequate heat transfer surfaces must also be provided to eliminate localized heating and facilitate both heating during regeneration and cooling during absorption.

Several process variables still require definition the most important of which is the absorption rates that can be achieved under the stated operating conditions. These data can best be obtained by empirical, evaluation of typical canisters in both the absorbing and desorbing modes of operation.

4.4.3 RESULTS

Table 4-3 lists the amount of electrical heat required as well as the electrical support power required for a nonintegrated life support system in which oxygen is recovered from the carbon dioxide. Figure 4-9 shows a block diagram of this system. Figure 4-10 illustrates a possible arrangement of the oxygen recovery system.

4.5 THERMAL INTEGRATION METHODS

4.5.1 THERMAL INTEGRATION WITH THE PRINCIPAL POWER SYSTEM

The use of the electrical power system waste heat in place of electric power to operate life support processes must be implemented with fluid connections between the cabin and trailer module. For minimum sizing a liquid should be used to transport the heat from heat exchangers on the power system heat rejection lines to the life support consoles. For the space between the cabin and trailer either flexible lines or flexible connections must be used. The concept presented here is to have two flexible lines attached on the trailer module near the hitch and on the cabin near the bottom. These lines are composed of:

a. Inner tube of DuPont "H" film which survives the 600^{o}F temperatures supplied by some of the power systems and the -240^{o}F extreme lunar night temperature during the dormant period.

b. Woven stainless steel hydraulic type hose surrounding the H-film for semi-rigidization and protection.

c. Aluminum foil - fiberglass mat as an external thermal coverage (mainly for crew contact protection)

It is estimated that the 8 ft of hose required for trailer-cabin attachment would weigh three pounds per trailer module.

TABLE 4-3. POWER REQUIREMENTS (WATTS) - NON INTEGRATED WITH OXYGEN RECOVERY

Cabin Environmental Control System		Weight (lb)	Average Process Power (Watts)	Average Support Power (Watts)
Consisting of:	Air Filter	0.5	0.0	0.0
	Blower	2.0	0.0	50.0
	Redundant Blower	2.0	0.0	0.0
	Container/Expendable Refrigerant	17.0	0.0	0.0
	Dormant Period Blower	0.5	0.0	0.0
	Evaporative Heat Exchanger	3.0	0.0	0.0
	Cabin Heat Exchanger	7.0	0.0	0.0
	Odor Control Canister	1.5	0.0	0.0
	Trace Gas Control Canister	2.0	50.0	0.0
	Germicidal Lamp/Ballast	1.0	25.0	5.0
	Glycol Pump	2.0	0.0	30.0
	Redundant Glycol Pump	2.0	0.0	0.0
	Dormant Period Pump	0.5	0.0	0.0
	Suit Blower #1	1.0	0.0	2.0
	Suit Blower #2	1.0	0.0	2.0
	Ducting	10.0	0.0	0.0
	Misc. Hardware & Plumbing	4.0	0.0	0.0
	Controls	1.0	0.0	2.0
		58.0 lb.	75.0 watts	91.0 watts
			166.0 watts	

Carbon Dioxide Removal System		Weight (lb)	Average Process Power (Watts)	Average Support Power (Watts)
Consisting of:	Desiccant Canister	3.0	364.0	0.0
	Desiccant Canister	3.0	0.0	0.0
	Gas Heat Exchanger	0.5	0.0	0.0
	Mole Sieve Canister	10.0	270.0	0.0
	Mole Sieve Canister	10.0	0.0	0.0
	Blower	1.0	0.0	14.0
	Redundant Blower	1.0	0.0	0.0
	CO_2 Gas Pump	6.0	0.0	33.0
	CO_2 Accumulator	2.0	0.0	0.0
	Sabatier Reactor	10.0	0.0	0.0
	Condenser	2.0	0.0	0.0
	Condenser Pump	1.0	0.0	1.0
	Electrolysis Unit	25.0	330.0	0.0
	CH_4 Pyrolyzer	10.0	43.0	0.0
	Four MgO Canisters (2 Spares)	26.0	0.0	0.0
	MgO Regeneration Equipment	5.0	110.0	0.0
	Heat Exchanger	2.0	0.0	0.0
	Condensate Tank	0.5	0.0	0.0
	Pump	0.5	0.0	7.0
	Redundant Pump	0.5	0.0	0.0
	Misc. Hardware & Plumbing	10.5	0.0	0.0
	Controls	3.5	0.0	6.0
	Heat Leakage	0.0	0.0	40.0
		132.0 lb.	1117.0 watts	101.0 watts
			1218.0 watts	

TABLE 4-3. POWER REQUIREMENTS (WATTS) - NON INTEGRATED WITH OXYGEN RECOVERY (CONT'D)

Cabin Environmental Control System (Cont'd)	Weight (lb)	Average Process Power (Watts	Average Support Power (Watts)
Water Recovery System			
Consisting of: Recovery From Urine			
Urine Storage	2.0	0.0	0.0
Solids Storage	1.0	0.0	0.0
Evaporator	5.0	81.0	0.0
Pyrolysis Unit	10.0	50.0	0.0
Condenser	2.0	0.0	0.0
Condensate Pump	0.5	0.0	0.5
H.E. Condensate Pump	0.5	0.0	1.0
H.E. Condensate Filter	1.5	0.0	0.0
Cold Water Storage	2.5	0.0	0.0
Hot Water Storage	2.5	0.0	9.0
Misc. Hardware & Plumbing	2.0	0.0	0.0
Controls	1.5	0.0	3.5
Heat Leakage	0.0	0.0	20.0
	31.0 lb	131.0 watts	34.0 watts
		165.0 watts	
Recovery From Wash Water			
Wash Water Storage	3.0	0.0	0.0
Solids Storage	1.0	0.0	0.0
Evaporator	7.0	147.0	0.0
Condenser	2.5	0.0	0.0
Condensate Pump	0.5	0.0	1.0
Wash Water Reservoir	3.0	0.0	0.0
Misc. Hardware & Plumbing	1.5	0.0	0.0
Controls	0.5	0.0	2.0
Heat Leakage	0.0	0.0	20.0
	19.0 lb.	147.0 watts	23.0 watts
		170.0 watts	
Food Management System			
Consisting of: Oven	5.0	3.0	0.0
Sterilizer	7.5	7.0	0.0
Misc. Hardware & Plumbing	2.0	0.0	0.0
Controls	0.5	0.0	2.0
Heat Leakage	0.0	0.0	10.0
	15.0 lb.	10.0 watts	12.0 watts
		22.0 watts	
Misc. Structure & Enclosure	20.0 lb		
TOTALS NON-INTEGRATED SYSTEM WITH OXYGEN RECOVERY	275.0 lbs	1480.0 watts	261.0 watts
		1741.0 watts	

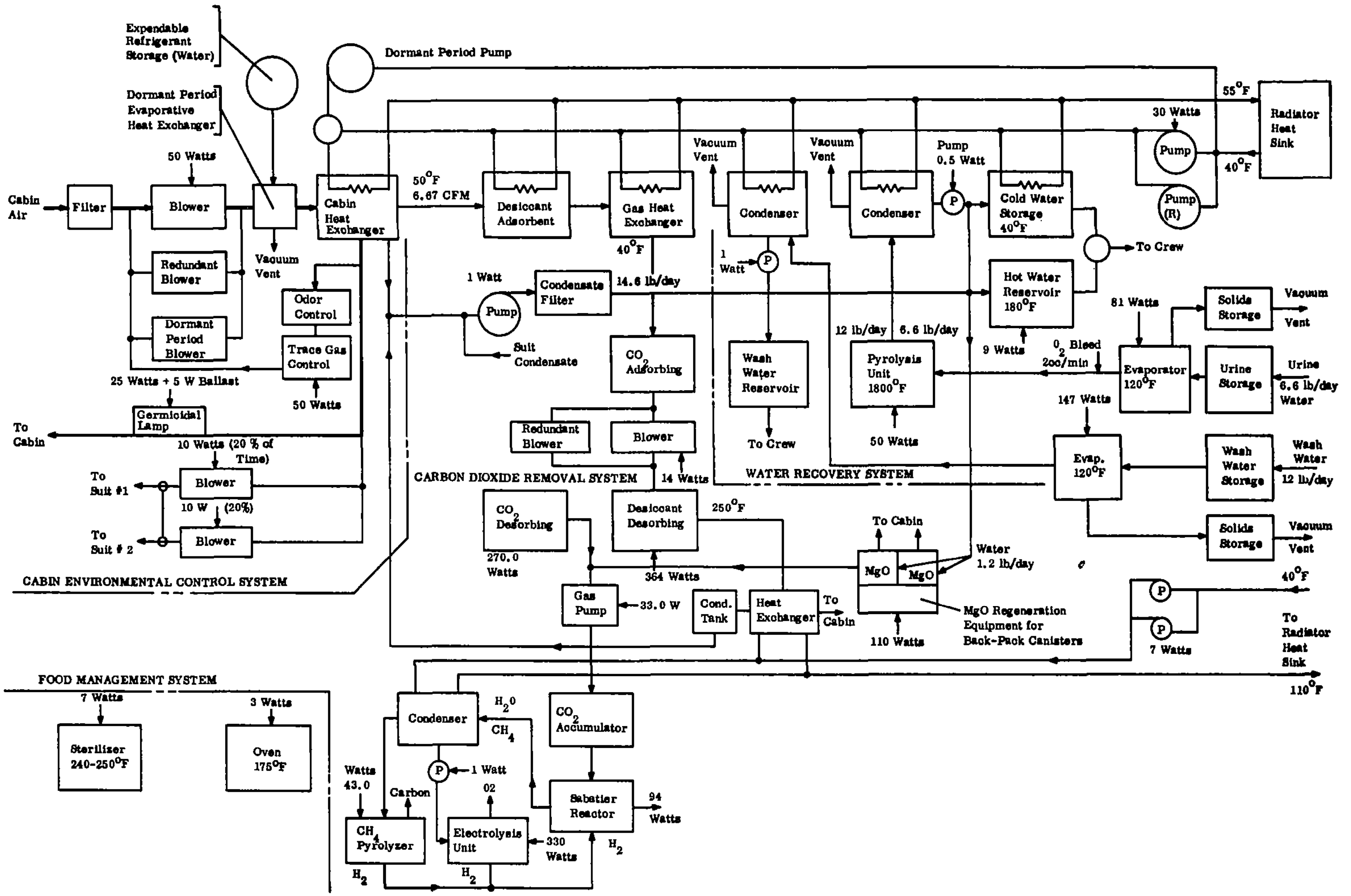
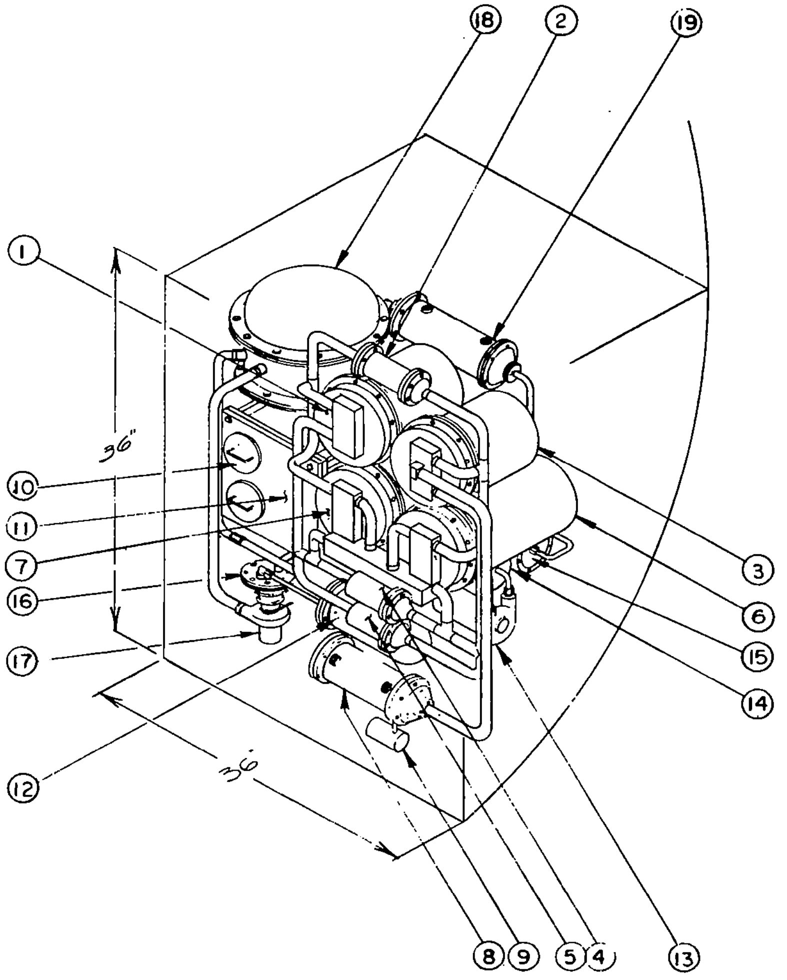


Figure 4-9 Block Diagram of Closed Loop Nonintegrated Life Support System



1 DESICCANT CANISTER
2 GAS HEAT EXCHANGER
3 DESICCANT CANISTER
4 BLOWER
5 REDUNDANT BLOWER
6 MOLECULAR SIEVE CANISTER
7 MOLECULAR SIEVE CANISTER
8 HEAT EXCHANGER
9 CONDENSATE TANK
10 MgO CANISTER
11 MgO REGENERATION EQUIPMENT
12 CH_4 PYROLYZER
13 CO_2 GAS PUMP
14 CO_2 ACCUMULATOR
15 SABATIER REACTER
16 CONDENSER
17 CONDENSER PUMP
18 ELECTROLYSIS UNIT
19 HEAT EXCHANGER (THERMALLY INTEGRATED SYSTEM ONLY)

Figure 4-10 Oxygen Recovery System

4.5.2 THERMAL INTEGRATION WITH THE RADIOISOTOPE THERMOELECTRIC GENERATOR

The 100-watt RTG mounted on the cabin wall rejects in the order of 2.8 kw thermal heat at 500^{o}F. The amount of heat and the level is sufficient to operate the life support processes. The method of heat transfer to implement this concept would be to use a heat pipe to this life support equipment. The boiler would be mounted on the flat side of the "D" configuration RTG.

4.5.3 COMPARISON OF METHODS

Thermal integration with the primary electrical power system has the advantage of not only reducing power system size but radiator size as well. For the application considered a reduction of 4.5 square feet would occur with the Rankine cycle, and approximately 30 square feet with the Brayton cycle. Thermal integration with the RTG eliminates the hoses between the cabin and the trailer module. Either method of course, reduces the primary electrical power requirements by the amount of electrical heat replaced with thermal heat.

4.6 INTEGRATED AND NO OXYGEN RECOVERY

The thermally integrated life support system which does not recover oxygen from the produced carbon dioxide is nearly identical to the nonintegrated system described in Section 4.3. The difference is that waste heat from the power system is utilized instead of electrical heat to initiate and sustain many of the life support functions. Practically all process heat is supplied thermally. The power system waste heat is transferred to a liquid heat transport medium (Therminol) in the life support heat exchanger and it pumped to the various life support subsystems. The pump requires 31 watts of electrical power for continuous operation and is backed up by a redundant pump, see Figure 4-11. The initial temperature of the liquid is a function of the power system used. Data is presented for two representative levels, 400^{o}F and 600^{o}F. Since some life support subsystems do not require the high temperature waste heat, a heat exchanger and a separate liquid loop is utilized to transfer low temperature (155^{o}F) waste heat to these components. In previous thermal integration studies,

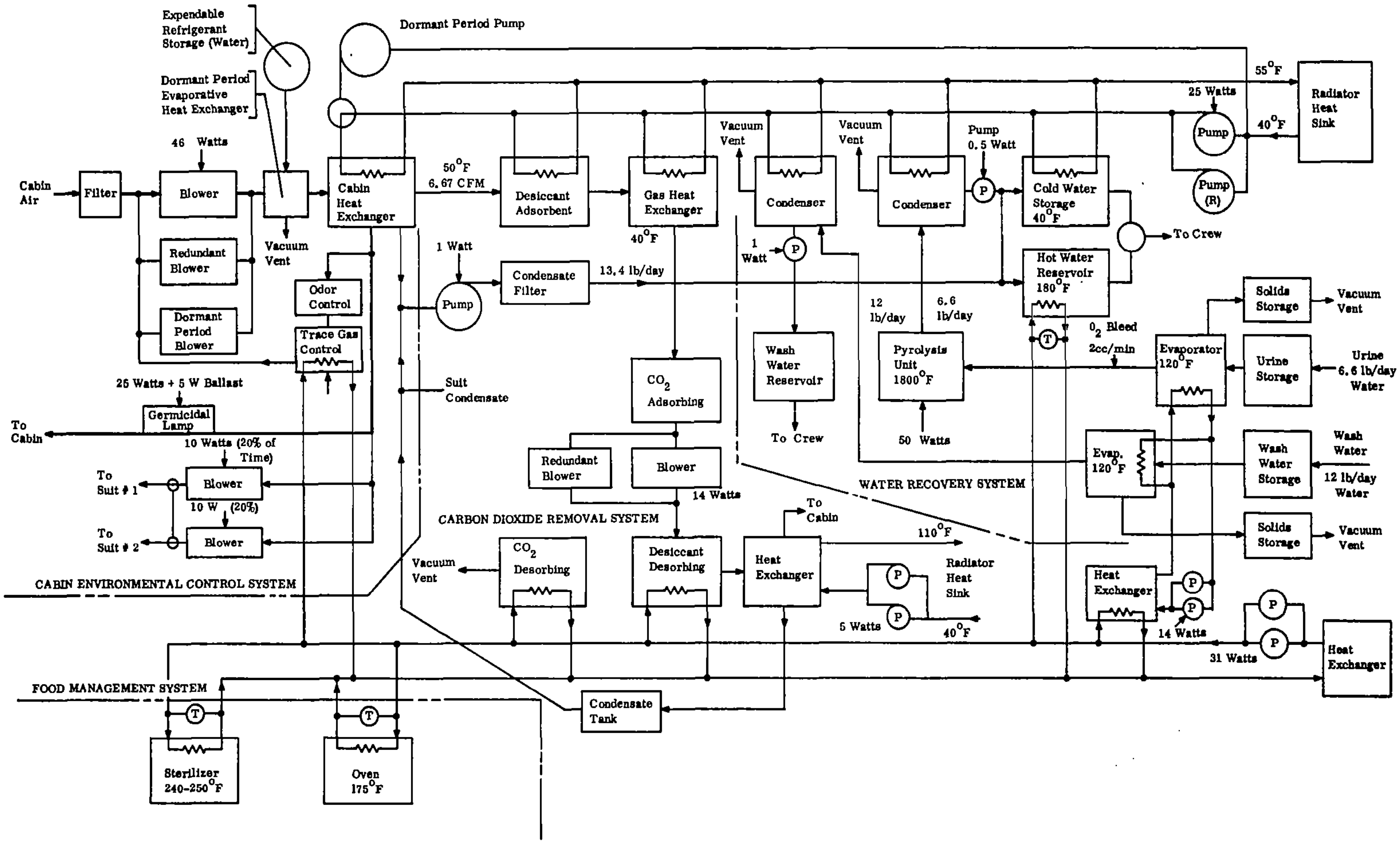


Figure 4-11 Block Diagram of Open Loop Integrated Life Support System

several components required a waste heat temperature between the high and low temperature loops. Consequently these components e.g. sterilizer, oven and hot-water reservoir were heated initially by the 155^{o}F liquid and finally to the desired temperature by electrical energy. However, for this study, all these components are heated from the high temperature liquid loop,and the component temperature is controlled by a thermostatically operated liquid bypass valve. As the component reaches the desired temperature the heating liquid is shunted around the component.

For each process described in the following paragraphs, thermal requirements will be indicated for two temperature levels, 400^{o}F and 600^{o}F, to indicate the dependence on temperature.

4.6.1 CABIN ENVIRONMENTAL CONTROL

Thermal integration of this subsystem results in the use of a catalytic oxidizer for trace gas control, that utilizes waste heat instead of electrical power (50 watts). The 600^{o}F heat transport liquid is utilized "as is"; however if the 400^{o}F liquid is to be used on additional 17 watts of electrical power is required to establish the proper operating temperature. The weight of the unit increases by 1.0 pound.

4.6.2 CARBON DIOXIDE REMOVAL

The desorbing heat required for the silica gel and molecular sieve of 43 and 364 watts is suplied by power system rejected energy in place of electrical energy. With rejected heat being supplied at a minimum of 400^{o}F, there is no need to periodically heat the molecular sieve to a higher temperature to prevent sieve contamination (poisoning). This heating is accomplished during every desorption cycle. This simplifies the controls and reduces their weight by one pound and their power by 2 watts. Heating the sieves to 250^{o}F for regeneration increases the desorption efficiency so that less sieve material is required. A total canister weight of 20 pounds is required when the sieves are regenerated with 600^{o}F waste heat. When 400^{o}F waste heat is utilized, more heat transfer surface is provided so that the total canister weight is 25 pounds. The amount of heat required to regenerate at a 600°F level is 920 btu/hr.

The amount of heat required to regenerate the silica gel desiccant is 64 watts for the 600°F temperature case. However, when lower temperature 400°F waste heat is utilized, additional heat transfer surface must be added to the canisters and a resulting additional 26 watts is required for a total of 390 watts.

4.6.3 WATER RECOVERY

As a result of thermal integration, the hot potable water reservoir is heated to 180°F by the 400°F or 600°F high temperature liquid. A thermostatically controlled liquid bypass valve maintains the reservoir at a maximum temperature of approximately 180°F. When the hot water is dispensed to the 7.0 psia cabin atmosphere, boiling may ensue. This may be a advantageous for proper mixing, however if it is not, the water temperature may be reduced

The urine and wash water evaporators do not require and cannot use the high temperature liquid for evaporation. Pressure control eliminates the requirement for high temperatures since these higher temperatures would tend to volatilize more organics. An intermediate lower temperature 155°F liquid is utilized, for the evaporation process. This liquid (propylene glycol/water, as for the coolant loop) is pumped thru a heat exchanger which collects heat from the high temperature liquid loop and transports the thermal energy to the evaporators. The pump operates continuously on 14 watts of electrical power. Since it is vital to the life support system, a redundant pump is provided.

4.6.4 FOOD MANAGEMENT

The food management oven and sterilizer (see Section 4.3.4) utilize the high temperature liquid for thermal energy instead of electrical power without a significant increase in the heat requirement to operate this equipment.

4.6.5 RESULTS

Table 4-4 summarizes the temperature and amount of waste heat utilized as well as the electrical power requirement for a thermally integrated life support system in which oxygen is not recovered from the carbon dioxide.

4.7 INTEGRATED AND OXYGEN RECOVERY

The thermally integrated life support system which recovers the oxygen from the produced carbon dioxide is nearly identical to the life support which is not thermally integrated (Section 4.4). The differences are the addition of tubing, pumps and heat exchangers necessary to supply heated fluid to the components rather than the use of electrical power for the heating. See Figures 4-12 and 4-13 for the system block diagram.

4.7.1 CABIN ENVIRONMENTAL CONTROL

The basic cabin environmental control system is the same as that described in Section 4.3.1. The modifications brought about by thermal integration are identical in this system to those discussed in Section 4.6.1.

4.7.2 CARBON DIOXIDE REMOVAL AND OXYGEN RECOVERY

The procedure for molecular sieve and silica gel regereration is the same as that described in Section 4.6.2, except the CO_2 gas is not discarded.

The carbon dioxide gas is removed from the sieve by a suction gas pump which stores the carbon dioxide in an accumulator. The pump requires 33 watts for operation and is not back-up by a redundant pump since the whole oxygen recovery system may be shut-down while the pump is repaired.

TABLE 4-4. POWER REQUIREMENTS - INTEGRATED AND NO OXYGEN RECOVERY

Cabin Environmental Control System		Weight (lb.)		Waste Heat (Watts)		Electrical Power (Watts)
		400°F	600°F	400°F	600°F	400°F /600°F
Consisting of:	Air Filter	0.5	0.5	0.0	0.0	0.0
	Blower	2.0	2.0	0.0	0.0	46.0
	Redundant Blower	2.0	2.0	0.0	0.0	0.0
	Container/Expendable Pefrig.	17.0	10.0	0.0	0.0	0.0
	Dormant Period Blower	0.5	0.5	0.0	0.0	0.0
	Evaporative Heat Exchanger	3.0	3.0	0.0	0.0	0.0
	Cabin Heat Exchanger	6.0	6.0	0.0	0.0	0.0
	Odor Control Canister	1.5	1.5	0.0	0.0	0.0
	Trace Gas Control Canister	3.0	2.0	50.0	50.0	17.0
	Germicidal Lamp/Ballast	1.0	1.0	0.0	0.0	30.0
	Glycol Pump	2.0	2.0	0.0	0.0	25.0
	Redundant Glycol Pump	2.0	2.0	0.0	0.0	0.0
	Dormant Period Pump	0.5	0.5	0.0	0.0	0.0
	Suit Blower # 1	1.0	1.0	0.0	0.0	2.0
	Suit Blower # 2	1.0	1.0	0.0	0.0	2.0
	Ducting	10.0	10.0	0.0	0.0	0.0
	Misc. Hardware & Plumbing	4.0	4.0	0.0	0.0	0.0
	Controls	1.0	1.0	0.0	0.0	2.0
		58.0	7.0 lb.	67.0	50.0 watts	124.0 watts

Carbon Dioxide Removal System		Weight (lb.)		Waste Heat (Watts)		Electrical Power (Watts)
		400°F	600°F	400°F	600°F	400°F /600°F
Consisting of:	Desiccant Canister	4.0	3.0	390.0	364.0	0.0
	Desiccant Canister	4.0	3.0	0.0	0.0	0.0
	Gas Heat Exchanger	0.5	0.5	0.0	0.0	0.0
	Mole Sieve Canister	12.5	10.0	334.0	270.0	0.0
	Mole Sieve Canister	12.5	10.0	0.0	0.0	0.0
	Blower	1.0	1.0	0.0	0.0	14.0
	Redundant Blower	1.0	1.0	0.0	0.0	0.0
	Heat Exchanger	2.0	2.0	0.0	0.0	0.0
	Condensate Tank	0.5	0.5	0.0	0.0	0.0
	Pump	0.5	0.5	0.0	0.0	5.0
	Redundant Pump	0.5	0.5	0.0	0.0	0.0
	Misc. Hardware & Plumbing	3.0	3.0	0.0	0.0	0.0
	Controls	1.0	1.0	0.0	0.0	3.0
	Heat Leakage	0.0	0.0	20.0	30.0	0.0
		43.0	36.0 lb.	744.0	664.0 watts	22.0 watts

TABLE 4-4. POWER REQUIREMENTS - INTEGRATED AND NO OXYGEN RECOVERY(CONT'D)

		Weight (lb.) 400°F/600°F		Waste Heat (Watts) 400°F/600°F		Electrical Power (Watts) 400°F /600°F
Water Recovery System						
Consisting of:	Recovery From Urine					
	Urine Storage	2.0	2.0	0.0	0.0	0.0
	Solids Storage	1.0	1.0	0.0	0.0	0.0
	Evaporator	5.0	5.0	81.0	81.0	0.0
	Pyrolysis	10.0	10.0	0.0	0.0	50.0
	Condenser	2.0	2.0	0.0	0.0	0.0
	Condensate Pump	0.5	0.5	0.0	0.0	0.5
	H. E. Condensate Pump	0.5	0.5	0.0	0.0	1.0
	H. E. Condensate Filter	1.5	1.5	0.0	0.0	0.0
	Cold Water Storage	2.5	2.5	0.0	0.0	0.0
	Hot Water Storage	2.5	2.5	9.0	9.0	0.0
	Misc. Hardware & Plumbing	3.0	3.0	0.0	0.0	0.0
	Controls	1.5	1.5	0.0	0.0	3.5
	Heat Leakage	0.0	0.0	20.0	20.0	0.0
		32.0	32.0 lb.	110.0	110.0 watts	55.0 watts
	Recovery From Wash Water					
	Wash Water Storage	3.0	3.0	0.0	0.0	0.0
	Solids Storage	1.0	1.0	0.0	0.0	0.0
	Evaporator	7.0	7.0	147.0	147.0	0.0
	Condenser	2.5	2.5	0.0	0.0	0.0
	Condensate Pump	0.5	0.5	0.0	0.0	1.0
	Wash Water Reservoir	3.0	3.0	0.0	0.0	0.0
	Misc. Hardware & Plumbing	2.5	2.5	0.0	0.0	0.0
	Controls	0.5	0.5	0.0	0.0	2.0
	Heat Leakage	0.0	0.0	20.0	20.0	0.0
		20.0	20.0 lb.	167.0	167.0 watts	3.0 watts
	Low Temperature					
	Liquid/Liquid H. E.	7.0	6.0	0.0	0.0	0.0
	Pump	2.0	2.0	0.0	0.0	14.0
	Redundant Pump	2.0	2.0	0.0	0.0	0.0
	Heat Leakage	0.0	0.0	10.0	20.0	0.0
		11.0	10.0 lb.	10.0	20.0 watts	14.0 watts
Food Management System						
Consisting of	Oven	5.0	5.0	3.0	3.0	0.0
	Sterilizer	7.5	7.5	7.0	7.0	0.0
	Misc. Hardware & Plumbing	2.0	2.0	0.0	0.0	0.0
	Controls	0.5	0.5	0.0	0.0	2.0
	Heat Leakage	0.0	0.0	10.0	20.0	0.0
		15.0	15.0 lb.	20.0	30.0 watts	2.0 watts
Misc. Structure & Enclosure		20.0	20.0 lb.	0.0	0.0 watts	0.0 watts
High Temp. Liquid/Gas H. E.		13.0	10.0	0.0	0.0	0.0
	Pump	2.5	2.5	0.0	0.0	31.0
	Redundant Pump	2.5	2.5	0.0	0.0	0.0
	Heat Leakage	0.0	0.0	30.0	50.0	0.0
		18.0	15.0 lb.	30.0	50.0 watts	31.0 watts

	Weight 400°F/600°F		Waste Heat 400°F/600°F		Electrical Power 400°F /600°F	
TOTALS INTEGRATED SYSTEM NO OXYGEN RECOVERY	217.0	205.0 lb.	1131.0	1091.0 watts	251.0	234.0 watts

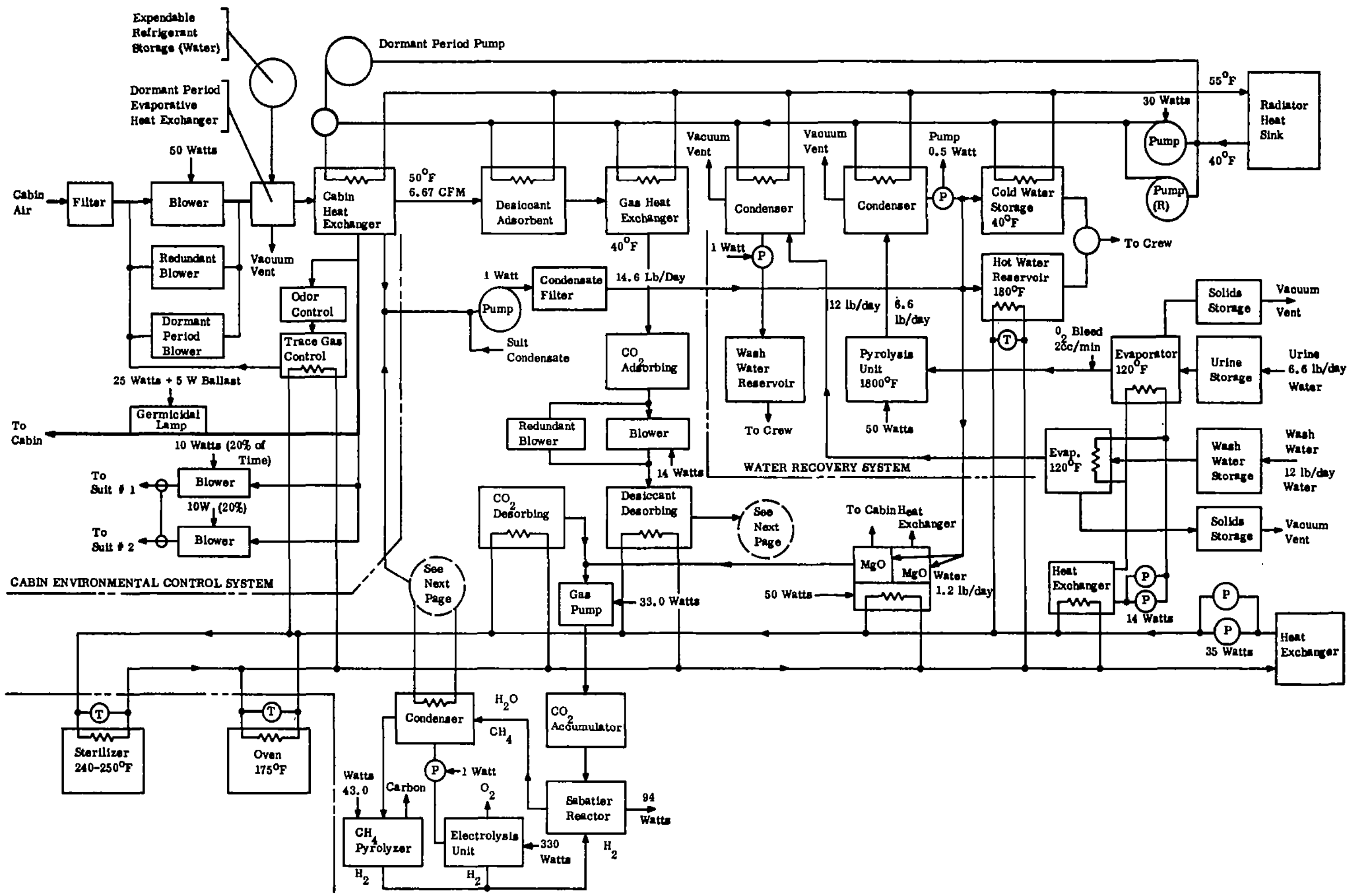


Figure 4-12. Block Diagram of Closed Loop Integrated Life Support System

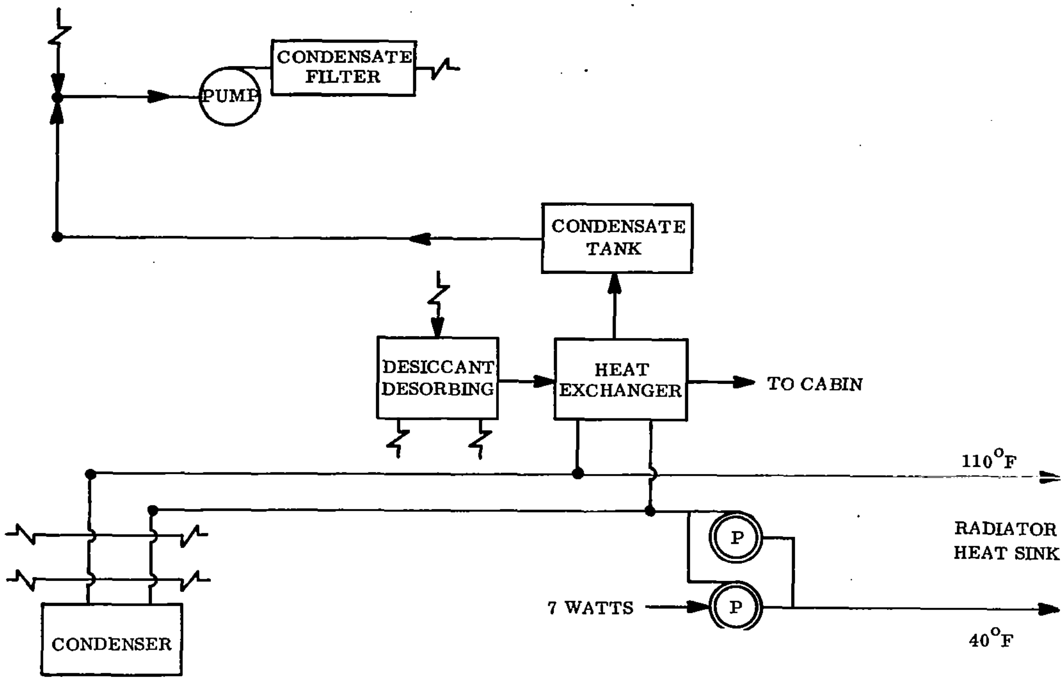


Figure 4-13. Block Diagram of Closed Loop Integrated Life Support System

The carbon dioxide is continuously bled into the Sabatier reactor where in the presence of a catalyst, the carbon dioxide is hydrogenated to form water and methane. The process is exothermic and liberates approximately 94 watts of thermal power. The water is separated from the methane in a condenser and is pumped to the electrolysis unit. The water is electrolyzed to form hydrogen and oxygen at the expenditure of 330 watts of electrical power. The oxygen is returned to the cabin for rebreathing and the hydrogen is returned to the Sabatier reactor for reuse.

The methane separated from the water in the condenser flows to a pyrolysis unit which thermally/catalytically cracks the methane to form carbon and hydrogen. The carbon is discarded while the hydrogen is reused in the Sabatier reactor. A more complete description of the process may be found in Reference 4-1.

4.7.2.1 SUIT CARBON DIOXIDE PROCESSING

The magnesium oxide canisters used to remove CO_2 from the suit atmosphere are the same as those described in Section 4.4.2.1. The regeneration processing (CO_2 removal) is identical to the process described in that section. Thermal heat is used in place of electrical heat to drive off the water (see Figure 4-8). The remaining heat required to separate the MgO and CO_2 is principally supplied by electrical energy.

From 4.4.2.1, the 24 hour average power is 110 watts to process two canisters using 400°F temperature level, 50 watts of this total can be obtained through thermal integration. When the 600°F temperature level is taken, then 60 watts of the 110 watt total can be supplied with waste heat. The remaining heat must be supplied electrically, since 900°F is the final temperature required to complete MgO/CO_2 separation.

4.7.3 WATER RECOVERY

Paragraphs 4.3.3 and 4.6.3 present discussions of the identical water recovery to the one used in this integrated closed loop system.

4.7.4 FOOD MANAGEMENT

The oven and sterilizer use thermal energy in place of electrical energy. The system is otherwise as described in 4.3.4.

4.7.5 RESULTS

Table 4-5 summarizes the temperature and amount of waste heat utilized as well as the electrical power requirement for a thermally integrated life support system in which oxygen is recovered from the carbon dioxide produced by the crew.

TABLE 4-5. POWER REQUIREMENTS (WATTS) - INTEGRATED WITH OXYGEN RECOVERY

		Weight (lb)		WASTE HEAT (WATTS)		ELECTRICAL POWER (WATTS)	
Cabin Environmental Control System							
		400°F	600°F	400°F	600°F	400°F	600°F
Consisting of:	Air Filter	0.5	0.5	0.0	0.0	0.0	0.0
	Blower	2.0	2.0	0.0	0.0	50.0	50.0
	Redundant Blower	2.0	2.0	0.0	0.0	0.0	0.0
	Container/Expendable Refrigerant	17.0	17.0	0.0	0.0	0.0	0.0
	Dormant Period Blower	0.5	0.5	0.0	0.0	0.0	0.0
	Evaporative Heat Exchanger	3.0	3.0	0.0	0.0	0.0	0.0
	Cabin Heat Exchanger	7.0	7.0	0.0	0.0	0.0	0.0
	Odor Control Canister	1.5	1.5	0.0	0.0	0.0	0.0
	Trace Gas Control Canister	3.0	2.0	50.0	50.0	17.0	0.0
	Germicidal Lamp/Ballast	1.0	1.0	0.0	0.0	30.0	30.0
	Glycol Pump	2.0	2.0	0.0	0.0	30.0	30.0
	Redundant Glycol Pump	2.0	2.0	0.0	0.0	0.0	0.0
	Dormant Period	0.5	0.5	0.0	0.0	0.0	0.0
	Suit Blower #1	1.0	1.0	0.0	0.0	2.0	2.0
	Suit Blower #2	1.0	1.0	0.0	0.0	2.0	2.0
	Ducting	10.0	10.0	0.0	0.0	0.0	0.0
	Misc. Hardware & plumbing	4.0	4.0	0.0	0.0	0.0	0.0
	Controls	1.0	1.0	0.0	0.0	2.0	2.0
		59.0	58.0 lb.	50.0	50.0 watts	133.0	116.0 watts
Carbon Dioxide Removal System							
		400°F	600°F	400°F	600°F	400°F	600°F
Consisting of:	Desiccant Canister	4.0	3.0	390.0	384.0	0.0	0.0
	Desiccant Canister	4.0	3.0	0.0	0.0	0.0	0.0
	Gas Heat Exchanger	0.5	0.5	0.0	0.0	0.0	0.0
	Mole Sieve Canister	12.5	10.0	334.0	270.0	0.0	0.0
	Mole Sieve Canister	12.5	10.0	0.0	0.0	0.0	0.0
	Blower	1.0	1.0	0.0	0.0	14.0	14.0
	Redundant Blower	1.0	1.0	0.0	0.0	0.0	0.0
	CO_2 Gas pump	5.0	5.0	0.0	0.0	33.0	33.0
	CO_2 Accumulator	2.0	2.0	0.0	0.0	0.0	0.0
	Sabatier Reactor	10.0	10.0	0.0	0.0	0.0	0.0
	Condenser	2.0	2.0	0.0	0.0	0.0	0.0
	Condenser Pump	1.0	1.0	0.0	0.0	1.0	1.0
	Electrolysis Unit	25.0	25.0	0.0	0.0	330.0	330.0
	CH_4 Pyrolyzer	10.0	10.0	0.0	0.0	43.0	43.0
	Four MgO Canisters (2 spares)	26.0	26.0	0.0	0.0	0.0	0.0
	MgO Regeneration Equipment	8.0	8.0	50.0	60.0	60.0	50.0
	Misc. Hardware & Plumbing	10.5	10.5	0.0	0.0	0.0	0.0
	Controls	4.5	0.0	0.0	0.0	8.0	8.0
	Heat Leakage	0.0	0.0	30.0	40.0	10.0	10.0
		143.0	136.0 lb.	804.0	734.0 watts	506.0	496.0 watts

TABLE 4-5. POWER REQUIREMENTS (WATTS) - INTEGRATED WITH OXYGEN RECOVERY (CONT'D)

CARBON DIOXIDE REMOVAL SYSTEM	Weight (lb)		WASTE HEAT (WATTS)		ELECTRICAL POWER (WATTS)	
Water Recovery System						
Consisting of: Recovery From Urine	400°F	600°F	400°F	600°F	400°F	600°F
Urine Storage	2.0	2.0	0.0	0.0	0.0	0.0
Solids Storage	1.0	1.0	0.0	0.0	0.0	'0.0
Evaporator	5.0	5.0	81.0	81.0	0.0	0.0
Pyrolysis Unit	10.0	10.0	0.0	0.0	50.0	50.0
Condenser	2.0	2.0	0.0	0.0	0.0	0.0
Condensate Pump	0.5	0.5	0.0	0.0	0.5	0.5
H.E. Condensate Pump	0.5	0.5	0.0	0.0	1.0	1.0
H.E. Condensate Filter	1.5	1.5	0.0	0.0	0.0	0.0
Cold water storage	2.5	2.5	0.0	0.0	0.0	0.0
Hot Water Storage	2.5	2.5	9.0	9.0	0.0	0.0
Misc. Hardware & Plumbing	3.0	3.0	0.0	0.0	0.0	0.0
Controls	1.5	1.5	0.0	0.0	3.5	3.5
Heat Leakage	0.0	0.0	20.0	20.0	0.0	0.0
	32.0	32.0 lb	110.0	110.0 watts	55.0	55.0 watts
Recovery From Wash Water	400°F	600°F	400°F	600°F	400°F	600°F
Wash Water Storage	3.0	3.0	0.0	0.0	0.0	0.0
Solids Storage	1.0	1.0	0.0	0.0	0.0	0.0
Evaporator	7.0	7.0	147.0	147.0	0.0	0.0
Condenser	2.5	2.5	0.0	0.0	0.0	0.0
Condensate Pump	0.5	0.5	0.0	0.0	1.0	1.0
Wash Water Reservoir	3.0	3.0	0.0	0.0	0.0	0.0
Misc. Hardware & Plumbing	2.5	2.5	0.0	0.0	0.0	0.0
Controls	0.5	0.5	0.0	0.0	2.0	2.0
Heat Leakage	0.0	0.0	20.0	20.0	0.0	0.0
	20.0	20.0 lb	167.0	167.0 watts	3.0	3.0 watts
Food Management System	400°F	600°F	400°F	600°F	400°F	600°F
Consisting of: Oven	5.0	5.0	3.0	3.0	0.0	:0.0
Sterilizer	7.5	7.5	7.0	7.0	0.0	0.0
Misc. Hardware & Plumbing	2.0	2.0	0.0	0.0	0.0	0.0
Controls	0.5	0.5	0.0	0.0	2.0	2.0
Heat Leakage	0.0	0.0	10.0	20.0	0.0	0.0
	15.0	15.0 lb	20.0	30.0 watts	2.0	2.0 watts
Misc. Structure & Enclosure	20.0	20.0 lb				
High Temp Liquid/Gas H.W.	400°F	600°F	400°F	600°F	400°F	600°F
Pump	13.0 2.5	10.0 2.5	0.0 0.0	0.0 0.0	35.0 0.0	35.0 0.0
Redundant pump	2.5	2.5	0.0	0.0	0.0	0.0
Heat leakage	0.0	0.0	30.0	50.0	0.0	0.0
	18.0	15.0 lb	30.0	50.0 watts	35.0	35.0 watts
Low Temp Liquid/Liquid H.E.	400°F	600°F	400°F	600°F	400°F	600°F
Pump	7.0 2.0	6.0 2.0	0.0 0.0	0.0 0.0	0.0 14.0	0.0 14.0
Redundant pump	2.0	2.0	0.0	0.0	0.0	0.0
Heat Leakage	0.0	0.0	10.0	20.0	0.0	0.0
	11.0	10.0 lb	10.0	20.0 watts	14.0	14.0 watts
Totals Integrated System with oxygen recovery	314.0lb.	[illegible] lb	1173.0 watts	1143.0 watts	741.0 watts	714.0 watts

4.8 SYSTEM CONSIDERATIONS AND DISCUSSION OF RESULTS

Tables 4-6 thru 4-9 itemize the vehicle heat loads for the four basic systems and defines the method by which the heat is rejected. Table 4-10 summarizes the cabin heat loads. Table 4-11 summarizes the electrical power requirements and Table 4-12 summarizes the life support equipment weights for the several systems. Also Table 4-13 itemizes the configuration and size of each component.

The basic system considerations are whether it is best to close the life support loop (e.g. recover oxygen) for the lunar roving vehicle and whether the waste heat from the power system can be efficiently utilized to replace electrical energy as a source of heat. In all comparisons the electrical power requirements are also of prime concern along with the weight/power penalty for the several power systems. It was early determined with the chemical systems that closed loop operation imposed a significant weight penalty. Therefore oxygen is carried in the oxidizer tanks when these power systems are used. The isotope systems do not consume fuel in the sense that the chemical systems do. It might be fruitful then to investigate the weight penalties associated with extra power required in the closed loop systems and compatible with equivalent open loop operation.

	Brayton Power System	Rankine Power System
Integrated	0.81 lb/watt	0.92 lb/watt
Nonintegrated	0.64 lb/watt	0.68 lb/watt

4.8.1 OXYGEN RECOVERY

If no oxygen recovery is utilized, the weight of oxygen consumed by the two men crew during the 45-day mission is:

$$\frac{2.78 \text{ lb oxygen}}{\text{man day}} \times \frac{45 \text{ day}}{\text{mission}} \times \frac{2 \text{ men}}{1} = \underline{250 \text{ lb oxygen}}$$

TABLE 4-6. VEHICLE HEAT LOAD - NONINTEGRATED SYSTEM

PROCESS	ELECTRICAL POWER REQUIRED (WATTS)	EXOTHERMIC (WATTS)	AIR COOLING (WATTS)	LIQUID COOLING (WATTS)
MEN	----	272	272	
Cabin Cooling				
Blower	46		46	
Trace Gas Control	50		50	
Germicidal Lamp	30		30	
Suit Blowers (2)	4		4	
Coolant Pump	25		25	
Controls	2		2	
CO_2 REMOVAL				
Desiccant (Adsorb)	--	89	---	89
Heat Exchanger	--	11	---	11
Sieve (Adsorb)	--	18	18	
Blower	14		14	
Desiccant (Desorb)	364		74	290
Sieve (Desorb)	43		43	
Controls	5		5	
Heat Leakage	20		20	
Pump	5		5	
WATER RECOVERY				
Condensate Pump	1		1	
Urine Evaporator	81		---	
Pyrolysis Unit	50		50	
Condenser	--	81	---	81
Pump	0.5		0.5	
Water Heater	9		9	
Wash Water Evaporator	147			
Condenser	--	147	---	147
Wash Water Pump	1		1	
Controls	5.5		5.5	
Heat Leakage	40		40	
FOOD MANAGEMENT				
Oven	3		3	
Sterilizer	7		7	
Controls	2		2	
Heat Leakage	10		10	
MISC.	600		120	480
Battery Charging				
Navigation and Control				
Communication				
Data Management				
Lighting				
	TOTAL RADIATOR LOAD		857 WATTS	1098 WATTS
			1955 WATTS	

TABLE 4-7. VEHICLE HEAT LOAD - NONINTEGRATED SYSTEM

PROCESS	ELECTRICAL POWER REQUIRED (WATTS)	EXOTHERMIC (WATTS)	AIR COOLING (WATTS)	LIQUID COOLING (WATTS)
MEN	----	272	272	
Cabin Cooling				
Blower	46	46		
Trace Gas Control	50	50		
Germicidal Lamp	30	30		
Suit Blowers (2)	4	4		
Coolant Pump	30	30		
Controls	2	2		
OXYGEN RECOVERY				
Desiccant (Desorb)	----	89	---	89
Heat Exchanger	----	11	---	11
Sieve Absorb	----	18	18	
Blower	14		14	
Desiccant (Desorb)	364		74	290*
Sieve (Desorb)	270		270	
MgO (Desorb)	110		110	
Gas Pump	33		33	
Sabatier Reactor	----	94	94	
Condenser	----	330	---	330*
Electrolysis Unit	330		---	
CH_4 Pyrolyzer	43		43	
Pump	1		1	
Controls	6		6	
Heat Leakage	40		40	
Pump	7		7	
WATER RECOVERY				
Condensate Pump	1		1	
Urine Evaporator	81		---	
Pyrolysis Unit	50		50	
Condenser	----	81	---	81
Pump	0.5		0.5	
Water Heater	9		9	
Wash Water Evaporator	147		---	
Condenser	----	147	---	147
Wash Water Pump	1		1	
Controls	5.5		5.5	
Heat Leakage	40		40	
FOOD MANAGEMENT				
Oven	3		3	
Sterilizer	7		7	
Controls	2		2	
Leakage	10		10	
MISC.	600	120	480	
Battery Charging				
Navigation and Control				
Data Management				
Lighting				
		1393 WATTS	1428 WATTS	
TOTAL RADIATOR LOAD		2821 WATTS		

*Heat dissipated to high temperature radiator = 620 watts

TABLE 4-8. VEHICLE HEAT LOAD, INTEGRATED SYSTEM. NO OXYGEN RECOVERY

PROCESS	ELECTRICAL POWER REQUIRED (WATTS) 400°F	600°F	WASTE HEAT (WATTS) 400°F	600°F	EXOTHERMIC (WATTS)	AIR COOLING (WATTS) 400°F	600°F	LIQUID COOLING (WATTS) 400°F	600°F
MEN					272	272	272		
CABIN COOLING									
Blower	46	46				46	46		
Trace Gas Control	17	0	50	50		67	50		
Germicidal Lamp	30	30				30	30		
Suit Blowers (2)	4	4				4	4		
Coolant Pump	25	25				25	25		
Controls	2	2				2	2		
CO_2 REMOVAL									
Desiccant (Adsorb)					89			89	89
Heat Exchanger					11			11	11
Sieve (Adsorb)					18	18	18		
Blower	14	14				14	14		
Desiccant (Desorb)			390	364		78	74	312*	290*
Sieve (Desorb)						334	270		
Controls	3	3	334	270		3	3		
Heat Leakage			20	30		20	30		
Pump	5	5				5	5		
WATER RECOVERY									
Condensate Pump	1	1				1	1		
Urine Evaporator			81	81					
Pyrolisis Unit	50	50				50	50		
Condenser					81			81	81
Pump	0.5	0.5				0.5	0.5		
Water Heater			9	9		9	9		
Wash Water Evaporator			147	147					
Condenser					147			147	147
Pump	1	1				1	1		
Controls	5.5	5.5				5.5	5.5		
Low Temp. H. E.									
Pump	14	14				14	14		
Heat Leakage			50	60		50	60		
FOOD MANAGEMENT									
Oven			3	3		3	3		
Sterilizer			7	7		7	7		
Controls	2	2				2	2		
Heat Leakage			10	10		10	20		
HIGH TEMP. H. E.									
Pump	31	31				31	31		
Heat Leakage			30	50		30	50		
MISC.	600	600				120	120	480	480
				TOTAL		1252	1217 WATTS	1120	1098 WATTS
				TOTAL RADIATOR LOAD		2372 WATTS AT 400°F & 2315 WATTS AT 600°F			

*Heat dissipated to high temperature radiator 400°F 600°F
312 290

TABLE 4-9. VEHICLE HEAT LOAD - INTEGRATED SYSTEM OXYGEN RECOVERY

PROCESS	ELECTRICAL POWER REQUIRED (WATTS)		WASTE HEAT (WATTS)		EXOTHERMIC (WATTS)	AIR COOLING (WATTS)		LIQUID COOLING (WATTS)	
	400°F	600°F	400°F	600°F		400°F	600°F	400°F	600°F
MEN					272	272	272		
CABIN COOLING									
Blower	50	50				50	50		
Trace Gas Control	17	0	50	50		67	50		
Germicidal Lamp	30	30				30	30		
Suit Blowers (2)	4	4				4	4		
Coolant Pump	30	30				30	30		
Controls	2	2				2	2		
OXYGEN RECOVERY									
Desiccant (Adsorb)					89			89	89
Heat Exchanger					11			11	11
Sieve (Adsorb)					18	18	18		
Blower	14	14				14	14		
Desiccant (Desorb)			390	364		78	74	312*	290*
Sieve (Desorb)			334	270		334	270		
MgO (Desorb)	60	50	50	60		110	110		
Gas Pump	33	33				33	33		
Sabatier Reactor					94	94	94		
Condenser					330	--	--	330*	330*
Electrolysis Unit	330	330				--	--		
CH_4 Pyrolyzer	43	43				43	43		
Pump	1	1				1	1		
Controls	8	8				8	8		
Heat Leakage	10	10	30	40		40	50		
Coolant Pump	7	7				7	7		
WATER RECOVERY									
Condensate Pump	1	1				1	1		
Urine Evaporator			81	81		---	---		
Pyrolysis Unit	50	50				50	50		
Condenser	--	--			81	--	--	81	81
Water Heater	--	--	9	9		9	9		
Pump	0.5	0.5				0.5	0.5		
Wash Water Evaporator			147	147		--	--		
Condenser					147	--	--	147	147
Pump	1	1				1	1		
Controls	5.5	5.5				5.5	5.5		
Low Temp. H. E.	--	--				--	--		
Pump	14	14				14	14		
Heat Leakage	--	--	50	60		50	50		
FOOD MANAGEMENT									
Oven	--	--	3	3		3	3		
Sterilizer	--	--	7	7		7	7		
Controls	2	2	--	--		2	2		
Heat Leakage	--	--	10	20		10	20		
HIGH TEMP H. E.									
Pump	35	35				35	35		
Heat Leakage	--	--	30	50		30	50		
MISC.	600	600				120	120	480	480
					TOTAL	1573	1538 WATTS	1450	1428 WATTS
					TOTAL RADIATOR LOAD	3023 WATTS at 400°F & 2966 WATTS at 600°F			

*Heat dissipated to high temperature radiator 400°F 600°F
642 620

TABLE 4-10. VEHICLE CABIN HEAT LOADS (WATTS)

	NO OXYGEN RECOVERY	OXYGEN RECOVERY
Nonintegrated	1955	2821
Integrated Brayton (400°F)	2372	3223
Integrated Rankine (600°F)	2315	2966

TABLE 4-11. VEHICLE CABIN ELECTRICAL POWER REQUIREMENT (WATTS)

	NO OXYGEN RECOVERY	OXYGEN RECOVERY
Non Integrated	965	1741
Integrated Brayton (400°F)	251	748
Integrated Rankine (600°F)	234	721

TABLE 4-12. LIFE SUPPORT EQUIPMENT WEIGHT (LB)

	NO OXYGEN RECOVERY	OXYGEN RECOVERY
Nonintegrated	184	275
Integrated Brayton (400°F)	217	318
Integrated Rankine (600°F)	205	306

TABLE 4-13. COMPONENT SIZE AND CONFIGURATION

CABIN ENVIRONMENTAL CONTROL SYSTEM	CONFIGURATION	SIZE (IN.)	VOLUME (IN.^{3})
Air Filter	Cylinder	6 Dia x 6 Lg	170
Blower	Cylinder	2 Dia x 4 Lg	13
Redundant Blower	Cylinder	2 Dia x 4 Lg	13
Container/Expendable Refrigerant	Domed Cylinder	8 Dia x 12	468
Dormant Period Blower	Cylinder	1 Dia x 2 Lg	2
Evaporative Heat Exchanger	Cylinder	2 Dia x 4 Lg	13
Cabin Heat Exchanger	Cylinder	6 Dia x 12 Lg	340
Odor Control Canister	Cylinder	4 Dia x 8 Lg	100
Trace Gas Control Canister	Cylinder	4 Dia x 2 Lg	25
Germicidal Lamp (Ballast)	Rectangular	2 x 2 x 6	24
Glycol Pump	Cylinder	3 Dia x 6 Lg	45
Redundant Glycol Pump	Cylinder	3 Dia x 6 Lg	45
Dormant Period Pump	Double Cylinder	3 Dia x 1 Lg 1 Dia x 2 Lg	9
Suit Blower #1	Cylinder	2 Dia x 4 Lg	13
Suit Blower #2	Cylinder	2 Dia x 4 Lg	13
Ducting	---	---	---
Miscellaneous Hardware and Plumbing	---	---	---
Controls	Rectangular	2 x 2 x 4	16
CARBON DIOXIDE REMOVAL SYSTEM			
Desiccant Canister	Cylinder	6 Dia x 8 Lg	220
Desiccant Canister	Cylinder	6 Dia x 8 Lg	220
Gas Heat Exchanger	Cylinder	2 Dia x 4 Lg	13
Mole Sieve Canister	Cylinder	8 Dia x 12 Lg	600
Mole Sieve Canister	Cylinder	8 Dia x 12 Lg	600
Blower	Cylinder	2 Dia x 4 Lg	13
Redundant Blower	Cylinder	2 Dia x 4 Lg	13
*CO_2 Gas Pump	Cylinder	4 Dia x 6 Lg	75
*CO_2 Accumulator	Domed Cylinder	4 Dia x 8 Lg	80
*Sabatier Reactor	Cylinder	4 Dia x 6 Lg	75
*Condenser	Conical	4 Dia x 4 Lg	17
*Condenser Pump	Double Cylinder	3 Dia x 1 Lg 1 Dia x 2 Lg	9
*Electrolysis Unit	Rounded End Cylinder	12 Dia x 10 Lg	1130
*CH_4 Pyrolyzer	Domed Cylinder	4 Dia x 8 Lg	80
*MgO Canisters (4)	Cylinder	4 Dia x 4 Lg x (4)	68
*MgO Regeneration Equipment	Rectangular	6 x 10 x 10	600
Heat Exchanger	Cylinder	4 Dia x 8 Lg	80
Condensate Tank	Cylinder	2 Dia x 6 Lg	19
Coolant Pump	Double Cylinder	3 Dia x 1 Lg 1 Dia x 2 Lg	9
Redundant Coolant Pump	Double Cylinder	3 Dia x 1 Lg 1 Dia x 2 Lg	9
Miscellaneous Hardware and Plumbing	---	---	---
Controls	Rectangular	4 x 4 x 4	64
WATER RECOVERY SYSTEM			
Recovery From Urine			
Urine Storage	Domed Cylinder	4 Dia x 8 Lg	80
Solids Storage	Domed Cylinder	6 Dia x 10 Lg	226
Evaporator	Double Cylinder	10 Dia x 4 Lg 2 Dia x 4 Lg	326
Pyrolysis Unit	Cylinder-Dome One End	6 Dia x 16 Lg	433
Condenser	Conical	4 Dia x 4 Lg	17
Condensate Pump	Double Cylinder	3 Dia x 1 Lg 1 Dia x 2 Lg	9
H. E. Condensate Pump	Double Cylinder	3 Dia x 1 Lg 1 Dia x 2 Lg	9
H. E. Condensate Filter	Cylinder	4 Dia x 6 Lg	75
Color Water Reservoir	Domed Cylinder	6 Dia x 10 Lg	226
Hot Water Reservoir	Domed Cylinder	6 Dia x 10 Lg	226
Miscellaneous Hardware and Plumbing	---	---	---
Controls	Rectangular	2 x 2 x 4	16
Recovery From Wash Water			
Wash Water Storage	Domed Cylinder	6 Dia x 10 Lg	226
Solids Storage	Domed Cylinder	4 Dia x 8 Lg	80
Evaporator	Double Cylinder	10 Dia x 5 Lg 2 Dia x 4 Lg	404
Condenser	Conical	4 Dia x 4 Lg	17
Condensate Pump	Double Cylinder	3 Dia x 1 Lg 1 Dia x 2 Lg	9
Wash Water Reservoir	Domed Cylinder	6 Dia x 10 Lg	226
Miscellaneous Hardware and Plumbing	---	---	---
Controls	Rectangular	2 x 2 x 4	16
FOOD MANAGEMENT SYSTEM			
Oven	Rectangular	6 x 6 x 8	288
Sterilizer	Rectangular	6 x 6 x 8	288
Miscellaneous Hardware and Plumbing	---	---	---
Controls	Rectangular	1 x 1 x 2	2
**HIGH TEMPERATURE LIQUID/GAS H. E.			
**High Temperature Liquid/Gas H. E.	Cylinder	4 Dia x 10 Lg	126
**Pump	Double Cylinder	3 Dia x 1 Lg 2 Dia x 3 Lg	17
**Redundant Pump	Double Cylinder	3 Dia x 1 Lg 2 Dia x 3 Lg	17
**Low Temperature Liquid/Liquid H. E.	Rectangular	4 x 4 x 4	64
**Pump	Double Cylinder	3 Dia x 1 Lg 2 Dia x 2 Lg	14
**Redundant Pump	Double Cylinder	3 Dia x 1 Lg 2 Dia 2 Lg	14

*For systems which require oxygen recovery only.
**For thermally integrated systems only.

With leakage and contingency allotments, the total oxygen requirement for 45 days would be 348, or 268 lb above the closed loop system. Based on data in Reference 4-3 a storage tank with insulation for oxygen would weigh approximately 80 lb. The Recovery of oxygen from CO_2 introduces additional equipment and requires more power (see Tables 4-11, 4-12). Figure 4-14 illustrates the time/weight relationship of the open loop process and the weight comparison when oxygen is recovered. In every case, the open loop system is lighter for the 45-day mission. However, if as much as two missions were desired during the life or any one lunar vehicle, then oxygen recovery represents a weight saving.

4.8.2 WATER RECOVERY

If no water were recovered from the urine and wash water the weight of water consumed and used by the two man crew during the 45-day mission is:

$$\frac{\text{2.8 LB URINE WATER RECOVERED}}{\text{man day}} \times \frac{\text{45 DAY}}{\text{mission}} \times \text{2 MEN} = \frac{\text{252 LB}}{\text{mission}}$$

$$\frac{\text{6 LB WASH WATER RECOVERED}}{\text{man day}} \times \frac{\text{45 DAY}}{\text{mission}} \times \text{2 MEN} = \frac{\text{540 LB}}{\text{mission}}$$

$$\text{TOTAL} \quad \frac{\text{792 LB}}{\text{45-DAY MISSION}}$$

Considering 20 percent of the liquid weight for tankage, the total weight equals 950 pounds.

The Water Recovery System weighs 50 pounds for a nonintegrated system and 52 pounds for an integrated system. The electrical power requirements are 335 watts and 58 watts respectively for nointegrated and integrated systems.

BRAYTON POWER SYSTEM WEIGHT:

NONINTEGRATED

$$\text{50 lb (equipment)} + \text{335 watts} \times \frac{\text{0.64 lb}}{\text{watt}} = \text{264 pounds}$$

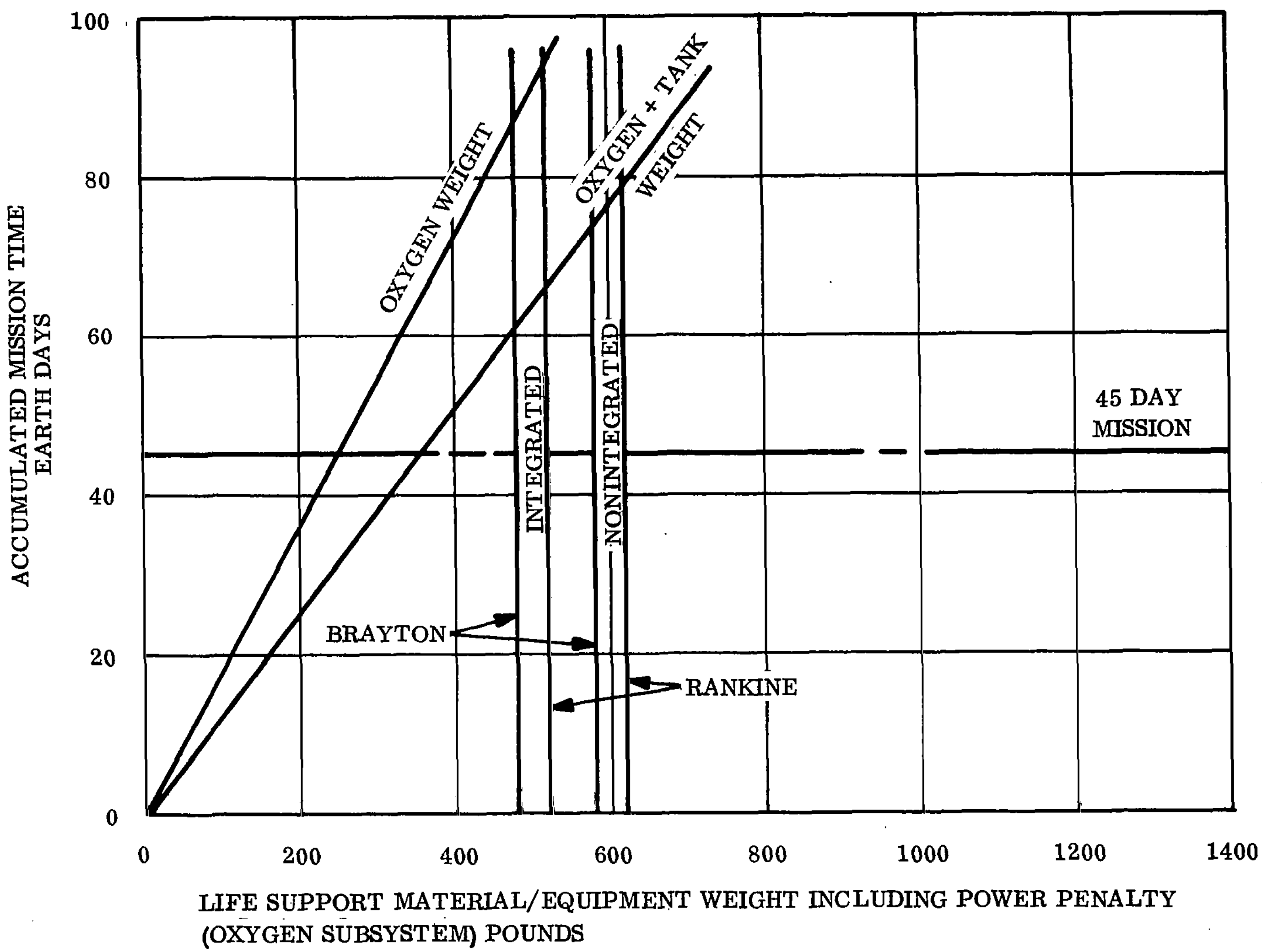


Figure 4-14 Comparison of Oxygen Recovery With Oxygen Storage for Isotopic Power Systems

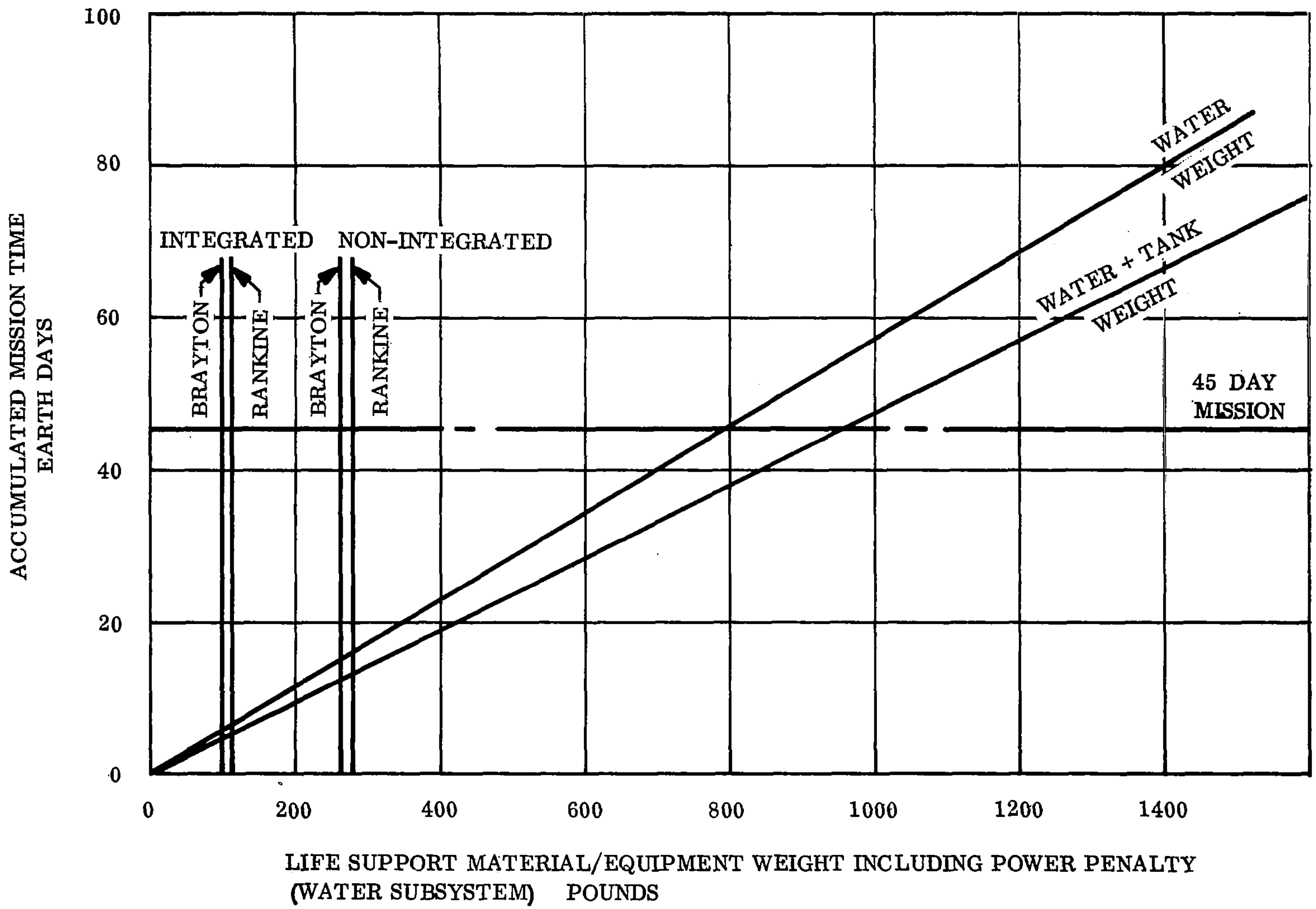


Figure 4-15. Comparison of Water Recovery With Water Storage For Isotopic Power Systems

INTEGRATED						
50 lb (equipment)		58 watts	x	$\frac{.81 \text{ lb}}{\text{watt}}$	=	99 pounds

WEIGHT:

NONINTEGRATED						
50 lb (equipment)		335 watts	x	$\frac{0.68 \text{ lb}}{\text{watt}}$	=	278 pounds
INTEGRATED						
52 lb (equipment)	+	58 watts	x	$\frac{0.92 \text{ lb}}{\text{watt}}$	=	105 pounds

Figure 4-15 illustrates the time/weight relationship of discarding water and the comparison of equipment and power weight for processing and re-using waste water. The time equivalence is around 13 days, comparing against nonintegrated systems and 5 days comparing against integrated systems. Clearly, water processing is much lighter than discarding water for the lunar roving vehicle 45-day mission.

4.8.3 Discussion

The increase in weight of life support equipment with thermal integration is minor compared to the decrease in power system size and weight. As can be seen in Figures 4-14 and 4-15; the decrease and power system equipment is on the order of 300 pounds for thermally integrated systems. From a long term viewpoint, the closed processes should generally be chosen over open CO_2 or water subsystems. Another consideration (not a part of this study) would be use of open loop processes to minimize isotope fuel requirements. For limited roving vehicle life, nonrecovery of oxygen would save 500 electrical watts, or two and four isotope thermal kilowatts in the Brayton and Rankine heat sources (integrated vehicles). Of course, approximately 8 lb/ day of consumed weight appears to supply oxygen.

The closed loop system has better flexibility for extending missions and for temporarily increasing crew size, for rescue or transport missions. Additional thermal energy is available especially from the Rankine system to operate at a higher process cycle rate, enabling additional CO_2 removal and oxygen recovery. The limit imposed for this mode of operation would be the cabin radiator heat rejection capability.

4.8.4 Cabin Heat and Temperature Transients

There are several times during the mission profile when the heat input into the cabin atmosphere and liquid coolant loops may exhibit a significant change. Several of the major heat transients are listed below:

a. Dormant to active period start-up.
b. When men exit from vehicle.
c. Period when men are less active (sleeping).
d. Variance of solar heat flux (Lunar day and nite).
e. Equipment failures.
f. Emergency decompression or fire.
g. Normal variation of equipment heat load.

4.8.4.1 Start-Up

During the lunar roving vehicle start-up (transition from dormant to active period) it is recommended that all systems be operating 8 hours before the men use the cabin life support system. This permits a less critical transition from the 100 watts RTG cabin heat load to the possible 3223 watts cabin heat load of normal vehicle operation. Also this period may be used to remotely checkout all systems before the men enter the vehicle. The vehicle thus has sufficient time to stabilize the cabin environment before the crew arrives.

4.8.4.2 All Other Normal Conditions

All other normal transient heat input conditions such as when the crew exits from the vehicle, periods of decreased crew activity (sleeping), variance of solar heat flux (lunar day and nite), etc. are dampened by the large thermal mass of the vehicle. The cabin weighs

4000 pounds and has an average specific heat of 0.2 btu/ lb$^{\circ}$F; consequently, a heat input of 235 watts (800 btu/hr) is required to raise the temperature 1°F. This is a significant thermal mass and will tend to dampen the effect of transient heat loads.

4.8.4.3 Equipment Failures and Emergencies

Equipment failures will generally decrease the thermal load but the vehicle thermal mass will dampen any rapid temperature change until the environmental control system stabilizes.

During emergencies such as fires or decompression all noncritical systems will be shut down. These noncritical systems are water recovery, oxygen recovery (if used) cooking, sterilizing, etc. Only the suit environmental control system, carbon dioxide removal and minimal lighting will be utilized until the emergency has been alleviated. Upon return to normal operation it may be necessary for the crew to remain suited until the cabin temperature has stabilized at an acceptable level since the cabin temperature can not be controlled if the cabin is voided of atmosphere.

4.8.4.4 Normal Variations

Normal variations in the equipment heat load is minimized by the continuous operation of the carbon dioxide removal, oxygen recovery (if used), and water recovery systems. These systems dissipate the bulk of the thermal energy into the cooling system at a nearly constant rate. If sufficient waste heat is not available from the power supply system to operate the life support systems at all times (periods when LRV is not mobile), the water recovery and regeneration of the MgO suit back pack canister may be delayed until sufficient waste heat is available. Approximately the same amount of energy is required so that more heat will be required for a shorter period of operation.

4.8.4.5 Summary

Generally the thermal mass of the vehicle will minimize rapid temperature transients. Special procedures are required for start-up and after emergencies to stabilize the cabin temperature at an acceptable level.

4.9 REFERENCES

4-1 Hanson, K. L., Thermal Integration of Electrical Power and Life Support Systems for Manned Space Stations, Missile and Space Division of General Electric Co., Final Report on Contract NAS 3-2799, issued as NASA Document CR316, November, 1965.

4-2 Woods, R. W. and Erlanson, E. P., "Study of Thermal Integration and Life Support Systems for Manned Space Stations", General Electric, MSD Document Number 66SD4231, Final Report for NASA Contract NAS 3-6478, January 1966.

4-3 Study of Human Factors and Environmental Control-Life Support Systems, Lunar Exploration Systems for Apollo, AiResearch Manufacturing Division of Garrett, Final Report for NAS 8-11447, April, 1965 (6 Volumes).

4-4 Colombo and Mills (Douglas), "Regenerative Separation of CO_2 via Metallic Oxides", AICLE preprint 26D, Philadelphia, Penna., 12/5 - 12/9, 1965.

4-5 ASD-TR-61-162 Part II, "Analytical Methods for Space Vehicles Atmospheric Control Processes", 11-62.

SECTION 5
CABIN HEAT REJECTION

5.1 INTRODUCTION

The lunar roving vehicle cabin during its mission will have heat generated in it by:

a. Life support equipment
b. Electronic equipment
c. The crew

This heat must be removed from the cabin and rejected to maintain cabin steady state conditions. The previous section has examined the heat balances for the life support equipment and crew. The cabin air is passed through an air to liquid heat exchanger to maintain its temperature. Much of the life support equipment heat is removed by circulation of a similar liquid through cooling coils. This section examines the radiator and control requirements following the heat exchangers.

The electronic equipment is to be cooled by the "cold plate" method. Absorption refrigeration is analyzed in conjunction with the electronic equipment to determine its potential for lunar roving vehicle use.

5.2 ELECTRONIC EQUIPMENT COOLING

The electronic equipment used in the lunar roving vehicle is contained in two consoles near the forward part of the cabin (see Figure 5-1). This equipment is part of the navigation, communications, command, telemetry and data management subsystems. The average dissipation of this equipment when the vehicle is active (during a 45-day mission) is 600 watts. The previous thermal integration study (Reference 5-1) considered cooling of electronic equipment by using cold plates with the component heat dissipators mounted on them. The component waste heat is transferred to the plate and then to a cooling fluid by conduction.

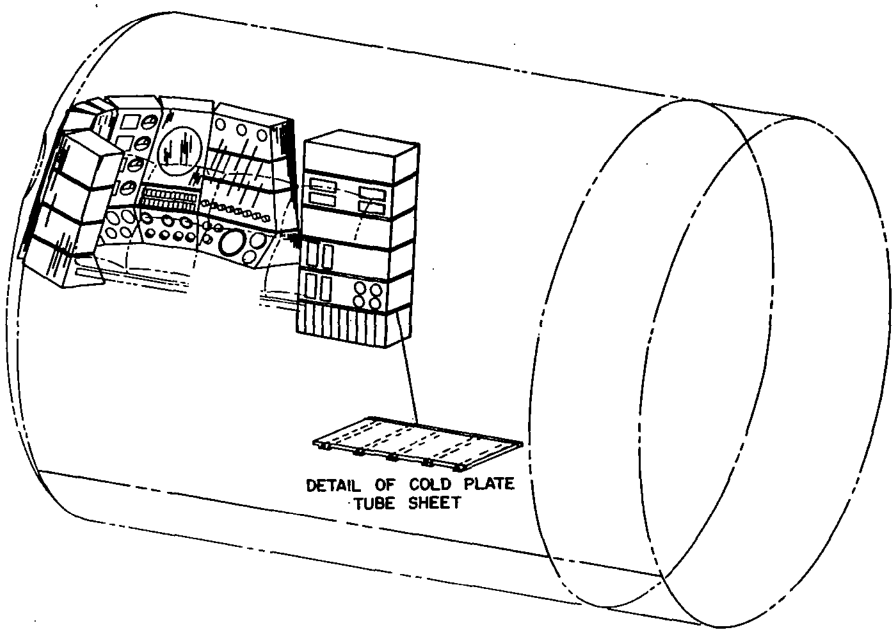


Figure 5-1. Electronic Equipment

The fluid is in turn pumped through the space-oriented cabin radiator to reject the heat. This concept is used with the lunar roving vehicle electronic equipment to reject 80% of its heat (480 watts). It is estimated that 20% of this heat (120 watts) is removed by the cabin air and rejected through the cabin environmental control heat exchanger and radiator.

The cold plate flow control is accomplished by temperature control between specified limits. The lower temperature limit is the cabin air dew point, in the neighborhood of 50^{o}F. The upper limit is probably set by the cabin air temperature, since it is desired to remove as much of this heat as possible directly by the liquid. The preferred operating point is then 50^{o}F, which is used in succeeding calculations in this section.

5.3 ABSORPTION REFRIGERATION SYSTEM

5.3.1 CYCLE ANALYSIS

This method of alleviating rejection temperature levels is here considered for use with the electronic equipment. It is assumed that the fluid to the cold plates will be maintained at 50°F with a 10°F temperature difference from the refrigeration system operating point of 40°F.

A mechanical vapor compression system requires a relatively large power input to compress the working fluid from the operating pressure of the evaporator to that of the condenser. If we introduce a liquid into the system before the compressor we can realize a significant reduction in compressor power requirements. Such a system is shown schematically in Figure 5-2.

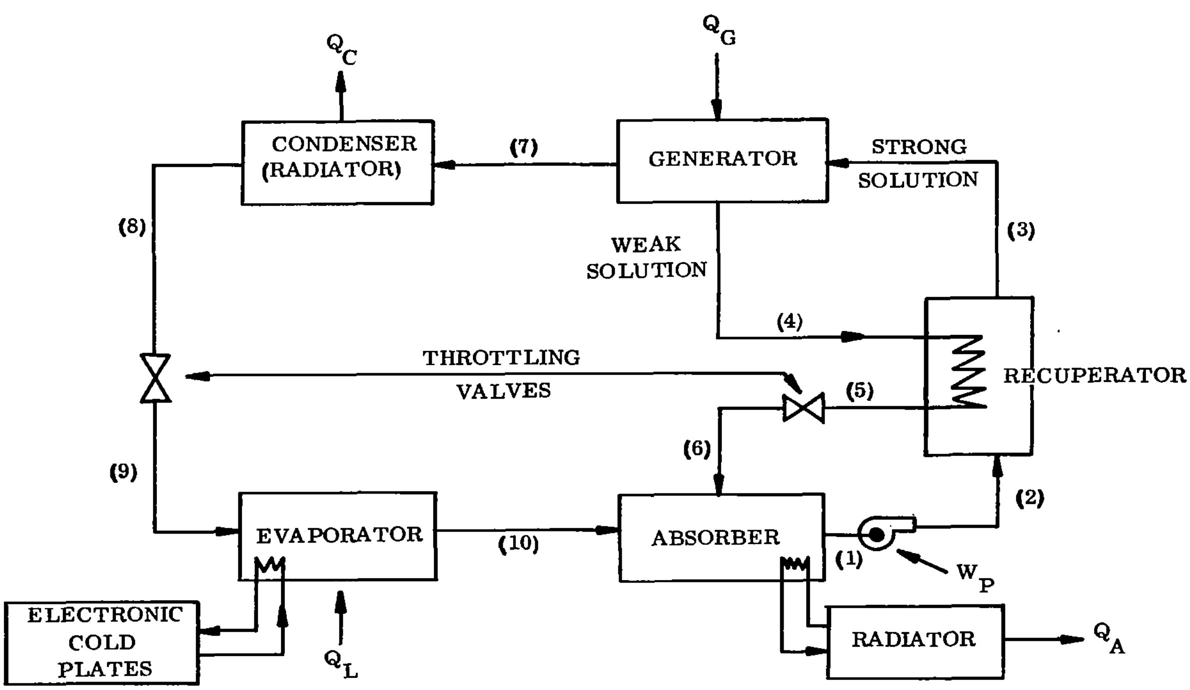


Figure 5-2. Schematic of Single Stage Absorption Refrigeration System

This system requires a significantly higher heat input as compared with a vapor compression system with the benefit of a much lower pumping power. The most commonly used working fluids are ammonia-water and lithium bromide-water. Several variations of these systems have been examined in Reference 5-1. The most efficient choice for this application is LiBr-H_2O because of its higher coefficient of performance (COP). The basic process is summarized below:

a. Saturated steam from the evaporator is absorbed by a weak LiBr-H_2O solution and compressed to the pressure level consistent with the generator saturation pressure. For the LiBr-H_2O system this is an exothermic process.

b. Part of the power system waste heat is used to evaporate a portion of the absorbed water; the remaining solution is returned through a throttling valve to the absorber. A portion of the heat content of the weak solution is utilized by means of a recuperator.

c. The steam leaving the generator is condensed to saturated water. This water is then throttled back to the evaporator saturation state.

d. The saturated water entering the evaporator is used to provide cooling to the main heat load.

The cycle conditions assumed are the same as those given in Reference 5-1 and are summarized below:

Evaporation temperature	=	$40^{\circ}F$
Condenser and absorber temperature	=	$100^{\circ}F$
Generator temperature	=	$200^{\circ}F$

(1) All of the throttling valves are isenthalpic.

(2) Cycle points 1, 3, 4, 7, 8 and 12 are saturated conditions.

(3) Negligible pressure losses in the components and lines.

(4) Pump work is negligible.

A summary of the thermodynamic state points for the assumed cycle are summarized in Table 5-1.

TABLE 5-1. ABSORPTION REFRIGERATION CYCLE POINTS (EVAPORATOR LOAD = 1636 Btu/Hr)

Point Cycle	Pressure (mm Hg)	Temperature (°F)	Quality (% LiBr)	Enthalpy (Btu/Pound)	Weight Flow Rate (Pounds/Hour)
1	6.29	100	.60	-70	20.97
2	49.1	100	.60	-70	20.97
3	49.1	180	.60	-35	20.97
4	49.1	200	.65	-27	19.35
5	49.1	110	.65	-65	19.35
6	6.29	110	.65	-65	19.35
7	49.1	200	0	1151	1.615
8	49.1	100	0	68	1.615
9	6.29	40	0	68	1.615
10	6.29	40	0	1079	1.615

The overall heat balance for this system is given below:

Component	Heat Input (Btu/Hr)	Heat Rejected (Btu/Hr)
Absorber		1749
Generator	2070	
Condenser		1953
Evaporator	1636	
Total	3706	3702

The calculated COP for this system is 0.79. Allowing for system heat losses, if we take a COP of 0.75 this results in a required generator heat load of 635 watts. This was the heat load used in sizing the generator in the integrated systems analyzed.

5.3.2 EQUIPMENT SIZING

The equipment size estimates are based primarily on extrapolating the size and weight figures presented in Reference 5-1.

In most instances we have assumed that the weight is linearly proportional to the cooling load level. The pump power is approximately proportional to the cooling load cubed. The following table summarizes the estimated system weight.

TABLE 5-2. ABSORPTION REFRIGERATION EQUIPMENT WEIGHT AND PUMP POWER

Unit	Weight (lb)	Pump Power (watts)
Evaporator	1.3	0.04
Condenser (Radiator)	8.0	0.01
Generator	0.9	0.01
Absorber	1.3	0.01
Recuperator	0.4	0.01
Pump	0.5	
Misc. Structure	0.5	
Absorber radiator	7.6	0.01
Totals	20.5	0.09
Saving of power system radiator	-3.0	
	17.5 (lb)	

Figure 5-1 shows a conceptual design of the electronics racks and the cold plates for cooling. The cold plates essentially serve as the evaporator. The electronics load was estimated to be on the order of 600 watts. A power density of 100 watts/cu ft is assumed for the purposes of showing a specific size. If the absorption refrigeration system is replaced with a low temperature ($T_{in} = 55^oF$, $T_{out} = 50^oF$) radiator with the OSR coating $\alpha_s = 0.05$, $\epsilon = 0.84$, it is estimated that the radiator would weigh on the order of 14.3 lb. It appears therefore that the absorption refrigeration system in this system has no particular advantage either in saving radiator area or weight. The reason for this is of course, the fact that the waste heat output of this system is so much higher than the heat load for a low temperature radiator. The sink temperature is also 150^oF less than the radiator outlet; the increase in the radiator temperature is too small to offset the increase in heat load.

5.4 CABIN HEAT REJECTION INTEGRATION

5.4.1 DESCRIPTION

The required cooling loops and their radiators for cabin heat rejection are described in the following paragraphs. In the previous study (Reference 5-1) all of the heat loads were combined into a single coolant loop. In view of the fact that the minimum radiator area is desirable, we can distinguish between the CO_2 removal system and the other elements in the life support system in order to maximize the radiating temperature. The basic cabin system elements, including the electronic equipment, are shown below.

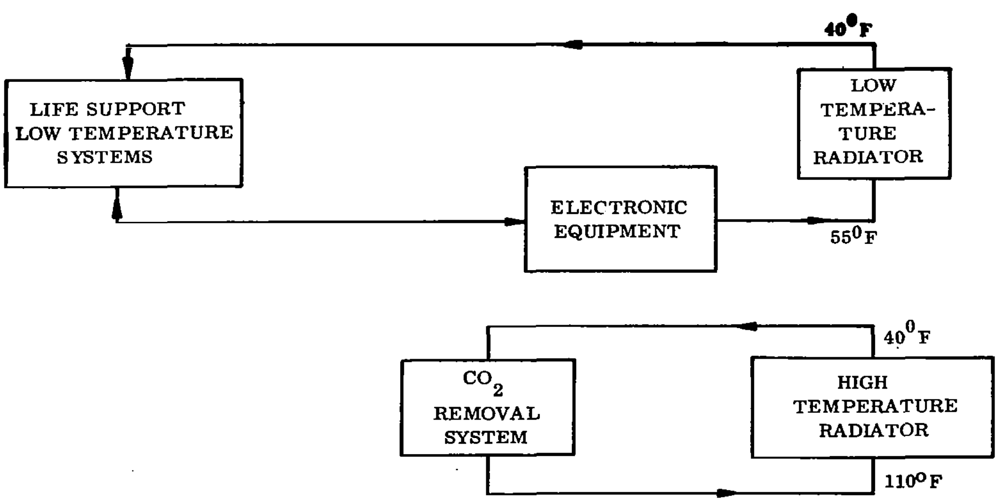


Figure 5-3. Cabin Heat Rejection System Block Diagram

Since the absorption refrigeration system does not provide advantages for this application, it is not considered as part of the heat removal loops.

The following table summarizes the heat loads for each of the aforementioned systems. A more detailed breakdown of these heat loads is given in Section 4.

TABLE 5-3. RADIATORS HEAT LOAD

	Electronic Equip. (watts)	Life Support Low Temperature (watts)	CO_2 removal System (watts)
Nonintegrated			
With O_2 recovery	480*	1695	621
Without O_2 recovery		1179	291
Integrated			
With O_2 recovery			
400° Heat Input		1876	642
600° Heat Input		1841	621
Without O_2 recovery			
400°F Heat Input		1556	312
600°F Heat Input		1521	291

*80% of the heat load is assumed to pass into liquid cooling loop, the remainder ends up as part of the cabin environmental control system heat load.

Figure 5-4 gives the estimated radiator weight and area as a function of heat load for both the low-temperature and high-temperature radiator loops. The assumed radiator characteristis are given below.

Coating: OSR Laminate
Solar Absorptance $(\alpha_s) = 0.05$
Emittance $(\epsilon) = 0.84$

Radiator specific weight (incl. coating) = 0.5 lb/ft^2
Overall radiator efficiency = 0.95

The following table summarizes the radiator area and weight for the various life support systems.

TABLE 5-4. RADIATOR AREAS AND WEIGHT

	Low Temperature Loop		High Temperature Loop	
Nonintegrated	Wt (lb)	Area (sq ft)	Wt (lb)	Area (sq ft)
With O_2 recovery	61.5	123	17.5	35.0
Without O_2 recovery	47.0	94	8.2	16.4
Integrated				
With O_2 recovery				
400°F Heat Input	66.5	133	17.8	35.6
600°F Heat Input	66.0	132	17.5	35.0
Without O_2 recovery				
400°F Heat Input	57.5	115	8.5	17.0
600°F Heat Input	57.0	114	8.2	16.4

5.4.2 CONTROL REQUIREMENTS

The cabin heat rejection system is susceptible to heat rejection load variation due to: (1) environmental temperature variations; and (2) cabin load variations. Several means are available to maintain stable system temperatures necessary to operate at peak system efficiency. These are summarized below:

a. Use of a bypass line to maintain a constant heat rejection element (e.g., radiator or heat exchanger) at a constant inlet temperature.

b. Use of a regenerative heat exchanger to maintain a fixed temperature drop across the radiator.

c. Allow selective freezing of the fluid in the radiator tubes (Apollo concept). This is ostensibly similar to the bypass concept.

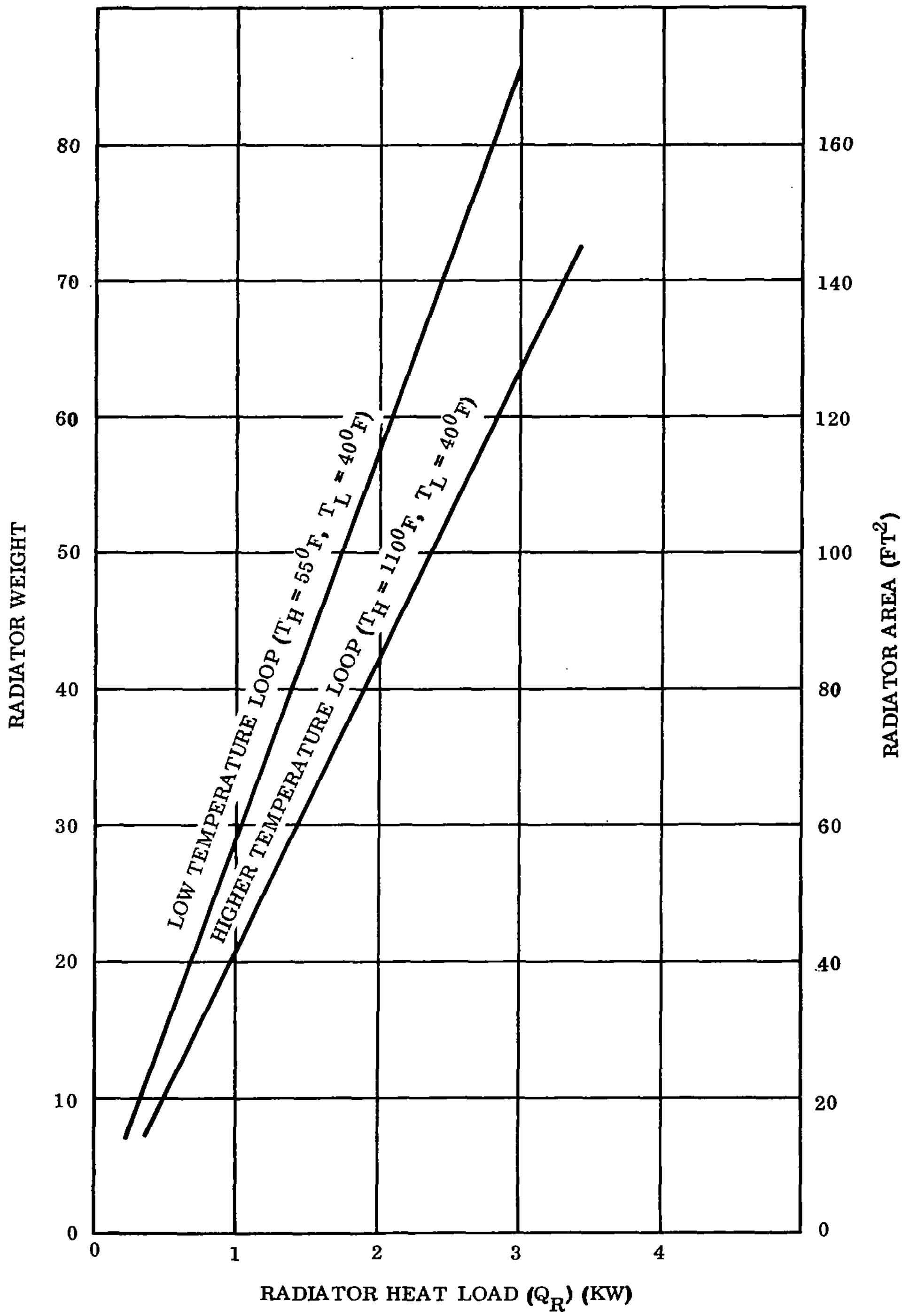


Figure 5-4. Life Support System Radiator Parameters

d. Use of a louvered radiator surface.

e. By suitable valving of a variable tube radiator.

f. Use of heat storage devices.

The use of a regenerative heat exchanger is in most cases the most accurately controllable system arrangement. Using a bypass line results in lowering the radiator outlet temperature at the expense of large viscosity variation in the fluid. Selective freezing of the fluid has innumerable problems associated with predicting performance, and in being able to recover full flow capacity when required in a relatively short period of time. The use of a louvered radiator surface is a poor choice for several reasons:

(1) Louvers add weight.
(2) They require large radiator area because of reduced effective surface emittance.

A variable tube radiator concept is of questionable reliability and complexity. In addition it represents a fairly severe weight penalty for the valving plus controls.

The use of the latent heat of fusion of a solid melting at or near the desired operating point, or the latent heat of vaporization or fusion is another means to accommodate load variations.

The major disadvantages of this device are:

(1) It allows potentially excessive variations in fluid viscosity with the attendant pump power penalties.

(2) Results in high weight for significant load-varying capability.

Reference 5-1 discusses in greater detail some examples of the comparison between a bypass configuration and regenerative configuration. In all of the system radiator designs analyzed, the radiator has been sized to accommodate the peak load since the majority of the power transients are of sufficiently long duration to prevent taking advantage of the heat capacity of the radiator. The regenerator must be sized in order to reduce the desired system outlet temperature to a value compatible with radiator design capability at the very lowest operating power level. In addition, the variation in radiator sink temperature between the peak value at the time of highest power dissipation and the lowest sink temperature at time of lowest power dissipation must be accounted for. The necessary operating conditions (e.g., regenerator flow rate) for intermediate conditions can be arrived at by the following method:

(1) Assume a cold side weight flow rate (hot side weight flow rate fixed by power level).

(2) Calculate the Reynolds number and Colburn modulus for the hot and cold sides.

(3) Calculate the hot and cold side film coefficients.

(4) Calculate the number of transfer units.

(5) From the number of transfer units the heat exchanger effectiveness can be estimated.

(6) With known hot side and cold side inlet temperatures and a known hot side outlet temperature, calculate the cold side outlet temperature.

From this procedure the precise relationship between system heat load, effective sink temperature and the required amount of regeneration can be established.

The preceding discussion is primarily applicable to the longer term variations such as encountered with the lunar environment. For short term transients, the cabin's large thermal mass will provide sufficient control (see Section 4).

5.5 REFERENCES

5-1. Woods, R. W. and Erlanson, E. P., "Study of Thermal Integration of Electric Power and Life Support Systems for Manned Space Stations, General Electric Company, Document No. 66SD4231, Final Report for Contract NAS 3-6478, January 26, 1966.

SECTION 6

HYDROGEN-OXYGEN ENGINE POWER SYSTEM

6.1 INTRODUCTION

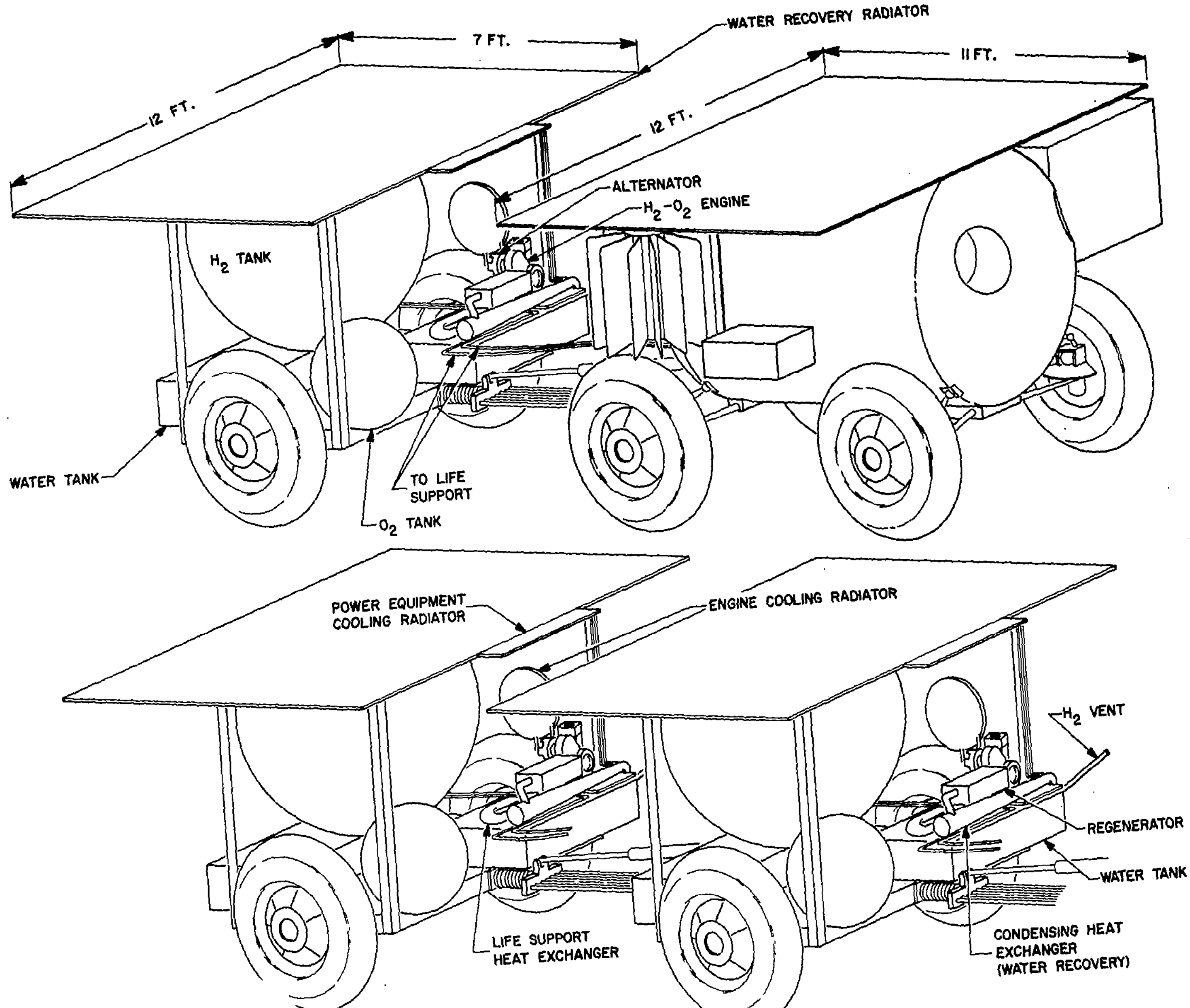
Of the possible chemical dynamic space power systems, the hydrogen-oxygen internal combustion engine has perhaps the most potential. The internal combustion engine has low weight and volume, a wide power range and a speed of rotation compatible with conventional alternators. The propellant choice has the advantages of high energy release per pound, high specific heat of hydrogen, and life support capabilities of oxygen with production of water as a by-product. Figure 6-1 depicts a possible power system equipment arrangement on the lunar roving vehicle trailer module.

6.2 NONINTEGRATED POWER SYSTEM

The nonintegrated system consists of a hydrogen-oxygen engine power plant that furnishes the entire power requirement in the form of electrical energy. Thermal energy obtained from cooling the engine is radiated to space.

6.2.1 DESCRIPTION

The H_2O_2 engine has been studied by Vickers Incorporated, Division of Sperry Rand Corporation (References 6-1 thru 6-6). The specific unit to be considered is described and test results are reported in Reference 6-1. It is a single-cylinder reciprocating, internal combustion engine operating on gaseous hydrogen as a fuel and gaseous oxygen as an oxidizer. Figure 6-2 shows a P-V diagram and the sequence of events in engine operation. The hydrogen and oxygen are not internally compressed, but are separately injected at high pressure (super-critical storage) near top-dead-center of the cycle. The propellants are regeneratively heated by the exhaust gases. The basic flow diagram is given in Figure 6-3. The particular engine designed and tested is rated at 2 kw average power, 3 kw maximum.


WATER RECOVERY RADIATOR
7 FT.
12 FT.
11 FT.
12 FT.
ALTERNATOR
H_2-O_2 ENGINE
H_2 TANK
WATER TANK
TO LIFE SUPPORT
O_2 TANK
POWER EQUIPMENT COOLING RADIATOR
ENGINE COOLING RADIATOR
H_2 VENT
REGENERATOR
WATER TANK
LIFE SUPPORT HEAT EXCHANGER
CONDENSING HEAT EXCHANGER (WATER RECOVERY)

Equipment

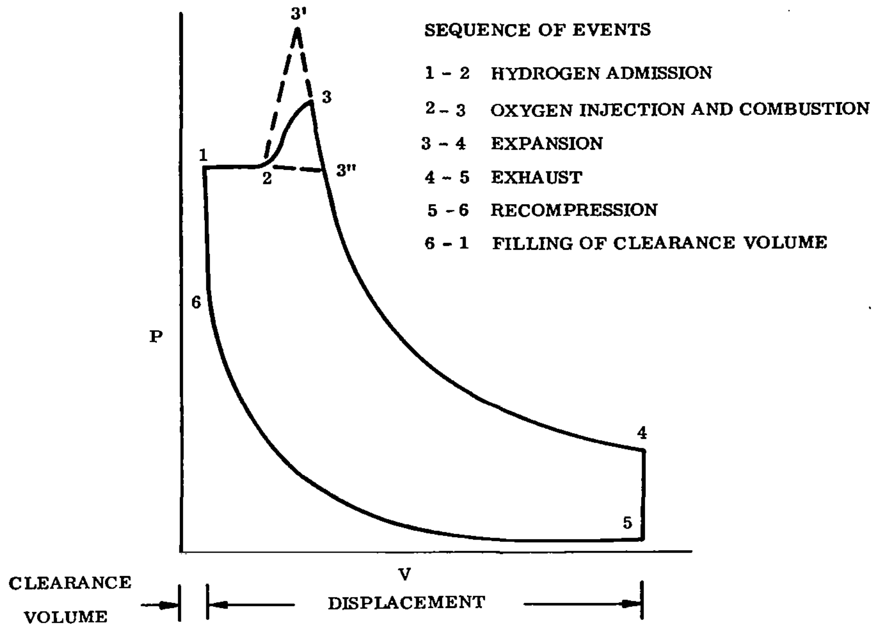


Figure 6-2. Typical Pressure-Volume Diagram for H_2-O_2 Engine

An analytical study has predicted a minimum specific propellant consumption (SPC) of 1.3 lb/kw-hr. The study was based on optimistic assumptions. The minimum value obtained in practice has been 2.2 lb/kw-hr and this is the value used in this study. However, to allow for future improvement in fuel consumption, the SPC will be based on net generator output (Reference 6-1). Typical generator efficiency for this application is 85%. Including a 5% control loss gives a shaft power of about 125% of the net generator output. Therefore, the SPC based on shaft power is about 1.8 lb/kw-hr.

The oxygen to hydrogen ratio (O/F) affects the specific propellant consumption, the engine cooling and propellant tankage. The value of the ratio used here is two since this gives the minimum propellant consumption.

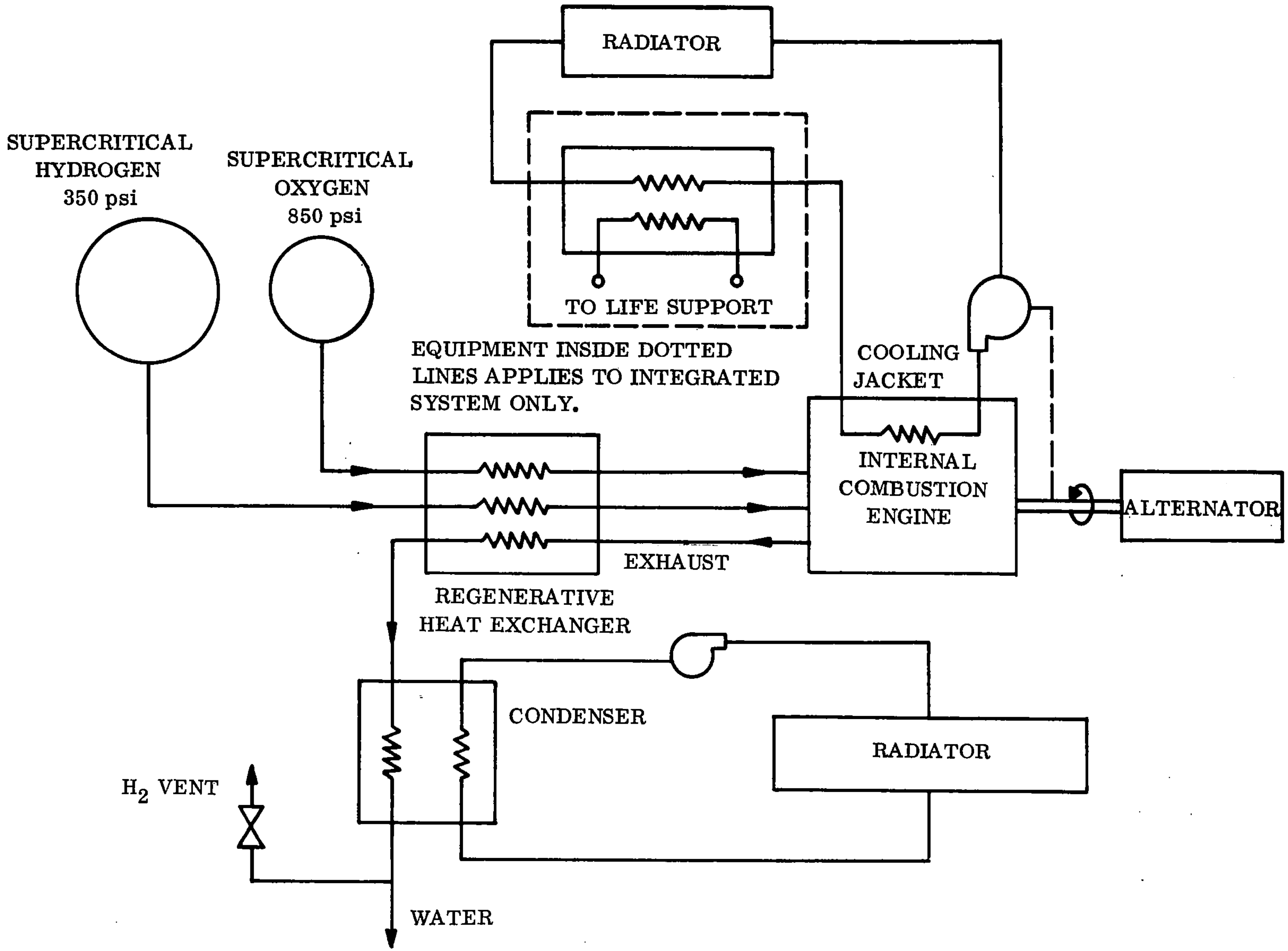


Figure 6-3. Hydrogen-Oxygen Internal Combustion Engine Flow Diagram

The engine cooling requirements are specified as equal to the brake engine power; 75% for cylinder wall cooling (500°F exit coolant temperature) and 25% for radiation cooling of the cylinder head (1500°F).

The following flight weights are given in Reference 6-1 for the 2 kw average, 3 kw maximum power engine system.

Engine	15 lb
Regenerator	10
Alternator	35
Battery	10
Controls and Plumbing	10
Total	80 lb

The radiator and propellant tank weight is not included.

6.2.2 ENERGY REQUIREMENTS

Refer to the typical power profile curve, Figure 3-5. The H_2-O_2 engine system uses the open loop (CO_2 unprocessed) profile since the oxygen needed for life support can best be obtained directly. This curve is the net power required for a configuration with one trailer. The power system considered here is too heavy for one trailer so additional trailers will be needed, resulting in increased locomotion power. Each trailer requires 800 watts of locomotion power for 5 hours out of every 24 during the 45-day operation (167 watts average). The power from the generator is ac. One third of this must be converted to dc at a conversion efficiency of 92%. To determine gross power, an average conversion factor may be calculated.

$$\frac{3}{0.92 + 2} = 1.03$$

Assuming that 'N' trailers are needed for the mission, the minimum, average and maximum gross power levels can be determined.

Minimum gross power	=	1.33 kw
Average gross power	=	(1.93 + .167N) (1.03) kw
Maximum gross power	=	(3.59 + .8N) (1.03) kw

The mission duration is 1080 hours so the gross energy is 1080 multiplied by average gross power.

Mission gross energy	=	2150 + 186N kw-hr

6.2.3 COMPONENT SIZING

6.2.3.1 Determination of Number of Trailers

First it is necessary to determine the number of trailers needed to transport the power system. Each trailer can weigh 3000 lb of which 2400 lb is payload. An estimate of the power system weight can be made by assuming one engine per trailer and an average tankage factor of 1.4. The system weight is approximately 100 N + (1.4 x SPC x Mission Energy).

$$N = \frac{100N + 1.4 \times 2.2 \text{ lb/kw-hr} \times 2150 + 186N}{2400 \text{ lb}}$$

$$N = \frac{100N + 6620 + 572N}{2400}$$

$$N = 3.8$$

Therefore, four trailers (N=4) will be required.

Using N = 4, the mission power levels are:

Minimum gross power	=	1.33 kw
Average gross power	=	2.68 kw
Maximum gross power	=	7.00 kw
Mission gross energy	=	2900 kw-hr

The maximum power is greater than can be supplied by two engines of the size of the test engine. In this case it is desirable to increase the engine size so that two engines are adequate for the peak power load. This would require a 3.5 kw maximum, 2.4 kw average power engine. It is assumed that the test engine may be linearly scaled as far as weight is concerned and that the operating parameters such as SPC and heat rejection remain the same. Two engines are available as backup units. See Table 6-1 for a summary of the energy requirements.

TABLE 6-1. NONINTEGRATED HYDROGEN-OXYGEN ENGINE SYSTEM ELECTRICAL ENERGY REQUIREMENTS

Mission gross energy		2900 kw-hr
Net energy (one trailer)	2260 kw-hr	
Energy for three extra trailers	540 kw-hr	
Conversion Loss (ac-dc)	100 kw-hr	
Minimum gross power		1.33 kw-hr
Average gross power		2.68
Maximum gross power		7.00

The net propellant is

$$2.2 \frac{\text{lb}}{\text{kw-hr}} \times 2900 \text{ kw-hr} = 6380 \text{ lb}$$

A 10% reserve increases this to 7020 lb. One third of this (2340 lb) is hydrogen and two thirds (4680 lb) is oxygen. This amount of propellant must be available after 30-day storage on the moon. Calculations were made to estimate insulation weight, boil-off, and tankage weight for the propellants assuming one hydrogen and one oxygen tank per tailer. A combination of foam and foil insulation is considered for thermal control. The foam is for launch conditions and the foil for the lunar storage environment. The method of calculation and the properties are from References 6-7, 6-8, and 6-9. The results are contained in Table 6-2.

6.2.3.2 Radiator Design

The following table summarizes the radiator parameters for the non-integrated design.

Q_R (kw)	T_{Ri} (°F)	T_{Ro} (°F)	Header (ft)	Tube Length (ft)	Area (ft^2)	Weight (lb)	No. of Tubes
2.5	500	400	2.75	2.8	8.3	5.9	17
2.7	500	400	3.0	2.8	8.9	6.6	15
3.0	500	400	3.25	2.9	10.0	7.4	17
3.3	500	400	3.35	3.2	10.8	8.4	18

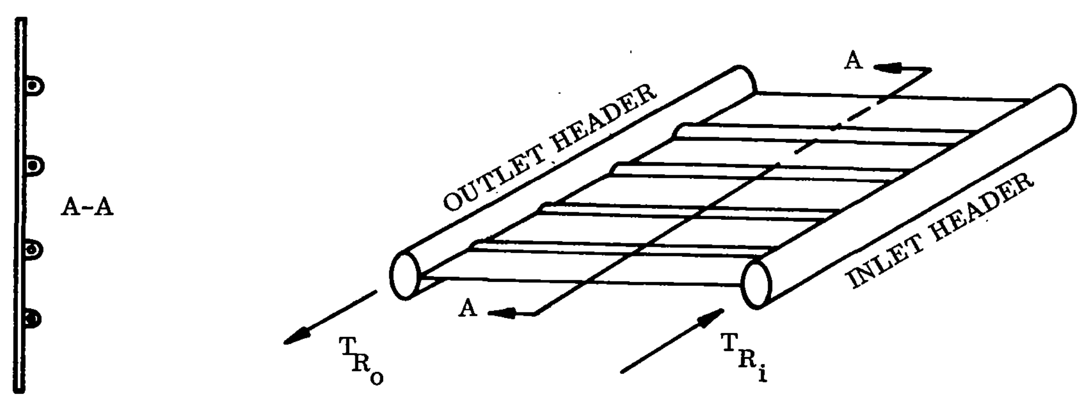


Figure 6-4. Radiator Configuration

TABLE 6-2. NONINTEGRATED HYDROGEN-OXYGEN ENGINE SYSTEM — SYSTEM WEIGHT

Engine system, 4 at 105 lb			420 lb
Radiators and heat exchangers			320
Total propellant and tankage			9330
Hydrogen and H_2 Tankage		3920	
Net hydrogen	2340 lb		
Boil-off	500		
Insulation	440		
Tankage	640		
Oxygen and O_2 tankage		5410	
Net oxygen	4680		
Boil-off	375		
Insulation	125		
Tankage	230		
Total power system weight			10070
Four trailers at 600 lb			2400
Life support oxygen			350
Gross power system weight			12820
H_2 tank diameter = 7.2 ft			
O_2 tank diameter = 3.5 ft			

6.2.3.3 Water Recovery System

The engine exhaust is a mixture of hydrogen and steam. A condensing heat exchanger in the exhaust stream makes possible the recovery of water by condensing the steam. Since the exhaust pressure is 3 to 4 psia, the condensing temperature is about $140^{\circ}F$. For each kilowatt of generator output, 940 watts must be rejected from the condensing heat exchanger. At maximum power this amounts to 3.3 kw for each of the two operating engines. A radiator of about 90 sq ft is required. At a weight of 0.78 lb/ft^2, the radiator will weigh about 70 lb.

6.3 INTEGRATED POWER SYSTEM

The primary purpose of integrating the power and life support systems is to reduce the electrical power requirements of the life support system by utilizing waste thermal power from the engine. This is accomplished by providing thermal energy for some of the life support system processes by means of a heat exchanger in the engine cooling loop.

6.3.1 DESCRIPTION

The integrated system is esentially the same as the non-integrated system except for the additional heat exchanger in the engine coolant loop (see Figure 6-3). The amount of thermal energy available in the coolant loop is equal to 75% of the engine brake power. The coolant exits the engine at $500^{\circ}F$.

6.3.2 ENERGY REQUIREMENTS

The amount of life support power that can be furnished thermally was determined in Section 4.6. The $400^{\circ}F$ values should be used when considering the H_2-O_2 engine. The life support electrical power requirement is reduced from 965 watts to 251 watts by thermal integration. The resulting power profile is shown in Figure 3-5; thermal power required is 1131 watts.

Using the power profile and assuming 'N' trailers:

Minimum gross power	0.60 kw
Average gross power	(1.22 + .167 N) (1.03)
Maximum gross power	(2.88 + .8 N) (1.03)
Mission gross energy	1355 + 186 N kw-hr

6.3.3 COMPONENT SIZING

6.3.3.1 Determination of Number of Trailers

The number of trailers required is determined in the same manner as was used for non-integrated sizing.

$$N = \frac{100N + 1.4 \times 2.2 \text{ lb/kw-hr} \times 1355 + 186\ N \text{ kw-hr}}{2400}$$

$$N = \frac{672\ N + 4180}{2400}$$

$$N = 2.5$$

Therefore, three trailers are needed and the resulting power levels are:

Minimum gross power	0.60 kw
Average gross power	1.77 kw
Maximum gross power	5.44 kw
Mission gross energy	1910 kw-hr

One engine (2 kw average power) can supply the power except during the peaks. Peak power can be supplied by two engines. One engine is used as a backup unit. See Table 6-3 for a summary of the integrated system energy requirements.

TABLE 6-3. INTEGRATED HYDROGEN-OXYGEN ENGINE SYSTEM ELECTRICAL ENERGY REQUIREMENTS

Mission gross energy		1910 kw-hr
Net energy (one trailer)	1490 kw-hr	
Energy for two extra trailers	360	
Conversion Loss (ac-dc)	60	
Minimum gross power		0.60 kw
Average gross power		1.77
Maximum gross power		5.44

6.3.3.2 Thermal Energy Availability

It is seen that the minimum electrical power (600 watts), occurring during the sleep periods, is less than the thermal power (1173 watts) required by the life support system. The maximum thermal power from cooling the engine is equal to the engine brake power. This is estimated by assuming a generator efficiency of 85% and a control loss of 5%. The brake power is then 600/0.8 = 750 watts. Seventy-five percent of this, 560 watts, is available from the cylinder wall cooling loop. If this is not sufficient, some of the power from the cylinder head can be used. This power is normally radiated from the 1500°F head.

During non-locomotion periods, the engine does not produce enough waste heat to supply the life support thermal power requirements. This problem can be eliminated by modifying the life support processing to a pulsed type of operation. Much of the desorbing and evaporative processing can be accomplished during the locomotion periods when adequate waste energy is available. The CO_2 process can be operated at 400°F, 100 watts, during locomotion and at 78°F, 18 watts, at other times. This saves 316 watts during non-locomotion. Storing the waste H_2O and processing it during locomotion can save 277 watts during non-locomotion periods. The result is that the thermal processing power can be reduced to 550 watts during non-locomotion periods (19 hr/day). During locomotion (5 hr/day) thermal power required is increased to 2500 watts.

6.3.3.3 Fuel and Oxidizer Tank Sizing

The net propellant required is:

2.2 lb/kw-hr x 1910 kw-hr = 4200 lb

A 10% reserve increases this to 4620 lb of which one third (1540 lb) is hydrogen and two thirds (3080 lb) is oxygen. Weights of the insulation, boil-off and tankage were calculated assuming one H_2 tank and one O_2 tank per trailer. Table 6-4 gives the results.

6.3.3.4 H_2-O_2 Engine Integrated Design Component Sizing

Figure 6-5 is a schematic presentation of the waste heat dissipation loop for the H_2-O_2 engine.

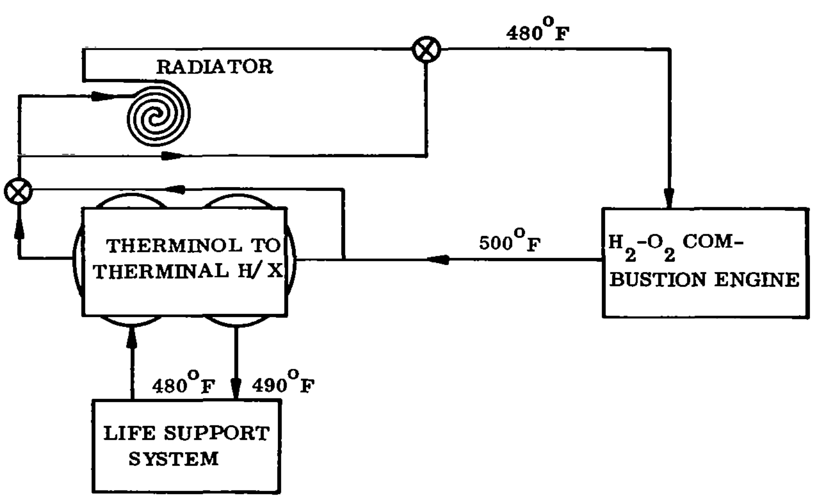


Figure 6-5. Schematic of Waste Heat Dissipation Loop for Engine H_2-O_2

The life support system heat exchanger has been sized to accommodate the peak load. During locomotion periods (5 hours/day) this load is on the order of 2500 watts, the remainder of the time the load is reduced to 575 watts. The peak waste heat load during locomotion

TABLE 6-4. INTEGRATED HYDROGEN-OXYGEN ENGINE SYSTEM

	System Weight (lb)		
Engine system, 3 at 80 lb			240
Radiators			230
Total propellant and tankage			6340
Hydrogen and H_2 tankage		2720	
Net hydrogen	1540		
Boil-off	370		
Insulation	330		
Tankage	480		
Oxygen and O_2 tankage		3620	
Net oxygen	3080		
Boil-off	280		
Insulation	90		
Tankage	170		
Total power system weight			6810 lb
Three trailers at 600 lb			1800
Life support oxygen			350
Gross power system			8960 lb
H_2 tank diameter = 7.12 ft			
O_2 tank diameter = 3.5 ft			

periods from the combustion engine is on the order of 2564 watts which matches the required life support power requirements. The peak load during non-locomotion periods is 850 watts so that the radiator size is based on carrying a peak load of 850-575 or 275 watts. A radiator for this purpose if estimated to weigh 0.62 lb with an area of 0.88 sq ft. A schematic of one possible arrangement is shown in Figure 6-6.

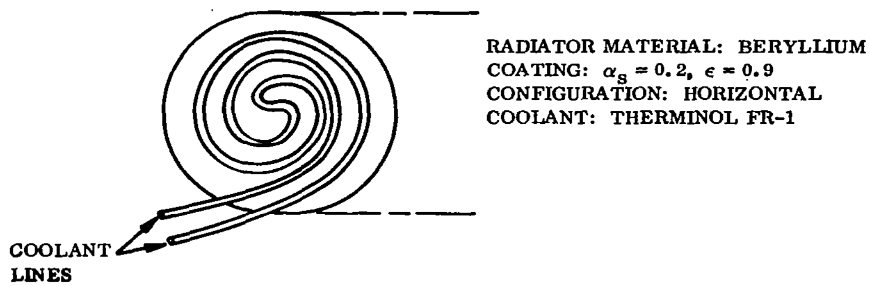


Figure 6-6. Integrated H_2-O_2 Combustion Engine Radiator Configuration

The heat exchanger (Figure 6-7) is a counterflow tube/shell heat exchanger. This heat exchanger has been sized for the peak load of 2500 watts. The circuit control is achieved by a series of bypass lines so as to: (1) allow a constant flow rate through the combustion engine; (2) maintain a fixed radiator outlet temperature; and (3) maintain constant temperature conditions at the inlet and outlet to heat exchanger. The basic design parameters for the weight optimized heat exchanger are given below.

H_2- O_2 Engine Heat Exchanger (H/X) Design Parameters

Heat exchanger weight	= 13.3 lb
Pump work	= 30.5 watts
Overall H/X diameter	= 2.26 in.
Overall H/X length	= 5 feet
Tube inside diameter	= 0.18 in.
Tube spacing	= 0.22
Tube wall thickness	= 0.15 in.

Shell wall thickness	=	0.1 in.
Feedline inside diameter	=	0.716 in.
Number of tubes	=	20

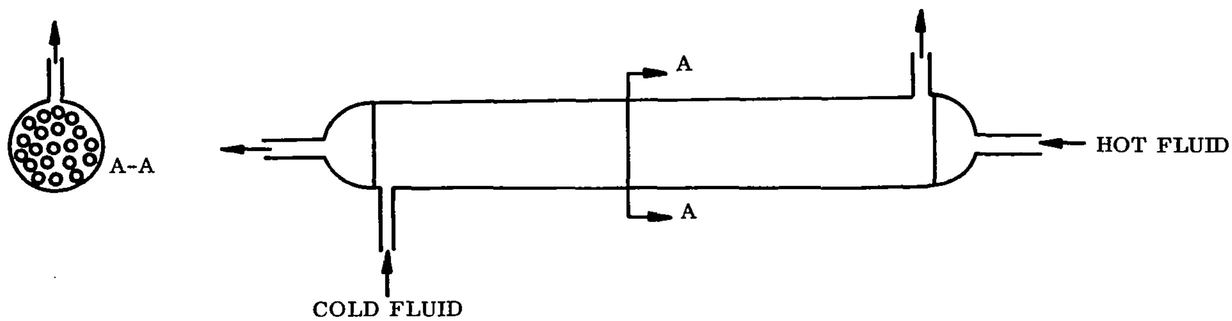


Figure 6-7. Heat Exchanger Configuration

6.3.3.5 Water Recovery System

The water is recovered by the same method as used in the non-integrated case. The maximum power per engine is $\frac{5.44}{2} = 2.72$ kw (see Table 6-3). At 940 watts rejection per kw the condenser load is 2.56 kw. This results in a radiator area of about 70 sq ft and a weight of about 55 lb.

6.4 DISCUSSION

Estimates have been made of the weights of the non-integrated and integrated hydrogen-oxygen engine systems for the 45-day lunar mission. The method of thermal integration has been indicated. Some additional considerations are:

a. The nonintegrated system requires four trailers and the integrated system three trailers. Problems of locomotion and interconnection of this number of trailers have not been considered in detail.

b. The development program for the H_2-O_2 engine had a goal of 350 hours of uninterrupted engine life. However, the components in the engine have not been proven in any prolonged engine run greater than 100 hours (Reference 6-1). A formidable problem is the design of the oxygen injector which must open and close in about 1.5 milliseconds in a highly oxidizing environment.

c. Oil consumption must be kept to a minimum. The exhaust gas (hydrogen and steam) will be contaminated by the oil and any water obtained from the exhaust must be filtered because of this contamination.

d. The engines are started with battery power and the pressure energy of the propellants. There may be a need to preheat the propellants with electrical heaters during start-up. The batteries must be temperature controlled and the engines may require protection from the hard vacuum to prevent cold welding of the engine parts.

e. The control system will consist of pressure regulators, valve timing mechanisms, temperature controlled valves, and other components. An optimum control system has not been defined.

6.5 REFERENCES

6-1. "Development of Hydrogen-Oxygen Fueled 3-Kilowatt Internal - Combustion Engine", Harry M. Cameron and Norman E. Morgan, AIAA Paper No. 64-756 presented at the Third Biennial Aerospace Power Systems Conference, Sept. 1964.

6-2. "Piston Engines are Reborn for Space Power", Walter K. Deacon and Norman E. Morgan, ASME Paper No. 62-AV-39 presented at the ASME Aviation Conference, June 1962.

6-3. "Development of Auxiliary Electric Power Supply System", NASA Contract NAS 3-2550, Progress reports, Jan. - Mar. 1963, R. C. Thomas and W. D. Morath, Vickers Incorporated.

6-4. "Power Systems for Lunar Roving Vehicles and Equipment", D. W. Brock, Vickers Incorporated, Interim Report No. 81601-426-1, Jan. 1963.

6-5. "Development of a Hydrogen-Oxygen Internal Combustion Engine Auxiliary Electric Power Supply System", N. E. Morgan, Vickers Incorporated, NASA Contract NAS 3-2550 Phase Report, Sept. 1963.

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6-7. " Lightweight Thermal Protection Systems for Space Vehicle Propellant Tanks", Richard H. Knoll and Jon C. Oglebay - Society of Automotive Engineers Paper 746C, Sept. 1963.

6-8. "Cryogenic Storage for Space Electrical Power", William A. Chandler, Astronautics and Aerospace Engineering, p. 97, May 1963.

6-9. "Cryogenic Propellant Feed Systems for Electrothermal Engines", Arthur D. Little, Inc., May 1964 NASA CR-60.

SECTION 7
HYPERGOLIC ENGINE POWER SYSTEM

7.1 INTRODUCTION

The light weight of small internal combustion engines and their ability to closely approach the optimum thermodynamic cycle suggests their application to space missions. The approach in this section concerns the use of hypergolic (immediate ignition on contact) propellants in this type of engine. Hypergolic propellants are highly energetic and require no ignition system, but are dependant on an exact injection system. The engines drive ac generators which furnish the mission energy requirements. Figure 7-1 depicts a possible power system equipment arrangement on the lunar roving vehicle trailer module.

7.1.1 DESCRIPTION OF SYSTEM

The hypergolic engine to be considered is a reciprocating, internal combustion engine operating on N_2O_4 oxidizer and 0.5 UDMH/0.5 N_2H_4 fuel. The engine, designated SPU-2, has been designed by the Marquardt Corporation (References 7-1 and 7-2). Hypergolic combustion occurs at approximately top-dead-center of the piston travel, the cycle being similar to that of the two-stroke Otto cycle (see Figure 7-2). The propellants are injected as liquids. The basic flow system is shown in Figure 7-3.

7.1.2 CRITICAL ENGINE PARAMETERS

An analytical study has predicted a minimum specific propellant comsumption (SPC) of 3.3 lb/kw-hr. The results of the engine tests indicate a SPC of 4.0 lb/hp-hr (5.36 lb/kw-hr). For the purposes of this study, the latter figure is used. This SPC occurs at an oxygen to fuel ratio (O/F) of 1.5. This is the value used here.

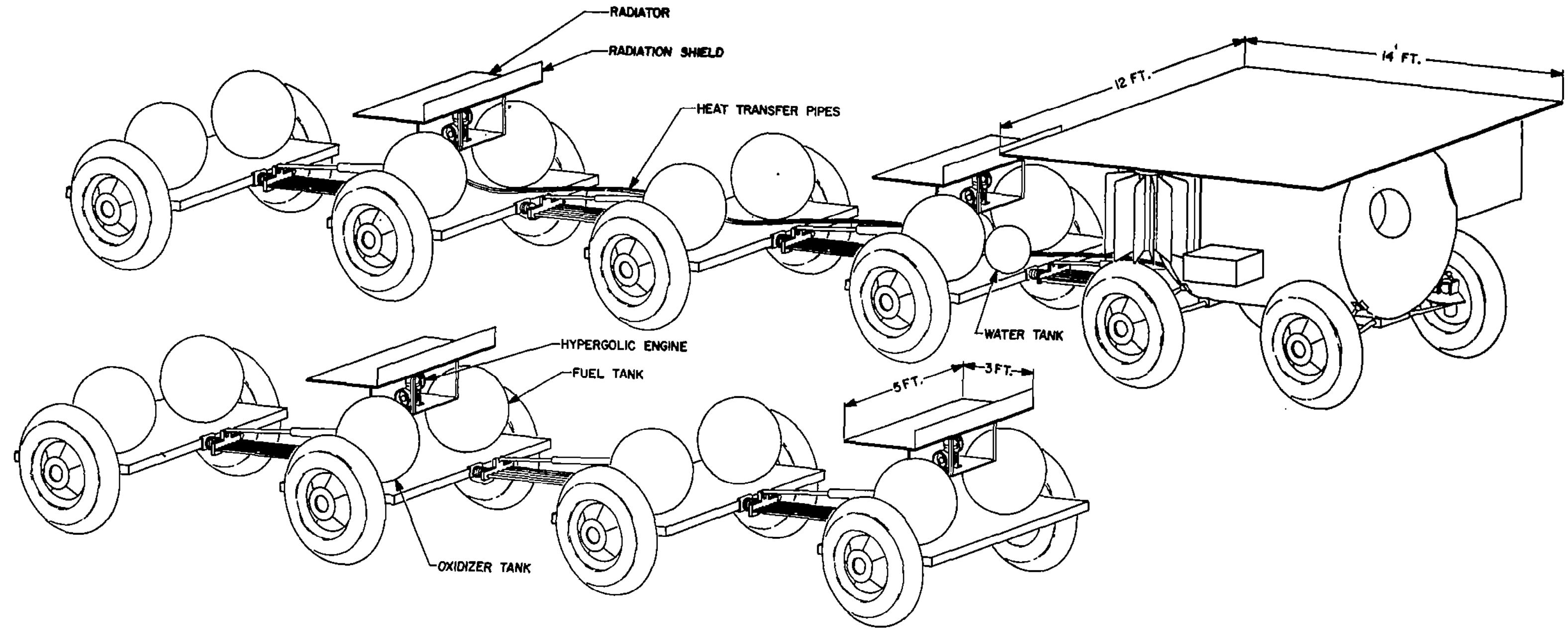

Figure 7-1. Hypergolic Power System Equipment

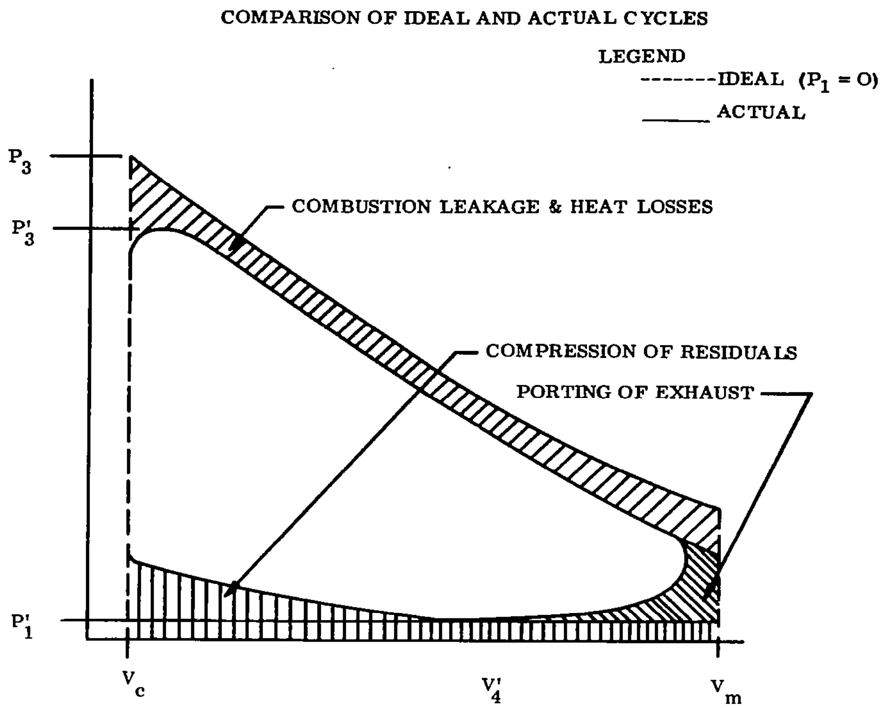

Figure 7-2. Pressure-Volume Diagram for Hypergolic Engine

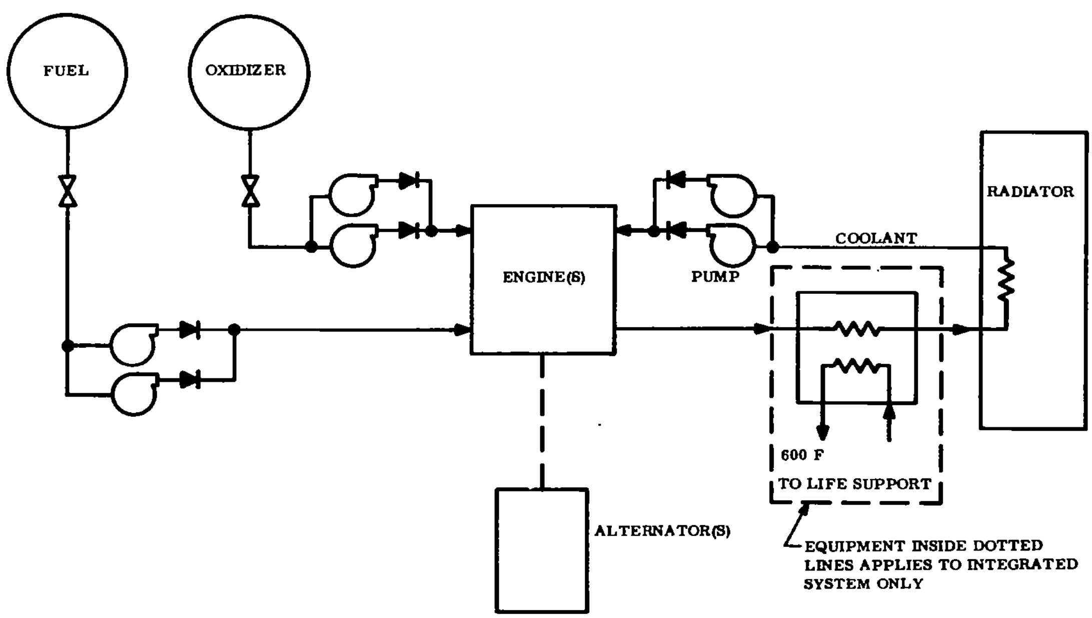

Figure 7-3. Hypergolic Engine Power System - Flow Diagram

7.2 SYSTEM SIZING

7.2.1 MISSION ENERGY REQUIREMENTS

The typical power profile is presented in Figure 3-4. The hypergolic engine system requires that the closed loop (processed CO_2) curve be used. However, this profile must be modified since it is based on a one trailer system. The hypergolic system will require 'N' trailers. Each additional trailer increases the power required during locomotion by 800 watts. Locomotion occurs five hours per day. Thus each additional trailer increases the average power by $5/24 \times 0.8 = 0.167$ kw. The average power for the one trailer system, from Figure 3-4, is 2.95 kw. One third of this is to be converted to dc with a conversion efficiency of 0.92. This gives a power conversion penalty of about 90 watts. It is assumed that there is no conditioning penalty associated with ac power (since it is included in the SPC calculation). The average gross power is then

$$2.77 + 0.167\,N \quad \text{kw}$$

The maximum gross power is

$$4.44 + 0.8\,N \quad \text{kw}$$

The minimum gross power is

$$2.14 \quad \text{kw}$$

7.2.2 ENGINE SYSTEM WEIGHT (EXCLUDING PROPELLANT)

The weight of the engine system, including radiator and electrical generator, for a 3 kw load is given as 150 lb. As a first approximation, it will be assumed that there is one engine for every two trailers, as long as this number can supply the peak power. Then the total engine weight is $150 \times N/2 = 75N$ lb.

7.2.3 PROPELLANT WEIGHT

The propellant required = SPC x P_{ave} x Time

$$= 5.36 \text{ lb/kw-hr} \times (2.77 + 1.67N) \text{ kw} \times 1080 \text{ hr}$$

$$= 16,050 + 965\ N \quad \text{lb}$$

$$\text{Fuel required} = \frac{16,050 + 965\ N}{1 + O/F} = 6,430 + 386N$$

$$\text{Oxidizer required} = (6430 + 386N) \times O/F = 9,640 + 578N$$

7.2.4. NUMBER OF TRAILERS REQUIRED

Each trailer can weigh 3000 lb of which 2400 lb is payload. The number of trailers required is then the power system weight divided by 2400.

$$N = \frac{75N + (16,050 + 965N)}{2400}$$

$$N = 11.8 \text{ trailers}$$

See Table 7-1 for the resultant weights.

7.2.5 INTEGRATED SYSTEM WEIGHT

The gross effect of thermally integrating this power system will be a reduction in the amount of propellant required. This reduction is possible because much of the life support power requirements can be obtained from the waste heat of the engine. When operating at an O/F of 1.5, the heat loss from the engine is approximately equal to the engine brake output (Reference 6-1). When the system is integrated, about 1.19 kw of waste thermal power from the engines can be substituted for about 0.99 kw of electrical power. This reduces the net electrical power and the previous calculations of Section 7.2 can be repeated taking this

TABLE 7-1. NONINTEGRATED HYPERGOLIC ENGINE SYSTEM

Configuration and Power		
Number of Trailers		12
Number of Engines		6
Mission Gross Energy		5150 kw-hr
Net Energy (One Trailer)	3080 kw-hr	
Additional Trailer Energy	1980	
Dc Conversion Loss	90	
Average Gross Power		4.77 kw'
Maximum Gross Power		14.04 kw
Minimum Gross Power		2.14 kw
Weights		
Engine System Weight		900 lb
Propellant Weight		27,600 lb
Fuel	11,000 lb	
Oxidizer	16,600 lb	
Total		28,500 lb

reduction of power into account. The results are given in Table 7-2.

7.3 DISCUSSION

There are a number of problems associated with the hypergolic engine system which were not considered in detail. Among these are the large number of trailers involved and the problem of transferring propellant and thermal integration fluid from trailer to trailer to cabin. This problem could best be eliminated by changing the mission constraints to allow refueling from a central location, and using the RTG for thermal energy to operate the life support processes.

There is also a problem of propellant storage. The propellants should be stored between $20^{\circ}F$ and $140^{\circ}F$. Passive thermal control (super-insulation) may be possible during storage. Active thermal control may be necessary as the tanks empty. In general, the larger the propellant tanks, the easier will be thermal control.

Additional weight penalties not included in Tables 7-1 and 7-2 include the 750 lb of water needed for space suit cooling. Any water produced by the hypergolic reaction is not collected as it is in the fuel cell system. Each trailer also has 600 lb of structural weight that must be considered as a mission weight penalty.

7.4 REFERENCES

7-1. Kessler, J. R., "Hypergolic Fueled Reciprocating Space Power Unit", AIAA Paper No. 64-755, The Marquardt Corp.

7-2. "Feasibility Study for Development of a Hypergolic Engine Space Power System", Phase 1 Final Report, NAS 9-857, Sept. 1964.

7-3. Sutton, George P., Rocket Propulsion Elements, 3rd Ed.

TABLE 7-2. INTEGRATED HYPERGOLIC ENGINE SYSTEM

Configuration and Power		
Number of Trailers		8
Number of Engines		4
Mission Gross Energy		3370 kw-hr
Net Energy (One Trailer)	2030 kw-hr	
Additional Trailer Energy	1260	
Dc Conversion Loss	80	
Average Gross Power		3.13 kw
Minimum Gross Power		9.86
Minimum Gross Power		1.16
Weights		
Engine System Weight		600 lb
Propellant Weight		18,100
Fuel	7240 lb	
Oxidizer	10,860	
Total		18,700 lb*
Diameter of tuel tank ~ 3.5 ft		
Diameter of oxidizer tank ~ 3.5 ft		

*Does not include weight of trailers

SECTION 8

ISOTOPE BRAYTON POWER SYSTEM

8.1 INTRODUCTION

Using weight as the criterion, the length and power requirements of this mission suggest the use of isotope power systems. Chemical systems require large amounts of reactants. Nuclear reactors producing the desired power level are probably too large or operate at too low a temperature, as discussed in Reference 8-3. Solar power systems are ruled out because of the long lunar nights. It is probable that dynamic systems would have a lower specific weight than static thermal energy converters. This section considers a dynamic system using the Brayton thermodynamic cycle with argon as the working fluid. Using an inert gas eliminates condensation and corrosion problems. Figure 8-1 depicts a possible power system equipment arrangement on the lunar roving vehicle trailer module.

8.2 NONINTEGRATED SYSTEM

8.2.1 DESCRIPTION

A basic temperature-entropy diagram for the Brayton cycle is shown in Figure 8-2. Heat is generated by the isotope source and rejected by a radiator with power being obtained from a turbine located in the working-fluid (argon) loop. The recuperator makes a higher efficiency possible. Systems of this type have been considered in References 8-1 and 8-2. General information is contained in Reference 8-3.

The thermodynamic cycle consists of the following operations. Referring to Figure 8-2, process 1-2 is the nearly isentropic compression of the argon. Processes 2-3 and 3-4 are the nearly constant pressure heating in the recuperator and heat source respectively. The argon is expanded through the compressor turbine, 4-5 and the power turbine, 5-6. Heat is given up to the recuperator during process 6-7 and to the radiator, 7-1.

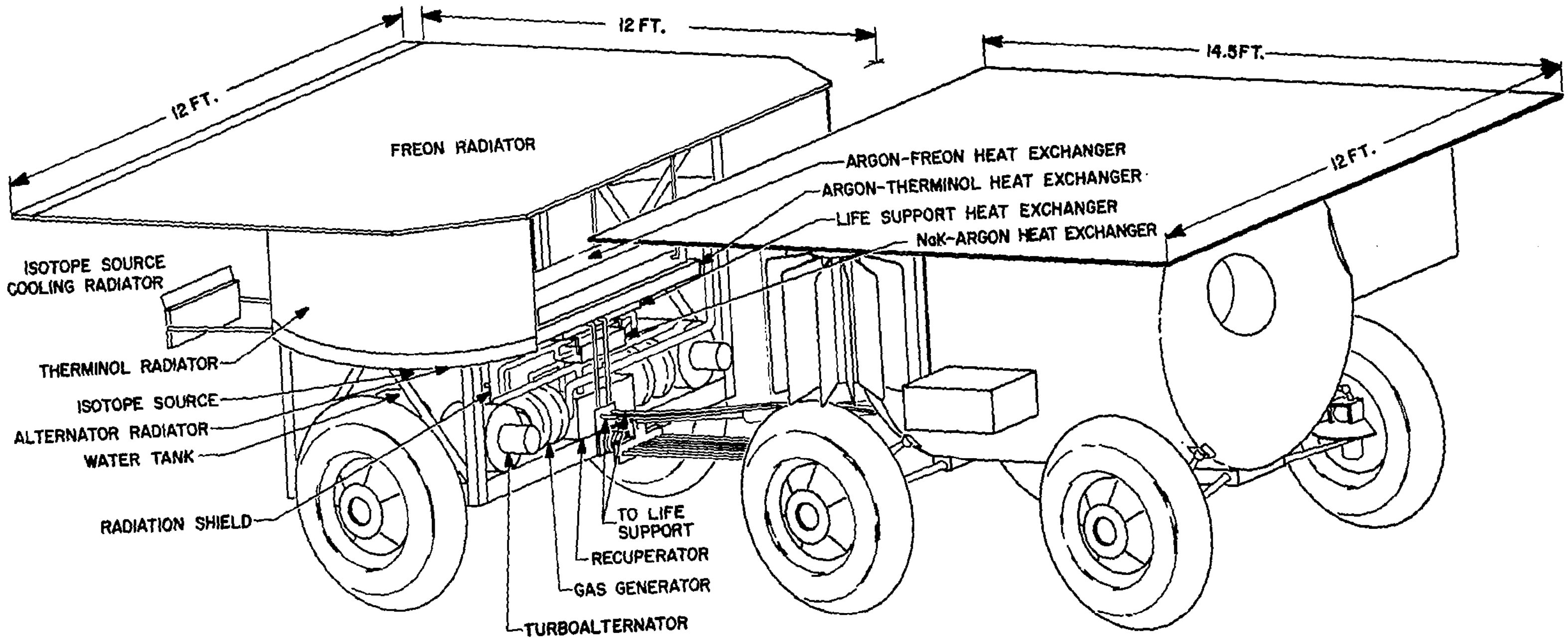

Figure 8-1. Brayton Power System Equipment

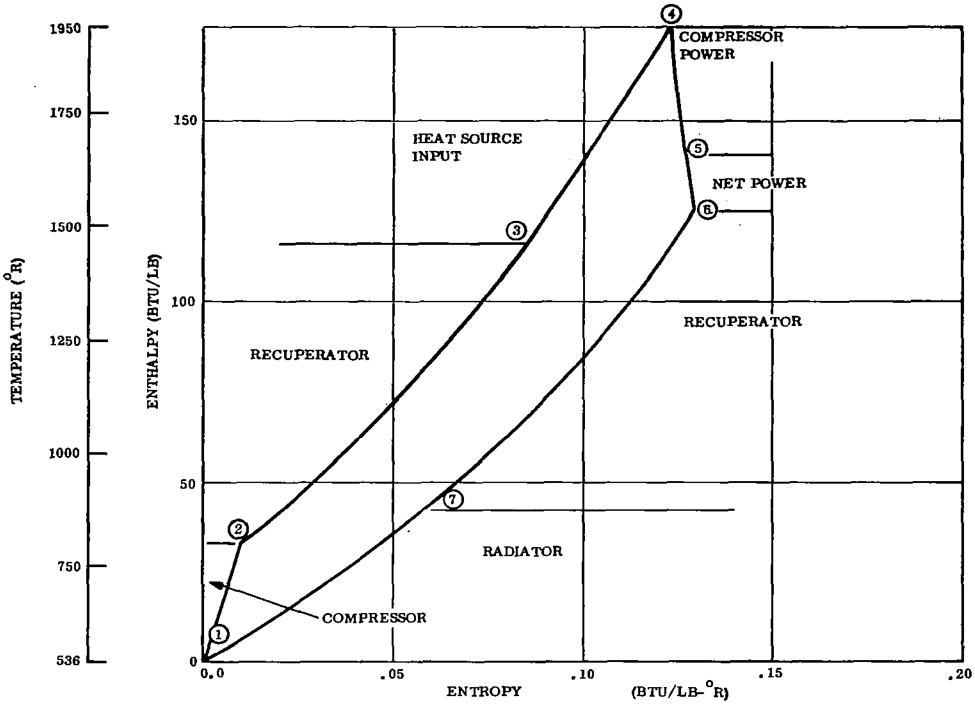

Figure 8-2. Brayton Cycle

Figure 8-3 is a functional diagram of the Brayton power system. The compressor and alternator work is supplied by separate turbines to take advantage of desirable operating speeds. The compressor rotates at 67,500 rpm, while the alternator operates at 12,000 rpm to generate 400 Hz power. The heat is removed from the isotope source by a NaK eutectic solution and is transferred to the argon working fluid through a heat exchanger. Use of NaK for heat removal minimizes isotope source size and weight. Following recuperation, (heat transfer from the low pressure side to the high pressure side of the working fluid) the remaining waste heat is removed by argon-therminol and argon-freon heat exchangers. Liquid tube radiators associated with each heat exchanger reject the waste heat to space. Liquid radiator systems are lighter and have smaller tubes than gas radiator systems.

Another cooling loop is provided for rejecting waste heat from the bearings, alternator and power conversion equipment. Speed control of the gas generator is accomplished by a gas bypass control and the speed of the power unit is controlled by a parasitic load on the alternator.

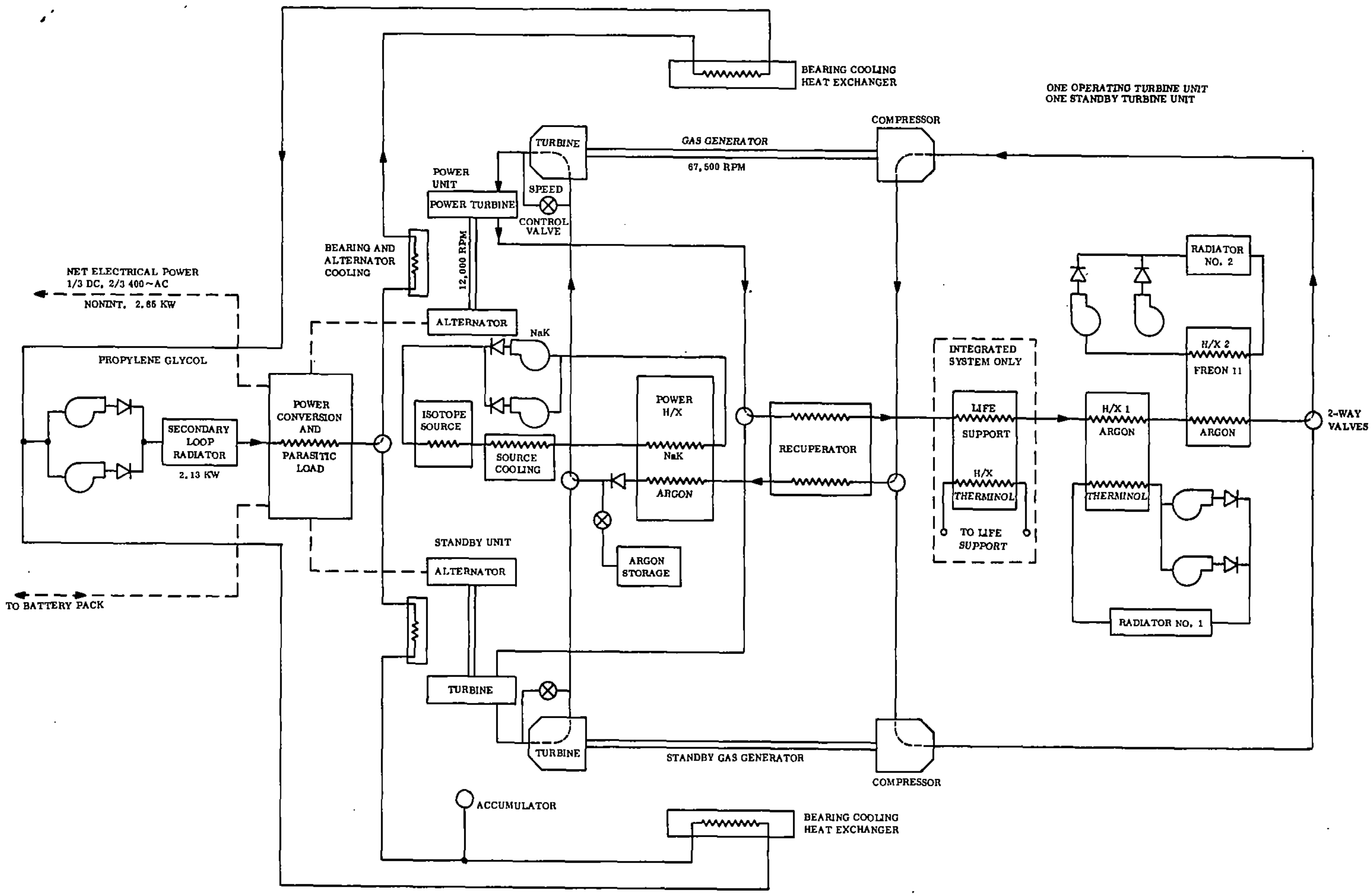

Figure 8-3. Brayton Cycle Functional Diagram

Reliability considerations require that two gas generators and alternators be provided for redundancy. It is possible to operate both units continuously at half load or to operate one at full load with the standby unit shut down. This latter mode of operation requires a smaller isotope source since the power units are more efficient at the higher power level. Primarily for this reason, the "one operating plus one stand-by" configuration was chosen. Argument can also be made that certain types of failure such as contamination of the argon could damage both turbines if both were on line. Batteries are available for power during switch over. Power for starting is obtained from argon stored at high pressure.

8.2.2 POWER REQUIREMENTS

The mission net power profile is given in the design guidelines, Figure 3-4. The closed loop CO_2 system is used with the isotope power systems. The average power level is 2.95 kw. However, 100 watts of this can be supplied by the RTG, leaving 2.85 kw net power to be supplied by the isotope dynamic system. The design guidelines are that one-third of the power is dc and two-thirds is 400 Hz ac. It is assumed that this division of power is constant during the mission.

The power profile of the mission presents problems to the isotope dynamic system. The isotope source produces an essentially constant amount of power. If a variable power load is to be drawn from the source, some thermal energy must be lost or provision must be made for thermal storage. If no storage is used, the source must be sized for the peak load. This would pose an excessive penalty. Thermal storage is to be avoided since the thermal efficiencies of the system require that four to six times as much energy would have to be stored as energy required for the load peaks. This type of operation would also require the turbine to operate at a varying power level. Operation at other than design point decreases efficiency and may result in other problems such as speed control. All these problems can be eliminated by averaging the profile before the turbine sees it. Batteries can accomplish this. The approach taken is to provide enough battery capacity so that the turbine-alternator works in to an essentially constant load. Therefore the isotope source size is reduced to a minimum and the turbine can be designed for maximum efficiency at a given operating point.

Batteries must be available for emergency conditions so all the battery weight penalty can not be charged to the constant alternator power type of operation. Power conditioning components were identified so that conditioning losses and power level could be calculated. The component efficiencies were that obtained from the design guidelines. The configuration and efficiencies are shown in Figure 8-4.

Additional power requirements include 5 percent of the alternator output for speed control. Also about 10 watts are required for NaK loop pump power, about 5 watts for the main radiator liquid loop and about 5 watts for the bearing cooling loop. The total shaft power is obtained by including bearing losses and alternator inefficiency. These were estimated from Reference 8-1. Table 8-1 is a summary of the power requirements.

TABLE 8-1. ISOTOPE BRAYTON POWER REQUIREMENTS (NONINTEGRATED)

Net Electrical Power		2.85 kw
Average Power Conditioning Loss		0.46
Battery	0.180	
Ac-dc	0.114	
Dc-ac	0.104	
BCR	0.050	
Diode	0.008	
Dc-ac	0.004	
Pump Power (Therminol and Freon)		0.01
NaK Pump Power		0.01
Speed Control		0.18
Total Electrical Power		3.51 kw
Alternator Loss (= 0.865)		0.57
Bearing Loss (power unit)		0.30
Total Shaft Power		4.38 kw

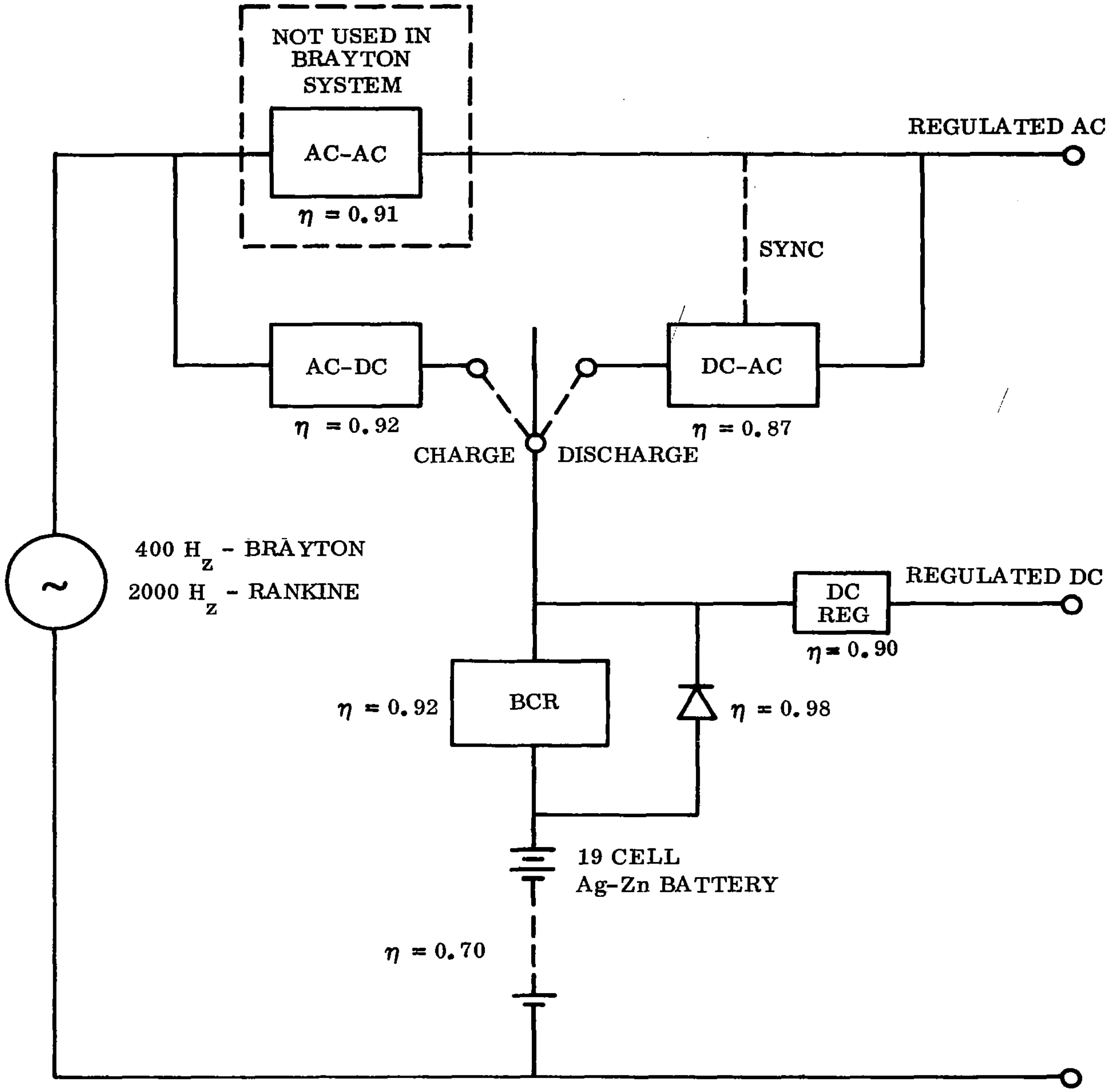

Figure 8-4. Power Conversion Configuration for Calculating Conditioning Losses in Isotope Systems

8.2.3 HEAT BALANCE

A cycle analysis can be made to determine the argon flow rate, the isotope source size and the heat rejection rate. The analysis is simplified since argon can be considered a perfect gas in this application. Table 8-2 is a summary of the cycle analysis. The given quantities are representative of systems under development and were used in References 8-1 and 8-2. The results give the power turbine work as 15.8 kw per lb of argon per sec. Since the shaft work is 4.38 kw, the required flow rate is 0.277 lb/sec. From the cycle analysis and flow rate, the heat added is 17.6 kw and the heat rejected from the power loop is 12.4 kw.

Table 8-3 is a power balance of the system. The net electrical power is dissipated elsewhere. Secondary cooling loops provide thermal control for the power conditioning equipment, the bearings, speed control and alternator. The batteries are to be mounted on the cabin unit. The source input must equal the power output.

8.2.4 ISOTOPE SOURCE

The isotope source for the Brayton system consists of radioactive material in a suitable container, a NaK heat transfer loop for transfering the heat from the source to a NaK-argon heat exchanger, and a shield to protect the crew from the nuclear radiation. Safety considerations are of primary importance. An analysis of the requirements and a conceptual design of the source can be found in Appendix C. From Table 8-3, the required source thermal power is 17.8 kw. Estimates of source subsystem characteristics from data in the appendix are as follows:

Source Size	Length "D"-17.9 inches, Diameter "C" 18.4 inches
Shield Weight	290 lb
Re-entry Protection Weight	227 lb
Total Subsystem Weight	915 lb

TABLE 8-2. BRAYTON CYCLE ANALYSIS (NONINTEGRATED)

Given Quantities	
Compressor Inlet Temperature, T_1	$536^{\circ}R$
Turbine Inlet Temperature, T_4	$1950^{\circ}R$
Compressor Inlet Pressure, P_1	6.0 psia
Compressor Pressure Ratio P_2/P_1	2.3
Compressor Efficiency	0.80
Compressor Turbine Efficiency	0.83
Power Turbine Efficiency	0.84
Recuperator Effectiveness	0.90
Gas Generator Bearing Losses	0.80 kw
Recuperator Pressure Drop, Cold Side, (P_2-P_3)	0.14 psi
Heat Source Pressure Drop, (P_3-P_4)	0.69 psi
Recuperator Pressure Drop, Hot Side, (P_6-P_7)	0.05 psi
Radiator Pressure Drop (P_7-P_1)	0.32 psi

Cycle Points

Station	T ($^{\circ}$R)	P (psia)
1	536	6.00
2	801	13.80
3	1467	13.66
4	1950	12.97
5	1663	7.97
6	1543	6.37
7	877	6.32

Power Turbine Work per lb $= C_p\,(T_5-T_6) = 15.8\,\frac{\text{kw-sec}}{\text{lb}}$

Heat Added per lb $= C_p\,(T_4-T_3) = 63.5\,\frac{\text{kw-sec}}{\text{lb}}$

Heat Rejected per lb $= C_p\,(T_7-T_1) = 44.8\,\frac{\text{kw-sec}}{\text{lb}}$

$\dot{W} = 0.277$ lb/sec

TABLE 8-3. BRAYTON SYSTEM POWER BALANCE (NONINTEGRATED)

Electrical Power Out		2.85 kw
Radiation from Secondary Radiator		2.13
Power Conditioning Loss	0.28 kw	
Speed Control	0.18	
Alternator Loss	0.57	
Bearing Cooling	1.10	
Radiation from Primary Radiator		12.40
Battery Heat Loss		0.18
Ducting Thermal Loss		0.24
Total Power Out		17.8 kw
Source Power Required		17.8 kw

8.2.5 HEAT REJECTION SYSTEM

The requirements of the primary heat rejection system of the nonintegrated Brayton system include the following:

a. It must dissipate 12.4 kw of waste thermal power. (See Table 8-3).

b. It must provide an argon compressor inlet temperature of 536°R.

c. It must be contained on the power system trailer. (See Figure 8-5 for the radiator envelope.)

d. It must operate in the lunar environment. (See Section 3.1.4 for a description of the lunar thermal environment.

Two types of heat rejection systems are considered that fulfill these requirements. Section 8.2.5.1 describes a system utilizing argon-to-liquid heat exchangers with fin and tube liquid radiators. Section 8.2.5.2 considers the use of a vapor fin radiator.

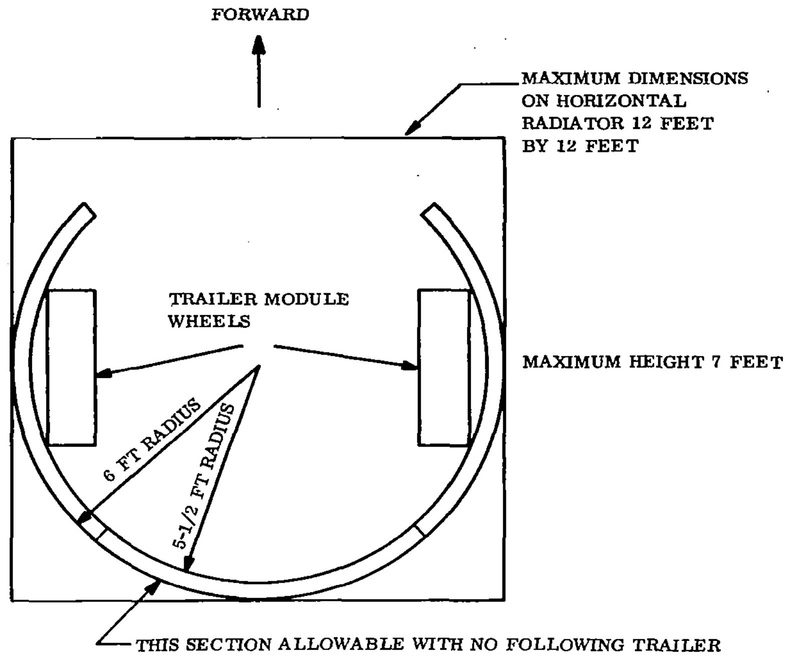

Figure 8-5. Radiator Envelope for Power System Mounted on Trailer Module

8.2.5.1 Liquid Loop Exchangers and Radiator

The approach taken is to provide two argon-to-liquid heat exchangers with corresponding liquid radiators. Figure 8-6 shows the arrangement. A discussion and design data can be found in Appendix D. The higher temperature radiator (No. 1) occupies the vertical portion of the radiator envelope and the lower temperature radiator (No. 2) is horizontal. The horizontal radiator has the lower sink temperature so maximum use is made of this part of the envelope. Based on the calculations in Appendix D, the following sizes and weights are determined for dissipation of the 12.4 kw.

	Size	Weight
Radiator No. 1	126 ft^2	31 lb
Radiator No. 2	144 ft^2	94 lb
Heat Exchanger No. 1	0.4 ft^3	19 lb
Heat Exchanger No. 2	1.0 ft^3	36 lb

Total radiator area = 270 ft^2, Total weight 180 lb

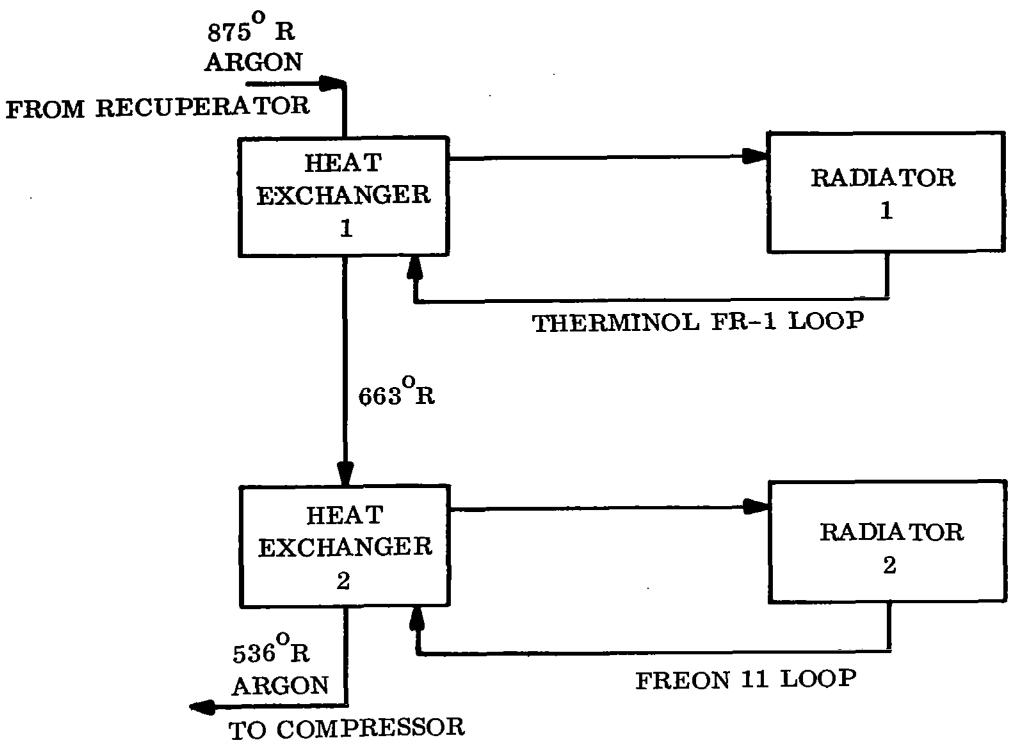

Figure 8-6. Brayton System Liquid Loop Radiator System

8.2.5.2 Vapor Fin Radiator

The vapor fin is potentially a very useful heat transfer device which has been under study and development for several years. The operating mechanisms involved in this device are indicated in Figure 8-7. Essentially, the vapor fin is a duct, partially filled with a working fluid, having one or more localized wall surfaces which are externally heated and also wall areas from which heat is removed. Through the combined agencies of working fluid evaporation, vapor flow, condensation and liquid refluxing, this duct functions as a heat transfer fin of near unity effectiveness. By properly selecting the fin working fluid in relation to the desired operating temperature level, the internal pressure can be kept low and the fin envelope can be very light in weight.

Factors of critical importance in vapor fin performance are the capillary structures which are attached to the internal heat input surfaces and also those attached to the surfaces along

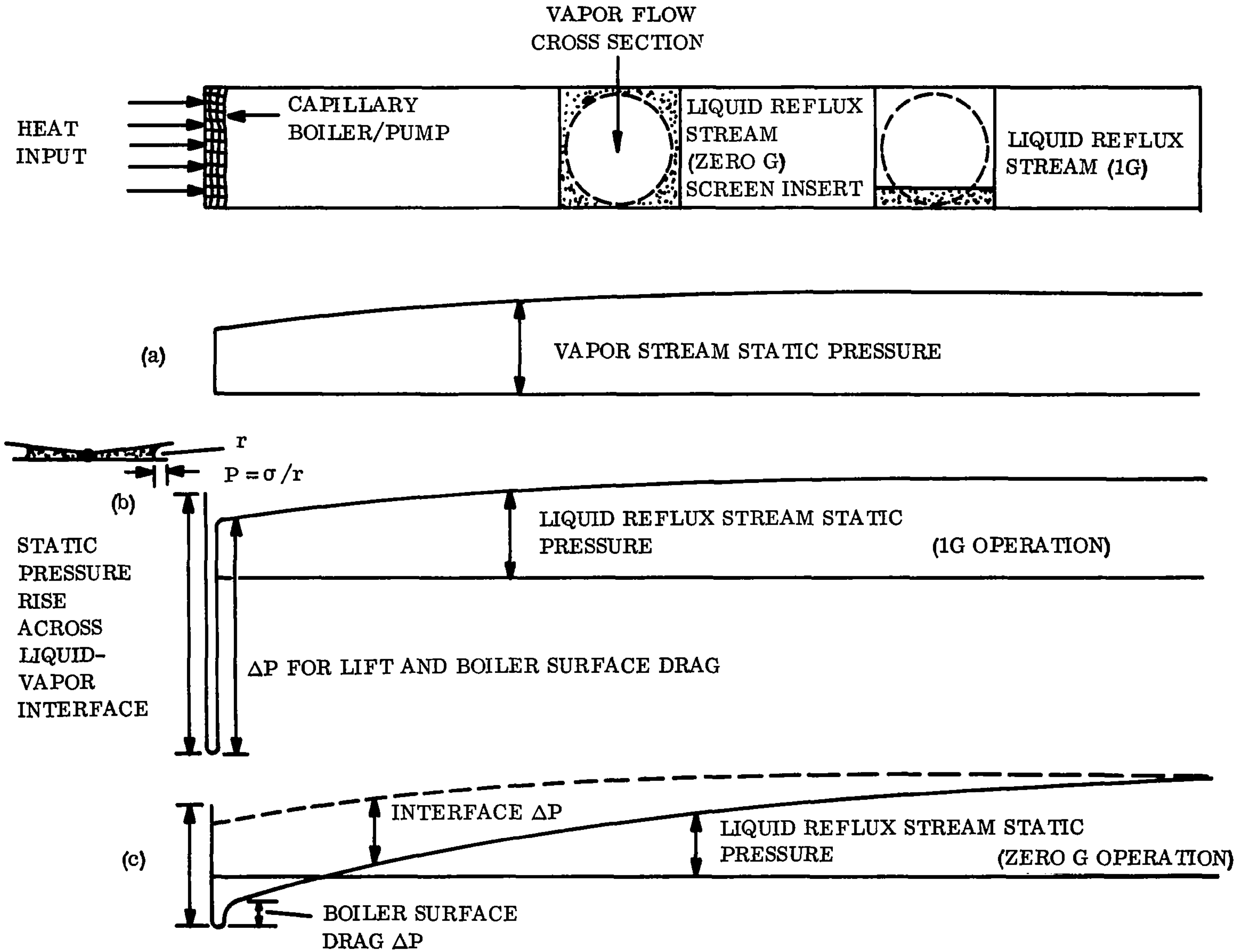

Figure 8-7. Vapor Chamber Flow Dynamics

which liquid condensate refluxing must occur. The heat input surface capillary structure is essential in all vapor fin applications. This structure is the boiler pump that distributes liquid over the heat input surface, regardless of its orientation in a gravity field, and which implements and stabilizes the evaporation process so as to achieve vapor generation with a minimum temperature difference between the heat input surface and the vapor saturation temperature. Further, the boiler pump structure stabilizes the location of vapor-liquid interfaces across which the pressure rise, necessary for the motivation of the two-phase flow is produced. For the heat transfer requirements of the radiators described herein, which are well below experimentally established limits, a simple square weave screen in two layers suitably interrupted for vapor release and spot welded to the vapor fin heat input surfaces, will suffice for the heat input surface capillary structure.

For static operation of a vapor fin in a gravity field, reflux surface capillaries are not required, as long as the fin tube axis is horizontal, or oriented with the vaporization surface below the condensing surface. However, in order to provide stabilization of the reflux stream in a moving vehicle and also to provide for refluxing in a tilted orientation of the radiator involving an uphill operation, reflux surface capillaries are required. For the reflux rates required in the low lunar gravity, even under conditions of 10 degrees tilt of the vehicle, a single layer screen insert, as shown in Figure 8-7 will be quite adequate. The vaporization surface capillary structure is a two-layer continuation of the single layer reflux surface screen.

Experimental data has been generated which shows the capillary boiler pump flow-head performance (using water as the working fluid) substantially in excess of that required by the study designs presented below. Some of this is presented in Figure 8-8. This overall temperature difference between the heat input and heat rejection surfaces of the vapor fins provided in the designs herein is on the order of $10^{\circ}F$.

The proposed vapor fin radiator design concept is illustrated in Figures 8-9 and 8-10. The overall configuration meets the space limitations specified in Figure 8-5.

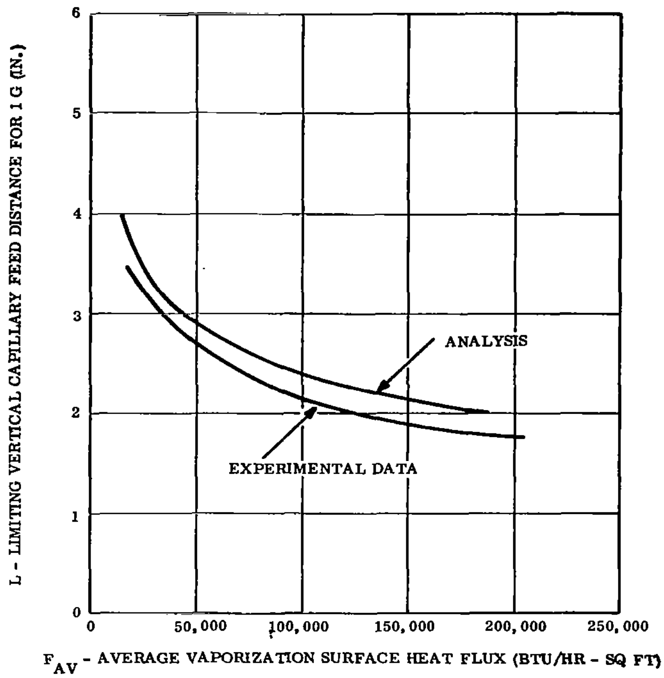

Figure 8-8. Correlation of Analysis and Experiment - SPPS Capillary Boiler Pump

As shown in Figure 8-9 the radiator structure consists of a 270^{o} cylindrical segment portion surmounted by a square plane. The high temperature portion of the radiator is the cylindrical portion, since this sees the highest effective sink temperature. The argon duct is continuous between the two sections.

The vapor fin radiator structure incorporates a single argon duct into which flow is introduced at both ends and from which flow is removed at the center. Internal fins receive heat by forced convection and conduct it to the duct outer wall surface, which forms the vaporization surface of each attached vapor fin, oriented at 90 degrees to the argon duct. There are a large number of individually sealed vapor fins, each of which operates at its own temperature level, corresponding to the temperature of the argon at the section of the argon duct to which the individual vapor fin is attached.

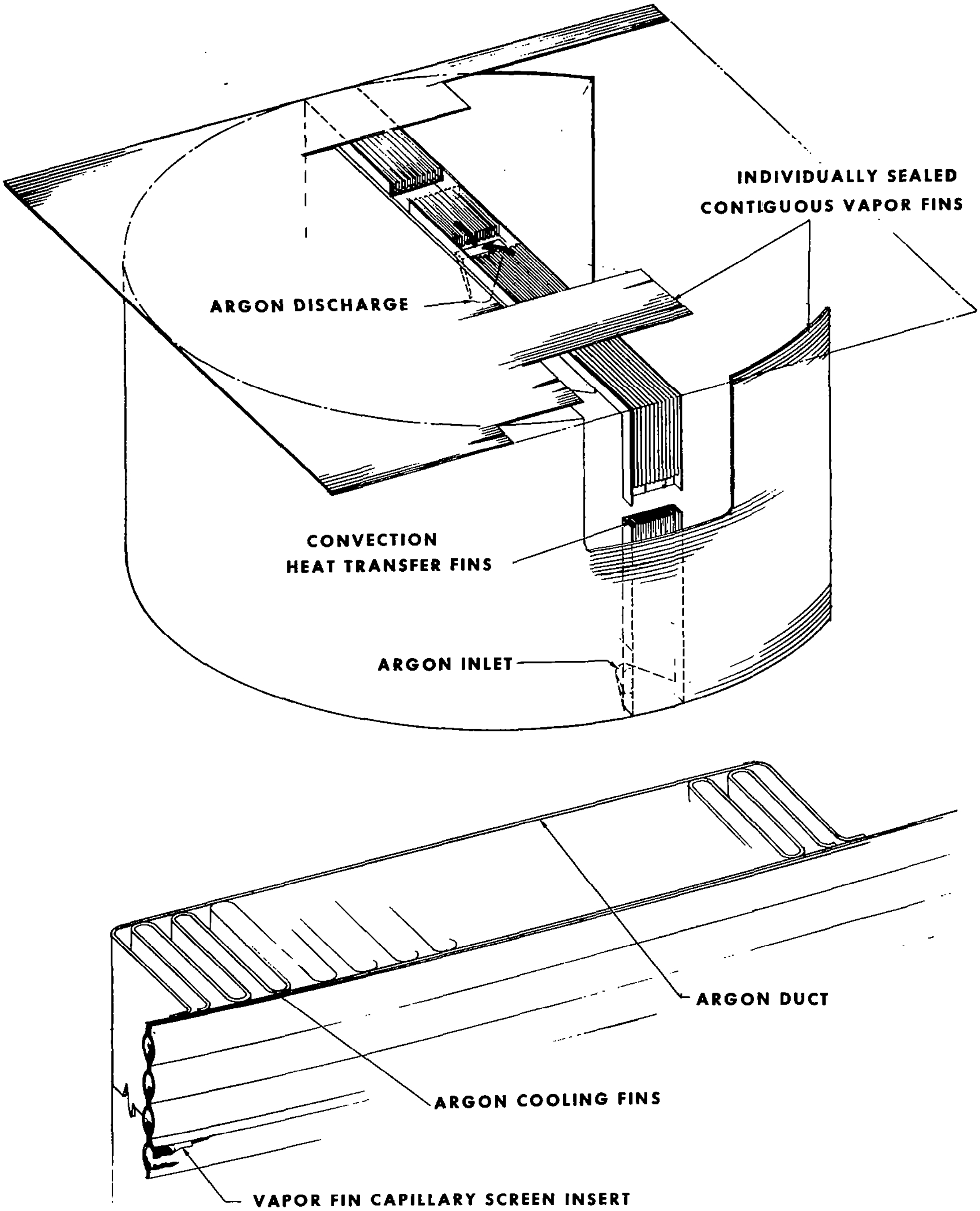

Figure 8-9. Vapor Fin Radiator Concept

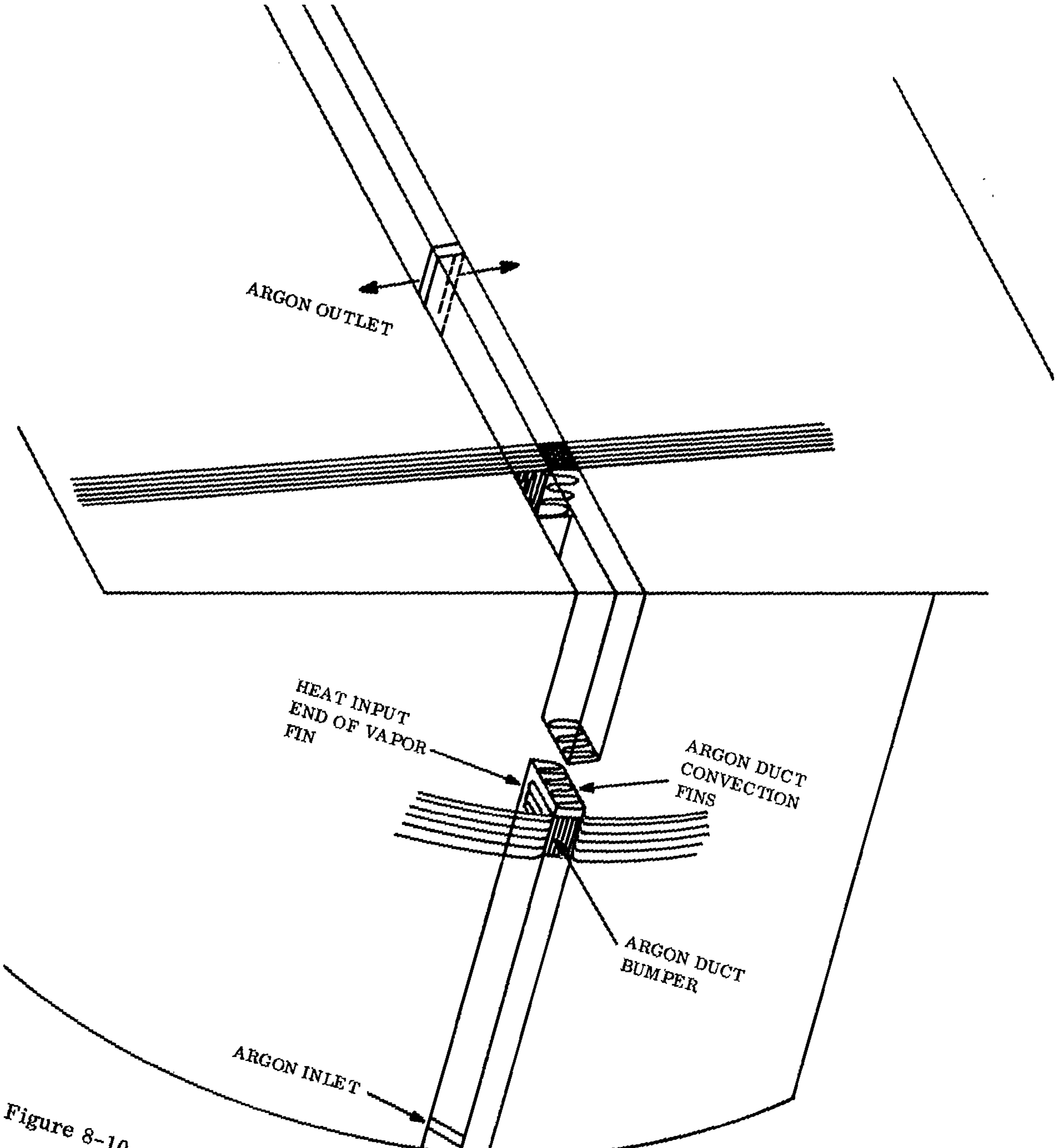

Figure 8-10. Vapor Fin Radiator Modification with Argon

The design modification shown in Figure 8-10 although it is somewhat more complex to fabricate than the design of Figure 8-9 has the advantage of higher convective fin effectiveness for a given fin thickness, and, also, less argon duct surface exposed to meteorite impact, with resulting reduction in duct weight.

The advantages offered to the lunar roving vehicle power plant design by the vapor fin concept include the following:

a. Minimum radiator size for given heat rejection rate and heat sink requirements. This results from the high effectiveness of the vapor fins, and also from the elimination of the argon-to-radiator fluid heat exchanger, ΔT.

b. System simplification by virtue of providing direct radiator cooling of the argon. By concentrating the argon in a single duct enclosed within the vapor fin structure, the high armor weight of the conventional argon radiator is eliminated.

The calculations and design of the vapor fin radiator are given in Appendix E. The results are that for 12.4 kw of heat rejection, the total area required is 227 ft^2 at a weight of 154 lb.

8.2.5.3 Comparison of Heat Rejection Methods

The vapor fin radiator is 26 pounds lighter and has 43 square feet less vertical radiator area than the fin tube design. This comes about because the vapor fin radiator fin efficiency approaches 1 with a relatively lightweight design, while the fin tube design achieves its lightest design at a lower fin efficiency. Also, the vapor fin has two-path direct argon flow, while the fin tube system has several tube runs and two argon to liquid heat exchangers.

The fin tube radiator has an advantage in folding ease for storage within the booster shroud envelope.

The vapor fin radiator has these additional advantages:

a. Does not require radiator loop pumps.

b. Self-compensation for variable load operation.

c. Provides greater parallel heat rejection reliability from meteorite punctures, due to fin heat rejection sharing.

d. Better adjustment to radiator fluid freezing such as in the dormant period.

8.2.6 OTHER COMPONENTS

8.2.6.1 Turbomachinery

Cycle thermodynamic efficiency is a function of compressor pressure ratio. Analysis shows that maximum efficiency at the desired inlet temperature occurs at a pressure ratio of about 2.3. A single stage centrifugal compressor rotating at 67,500 rpm, with an inlet pressure of 6 psia and the desired pressure ratio of 2.3, satisfies the compressor requirements. The compressor turbine is the radial inflow type integral with the shaft. The estimated weight of the turbocompressor gas generator unit is 34 lb. Its approximate size is 12 inches in diameter and 10 inches in length.

The turboalternator consists of a single stage turbine and a 4-pole homopolar alternator. The rotation speed is established by the frequency and power requirements. Power at 400 Hz can be generated with 12,000 rpm shaft rotation. Alternator efficiency as a function of design power is given in Figure 6-23 of Reference 8-1. Weight estimates are given in Figure 6-24 of Reference 8-2. At the 3.51 kw level, the alternator efficiency is 86.5%. The power unit weight is 52 lb for 2.85 kw net power.

8.2.6.2 Recuperator

The recuperator is based on a design given in Reference 8-1. It has an effectiveness of 0.9. The pressure drop in the hot side is 0.14 psi and in the cold side is 0.05 psi. Its estimated size is 24 x 24 x 12 inches, or 1152 cubic inches. It weighs about 140 lb (Figure 11-27 in Reference 8-2).

8.2.6.3 Battery

The battery supplies the energy storage capability so that the alternator can operate at a continuous constant load. Referring to the power profile, Figure 3-4, the maximum battery storage will be needed for the three peaks including the experiment activity. The net energy storage for these three peaks is 14.0 kw-hr. If a 50 percent depth of discharge is used, the required battery size is 28.0 kw-hr. The batteries will be discharged to this 50 percent level once every three days or 15 times during the mission. Much lesser depths of discharge will occur at each power peak or 120 times. The effective number of cycles corresponding to a 50 percent depth of discharge will be between 15 and 120. Silver-zinc batteries appear to be capable of operation under these conditions. It is desirable to use this type of battery since Ag-Zn has a high energy density.

A reasonable energy density, allowing for near term future development, is 50 $\frac{\text{watt-hr}}{\text{lb}}$. This figure gives a battery weight of 560 lb. Its approximate size is 12 x 12 x 36 inches, or 5184 cubic inches.

The battery is mounted on the cabin to be readily available for emergency power in case of a coupling failure between the cabin and power trailer.

8.3 INTEGRATED SYSTEM

8.3.1 DESCRIPTION

Waste heat from the nonintegrated Brayton system is lost to space through the main radiator. It is the purpose of thermal integration to make use of some of this thermal energy in the life support system. This is accomplished by the addition of a heat exchanger to the argon loop at a point where the argon exits the hot side of the recuperator (see Figure 8-3). A temperature of about 415^{o}F is available. The thermal energy extracted from this heat exchanger is used in the life support system. This results in a reduction in the electrical power requirements. Also, the power system radiator size is reduced. With the exception of the life support heat exchanger, the basic integrated power system configuration is the same as the nonintegrated system. The integrated system is sized for the reduced electrical load. Table 8-4 presents component weights for the nonintegrated Brayton system.

TABLE 8-4. ISOTOPE BRAYTON SYSTEM WEIGHT (NONINTEGRATED)

Isotope source subsystem	915* lb
Turbocompressor (2)	68
Turboalternator (2)	104
Recuperator	140
Controls subsystem	108
Support structure and piping	260
Main radiators	125
Heat exchangers (heat rejection system)	55
Secondary heat rejection system	38
Power system weight	1813 lb
Water for suit cooling	750
Locomotion subsystem	600
Total trailer weight	3163 lb

*Approximate

8.3.2 POWER REQUIREMENTS

The mission net power profile for the nonintegrated system is given in Figure 3-4. The integrated system also uses the closed-loop CO_2 system. Referring to Sections 4.4 and 4.7, the electrical power requirements of the life support system are 1741 watts for the non-integrated system and 748 watts for the integrated $400^\circ F$ system. Therefore the integrated electrical power is (1741-748) or 993 watts less than the nonintegrated case. Subtracting this amount and the 100 watts of RTG power from the curve of Figure 3-4, the average net electrical power required for the integrated system is 1.88 kw. One-third of the power is dc and two-thirds is 400 Hertz ac.

The same approach is taken with the integrated as was taken with the nonintegrated system in that batteries are provided to average the load. The same power conditioning configuration and efficiencies are used, (see Figure 8-3).

Table 8-5 is a summary of the power requirements. The average power conditioning loss is 410 watts. The losses of the different components are itemized in the table. The pump power is about the same as the nonintegrated case. Speed control amounts to 5 percent of the total electrical power. Alternator efficiency is estimated from Figure 6-23 of Reference 8-1. The rotation speed is the same as the nonintegrated case, so it is estimated that the bearing losses will be approximately the same. The total shaft power is 3.15 kw.

8.3.3 HEAT BALANCE

The cycle analysis is repeated for the integrated system using slightly lower component efficiencies. The compressor, compressor turbine and the power turbine efficiencies were all reduced by one percentage point since these components are usually less efficient as their size is decreased. The gas generator and power unit bearing losses were kept the same as the nonintegrated values. The temperatures and pressures at the given cycle points were not changed. Table 8-6 is a summary of the cycle analysis. The results give the power turbine work as 14.45 kw per lb of argon per second. Since the shaft work is 3.15 kw, the required flow rate is 0.218 lb/sec. From the cycle analysis and flow rate, the heat added is 13.7 kw and the heat rejected is 9.77 kw.

TABLE 8-5. ISOTOPE BRAYTON POWER REQUIREMENTS (INTEGRATED)

Net electrical power		1.88 kw
Average power conditioning loss		0.41
Battery	0.18 kw	
Ac-dc	0.09	
Dc-dc	0.07	
BCR	0.05	
Diode	0.01	
Dc-ac	0.01	
Pump power (Therminol and Freon loops)		0.01
Pump power (NaK)		0.01
Speed control		0.12
Total electrical power		2.43 kw
Alternator loss (η = 0.855)		0.42
Bearing loss (power unit)		0.30
Total shaft power		3.15 kw

Table 8-7 is a power balance on the system. The net electrical power and battery loss are dissipated elsewhere. A secondary cooling loop removes the power conditioning losses, speed control, alternator losses and bearing cooling. This amounts to 1.87 kw. Section 4.6 gives the thermal power to the life support system as 1191 watts for the 400°F case. This amount of power is subtracted from the cycle waste heat to obtain the required power rejection from the primary radiator, (9.77-1.19 = 8.58 kw).

The ducting loss is assumed to be one to two percent. The total power out is 13.9 kw which must equal the source power.

TABLE 8-6. BRAYTON CYCLE ANALYSIS (INTEGRATED)

Given Quantities	
Compressor Inlet Temperature, T_1	$536^{\circ}R$
Turbine Inlet Temperature, T_4	$1950^{\circ}R$
Compressor Inlet Pressure, P_1	6.0 psia
Compressor Pressure Ratio P_2/P_1	2.3
Compressor Efficiency	0.79
Compressor Turbine Efficiency	0.82
Power Turbine Efficiency	0.83
Recuperator Effectiveness	0.90
Gas Generator Bearing Losses	0.80 KW
Recuperator Pressure Drop, Cold Side $(P_2 - P_3)$	0.14 psi
Heat Source Pressure Drop, $(P_3 - P_4)$	0.69 psi
Recuperator Pressure Drop, Hot Side, $(P_6 - P_7)$	0.05 psi
Radiator Pressure Drop $(P_7 - P_1)$	0.32 psi

Cycle Points

Station	T ($^{\circ}R$)	P (psia)
1	536	6.00
2	804	13.80
3	1472	13.66
4	1950	12.97
5	1657	7.84
6	1547	6.37
7	877	6.32

$$\text{Power Turbine Work per lb} = C_P (T_5 - T_6) = 14.45 \ \frac{\text{kw-sec}}{\text{lb}}$$

$$\text{Heat Added per lb} = C_P (T_4 - T_3) = 62.9 \ \frac{\text{kw-sec}}{\text{lb}}$$

$$\text{Heat Rejected per lb} = C_P (T_7 - T_1) = 44.8 \ \frac{\text{kw-sec}}{\text{lb}}$$

$$\dot{m} = 0.218 \text{ lb/sec}$$

TABLE 8-7. BRAYTON SYSTEM POWER BALANCE (INTEGRATED)

Electrical power out		1.88 kw
Radiation from secondary radiator		1.87 kw
Power conditioning loss	0.23	
Speed control	0.12	
Alternator loss	0.42	
Bearing cooling	1.10	
Battery heat loss		0.18 kw
Thermal power to life support		1.19 kw
Radiation from primary radiator		8.58 kw
Ducting thermal loss		0.20 kw
Total power out		13.9 kw
Source power required		13.9 kw

8.3.4 ISOTOPE SOURCE

The required amount of isotope heat for the integrated system is 13.9 kw, see Table 8-7. The basic design, source configuration and safety requirements are identical to the non-integrated system and may be found in Appendix C. The following estimates can be made for a 13.9 kw source.

Shield weight	270 lb
Re-entry protection weight	221 lb
Total subsystem weight	795 lb

8.3.5 HEAT REJECTION SYSTEM

To obtain thermal energy for the life support system, a heat exchanger is added in the argon loop between the recuperator and the heat rejection system. The argon exits the recuperator at 415^{o}F. Allowing for a temperature drop between the argon and the thermal integration fluid, thermal energy at the 400^{o} level can be supplied to the life support system. Section 4.7 gives the required amount of thermal power as 1191 watts at $400^{o}F$. The closed loop (oxygen recovery) system is used with the isotope power systems. From Table 8-7, the cycle waste power (after subtracting the 1191 watts) tobe dissipated by the radiator is 8.58 kw.

The radiator envelope and operating constraints for the integrated heat rejection system are identical to the nonintegrated case. The two types of radiators, liquid loop and vapor fin, are also considered for the integrated heat rejection.

8.3.5.1 Liquid Loop Exchanger and Radiator

The integrated arrangement is the same as the nonintegrated except that an additional heat exchanger (for life support) is in the argon loop between the recuperator and the first radiator heat exchanger. The temperature of the argon leaving the life support heat exchanger is about $835^{o}R$. The amount of heat to be radiated is 8.6 kw. Based on the material in Appendix D, the following sizes and weights were determined for the integrated heat rejection system.

	Size	Weight
Radiator No. 1	51 ft^2	12.6
Radiator No. 2	140 ft^2	73.0
Heat Exchanger No. 1	1.5 ft^3	35.0
Heat Exchanger No. 2	2.2 ft^3	31.4

Total radiator area = 191 ft^2 Total weight = 152 lb

8.3.5.2 Vapor Fin Radiator

The vapor fin radiator for the integrated case is the same design as for the nonintegrated case except it is sized for a 8.6 kw load. The results (see Appendix E) are that for 8.6 kw of heat ejection, the area required is 158 sq ft and the weight is 110 lb. The heat exchanger for the life support thermal power used with the vapor fin radiator is identical to the one used with the liquid loop radiator.

8.3.5.3 Comparison of Heat Rejection Methods for Integrated System

The vapor fin rejectór system is 33 sq ft smaller and 42 pounds lighter than the fin tube radiator design. The same advantages discussed in 8.2.5.3 for the nonintegrated system are applicable to the integrated system.

8.3.6 OTHER COMPONENTS

The integrated turbomachinery is practically the same as the nonintegrated. The turbocompressor operates at the same pressure ratio and same inlet pressure and temperature. Its size and weight will be essentially the same as the nonintegrated case.

The turboalternator is the same as used in the nonintegrated case. Figure 11-26 of Reference 8-2 gives a weight estimate of 44 lb for a 1.88 kw net power unit.

The recuperator is scaled from data in Reference 8-1. Its estimated weight is 110 lb.

The battery is the same type as used in the nonintegrated system. The procedure for sizing is identical. The required energy storage is 13.5 kw-hr. Using a 50 percent depth discharge, the required capacity is 27.0 kw-hr. At 50 watt-hr/lb the battery weight is 540 lb.

8.4 DISCUSSION

This section has considered the Isotope Brayton Power System for application to a lunar roving vehicle. System configurations and weights were obtained for the nonintegrated case and for the case of thermally integrating the power system with the life support system. There is no significant problem in matching a significant portion (about 60 percent) of the life support energy requirements to the power system waste heat. Using the power levels of the study guidelines, a 16 to 17 percent reduction in power system weight is possible.

TABLE 8-8. ISOTOPE BRAYTON SYSTEM WEIGHT (INTEGRATED)

Isotope source subsystem	795* lb
Turbocompressor (2)	68
Turboalternator (2)	88
Recuperator	110
Controls subsystem	80
Support structure and piping	210
Main radiators	86
Heat exchangers (heat rejection system)	33
Heat exchanger (life support)	15
Secondary heat rejection system	34
Power system weight	1519 lb
Water for suit cooling	750
Locomotion subsystem	600
Total trailer weight	2869 lb

*Approximate

8.5 REFERENCES

8-1 Hanson, K. L., "Thermal Integration of Electrical Power and Life Support Systems for Manned Space Stations," General Electric Company, NASA CR-316, November 1965.

8-2 Erlanson, E. and R.W. Woods, "Study of Thermal Integration of Electric Power and Life Support Systems for Manned Space Stations," General Electric Company, Preliminary draft of final report, Contract NAS 3-4678, November 1965.

8-3 Glassman, Arthur J., "Summary of Brayton Cycle Analytical Studies for Space Power System Applications," NASA TN D-2487, September 1964.

SECTION 9

MERCURY RANKINE POWER SYSTEM

9.1 INTRODUCTION

The previous section considered an isotope dynamic system utilizing a Brayton thermodynamic cycle. This section is concerned with an isotope dynamic system utilizing a Rankine thermodynamic cycle with mercury as the working fluid. The advantages of this system are that the ideal Rankine cycle approaches Carnot efficiency and the waste heat can be rejected at a relatively high temperature. Liquid metals have good heat transfer characteristics and high mass flow so the components can generally be made smaller than equivalent components dealing with gas or other fluids. Figure 9-1 depicts a possible power system equipment arrangement on the lunar roving vehicle trailer module.

9.2 NONINTEGRATED SYSTEM

9.2.1 DESCRIPTION

The power system consists of an isotope heat source, turbo alternator, pump and condenser-radiator. A temperature-entropy diagram is given in Figure 9-2. The liquid mercury is pumped to boiler pressurė, 1-2. From 2-3, the mercury is heated and evaporated in the boiler by the isotope source. The vapor is heated 3-4 and expanded through the turbine, 4-5. The mercury vapor condenses and waste heat is rejected from the condenser-radiator, 5-1.

A functional diagram is shown in Figure 9-3. Similar systems are considered in Reference 9-1 and 9-2. Much of the work is based on Sunflower technology, References 9-3 and 9-4. The turbo alternator rotates at 40,000 rpm with the alternator producing 2000 Hz power. For comparison purposes, the ac power is converted to 400 H_z. This is a more common power frequency and is the frequency of the Brayton system. The configuration is a "one

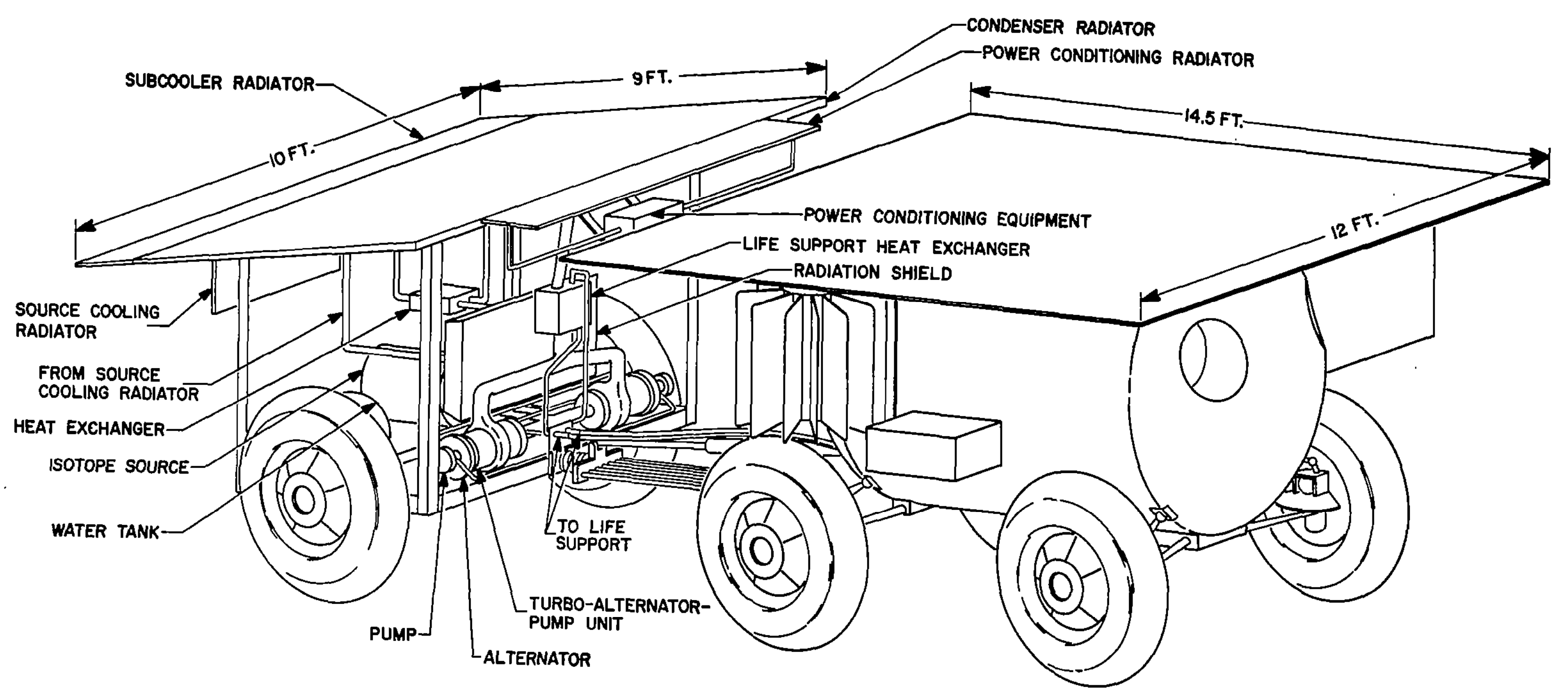

Figure 9-1. Mercury Rankine Power System Equipment

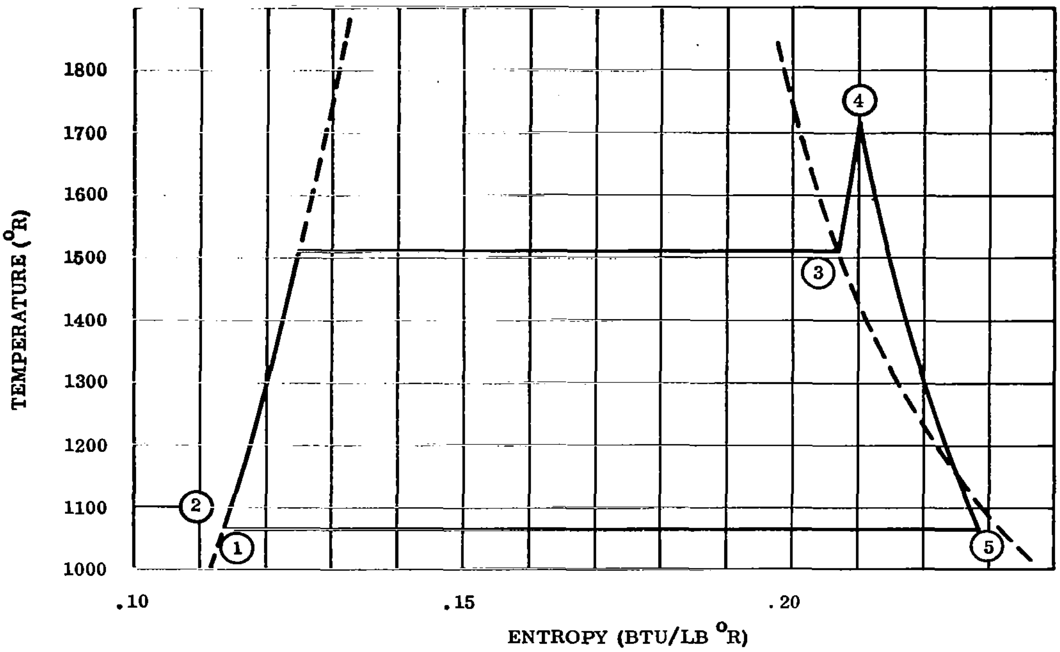

Figure 9-2. Idealized Rankine Cycle

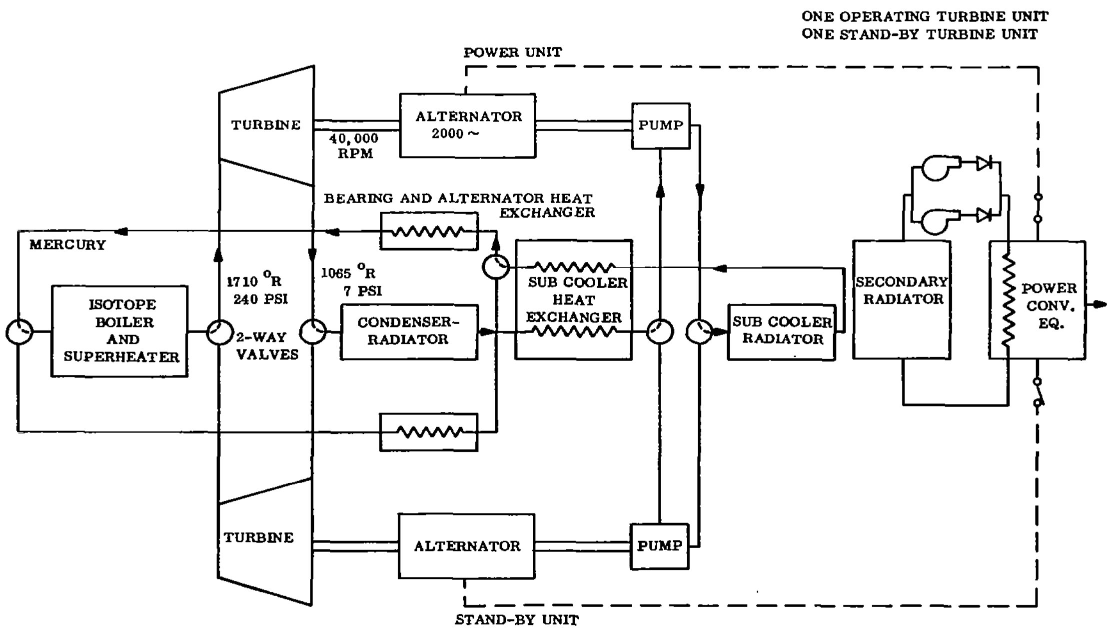

Figure 9-3. Rankine Cycle Functional Diagram

operating plus one standby" turbo alternator-pump unit. This provides 100% redundancy of the rotating machinery. A number of valves are used so that either turboalternator can be operated. Start-up auxiliaries are provided. The bearings and alternator are cooled by the mercury to about 600^{o}F. The power conversion equipment must be cooled to about 185^{o}F so a separate cooling loop is provided.

9.2.2 POWER REQUIREMENTS

The net power required for the Rankine system is the same as for the Brayton system, Figure 3-4, using the closed loop CO_2 processing equipment. Subtracting the 100 watts supplied by the RTG, the average net power is 2.85 kw. One third of the power is dc and two thirds is ac (400 H_z). A battery is used so the turboalternator can operate at approximately constant load, as in the Brayton system. The power conditioning configuration is shown in Figure 8-4. A 2000 Hz to 400 Hz frequency converter is used. Otherwise the configuration is the same as the Brayton system. Average power conditioning losses can be calculated using the efficiencies shown. Speed control requires an average of 5% of the alternator output. Additional shaft power consists of the alternator loss, bearing loss and Hg pump power. These are determined from References 9-3 and 9-4. Table 9-1 lists the power requirements.

9.2.3 HEAT BALANCE

A cycle analysis can be made to determine the mercury flow rate, the isotope source power requirements and the power radiated. Table 9-2 shows the given quantities and the results of the analysis. Cycle operating points were obtained from References 9-3 and 9-4. Mercury properties are from Reference 9-5. The analysis is simplified in that the heating and condensing processes are assumed constant pressure, the subcooling is not shown and the pump is idealized. The mercury flow rate is determined by dividing the shaft power by the turbine energy per lb of Hg per second. In this case, 5.15kw/22.5kw-sec/lb = 0.229 lb/sec.

TABLE 9-1. ISOTOPE RANKINE POWER REQUIREMENTS (NONINTEGRATED)

Net electrical power		2.85 kw
Average power conditioning loss		0.68
Ac-Ac	0.18 kw	
Battery	0.19	
Ac-Dc	0.12	
Dc-Dc	0.11	
BCR	0.055	
Diode	0.01	
Dc-Ac	0.005	
Speed control		0.18
Total electrical power		3.71
Alternator loss (η = 0.87)		0.56
Pump power		0.18
Bearing losses		0.70
Total shaft power		5.15 kw

TABLE 9-2. RANKINE CYCLE ANALYSIS (NONINTEGRATED)

GIVEN QUANTITIES

Boiler pressure, P_2	240 psia
Turbine inlet temperature, T_4	1710° R
Turbine outlet temperature, T_5	1065° R
Condensing pressure, P_5	7 psia
Turbine efficiency	52 %

CYCLE POINTS

Station	T (° R)	P (psia)	H $\frac{\text{(kw-sec)}}{\text{lb}}$
1	1065	7	43.4
2	1071	490	43.6
3	1510	240	189.5
4	1710	240	194.8
5	1065	7	172.3

Turbine energy per lb of Hg per sec = $(H_4-H_5) = 22.5 \frac{\text{kw-sec}}{\text{lb}}$

Heat rejected per lb of Hg per sec = $(H_5-H_1) = 128.9 \frac{\text{kw-sec}}{\text{lb}}$

Heat added per lb of Hg per sec = $(H_4-H_2) = 151.2 \frac{\text{kw-sec}}{\text{lb}}$

$\dot{W} = 0.229$ lb/sec

9.2.4 ISOTOPE SOURCE

The isotope sources for the Rankine Systems are used to boil the mercury directly. There is no need for a NaK loop to transfer heat to the mercury since a compact isotope heat source can be made using radiation heat transfer to tubes containing the mercury. However, a NaK loop and radiator are provided for supplementary cooling in case of mercury loop malfunction. A conceptual design of this type of source may be found in Appendix C. The source for the nonintegrated system must provide 35 kw of thermal power and has the following estimated characteristics.

Size:	Length, D, 20.9 inches, Diameter, C, 20.7 inches
Radiation shield weight:	435 lb
Re-entry protection:	390 lb
Total source subsystem:	1520 lb

A heat balance for the nonintegrated system is presented in Table 9-3. The net electrical power is delivered to the roving vehicle cabin. The battery is mounted on the cabin unit for effective temperature control and to be readily available for emergency use. The cycle waste power is the flow rate multiplied by $(H_5 - H_1)$. The other waste power is itemized in the table. The ducting thermal loss is on the order of 1%. The source power must supply the cycle requirements.

9.2.5 HEAT REJECTION

The condensing radiator design is taken from Reference 9-2. The design has the following parameters:

a.	Absorptivity to solar radiation	0.25
b.	Emissivity at operating temperature	0.90
c.	Fin effectiveness	0.545
d.	Weight (lb/ft^2)	0.78

TABLE 9-3. RANKINE SYSTEM POWER BALANCE (NONINTEGRATED)

Net electrical power			2.85 kw
Battery heat			0.19
Radiated power			31.63
Cycle waste power		29.50 kw	
Other waste power		2.13	
Power conditioning loss	0.49		
Speed control	0.20		
Alternator loss	0.56		
Pump power (Hg)	0.18		
Bearing loss	0.70		
Ducting thermal loss			0.33
Total power out			35.0 kw
Source power required			35.0 kw

The operating temperature of the radiator is 1060°R. At lunar noon, this gives 0.275 kw/ft^2 of heat rejected. The total amount of heat rejected through this radiator is 31.4 kw. This includes the cycle waste heat, bearing losses, and pump power. The required radiator area is 114 ft^2 at a weight of 89 lb. The radiator is approximately horizontal, forming a roof over the power system trailer.

An additional radiator is provided for thermal control of the power conditioning equipment. The amount of waste heat is equal to the power conditioning loss (0.68 kw), minus the battery loss (0.10 kw), or 490 watts. The radiator will have the following characteristics:

a.	Absorptivity to solar radiation	0.2
b.	Emissivity at operating temperature	0.9
c.	Fin effectiveness	0.8
d.	Weight (lb/ft^2)	1.0

The operating temperature of the radiator is 180°F. Using the above numbers, the heat rejected at lunar noon is 34 watts/ft^2. This gives a radiator area of 15 ft^2 with a weight of about 15 lb. The radiator is horizontal above the power system trailer in front of the main radiator.

9.2.6 OTHER COMPONENTS

The turboalternator-pump unit is a single, rotating assembly using hydrodynamic bearings. The size of the unit is about 13 inches long by 6 inches in diameter. Its weight is 32 lb. The speed control, start auxiliaries, structure and piping, and mercury inventory weights were estimated from References 9-3 and 9-4.

The battery is Ag-Zn sized for a 50% depth-of-dischárge at the completion of the 3 power peaks including the experiment peak. The required size is 27.8 kw-hr. At 50 $\frac{\text{watt-hr}}{\text{lb}}$ the weight is 560 lb. The battery is mounted on the cabin for thermal control and to be readily available in emergency situations.

TABLE 9-4. ISOTOPE RANKINE SYSTEM WEIGHT (NONINTEGRATED)

Isotope source subsystem	1520 *
Turboalternator (2)	64
Speed control	30
Start auxiliaries (2)	140
Mercury inventory (2)	36
Main radiator	89
Secondary radiator	15
Structure	50
Power system weight	1944 lb
Trailer locomotion system	600
Water for suit cooling	750
Total trailer weight	3294 lb

* Approximate

9.3 INTEGRATED SYSTEM

9.3.1 DESCRIPTION

The Rankine system can be thermally integrated with the life support system by incorporating a heat exchanger in the mercury loop between the turbine outlet and the radiator inlet. Thermal energy at 600°F is supplied from the heat exchanger to the life support system. This results in a reduction in the electrical power requirements of the life support system and a reduction in the size and weight of the power system radiator. With the exception of this additional heat exchanger, the power system configuration is identical to the non-integrated system. The components are sized for the reduced electrical load.

9.3.2 POWER REQUIREMENTS

The power profile of Figure 3-4 (closed-loop CO_2) represents the integrated Rankine power system requirements by subtracting a constant amount of power. This amount is the 100 watts supplied by the RTG and the difference between the electrical power requirements of the nonintegrated and integrated life support system. Referring to Sections 4.4 and 4.7, this difference is (1741 - 721) or 1.02 kw (the 600°F closed-loop values are used). Therefore, the net average electrical power required for the integrated system is 1.85 kw. One-third is dc and two-thirds is 400 Hz ac. Batteries are provided to average the load. The power conditioning configuration is the same as the nonintegrated case.

Table 9-5 is a summary of the power requirements. The average power conditioning losses were calculated using Figure 8-4. The total average conditioning loss is 490 watts. Speed control requires 5% of the alternator output.

Alternator efficiency, pump power and bearing losses are estimated from data in References 9-3 and 9-4. The total shaft power is 3.76 kw.

TABLE 9-5. ISOTOPE RANKINE POWER REQUIREMENTS (INTEGRATED)

Net electrical power		1.85 kw
Average power conditioning loss		0.49
Battery	0.18	
Dc-dc	0.07	
Ac-ac	0.06	
BCR	0.055	
Ac-dc	0.10	
Dc-ac	0.015	
Diode	0.01	
Speed control		0.13
Total electrical power		2.47 kw
Alternator loss (η = 0.86)		0.41
Pump power		0.18
Bearing losses		0.70
Total shaft power		3.76 kw

9.3.3 HEAT BALANCE

The cycle analysis of the integrated system yields slightly different values of turbine energy and heat rejected per lb of Hg because a lower turbine efficiency is used. Data from References 9-3 and 9-4 indicate the turbine efficiency decreases as the units are made smaller. The turbine inlet and outlet temperatures and pressures are the same as those used in the nonintegrated system. Table 9-6 shows the given quantities and the results of the analysis. The mercury flow rate is 3.76 kw divided by 20.3 $\frac{\text{kw-sec}}{\text{lb}}$ or 0.185 lb/sec. The cycle waste heat is H_5-H_1 multiplied by the flow rate of about 24.3 kw. The net cycle waste is obtained by subtracting the life support thermal power (1143 watts). Table 9-7 is the power balance. The required source power is 28.4 kw.

9.3.4 ISOTOPE SOURCE

The isotope heat source for the integrated Rankine system is a slightly scaled down version of the nonintegrated source. It must furnish 28.4 kw of thermal power and its estimated characteristics are:

Size:	Length, D, 20.4 inches; Diameter, C, 19.9 inches
Radiation shield weight:	400 lb
Re-entry protection:	382 lb
Total source subsystem:	1360 lb

9.3.5 HEAT REJECTION

The condensing radiator for the integrated system is a smaller version of the nonintegrated system radiator. It can radiate 0.275 kw/ft^2 at lunar noon. The necessary area is then 24.6 kw/0.275/kw/ft^2 or 90 ft^2. The 24.6 kw is the net cycle waste heat, bearing losses, speed control, alternator loss, and pump power.

The power conditioning loss (minus the battery heat loss) is dissipated in a secondary

TABLE 9-6. RANKINE CYCLE ANALYSIS (INTEGRATED)

Given Quantities	
Boiler pressure, P_2	240 psia
Turbine inlet temperature, T_4	$1710^{\circ}R$
Turbine outlet temperature, T_5	$1065^{\circ}F$
Condensing pressure, P_5	7 psia
Turbine efficiency	46 %

Cycle Points

Station	T ($^{\circ}$ R)	P (psia)	H $\frac{\text{kw-sec}}{\text{lb}}$
1	1065	7	43.4
2	1071	490	43.6
3	1510	240	189.5
4	1710	240	194.8
5	1065	7	174.5

Turbine energy per lb of Hg per sec = $(H_4-H_5) = 20.3 \ \frac{\text{kw-sec}}{\text{lb}}$

Heat rejected per lb of Hg per sec = $(H_5-H_1) = 131.1 \ \frac{\text{kw-sec}}{\text{lb}}$

Heat added per lb of Hg per sec = $(H_4-H_2) = 151.2 \ \frac{\text{kw-sec}}{\text{lb}}$

$\dot{m} = 0.185$ lb/sec

TABLE 9-7. RANKING SYSTEM POWER BALANCE (INTEGRATED)

Net electrical power			1.85 kw
Battery heat (to cabin)			0.18
Thermal power to life support			1.14
Radiated power			
Net cycle waste power		23.17 kw	
Other waste power		1.73	
Power conditioning loss	0.31		
Speed control	0.13		
Alternator loss	0.41		
Pump power	0.18		
Bearing loss	0.70		
Ducting thermal loss			0.33
Total power out			28.4 kw
Source power required			28.4 kw

TABLE 9-8. ISOTOPE RANKINE SYSTEM WEIGHT (INTEGRATED)

Isotope source subsystem	1360 * lb
Turboalternators (2)	54
Speed control	28
Start auxiliaries (2)	130
Mercury inventory (2)	28
Main radiator	70
Secondary radiator	10
Structure	35
Life support heat exchanger	5
Power system weight	1720 lb
Trailer locomotion system	600
Water for suit cooling	750
Total trailer weight	3070 lb

* Approximate

radiator of the same type as the nonintegrated system secondary radiator. Its area is 310 watts/ 34 watts/ft^2 or about 9 ft^2.

9.3.6 OTHER COMPONENTS

The turboalternator-pump unit is the same type, but slightly smaller than the nonintegrated unit. Weights are obtained from Figure 11-32 of Reference 9-2.

The Ag-Zn battery is sized for 50% depth-of-discharge at the completion of the three power peaks including the experiment peak. A 26.3 kw-hr battery weighing about 530 lb is required.

9.4 DISCUSSION

The configuration and weight of an isotope Rankine power system was determined for the nonintegrated case and for the case of the thermal integration with the life support system. About 60% of the life support system power, or essentially all of the process thermal power, is obtained from the 600°F waste heat of the power system. The net electrical power reduction of 35% allows a power system weight reduction of 1944 to 1720 pounds - about an 11 percent change.

9.5 REFERENCES

9-1 Hanson, K.L., "Thermal Integration of Electrical Power and Life Support Systems for Manned Space Stations", General Electric Company, NASA Contractors Report CR-316, November 1964.

9-2 Erlanson, E. and R.W. Woods, "Study of Thermal Integration of Electric Power and Life Support Systems for Manned Space Stations," General Company, MSD, Document Number 66SD4231, Final Report for NASA Contract NAS 3-4678, January 1966.

9-3 Picking, J.W. and D.L. Southan, "Sunflower Status Review", Proceedings of the Solar Dynamic Systems Symposium, PIC-SOL 209/4, Sept. 1963.

9-4 "Sunflower Status and Applications Considerations", Thompson Ramo Wooldridge Electromechanical Division, TRW Bulletin 311-MRD-3.

9-5 Meisl, C.J., "Thermodynamic Properties of Alkali Metal Vapors and Mercury", General Electric Company, R60FPD358-A, Nov. 1960.

SECTION 10
FUEL CELL POWER SYSTEM

10.1 INTRODUCTION

A fuel cell power system is a possible choice for the 45-day lunar roving vehicle. The fuel cell converts chemical energy directly into electrical energy without going through an intermediate working fluid. Elimination of this step results in high theoretical efficiencies. Water is the product of H_2-O_2 fuel cells. These advantages and their selection for Gemini, Apollo and LEM suggest that fuel cells will be a major source for future missions (see Reference 10-1). Previous studies of lunar surface vehicles have considered fuel cell power supplies (References 10-2, 10-3, and 10-4). Apollo-type fuel cell modules were chosen to take full advantage of a developed system. The specific module to be considered is the PC3A developed by Pratt & Whitney Aircraft. Figure 10-1 depicts a possible power system equipment arrangement on the lunar roving vehicle trailer module.

10.2 DESCRIPTION

10.2.1 OPERATION DESCRIPTION

A fuel cell is an electrochemical device for converting chemical energy directly into electrical energy. The basic device consists of an electrolyte and two electrodes. The fuel is oxidized at one electrode and the oxidant is reduced at the other. The electrons involved are required, by suitable construction of the cell, to pass through an external circuit. In the cell considered, hydrogen is used as fuel and oxygen as the oxidizer. A practical fuel cell system contains subsystems for product (water) removal, heat removal, reactant conditioning and power plant control. Figure 10-2 shows the flow diagram of a fuel cell system.

The hydrogen-oxygen fuel cell operates at a temperature of about 400^{o}F and uses potassium hydroxide as an electrolyte. The electrodes are biporous nickel and nickel oxide. The primary loop comprises the H_2 side of each of the cells, the H_2 regenerator, a bypass

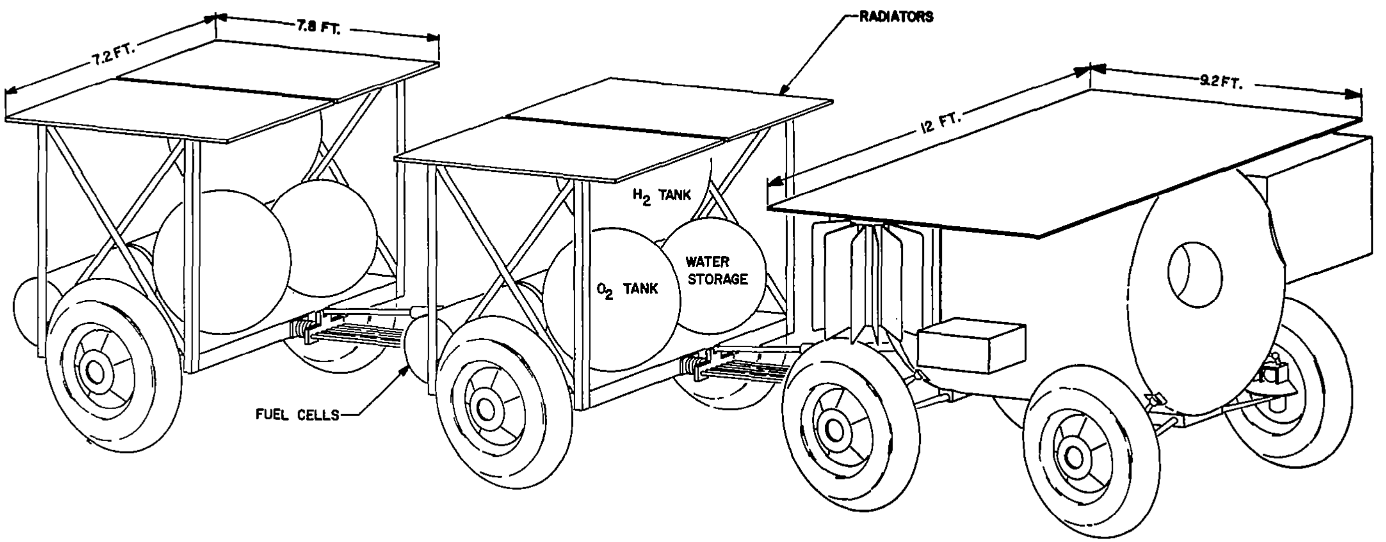

Figure 10-1. Fuel Cell Power System Equipment

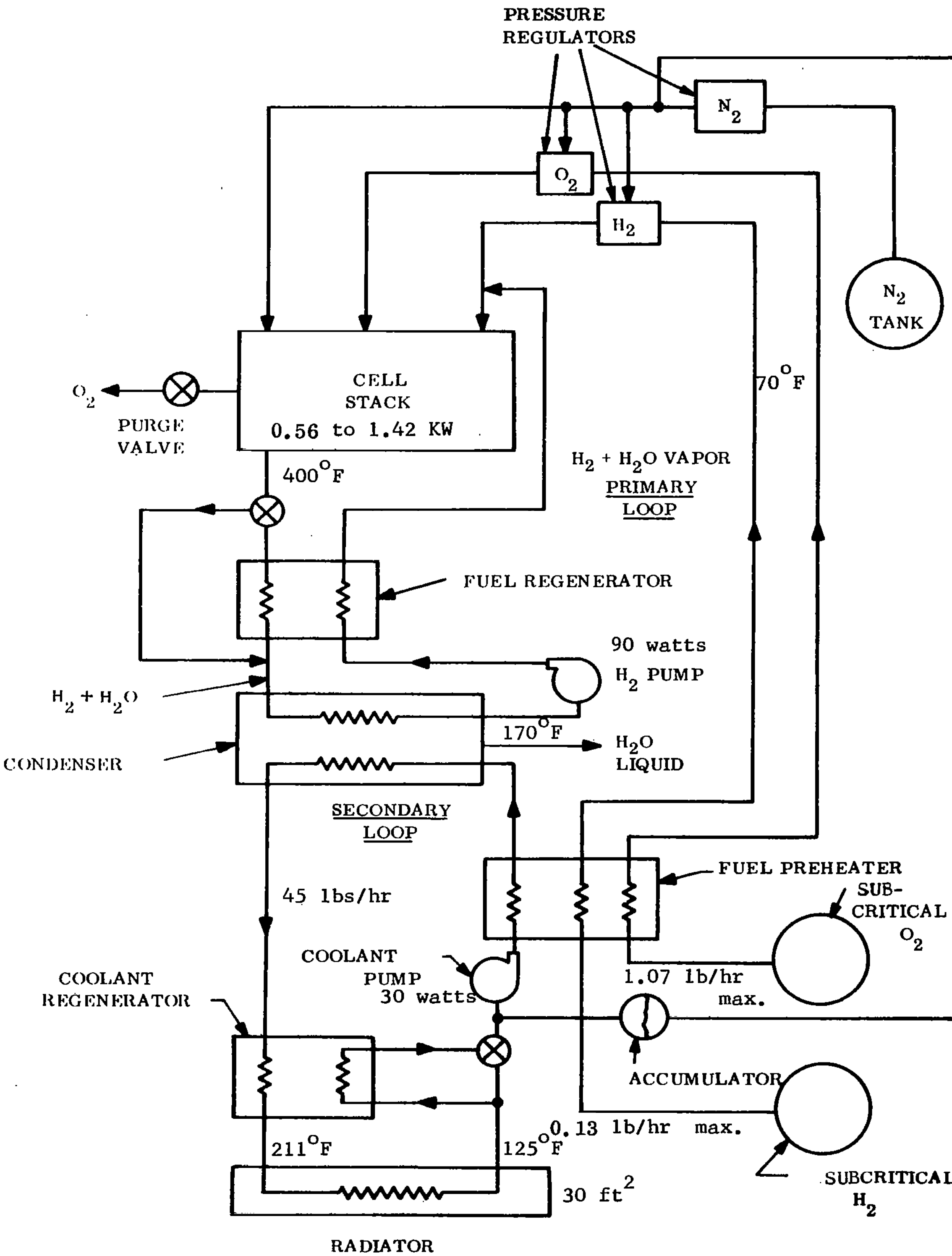

Figure 10-2. High Temperature Fuel Cell Flow Diagram

valve controlled by a temperature sensor, a condenser which exchanges heat with the secondary loop, and a H_2 pump. The temperature of fluid (steam + H_2) exhausted by the cell stack controls the setting of the bypass valve, and thus divides the exhaust gases. Part passes through the regenerator, exchanging heat with the fluid entering the stack, and is combined with the other part prior to entering the condenser. The condenser lowers the temperature of the steam-H_2O mix to liquify H_2O which is partially extracted from the condenser. Pressure losses due to condensation and fluid friction are restored by the H_2 pump. Additional H_2 is fed into the system at this point to replace consumed fuel and the H_2 rich mixture flows through the regenerator to the cell stack to complete the primary circuit. If the temperature of the exhaust is low, the regenerator bypass valve will allow more exhaust fluid to flow through the regenerator to increase the temperature of the incoming fuel, tending to offset the drop in cell temperature.

The secondary loop carries waste heat extracted at the condenser to the space radiator. The loop consists of a condenser (glycol side), the coolant regenerator and sensor controlled bypass valve, the coolant pump, an accumulator to maintain system pressure and absorb volumetric changes, the oxygen and hydrogen preheaters, and the space radiator.

10.2.2 MODULE PARAMETERS (FUEL CELL MODULE, PC3A)

The fuel cell module is composed of a 31 cell stack and associated plumbing. The fuel, fuel tanks and radiator are not included. It is 22.5 inches in diameter and 44 inches long. The module weight including plumbing is 204 lb. It requires 130 watts of parasitic power of which 90 watts are for hydrogen pump power, 30 watts are for glycol pump power and 10 watts are for sensors.

The output voltage is a function of the module operating power level (see Figure 10-3). The module may operate at any power level between 0.56 kw and 1.42 kw.

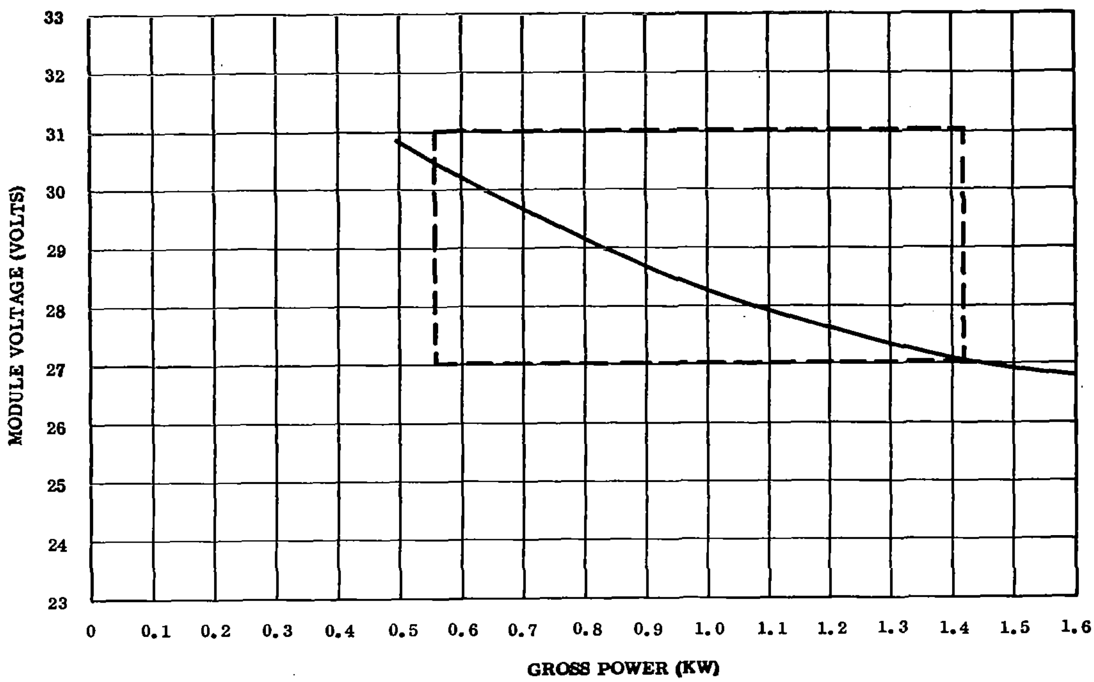

Figure 10-3. Fuel Cell Module Voltage Versus Power
(Apollo C/SM 31-Cell Module)

10.2.3 SYSTEM CONFIGURATION

Several system configurations are possible using fuel cells, batteries and standby fuel cells to furnish peak loads. Start up of standby fuel cells wastes energy, probably poses a reliability penalty and limits mission flexibility because of the start up time involved. The power profile shows that the periods of peak power are frequent and evenly distributed which would require an almost continuous cycling. For these reasons, the use of standby fuel cells under normal conditions is not recommended. Storage of fuel cells at elevated temperatures or operation at very low power levels so that power build up may be accomplished quickly and with little energy penalty might be feasible. Little data is available on this type of operation and it is not considered here. Reliability, parasitic load and threshold load situation will seriously limit this mode of operation.

Fuel cell module data depicts a normal operating range of between 0.56 kw to 1.42 kw. The operation of the roving vehicle modules will be confined to this range at all levels on the power profile curve. Batteries are required in order that this can be done economically. It should be pointed out that considerable battery energy is needed for fuel cell start up after initial storage and that it is desirable to have battery energy available for emergency situations. Thus, the power and weight penalty of using batteries is not truly a part of this system configuration. The chosen fuel cell-battery configuration is designed to rely as little as possible on battery energy because of heavy battery weight and the power penalty associated with battery charge-discharge (assumed efficiency of 0.70). Advantage is taken of the variable power capability of the modules within the indicated limits. Calculations, as well as Reference 10-1, indicate that the optimum number of modules from a weight standpoint is the lowest number that can supply peak loads. The parasitic load, threshold load and weight of the modules negate the efficiency gains obtained by using a larger number of modules than necessary to supply peak loads. The configuration chosen then is N modules sharing the load equally and operating between 0.56 kw and 1.42 kw per module with the minimum battery capacity necessary.

10.3 POWER REQUIREMENTS

The power profile curve for the open loop CO_2 condition (Figure 3-5) is used for the fuel cell system. The fuel cell system combines H_2 and O_2 to form water. A step in the closed loop CO_2 process decomposes water to obtain O_2. Taking into account the energy losses during each process, it is evident that it is more economical to furnish the life support O_2 directly, rather than supplying electrical power for that life support process.

The results of weight estimates of the fuel cell system show that it is considerably heavier than the weight that one trailer can carry. Therefore, two trailers will be needed, thereby increasing the locomotion power by 800 watts. The open-loop power profile (Figure 3-5) must be modified for this use by adding 0.8 kw to all the locomotion peaks. The RTG furnishes 100 watts of the required power. The net energy required for the mission can be obtained by integrating the modified power profile over the three typical days and multiplying by 15. The result is 2440 kw-hr, or an average net power of 2.26 kw.

The required power is to be half ac and half dc with conditioning efficiencies of 0.87 and 0.90 respectively. The factor for converting nonconditioned power to conditioned power is then $\frac{1}{2}\left[\frac{1}{0.87} + \frac{1}{0.90}\right] = 1.13$. This factor is assumed to be constant regardless of power level.

The parasitic power is 130 watts of conditioned power per module. If the same number of modules are in operation through the mission, the parasitic energy is 140 kw-hr per module.

A battery charge-discharge efficiency of 0.7 has been assumed. Therefore, there is a battery energy penalty of $\frac{E_B}{0.7} - E_B = 0.43\ E_B$ where E_B is the mission energy supplied by batteries.

The gross energy is:

$$(2440 + 140\ N)\ 1.13 + 0.43\ E_B \text{ kw-hr}$$

where N is the number of modules.

The power peaks have a duration of 18 hr out of 72 or 25% of the mission time. The maximum net power peak is 5.19 kw or (5.19 + 0.13 N) x 1.13 kw gross power. The minimum net power is 1.29 kw or $(1.29 + 0.13\ N) \times 1.13 + \frac{E_B}{810 \times 0.7}$ gross power, assuming an average charging power during non-peak periods. The charging time is 810 hours for the mission.

10.4 ENERGY BALANCE

10.4.1 DETERMINATION OF NUMBER OF MODULES

The number of modules determines the energy supplied by batteries since each module can supply a maximum of 1.42 kw and the battery must supply the power above the 1.42 kw level. The battery energy, average gross power and minimum power level can be determined using the data in Section 3.

Number of Modules	Battery Energy (E_B)	Average Gross Power	Minimum Power
2	736 kw-hr	3.15 kw	3.05 kw
3	393	3.15	2.60
4	61	3.16	2.15

Two modules can supply only 2.84 kw and thus cannot supply the mission power requirements since the average required power is 3.15 kw. In the four module case, the minimum power level is 2.15 kw or 0.54 kw per module. This is outside the stipulated operating range of the modules. Therefore, systems containing four or more modules must be ruled out. The only possible choice under the imposed constraints is a three module system. The three modules operate at maximum power (1.42 kw each) during peak loads and down to a minimum of 0.866 kw during minimum power load.

10.4.2 BATTERY SIZING

A three module system will produce a maximum of 3 x 1.42 = 4.26 kw. The peaks in load above this level must be supplied by battery power. Examination of the typical power profile shows that maximum battery power will be required in the time interval containing three power peaks starting with the experiment activity peak during day two. Consider the three peak series:

Discharge during experiments: (4.99 + 0.39) 1.13 - 4.26 x 3 hr = 5.46 kw-hr

Net charging during next 2 hours: 0.7 x 0.69 x 2 = 0.965 kw-hr

Discharge during next 3 hours: (3.79 + 0.80 + 0.39) 1.13 - 4.26 x 3 hr = 3.99 kw-hr

Net charging during next 3 hours: 0.7 x 0.69 = 3 = 1.45 kw-hr

Discharge during next 2 hours: (5.59-4.26) x 2 hr = 2.66 kw-hr

Net discharge during interval = 9.70 kw-hr

Ag-Zn batteries have a high energy density and sufficient cycling life (mission requirement of 120 cycles) to be used for this mission. A 50% depth of discharge is reasonable. This requires a battery capacity of 19.4 kw-hr (680 amp-hr, 28.5 v). Using an energy density of $\frac{50 \text{ w-hr}}{\text{lb}}$, the required weight is 390 lb.

The battery charging power is 0.69 kw. The charging voltage is 37.4 volts (19 cells x 1.97 volt/cell). This corresponds to a charging current of about 18 amps or a C/37 rate.

10.5 H_2, O_2 SUPPLY SIZING

The specific propellant consumption (SPC) is a function of cell voltage or the power level at which the module is operating. Thermodynamic considerations give

$$SPC = \frac{0.741}{V} \text{ lb/kw-hr} ,$$

where V is the volts per cell which can be obtained by dividing the module voltage, Figure 10-3 by 31, the number of cells per module. See Figure 10-4 for a plot of SPC versus power level. The energy-averaged SPC can be obtained using the power profile curve and found to be 0.824 lb/kw-hr. The net propellant weight is SPC x P_{ave} x Time = 0.824 lb/kw-hr x 3.15 kw x 1080 hr = 2800 lb. Allowing 3% for purging, the required weight is 2880 lb. Approximately 8/9 ths of this oxygen (2560 lb). The remaining is hydrogen (320 lb).

This amount of propellant must be available for use after storage on the moon. An analysis of tankage weight and boil off for subcritical storage on the moon was made in Reference 9-2. The thermal protection system consists of an inner wall, superinsulation, a vapor-cooled shield, more superinsulation and an outer wall. Load factors (the ratio of liftoff propellant and tankage weight to usable-propellant weight) were estimated from data in this reference. Storage is for a maximum of 720 hours. Four tanks are used, two containing half the H_2 each and two containing half the O_2 each. The load factor for H_2 is 2.5 and for O_2 is 1.15. These give a propellant and tankage weight of 3740 lb.

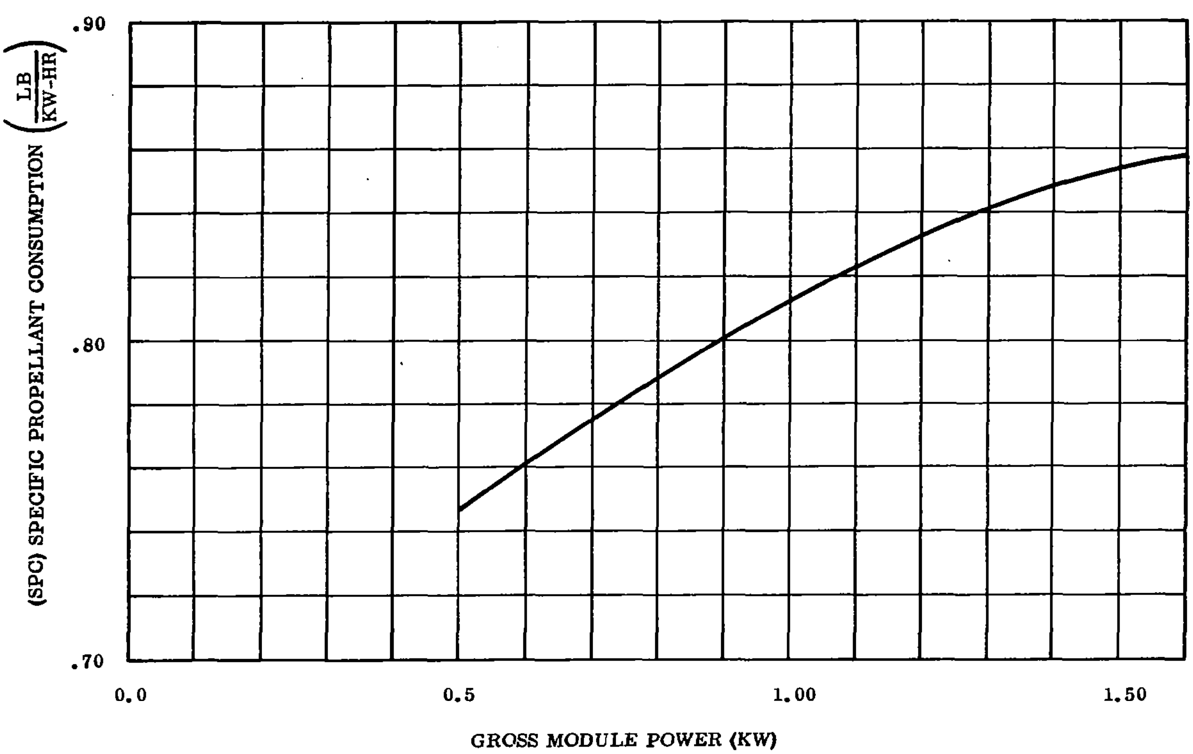

Figure 10-4. Variation of Specific Propellant Consumption with Module Output Power

10.6 HEAT REJECTION SYSTEM

10.6.1 WASTE HEAT AT MAXIMUM POWER

Maximum waste heat generation will occur during the load peaks when the modules are operating at the 1.42 kw level. The SPC at this level is 0.848 lb/kw-hr. Thermodynamic considerations give the thermal efficiency (η) of the H_2 - O_2 reaction as approximately $\frac{V}{1.48}$, where V is the cell voltage, 0.874 volts. The exact efficiency depends on operating temperature and propellant inlet conditions, and any loss of neutral molecules across the electrolyte. For this case, $\eta = 0.59$ at peak power.

The heat rejection per module = $1.42\ (\frac{1-\eta}{\eta}) = 1$ kw or 3400 btu/hr. The propellant flow rate is $0.848 \times 1.42 = 1.2$ lb/hr. Taking the propellant inlet temperature as $75^{o}F$, it is estimated that an average of about 300 btu/hr (see Figure 10-5) may be used to preheat the incoming propellant. Therefore, 3100 btu/hr per module must be rejected to space during the power peaks.

10.6.2 RADIATOR LOOP DESIGN

Figure 10-6 shows the fuel cell condenser characteristics on which the radiator design was based. The two basic radiator coatings were considered: (1) OSR, Solar Absorptance (α_s) = 0.05, emittance (ϵ) = 0.84; and (2), Solar Absorptance (α_s) = 0.2, (ϵ) = 0.90. In the first mentioned coating the radiator analysis does not per se completely account for the presence of this coating. In particular, the density, thermal conductivity and strength characteristics are not considered in determining a weight optimized radiator.

Once having determined a weight optimized radiator, the additional weight of the coating is added to the weight of the optimized radiator. The justification for this procedure is based on the following considerations and assumptions.

a. The thermal conductivity of the coating is sufficiently low that its presence does not alter the fin efficiency significantly.

b. The strength characteristics of the coating do not significantly improve the resistance to micro-meteoroid penetration.

Reference 10-5 discusses the effect of a coating on the fin efficiency of a radiator. The data in this reference substantiates the first assumption. Table 10-1 presents a summary of the radiator parameters examined. The effective sink temperature was taken from Figure 3-3.

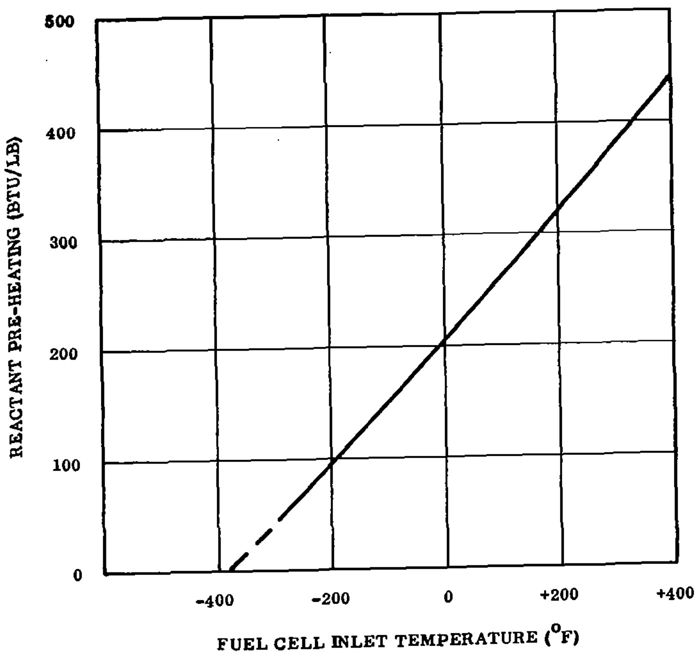

Figure 10-5. Fuel Cell Inlet Temperature

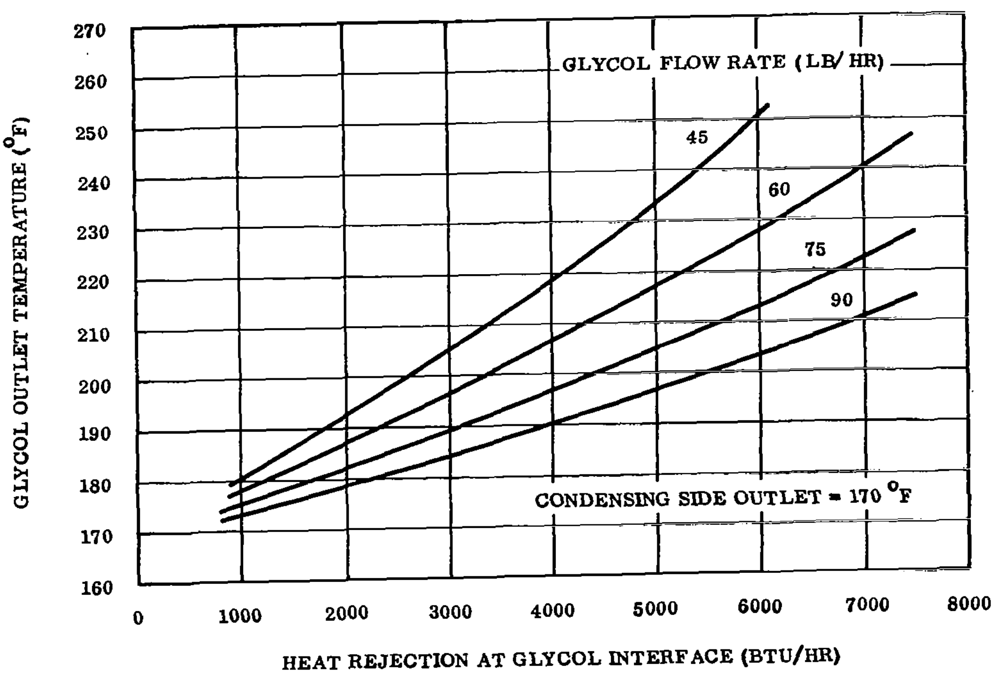

Figure 10-6. Heat Rejection at Glycol Interface - Btu/hr

TABLE 10-1. FUEL CELL RADIATOR DESIGN PARAMETERS

Case No.	(Btus/hr)	Radiator Inlet Temperature T_{Ri} (°F)	Radiator Outlet Temperature T_{Ro} (°F)	Area (ft^2)	Weight (lb)	Weight Flow Rate W (lb/hr)	Header (ft)	Tube Length (ft)
1	3100	211	125	20.0	10.2	45	1.50	13.32
2	2500	211	142	15.3	7.7	45	1.25	12.21
3	2000	211	155	11.7	6.1	45	1.25	9.34
4	2000	193	159	12.2	6.3	75	1.25	9.72
5	2500	193	151	15.6	7.8	75	1.25	12.44
6	3100	193	141	19.9	10.2	75	1.50	13.22
8	2500	188	149	15.9	8.0	90	1.25	12.72
9	2000	188	157	12.4	6.5	90	1.25	9.94

Coolant: Propylene Glycol (62.5% by weight) - water
Material: Aluminum
Coating: α_s = 0.05, ϵ = 0.84
Configuration: Horizontal folded

The use of the OSR optical coating system may or may not in all instances result in a lower weight radiator. Table 10-2 shows the radiator design characteristics for a coating with the assumed properties of $\alpha_s = 0.2$, $\epsilon = 0.9$. The radiator is assumed to be a horizontal radiator.

TABLE 10-2. FUEL CELL RADIATOR DESIGN PARAMETERS

Case No.	Q_R (Btu/hr)	T_{Ri} (°F)	T_{Ro} (°F)	Area (ft^2)	Weight (lb)	lbW/hr (ft)	Header (ft)	Tube Length (ft)
1	3100	211	125	27.5	10.8	45	1.75	15.7
2	2500	211	142	19.6	7.4	45	1.50	13.0
3	2000	211	155	16.1	5.7	45	1.50	10.6
4	2000	193	159	16.3	6.1	75	1.25	13.0
5	2500	193	151	20.2	7.6	75	1.25	16.2
6	3100	193	141	26.5	10.1	75	1.50	19.4
7	3100	188	140	30.0	10.5	90	1.25	21.7
8	2500	188	149	20.9	7.9	90	1.25	16.7
9	2000	188	157	16.9	6.3	90	1.25	13.5

Coolant: Propylene Glycol (62.5% by weight) - water
Material: Aluminum
Coating: α_s = 0.2, ϵ = 0.9
Configuration: Horizontal folded

From the above table it is evident that at the higher power level the use of a more conventional coating is slightly heavier than the Lockheed OSR coating, however, at the lower power levels, the more conventional coating system is slightly lighter.

10.7 DISCUSSION

The battery storage temperature range is $20^{o}F$ to $+ 100^{o}F$. Temperature control in this range can best be obtained by using a horizontal radiator mounted on the cabin. A small heat exchange loop must be provided for heat transfer from the batteries to the radiator. The extra load during storage will be small if the batteries are insulated from the lunar environment.

The minimum temperature that fuel cells should reach is minus $20^{o}F$. Provision must be made for heating the cell modules during a large portion of the lunar night while in the dormant and other inactive periods. This requires that the cell modules be well insulated and provided with electrical heating. The small amount of electric power is supplied from RTG.

It is anticipated that no upper temperature limit control will be needed by the fuel cells since their normal operating temperature is higher than any temperature exposure during storage.

10.7.1 REDUNDANCY CONSIDERATIONS

The demonstrated probability of a single fuel cell module operating for 45 days is not high at the present time. In order to get meaningful redundancy in this case, standby units are required that can be started up as needed. Allowing for increased module reliability in the future, one standby module will be assumed to be sufficient. There are, therefore, three operating units and one standby unit. Each unit is independent (separate radiators, pumps, etc.) with the exception of propellant tanks.

The fuel cell power system will produce a total of 2800 lb of water. Approximately 750 lb of water are required for thermal control of suits. The remaining 2050 lb of water can be used for cabin thermal control during peak heat loads. However, base roving vehicle complex studies indicate that it may be best to return the water to a base for processing back to hydrogen and oxygen (see Reference 10-2).

Since the power system weight is about 5000 lb, two trailers are needed. Each trailer will weigh about 3000 lb loaded, including the structure, locomotion equipment, etc. Two fuel cell module systems, one hydrogen tank and one oxygen tank will be on each trailer.

Tables 10-3 and 10-4 present a summary of fuel cell power system characteristics and of fuel cell system weights.

TABLE 10-3. FUEL CELL POWER CHARACTERISTICS

Mission gross energy		3400 kw-hr
Mission net energy	2440 kw-hr	
Fuel cell parasitic loss	420	
Battery charge-discharge loss	175	
Power conditioning loss		
50% dc at efficiency of 0.92	140	
50% ac at efficiency of 0.087	225	
	3400 kw-hr	
Average gross power		3.15 kw
Number of modules (includes one standby)		4
Battery capacity		19.4 kw-hr
Minimum power output (3 modules)		2.60 kw
Maximum power output (3 modules)		4.26 kw
Module voltage at minimum power		28.8 volts
Module voltage at maximum power		27.1 volts
Average specific propellant consumption		0.824 lb/kw-hr

TABLE 10-4. FUEL CELL SYSTEM WEIGHTS

Module power plant weight, 4 at 223 lb			732 lb
Module plumbing, mounting, etc. 4 at 21 lb			84
Battery weight			390
Net propellant		2880 lb	
Hydrogen	320 lb		
Oxygen	2560		
Tankage		830	
Hydrogen	480		
Oxygen	380		
Total propellant and tankage			3740
Radiator weight,4 at 11 lb			44
		Total	4990
Water production	2690 lb		
Weight penalty for extra trailer	600 lb		
H_2 Tank = 5 ft diameter			
O_2 Tank = 6 ft diameter			

10.8 REFERENCES

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10-4. "Apollo Logistics Support System," The Boeing Company, D2-36072-3, April 1965.

10-5. Plamodon, J. A. "Thermal Efficiency of Coated Fins" ASME Transactions, Journal of Heat Transfer, November 1962, P. 279-281.

SECTION 11
RTG POWER SUPPLY

11.1 INTRODUCTION

During the dormant operational phase of a lunar roving vehicle, power for the necessary vehicle standby systems is to be provided by a radioisotope thermoelectric generator (RTG). A total of 100 watts of dc power will be required full time during lunar day or night. The method used in analysis and configuration selection was to apply SNAP-27 criteria wherever possible since they represent:

a. State of the art design
b. Lunar environment considerations

The 100-watt power output is twice the SNAP-27 requirement allowing either simple scaling or the use of two SNAP-27 units. The RTG is to be mounted on one side near the rear of the vehicle personnel cabin. This section is a condensation of the analysis of a 100-watt RTG documented in the supplement to this report.

11.2 DESCRIPTION

Two possible RTG designs have been identified: a 100-watt D configuration unit or two 50-watt modified SNAP-27 units. Figure 11-1 shows the dimensions for the two configuration possibilities. A D configuration installation on the cabin is shown in Figure 11-2. Note that a reflector is provided to protect the cabin from RTG thermal radiation. Either of the RTG concepts would weigh on the order of 85 to 95 pounds, including supports, reflector and insulation (but excluding shielding).

Each system is sized to deliver 112 watts of unconditioned dc power to the vehicle. This permits incorporating a converter-regulator with an efficiency of 90%, while meeting the vehicle power requirement. By proper design, this regulator can also serve as the necessary generator protective circuit, preventing partial load operation of the RTG.

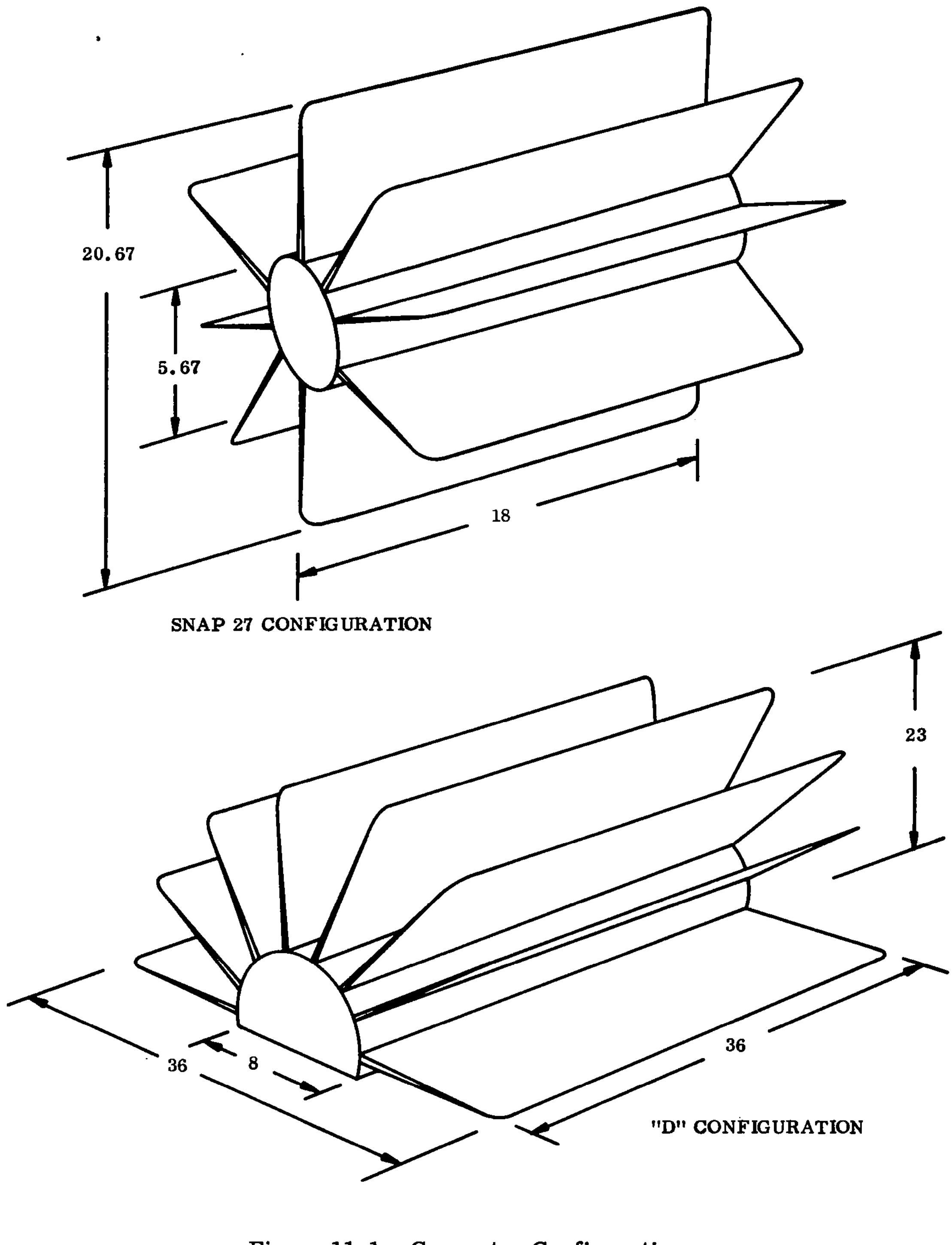


Figure 11-1. Generator Configurations

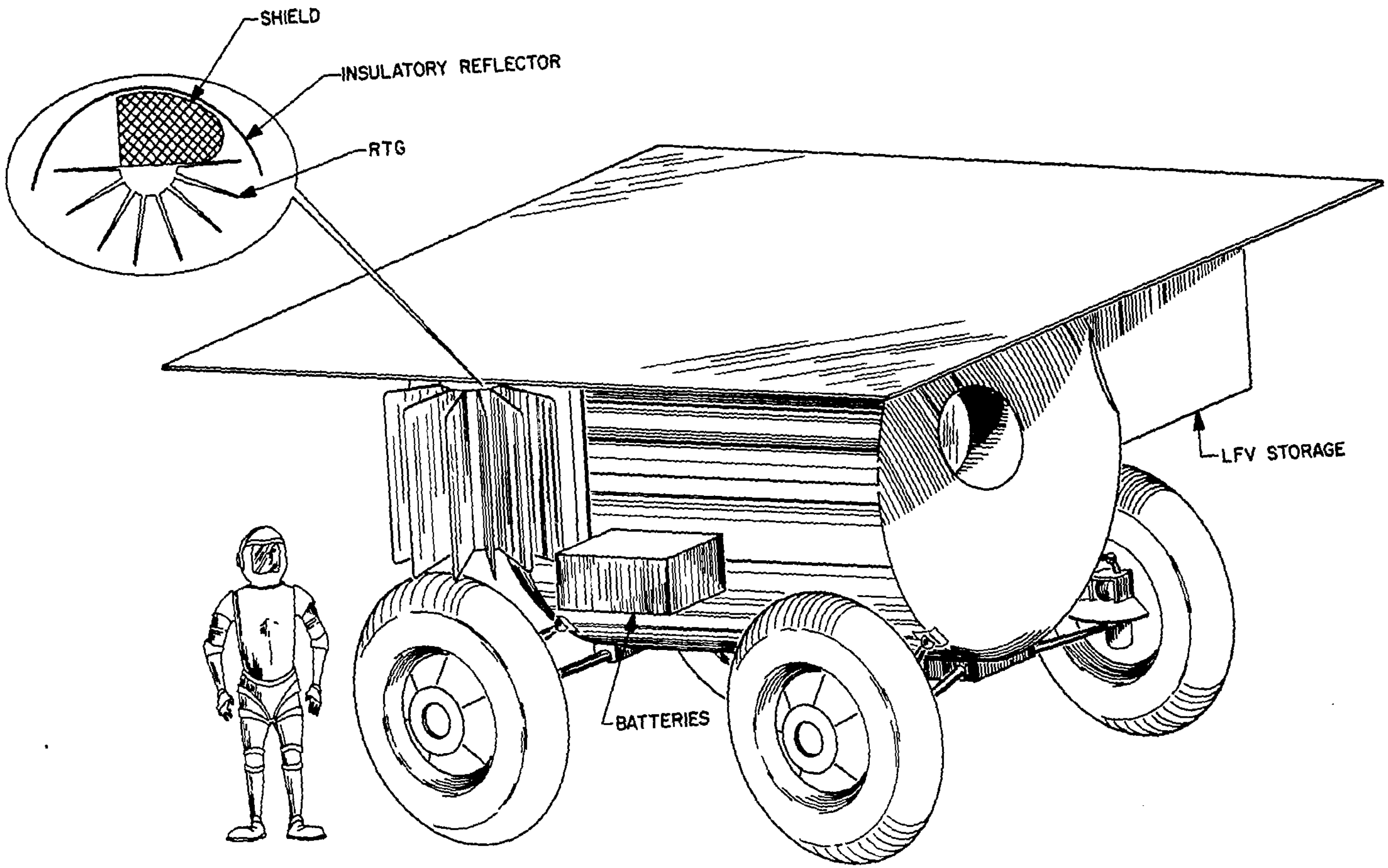


Figure 11-2. 100 Watt RTG Mounted on Lunar Roving Vehicle

Radiation dose rates for the unshielded systems mounted on one side of the vehicle are approximately 134 mrem/hour at one meter. It is clear that radiation shielding must be provided so that the crew dose does not exceed 12-1/2 rem for the 45-day mission. Since the amount of shielding is dependent on the type of power system (isotope or chemical) used, analyses were conducted to determine the proper RTG/dynamic power system shielding ratio. The results are presented in Appendix C. For a chemical power system type roving vehicle, the RTG shield weight is 260 pounds.

The generator and fuel capsule is designed to completely contain the isotope following any plausible abort situation. The use of beryllium for the generator structure provides the necessary heat shield to protect the fuel capsule following an abort which results in re-entry into the earth's atmosphere.

APPENDIX A

OVERALL RADIATOR HEAT BALANCE

$$Q_R = \epsilon_R A_R \eta_o \sigma T_R^4 - \eta_o \alpha_{SR} A_P Si - \eta_o \epsilon_R \epsilon_M F_{RM} A_R \sigma T_M^4 - \eta_o \alpha_{SR} a_m Si F_{RM} A_R$$

$$= \epsilon_R A_R \eta_o \sigma \left(T_R^4 - T_S^4\right)$$

where

a_m	=	Lunar albedo
A_R	=	Radiator heat transfer area
A_P	=	Radiator projected area to plane perpendicular to sun's rays
F_{RM}	=	Configuration factor of lunar surface relative to radiator
Q_R	=	Radiator heat load
Si	=	Incident solar flux
T_R	=	Radiator effective root temperature
T_S	=	Radiator effective sink temperature
α_{SR}	=	Radiator solar absorptance
ϵ_R, ϵ_M	=	Radiator and moon surface emittance response
σ	=	Boltzmann's constant
η_o	=	Overall radiator efficiency

so that

$$\sigma T_S^4 = \frac{1}{\epsilon_R} \left[\alpha_{SR} Si + \epsilon_R \epsilon_M F_{RM} \sigma T_M^4 \alpha_{SR} a_m Si F_{RM}\right]$$

The incident solar radiation is taken as

$$Si = S \cos \theta$$

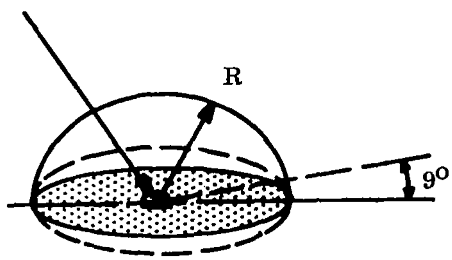
where S = Solar constant = 442 $btu/hr\text{-}ft^2$

θ = Angle of incidence

The effective sink temperature is shown for both a horizontal planar radiator and a vertical, cylindrical radiator. The configuration factors were determined by using Nusselt's method.

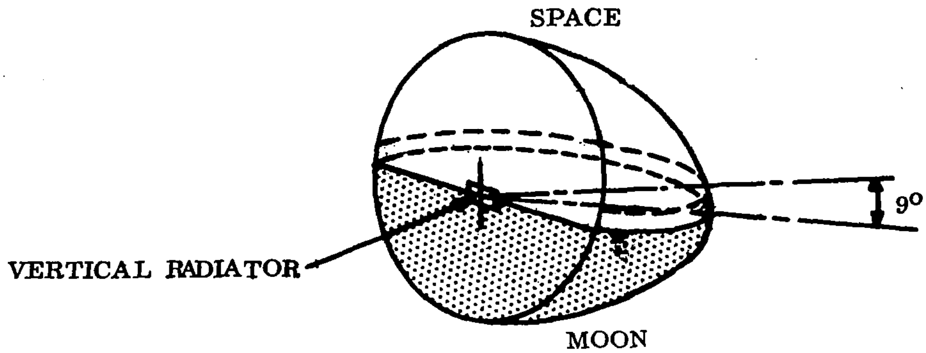
For the horizontal radiator the configuration factor can be determined simply by considering the following construction.

HORIZONTAL RADIATOR



$$F_{RM} = \frac{\pi[R^2 - R^2 \cos^2(9^\circ)]}{\pi R^2} = 1 - \cos^2(9^\circ) = 0.025$$

A similar expression can be derived for a vertical radiator.



$$F_{RM} = \frac{\pi R^2 - \frac{\pi R^2\, 162}{360} + \frac{R^2}{2}\ \mathrm{SIN}\ 162}{\pi R^2} = 0.60$$

APPENDIX B

LSS DORMANT PERIOD CONTROL

B.1 TRANSIT PERIOD

Since the shroud on the vehicle during the transit period will not permit influx of heat into the vehicle or dissipation of heat from the vehicle, then the RTG electrical heat will be the only cabin heat input. The cabin weighs 4000 lb and has an average specific heat of 0.2 btu/lb/^{o}F, then approximately 800 btu are required to raise the cabin temperature 1^{o}F. The electrical heat load is (100 watts) 340 btu/hr, thus the cabin temperature increases 1^{o}F every 2.36 hours or a total of 30.5 ^{o}F during the transit period.

Prelaunch cooling and thermal jackets, if necessary, are assumed to keep the vehicle at 75^{o}F. Thus the final temperature at the end of the 3-day transit period is 105^{o} or 5^{o}F in excess of the limit. Some form of cooling may then be provided during the transit period. An ideal cooler for this application is an expendable type of refrigerant which utilizes the evaporation of water into spatial vacuum to provide cooling. Assuming that the evaporative cooler is activated when the cabin temperature is 100^{o}F (approximately 65 hours after launch) approximately 4 pounds of water will be required.

The cooler is a gas to flash the boiler heat exchanger. The expendable refrigerant (water) is sprayed into the low pressure boiler where sufficient heat is transferred to the low-steam-quality vapor to produce superheated steam. The spray is essentially a throttling process (constant enthalpy) and the heat transferred from the cabin-environment gas produces a vapor enthalpy change (h_{fg}) of approximately 1000 btu per pound of water evaporated.

The container utilized to store the water is a pressurized bottle with a bladder to expel the water in the zero gravity environment. Either the vehicle gas supply or a separate Freon C318 source is utilized to pressurize the water. The container weighs approximately 1 pound, and the filled container weighs 5 pounds. Boiling rate of the heat exchanger is controlled by a modulating expansion valve and an absolute pressure regulator. The expansion valve modulates (varies the spray nozzle opening) in accordance with the superheat steam temperature. As the superheat temperature increases the orifice size increases and vice versa to shut off all water flow. The absolute pressure regulator at the steam outlet provides pressure control within a set limit to assure that the heat exchanger does not freeze. A freeze could otherwise occur if large amounts of water are injected into the heat exchanger during transient heat loads.

This same cooler may also be utilized as an emergency cooler when the crew is occupying the vehicle.

B. 2 LUNAR DAY

B. 2. 1 FLOW CALCULATIONS

The total heat load in the vehicle during the lunar day period is the electrical load from the power system plus the heat leakage thru the cabin walls. This amounts to 340 + 69 = 409 btu/hr.

a. Heat Removed

$$(Q)\ \text{total} = Wg\ Cp\ \Delta\ t\ (\Delta\ t = 240^{o}F,\ Cp = 2.22\ \text{btu/lb}^{o}F)$$

$$(Q)\ \text{equipment} + Q\ \text{wall} = Wg \times 0.22 \times 40^{o}F$$

$$(Q)\ \text{total} = 8.8\ Wg\ \text{or}\ Wg = 0.113\ Q\ \text{total} \qquad (1)$$

The density of the cabin oxygen is approximately 0.02 lb/ft^3 at 60^{o}F and 180 mm Hg pressure. The flow rate (cfh) thru the heat exchanger is then: $cfh = 5.65\ ^{Q}\ \text{total}$ (2)

The total heat load is dissipated to a liquid heat transport medium via a gas-to-liquid heat exchanger and then is rejected from the vehicle by a radiator.

Blower electrical power requirement (B_e watts) is a function of the gas flow rate (Wg lb/hr), the heat exchanger system pressure drop (ΔP_g feet of gas head) and blower efficiency (E_b):

$$B_e = \frac{Wg \times \Delta P_g}{2650 \times E_b} = \frac{0.113 \, Q_{total} \times \Delta P_g}{2650 \times E_b} \tag{3}$$

The liquid coolant pump electrical power requirements is determined in a similar manner:

$$\frac{P_e = W_f \times \Delta P_f}{2650 \times E_p} \tag{4}$$

where ΔP_f = feet of liquid head

A tentative selection of the coolant liquid is an aqueous solution of 40%-by-weight propylene glycol. This liquid is selected on the basis of the lunar shelter study and the minimal oral toxicity hazard of the liquid.

The liquid properties at 50°F are:

Specific gravity	= 1.04
Viscosity	= 6.0 centipoises
Specific heat	= 0.89 btu/lb °F
Thermal conductivity	= 0.24 btu/hr ft^2 F°

The temperature increase of the coolant in the heat exchanger is assumed to be 30°F. Thus the liquid enters at 40°F and exits at 70°F and the gas enters at 90°F and exits at 50°F. The collant flow rate (W_f,lb/hr) is then calculated from:

$$Q \ total = W_f \times 0.89 \text{ btu/lb } ^{o}F \times 30^{o}F$$

$$Q \text{ total} = 26.7 \text{ W} \quad \text{or} \quad W = 0.0375 \text{ Q total} \tag{5}$$

Combining equations (4) and (5)

$$P_e = \frac{0.0375 \text{ Q total} \times \Delta P_f}{2650 \times E_p} \tag{6}$$

The efficiency of the blower E_b will be low (assumed $E_b = 2.5\%$) due to the impeller slippage in the low density gas and the small size of the unit which cause the inherent frictions to become a major restriction. The liquid pump will also utilize a small motor but slippage will not be a problem; consequently an efficiency, E_p, of 5% is assumed.

Figure B-1 illustrates the gas flow rate and blower power for various heat loads and Figure B-2 illustrates coolant liquid flow rate and pump power for various heat loads as derived by equations (1) (3) and (6). Note several pressure drops are given for both the blower and pump systems to aid in the design.

B. 2. 2 LUNAR DAY DORMANT CONTROL DESCRIPTION

The thermal control for the lunar day dormant period will use much of the thermal control system utilized during crew occupancy. A block diagram of the system is shown in Figure 4-2. The dormant period blower circulates the gas thru the existing ducts and heat exchanger. Consider the gas pressure drop as similar to that of the lunar shelter (Reference 4-1, 0.6 inches Wg pressure drop for 330 cfm at 7 psia) then the pressure drop thru the system during the dormant period will be an inverse function of the densities and the velocities squared. Thus for 37.6 cfm (45.2 lb/hr, sufficient for a heat load of approximately 400 btu/hr) the system pressure drop will be

$$\frac{0.02 \text{ lb/ft}^3}{0.04 \text{ lb/ft}^3} \times \left[\frac{37.6 \text{ cfm}}{330 \text{ cfm}}\right]^2 \times 0.6 \text{ inch Wg} = 0.0039 \text{ inch Wg}$$

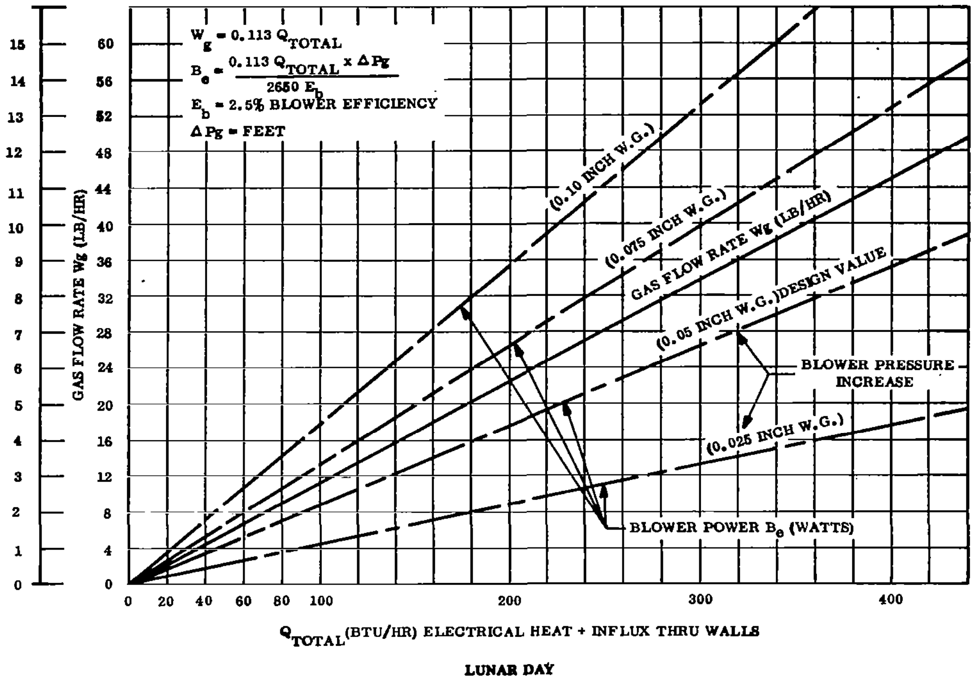


Figure B-1. Gas Flow - Thermal Control - Dormant Period

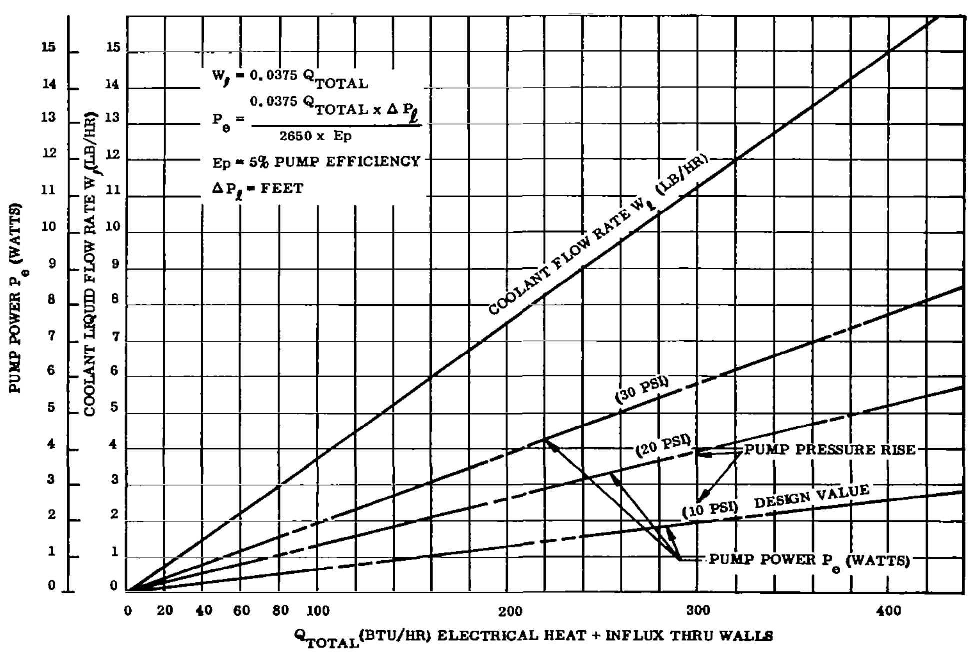


Figure B-2. Coolant Flow - Thermal Control - Dormant Period

This pressure drop is negligible for design consideration thus a pressure drop of 0.05 inch Wg friction loss is assumed for all flow rates shown in Figure B-1.

The dormant period pump circulates the liquid coolant through the heat exchanger and the radiator. The same condition exists (calculated pressure negligible) in the liquid circuit as in the gas circuit for determining pressure drop. Thus a coolant pressure drop of 10 psi is assumed for all flow conditions shown in Figure B-2.

Based on the above calculations and the assumption that the total vehicle electrical load is 100 watts (340 btu/hr) and that the vehicle maximum solar heat influx is the same as the lunar shelter (68 btu/hr), then the total maximum heat load is 408 btu/hr. From Figure B-1 the blower will circulate:

$$\frac{46\text{ lb}}{\text{hr}} \times \frac{\text{ft}^3}{0.02\text{ lb}} \times \frac{1\text{ hr}}{60\text{ min}} = 38.3\text{ cfm gas}$$

at an expenditure of 9 watts of electrical power. From Figure B-2 the pump will circulate:

$\dfrac{15.2\text{ lb}}{\text{hr}} \times \dfrac{\text{gal}}{8.7\text{ lb}} \times \dfrac{1\text{ hr}}{60\text{ min}} = 0.029$ GPM coolant at an expenditure of 2.7 watts of electrical power.

B.2.3 LUNAR NIGHT DORMANT CONTROL

The total heat load in the vehicle during the lunar night period is the electrical heat load from the power system minus the heat lost through the cabin walls. Operation of the radiator heat rejection circuit is not anticipated except as discussed. If the loss thru the walls exceeds the electrical load (maximum 100 watts or 340 btu/hr) then an additional source of thermal energy may be required to maintain the cabin temperature within acceptable limits. Such heat sources as the thermal energy from the isotope power system may be required. The lunar shelter study established a maximum heat leakage of 385 btu/hr which exceeds the electrical heat by 45 btu/hr or 40.3^{O}F during a 15-day lunar night. This temperature drop of 0.112^{O} F/hr may be acceptable if the cabin initial temperature is high.

More flexibility may be added to the system by conducting some heat from the RTG source into the cabin and by regulating the temperature through modulation of the cooling circuit. This is easily accomplished since the power source is located on the exterior of the right rear cabin wall. A heat leakage (via a heat pipe) of 60 btu/hr into the cabin from the isotope power system will permit cabin thermal control for heat losses up to 400 btu/hr during the lunar night period. This heat leakage could also occur during the lunar day but would increase the blower and pump power by less than 2 watts.

APPENDIX C

ISOTOPE SOURCES

C.1 INTRODUCTION

This appendix presents the conceptual design for the module containing the isotope heat source, the heat exchanger for extracting the heat, and the associated biological shielding for the nonintegrated Brayton cycle and Rankine cycle power conversion systems. The sources and heat exchangers are optimized to meet specified cycle conditions of temperature, flow rate, and pressure. This is a condensation of the analysis and design of these isotope sources which appears in the report Supplement.

C.2 SHIELDING

C.2.1 INTEGRATING FACTORS

Maximum benefit of distance for shielding is obtained by locating the heat source on the rear of the trailer module as shown in Figure C-1. This location has other advantages which include:

a. The heat exchanger is accessible for loading of the isotope module when the vehicle is on the launch pad.

b. Advantage may be taken of the intervening equipment for shielding of the vehicle cabin.

c. The heat source module can be ejected from the heat exchanger so that it can radiate its heat to space as a "last-ditch" method of cooling. Ejection of the module is not impaired by other equipment or structural members.

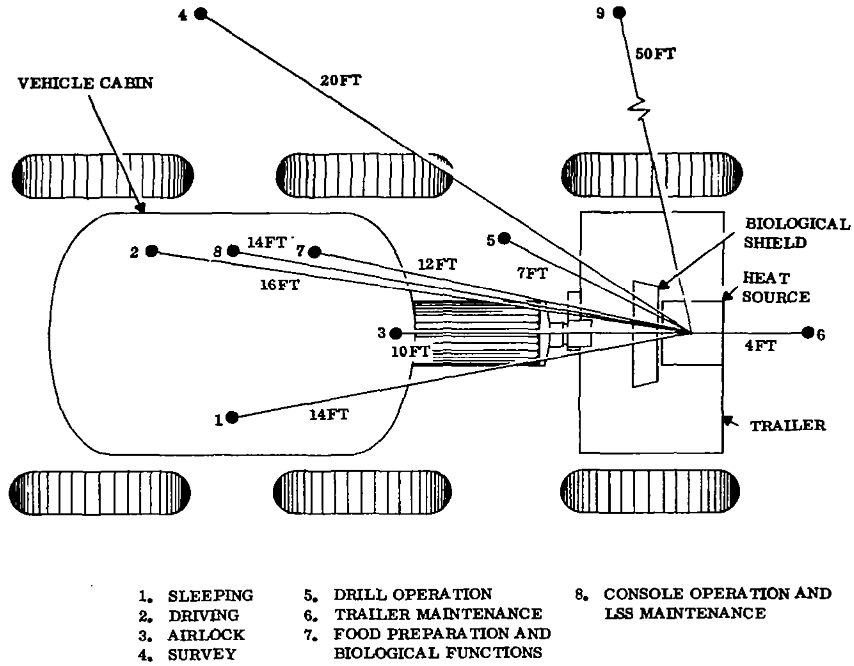


Figure C-1. Factors Considered in Shield Design

The shielding is designed to limit the total radiation dose to be received by each astronaut for the entire mission from earth launch to earth return to 25 rem. The dose partition is:

Roving vehicle mission profile	12.5 rem
Cosmic and solar radiation transit (radiation belts)	12.5 rem
Total	25 rem

C.2.2 SHIELD DESIGN

As indicated, the shield design for each heat source is based on a total dose of 12.5 rem for the roving vehicle mission. Since the heat source will be located relatively close to the crews quarters, even at the location on the rear of the trailer, the fraction of time spent by each crew member at the various duty stations both within and outside the vehicle cabin can have significant affect on the shielding required. This factor is taken into account by estimates of the time spent at each duty station. These estimates, given in Table C-1, are based on the average time spent per 24-hour period at each position. Corresponding distances which are considered average for each position are also listed in Table C-1 and are shown in Figure C-1. The entries in Table C-1 in the column headed Shielding represent the contribution to shielding afforded by the cabin walls and/or intervening equipment for the purpose of estimating the gamma contribution only. In addition, the heat source itself is considered to provide the equivalent of one centimeter of steel. The quantity, f_t, shown in the table is the fraction of the total mission time spent at each duty station.

TABLE C-1. CREW POSITIONS FOR SHIELDING ANALYSIS

Location	Function	Distance	Shielding	Time	f_t
1	Sleeping	14 ft	1 in. Al	8 hr	0.333
2	Driving	16 ft	1 in. Al	5 hr	0.2083
3	Airlock	10 ft	1 in. Al	1/3 hr	0.0139
4	Survey	20 ft	none	1 hr	0.0417
5	Drill operation	7 ft	0.5 in. Al	1 hr	0.0417
6	Trailer maintenance	4 ft	none	1/3 hr	0.0139
7	Food Prep.	12 ft	1 in. Al	2 2/3 hr	0.1111
8	LSS maintenance	14 ft	1 in. Al	2 2/3 hr	0.1111
9	Exploration	50 ft	none	3 hr	0.1250
				24 hr	1.0000

Configurations considered for the heat source include the 4π shield which will effectively surround the source and the slab or shadow shield which will protect the cabin only. The shield for the RTG is a slab. While the 4π shield would be thinner than the shadow shield, it would probably incur a much larger weight penalty and would impair passive heat rejection for emergency cooling.

Analysis indicated that the shadow shield could not be used with principal outside duty stations being left unprotected. Thus a scheme was chosen in which most of the duty stations would be protected by a shadow shield (which in effect simulates a 4π shield).

A reduction in the RTG (radioisotope thermoelectric generator) contribution to the total dose is effected by moving the unit from the forward right side of the cabin to the right rear. This also greatly reduces the weight of the RTG shield because of the angle subtended by the cabin as well as reducing weight of the shield for the heat source.

With duty stations and associated time allocations as given in Table C-1 for the 45-day mission of the lunar roving vehicle, weights of the shields for the nonintegrated and integrated systems are given in Table C-2.

TABLE C-2. ESTIMATED SHIELDING FOR NONINTEGRATED SYSTEMS
(12.5 Rem Dose in 45 Days)

	Shield Weight-lb	
	RTG	Heat Source
15 KWt Brayton System	330	275
31 KWt Rankine System	350	415

The weight of the RTG shield for those power systems other than the isotope source systems is 260 pounds.

C.3 HEAT SOURCE CONCEPT

Selection of concepts for the heat source configurations is based on a number of factors including the design guidelines, safety considerations, and the mission restraints which follow.

a. The fuel package shall be designed to survive and maintain containment of the fuel for any credible accident either on the launch pad, during launch, in transit, and during landing on the moon and mission utilization.

b. The fuel package shall be designed to contain the fuel for the length of time required for nuclear safety considerations in the event that it is jettisoned at sea and cannot be recovered.

c. The fuel package shall be designed for separation, atmospheric re-entry and recovery following an aborted mission in which it returns to earth.

d. The fuel module shall be loaded into its heat exchanger aboard the spacecraft prior to launch with no provision for refueling during the entire mission or for removal from the heat exchanger except for "last-ditch" emergency cooling.

e. The most reactive configuration of fuel, regardless of the situation, must not be critical.

Comprehensive preliminary design and safety studies (Reference C-1) of flexible heat source subsystems performed by General Electric considered various isotopes and their compounds in conjunction with thermoelectric, Rankine, and Brayton power conversion subsystems that would lead to safe and reliable power supplies for manned and unmanned missions requiring 1 to 10 KWe. The studies developed and analyzed overall heat source subsystems for Pu-238, Po-210, and Cm-244 fuels in conjunction with power system/heat exchanger combinations.

Based on the results of the previous large heat source studies, the torus configuration is selected as the design concept which most effectively meets the guidelines and constraints of the heat sources for use with the 45-day lunar rover. The design of the heat source and heat exchanger module is shown in Figures C-2 and C-3 for the Brayton cycle system. The heat source in this configuration provides a single package for handling and for interfacing with a cylindrical heat exchanger matrix as shown in Figure C-4. Generally, the heat source

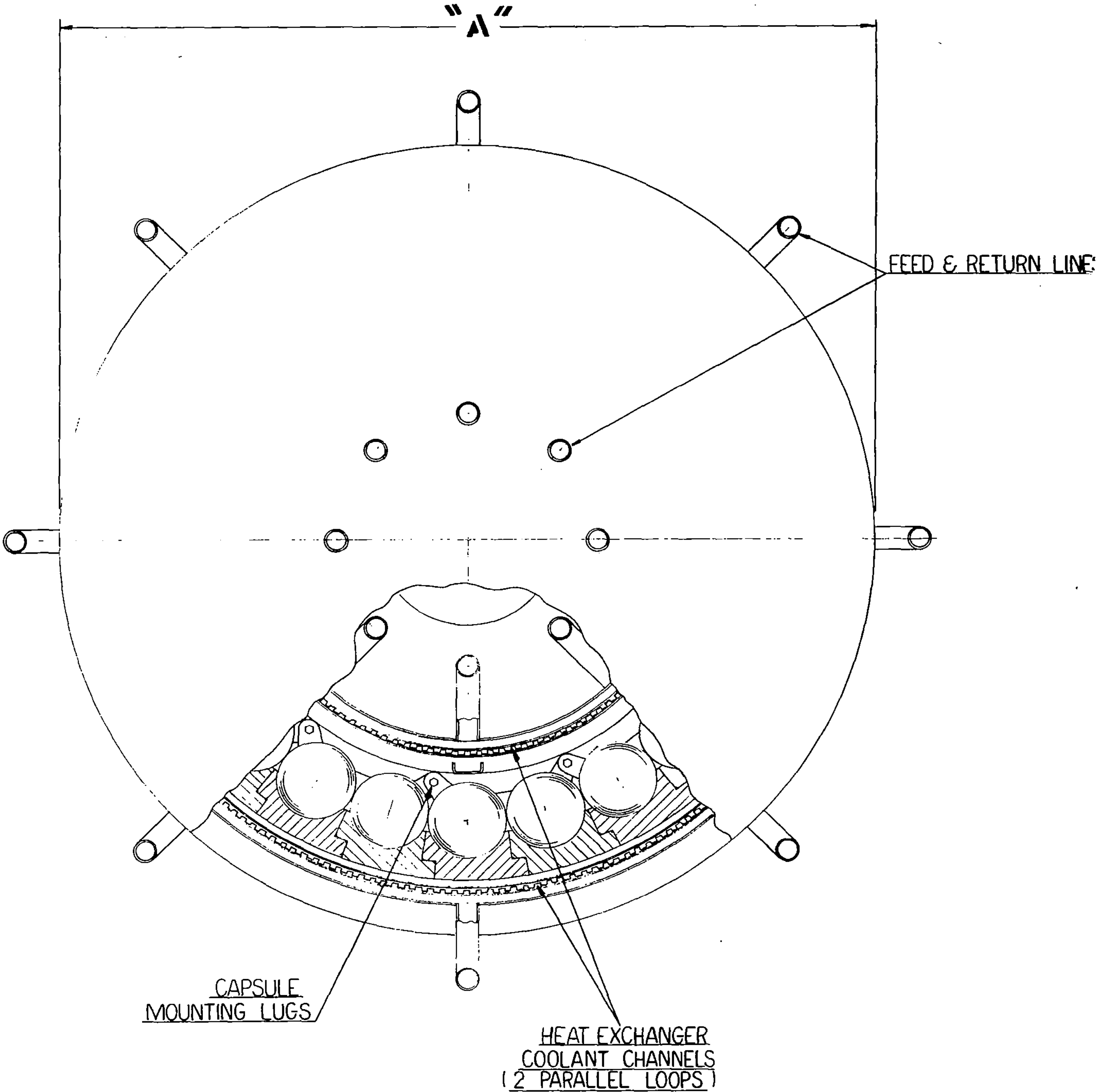


Figure C-2. Heat Source and Heat Exchanger Module-End View

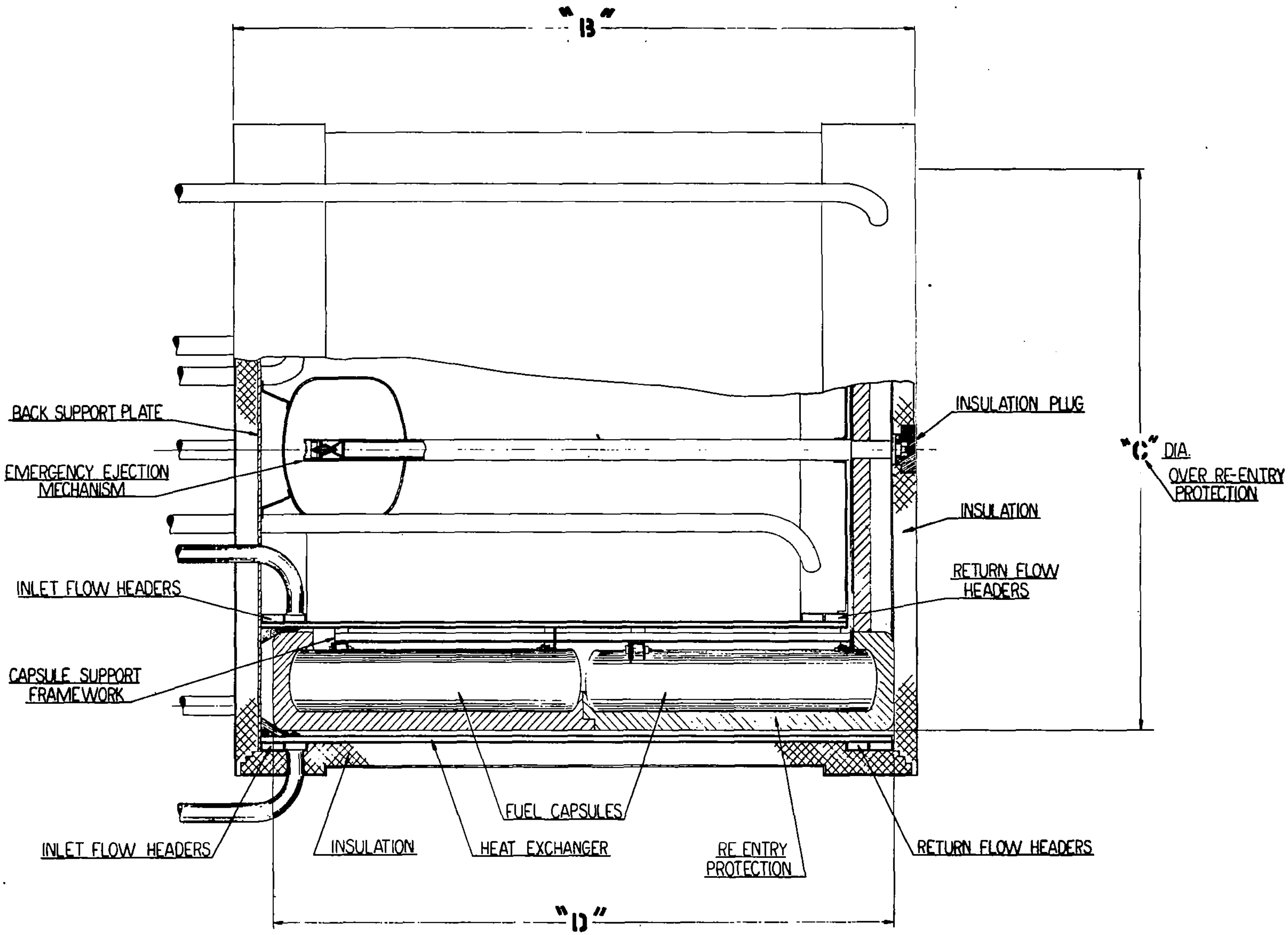


Figure C-3. Heat Source and Heat Exchanger Module-Side View

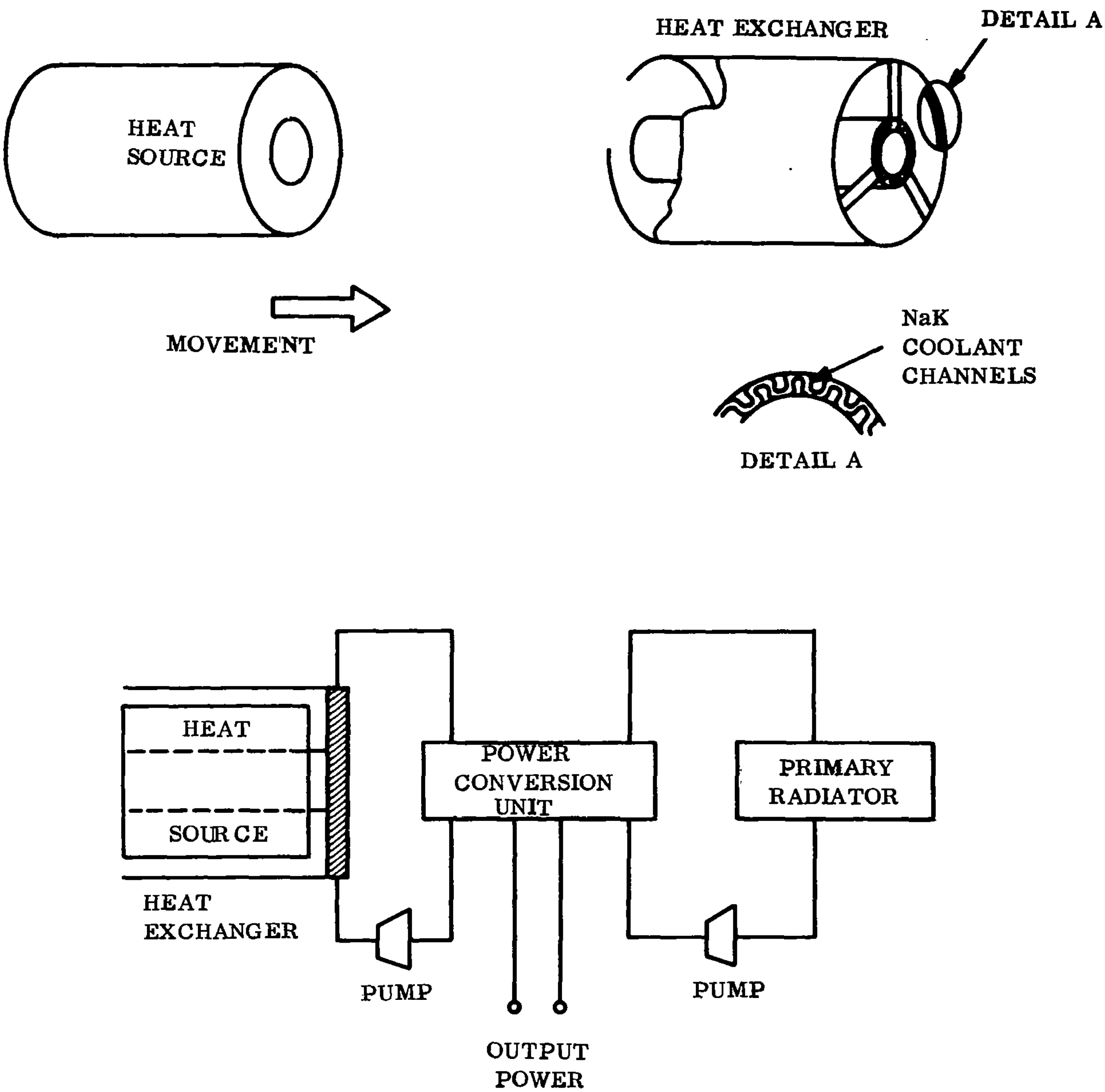


Figure C-4. Heat Source - Heat Exchanger Interface proper will consist of either one or two rings of capsules enclosed by a torus of re-entry protection material in the form of a foamed metal impregnated with a salt.

The fuel capsules are mounted to a skeleton framework in a circular arrangement. The framework consists of channel rings with circular segments cutaway for location of the capsules. Angle rings are provided at one end to hold the spring clip in place. The heat source module is supported by the channel rings and angle rings which are attached to the closure plate at one end of the framework. Re-entry protection is provided over the entire module. The foamed metal is brazed to the individual capsules, impregnated prior to assembly, and is stepped to provide an interlocking assembly. A closure cap of re-entry material on the capsules completes the protective shroud. Thermal insulation is provided over the closure cap. Heat rejection is by conduction from the outside walls of the capsules through the re-entry protection and with radiation to the outside heat exchanger. Heat transfer to the inner heat exchanger is by direct radiation from the interior walls of the fuel capsules. Two independent cooling loops are provided in the heat exchanger within the heat source, each of which can remove the entire energy output of the source.

In addition to the redundancy of coolant loops, a "last-ditch" cooling capability is provided by an emergency ejection mechanism. This mechanism, which is designed to expose the fuel capsule module for environmental cooling in the event of a loss of cooling to the heat source, ejects the fuel capsule module to the rear of the roving vehicle trailer.

Heat is transferred to the gas loop by a compact NaK to argon heat exchanger. Integral with the NaK loops is an insulated tube and fin radiator which is used as a secondary means of heat rejection should the power conversion loop fail. The heat exchanger is exposed by insulated shutters operated by a vapor pressure bellows from a temperature probe in the heat source.

For the Mercury-Rankine system, the heat source and heat exchanger module is shown in Figures C-5 and C-6. Mating of the heat source with the exchanger occurs also as shown in Figure C-4. The arrangement of capsules and re-entry material is the same as the design

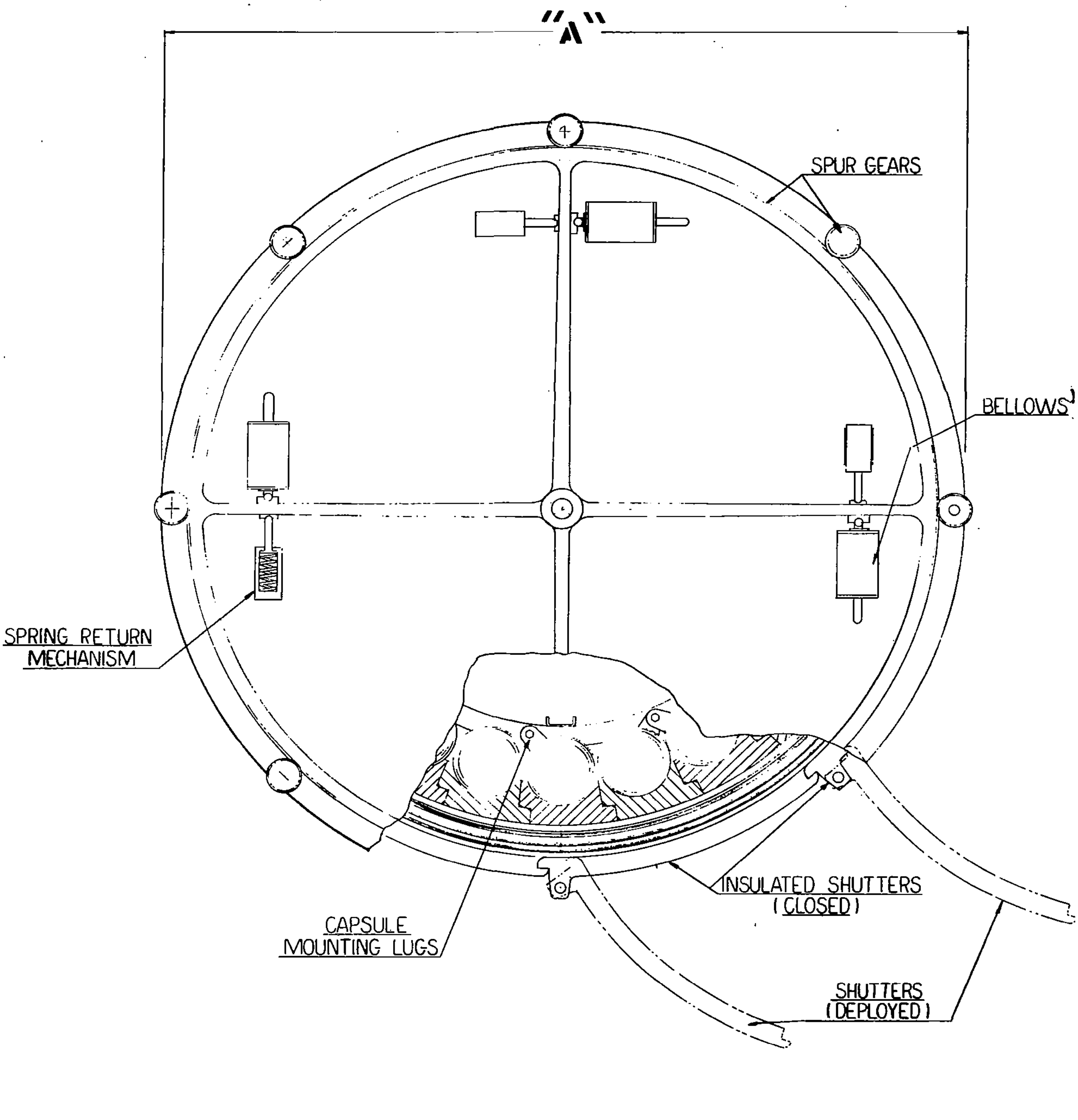


Figure C-5. Mercury-Rankine Heat Source and Heat Exchanger Module - End View

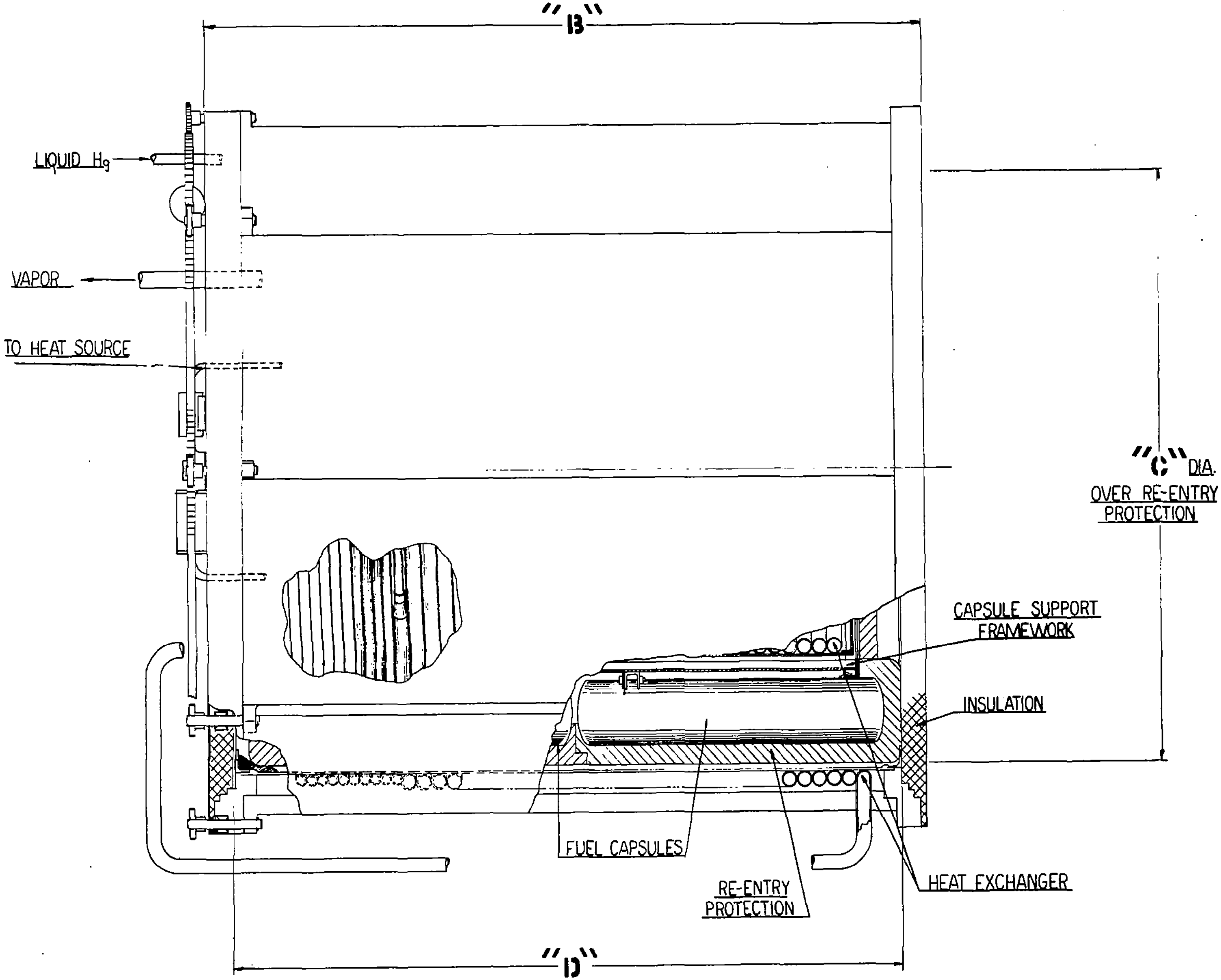


Figure C-6. Mercury-Rankine Heat Exchanger Interface Module - Side View for the Brayton systems as is the method of securing the individual capsules (with their segment of re-entry material) to the framework forming the torus. Heat transfer to the inner and outer sections of the heat exchanger is also by radiation as described for the Brayton systems.

In the event of failure of the power conversion loop, a possible secondary means of heat rejection would be periphal insulation in the form of shutters operated by bellows activated by vapor pressure generated in a temperature probe within the heat source. The shutters would open allowing rejection of heat to space by direct radiation if the temperature of the fuel capsules increases. This system was considered for possible use with the Rankine systems since the high temperature radiator for waste heat rejection might not require such a large area on the trailer and might therefore provide a sufficient view of space. System analyses, however, indicated that the view of space afforded by the Rankine cycle radiator would also be severely restricted. Therefore a separate cooling loop is provided in the exchanger within the source which is used to transfer heat to a temperature control radiator. Heat rejection is by direct radiation to space and is controlled by a system of insulated shutters on the radiator from a temperature probe within the source. As was incorporated in the Brayton cycle heat sources, a "last-ditch" capability is provided by an emergency ejection mechanism which will expose the fuel capsule module for environmental cooling.

C.4 CAPSULE MODULE

C.4.1 FUEL CAPSULE DESIGN

Design considerations for fuel capsules based on previous studies of large isotope heat sources included such factors as pressure containment, (i.e., pressure increase caused by the buildup of helium gas from the alpha-particle emitter), long term stress rupture, impact survival, fabrication, thermal performance, and fuel containment. In addition, for the present application, other criteria were imposed which included:

a. Maximum cladding temperature of the fuel shall not be exceeded during normal operation.

b. The most fissionably reactive configuration of all capsule fuel for any one heat source shall not exceed a reactivity of 0.9.

c. The fuel shall be contained for any credible accident.

C.4.1.1 Venting

Although there are currently no tried and proven vent designs which can be incorporated into capsule design, the choice of a vented capsule for the heat sources is based on the following considerations:

a. Stress rupture limitations are severe on nonvented plutonium capsules due to the long half life of the fuel.

b. Fabrication of a vented capsule should be much simpler than an unvented design and is therefore important for its benefit to the development of a single-wall capsule (other than a thin wall fuel liner).

c. Venting eliminates many unknowns introduced by the composite wall concept for nonvented designs in the areas of materials compatibility and design interactions.

d. The volume required for poisons in plutonium capsules competes with the void volume requirements in nonvented designs and aggravates the stress-rupture problem.

e. Uncertainties in the available data on sea water corrosion and long term effect on material properties is an added incentive for venting.

f. Vented designs offer the decided advantage of eliminating the need for pressure containment within the post-impact configuration.

C.4.1.2 Criticality

The design of the fuel capsule is such that the most fissionably reactive configuration of all fuel and structural material for any one heat source will not exceed a reactivity of 0.9.

C.4.2 MODULE DESIGN

The dimensions of the capsule module and of the fuel capsules are related to the overall system design through consideration of heat transfer requirements. Equating the heat generation rate of all capsules for a given heat source to the total heat flux which must be removed from the surface of the enclosing torus yields a relation between the dimensions of the torus and the capsule dimensions as a function of the sink temperature (heat exchanger wall temperature). The resulting dimensions for the nonintegrated and integrated heat sources are shown in Table C-3.

C.5 RE-ENTRY PROTECTION

Under normal conditions, the heat source subsystem will be located on the lunar roving vehicle after having experienced the launch from earth, translunar flight and landing on the moon without being exposed to unprotected re-entry conditions. However, the fuel module may possibly experience random re-entry as a result of an accident or an abort. Therefore, backup re-entry protection is provided for the fuel capsule module.

TABLE C-3. NONINTEGRATED HEAT SOURCE DIMENSIONS

	Nonintegrated Brayton	Integrated Brayton	Nonintegrated Rankine	Integrated Rankine
Torus Length (in.)	15.7	14.0	17.2	16.2
Torus Diameter (in.)	16.2	14.5	17.9	15.8

Based on the Apollo mission profile, an altitude of 450,000 ft (85 miles) and a velocity of 36,000 ft/sec (24,600 mph) are reference conditions for superorbital re-entry and present the worst conditions on the basis of the total integrated heat pulse experienced by the body re-entering the earth's atmosphere. With these conditions and for a typical range of values for the quantity, W/C_DA, where W = weight of body, C_D = drag coefficient, A = frontal area of body, the smallest angle of re-entry is found to be approximately 6.5^o. The module will not re-enter the earth's atmosphere at smaller angles.

Aerothermodynamic effects of superorbital re-entry are analyzed by means of a GE-MSD digital computer code (the Round Earth Point Mass Code), which calculates instantaneous heating rates based on a computed trajectory and employs appropriate body drag subroutines dependent on density and Mach number relationships coupled to a reference area. An instantaneous stagnation heating rate (based on a reference sphere of 1 foot diameter) is computed according to the relationship

$$\dot{q}_{stagnation} = \frac{17,600}{\sqrt{R_n}} \left[\frac{\rho_\infty}{\rho_s}\right]^{1/2} \left[\frac{V_\infty}{V_c}\right]^3$$

where R_n = Reference nose radius
ρ_∞ = Free stream density
V_∞ = Free stream velocity
ρ_s = Sea level density
V_c = Orbital velocity at re-entry

Since the heat source configuration is of a cylindrical geometry, the computed spherical heating rate is modified by the following multiplier to determine the actual heating rate.

To Obtain	Multiply Spherical Stagnation Heating Rate by
$\dot{q}_{tumbling\ cylinder}$	$\frac{0.554}{\sqrt{2}}\sqrt{\frac{24}{D_{ref}}}$

where D_{ref} is the reference diameter of the re-entering body. The drag coefficient used in the program is calculated by the relation $C_D = 2(0.398 + 0.178D/L)$ where D and L are the diameter and length respectively of the body.

The total integrated heat pulse, which is important to the re-entering body, is shown in Figure C-7 as calculated by the code for various angles of re-entry and for the given conditions for superorbital re-entry. Angles of re-entry larger than 6.5° are seen to result in lower total heat pulses. The total time of the heat pulse is given in Figure C-8 as a function of W/C_DA with re-entry angle as the parameter.

Since the ablative material must be bonded to the fuel capsules in the reference design to act as part of the heat conduction path, the use of graphite as the ablative material is prohibited because of its extremely high ablation temperature as compared with the clad and fuel materials. Foamed metal systems that may be bonded to the fuel capsules and impregnated with suitable ablative materials appear to offer a potential solution. Evaluations of several materials show that aluminum trifluoride is a promising candidate for the impregnating material, and the heat source designs and analyses are based on this material.

Heating rates employed to calculate the average heat shield thicknesses are based on the tumbling mode (cylinders with an L/D approaching unity will very likely tumble), and the heat shield material used in the evaluation is a porous nickel impregnated with aluminum trifluoride (90% AlF_3 + 10% Ni by volume). The AlF_3 vapor pressure-temperature characteristics indicate a potential sublimer with a heat vaporization of 1632 btu/lb and a subliming temperature of approximately $2300^{\circ}F$. This heat shield absorption capability and a body re-radiation temperature of $2200^{\circ}F$ with an effective emissivity of 0.7 result in the approximate heat shield thicknesses shown in Figure C-9. However, for each established design of the heat source with given values of L and D along with the weights of the capsules, the exact thickness of ablative material is determined by an iterative subroutine of the main program. The values found for the nonintegrated and integrated designs are given in Table C-4 with the letters referring to the corresponding dimensions in Figures C-2 and C-4.

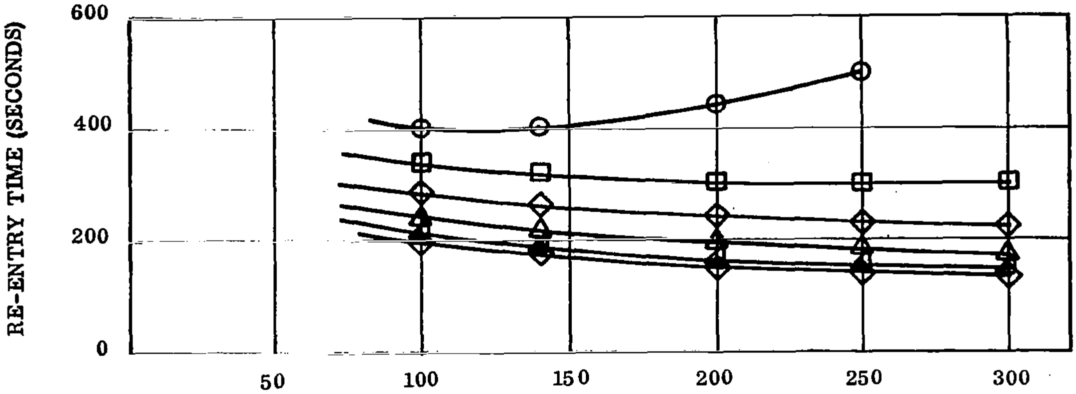


Figure C-7. Total Heat Pulse Time as a Function of W/C_DA

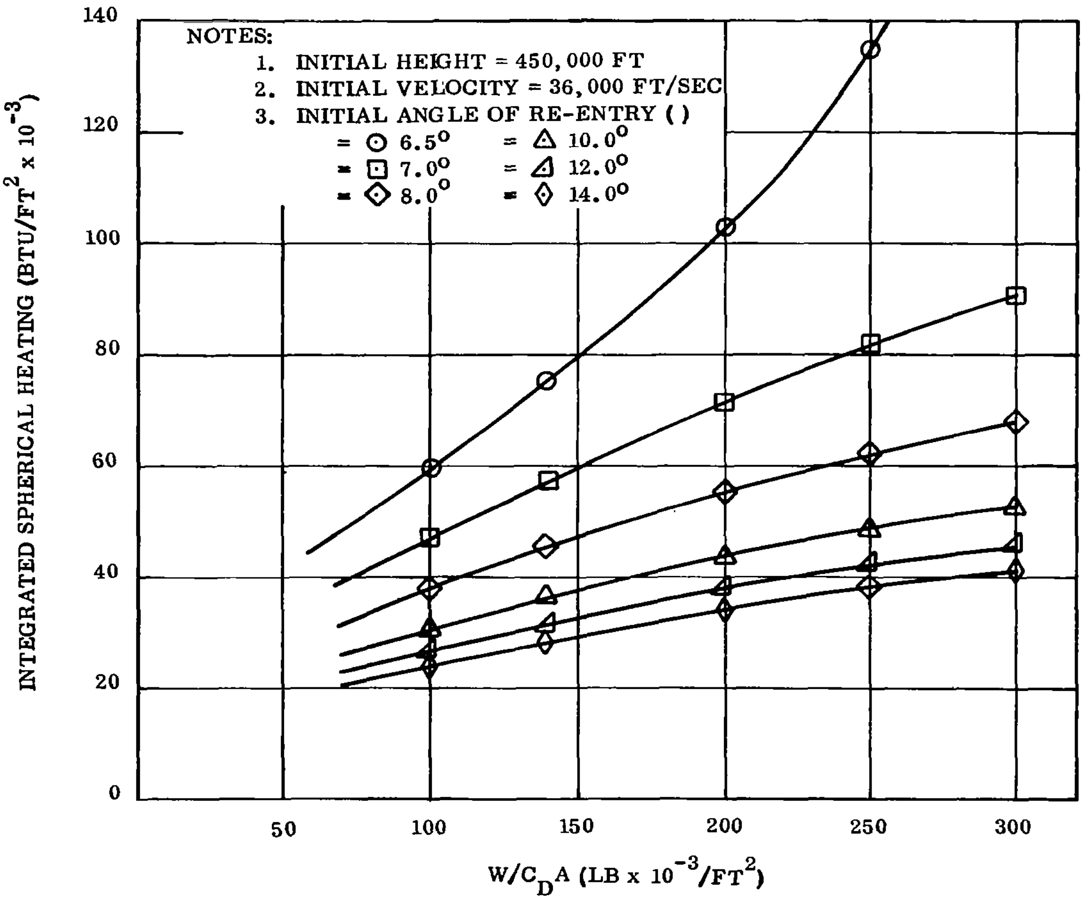


Figure C-8. Total Integrated Heat Pulse as a Function of W/C_DA

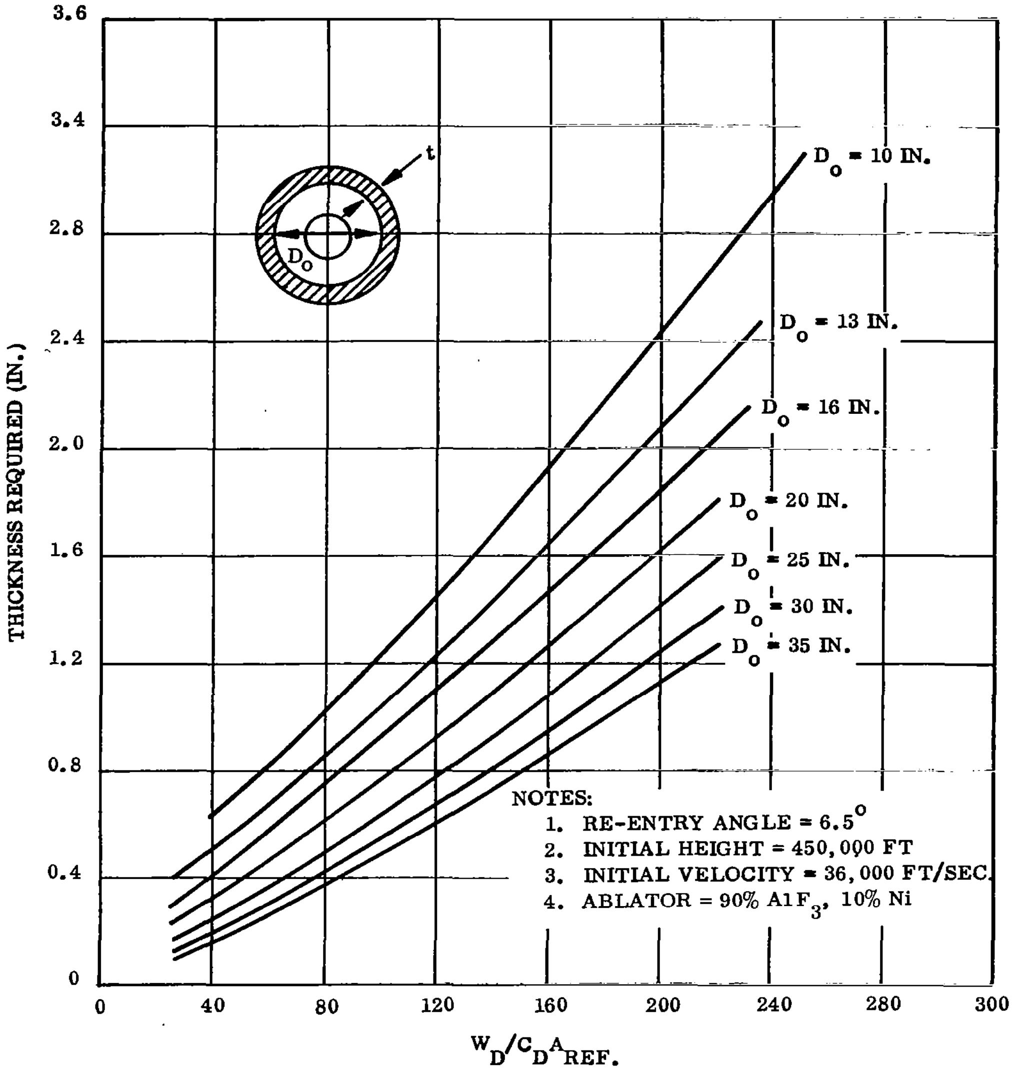


Figure C-9. Required Re-entry Material Thickness

C.6 SOURCE SUBSYSTEM WEIGHT

Additional calculations were made to determine the weights of the other components of the source subsystem such as the heat exchangers and radiators. Estimates of total subsystem weight are shown in Figure C-10.

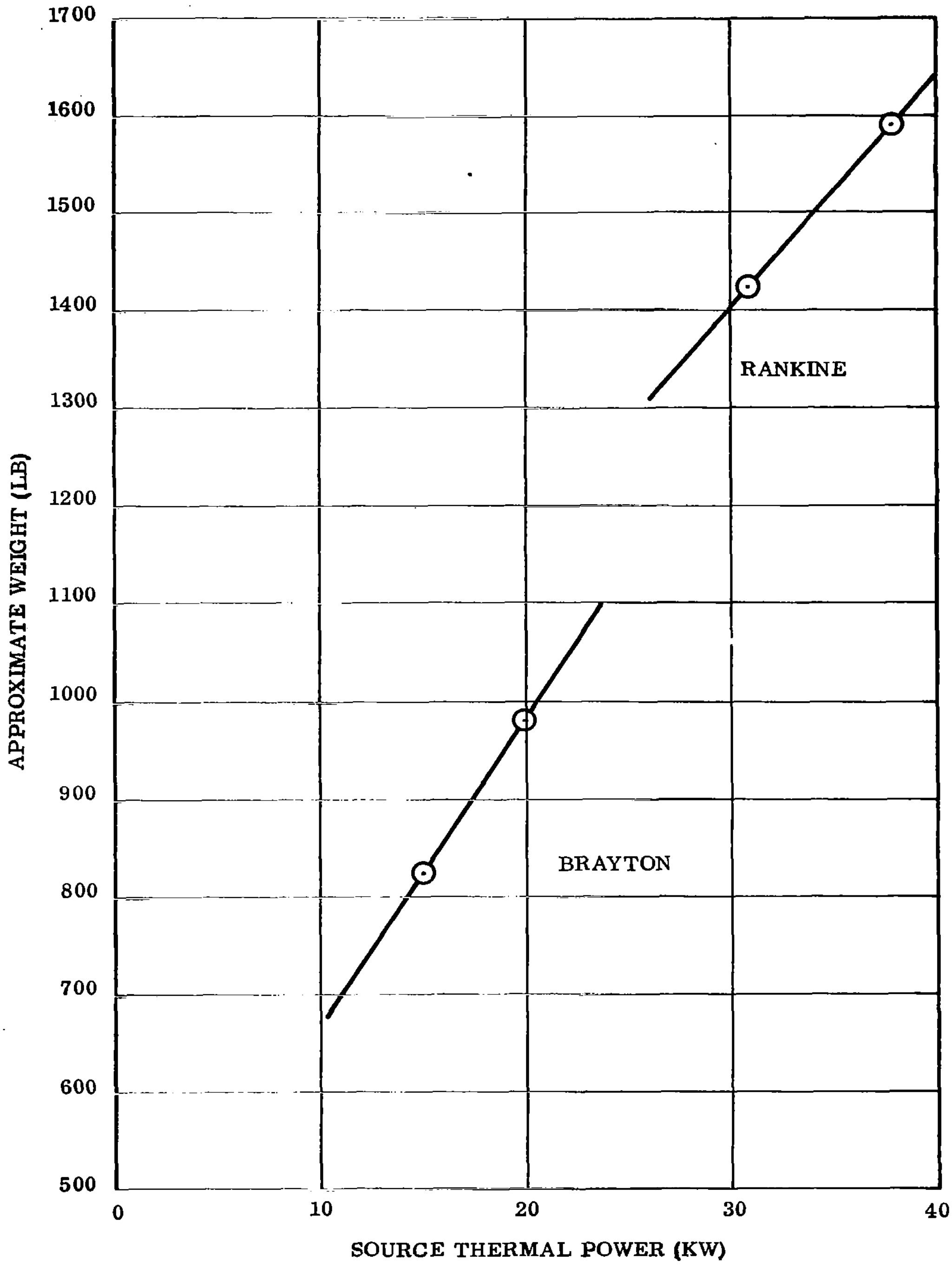


Figure C-10. Isotope Heat Source Subsystem Weight

TABLE C-4. SYSTEMS RE-ENTRY MATERIAL SPECIFICATIONS

Parameter	Brayton (Figure C-2)		Rankine (Figure C-4)	
	Nonintegrated	Integrated	Nonintegrated	Integrated
Heat Source Power (KWt)	20	15	38	31
Re-entry Material				
Thickness (in.)	1.44	1.49	2.02	2.18
Torus Length (in.) D	18.6	17.0	21.2	20.6
Torus Diameter (in.) C	19.1	17.5	21.0	20.2
Weight	231	223	395	386

In Figure C-9 the marked effect that body diameter has on the re-entry heating rate and, as a consequence, on the re-entry material thickness is apparent. Protection of individual capsules for the duration of re-entry was found from previous studies to result in higher re-entry protection weights than protection of groups of capsules in the torus configuration.

C.7 REFERENCES

C-1. Preliminary Design and Safety Studies of Large Isotope Heat Sources for Space Power, Final Technical Progress Report, General Electric - Advanced Nuclear Systems Engineering, GE Document Nos. 65SD4508-1A and 1B, January 1966.

APPENDIX D

LIQUID LOOP EXCHANGER AND RADIATOR FOR BRAYTON CYCLE

D.1 WORKING FLUID SELECTION

Reference D-1 discusses two criteria for the selection of a working fluid for the Brayton cycle cooling loop. The major problem is, of course, to find one single fluid to span the entire temperature range. Additional research has resulted in limiting the use of Freon 11 to temperatures less than about 300°F. This is necessary to limit the rate of chemical decomposition to which Freon 11 is susceptible. Two other Freon series products, Freon E1 and Freon E3 show greater potential than Freon 11 as high performance heat transfer fluids; no detailed thermophysical data is available, however, at the present time.

A schematic of the revised Brayton cycle cooling loop is shown in Figure D-1.

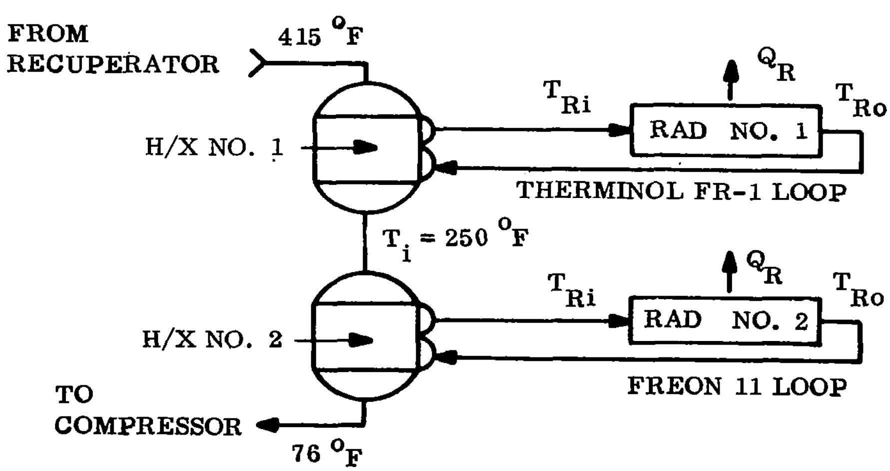


Figure D-1. Arrangement of Non-Integrated Cooling Loop for Brayton Cycle

Therminol FR-1 was chosen as the high temperature loop heat transfer fluid on the basis of its relatively low viscosity coefficient in the temperature range of use.

D.2 RADIATOR DESIGN

Two basic configurations have been chosen for a horizontal radiator.

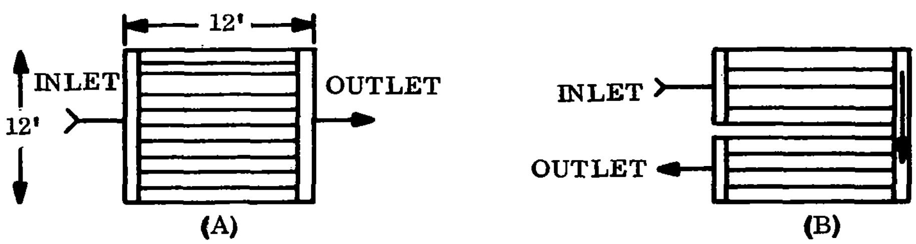


Figure D-2. Isotope Brayton Radiator Configuration

These particular configurations are not the precise dimensions of the weight optimized radiators, however, their choice is dictated by the necessity of fitting within 144 sq ft.

As discussed in Section 8, the Brayton Cycle radiator was divided into two sections; one a cylindrical segment, the other a horizontal radiator, to minimize the planar area and resulting booster shroud problems. The maximum horizontal area permitted is used in each case, since it faces the lowest sink temperature.

In order to fit the lower ($\alpha_s = 0.05$, $\epsilon = 0.84$) temperature radiator within the maximum allowable envelope the intermediate temperature in the Argon loop has been selected as 203°F. Table D-1 summarizes the optimum radiator parameters, and Table D-2 presents configuration parameters for a vertical radiator.

TABLE D-1. FREON 11 RADIATOR DESIGN PARAMETERS (HORIZONTAL - α_s=0.05, ϵ=0.84)

Configuration	Q_R (kw)	T_{Ri} ^oF	T_{Re} ^op	Header (ft)	Tube Length (ft)	Weight (lb)	Area (ft^2)
B	4.87	188	61	6.0	23.9	94.7	144

TABLE D-2. THERMINOL RADIATOR DESIGN PARAMETERS (VERTICAL, α_s=0.2, ϵ=0.9)

Configuration	Q_R (kw)	T_{Ri} ^oF	T_{Ro} ^oF	Header (ft)	Tube Length (ft)	Weight (lb)	Area (ft^2)
B	8.13	400	188	3.5	40.8	35.1	142.8
A	8.13	400	188	6.5	23.8	38.2	155.2

D.3 HEAT EXCHANGER DESIGN

The basic heat exchanger configuration chosen was a counter flow plate-fin heat exchanger. This arrangement represents the smallest number of transfer units and consequently the smallest weight. A summary description of the method of analysis is given in Reference D-1. Figure D-3 shows a schematic of this configuration. Figure D-4 depicts the radiator weight as a function of power system waste-heat rejector.

Table D-3 summarizes the basic heat exchanger parameters for the nonintegrated designs and integrated designs.

Figure D-5 depicts the heat exchanger weights as a function of waste heat rejection for the nonintegrated Brayton Power System.

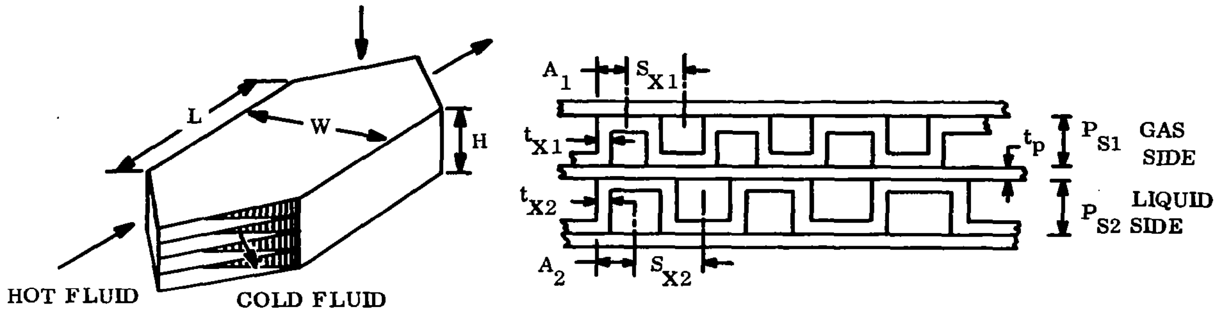


Figure D-3. Counterflow Plate Fin Heat Exchanger Configuration

D.4 INTEGRATED BRAYTON SYSTEM

Design data for heat exchanger can be found in Table D-3. The heat exchanger weights as a function of Brayton Cycle waste heat are shown in Figure D-6 for the case in which absorption refrigeration is used and D-7, for the design where absorption refrigeration is not used. For each Figure the H/X numbers corresponds to the application call outs in Table D-3.

The radiator design parameters are presented in Tables D-4 and D-5 for the integrated Brayton systems. Note that only a horizontal radiator is required for Brayton cycle waste heat less than about 7.5kw. Approximately 1.2 kw is transported to the life support system, so that this would represent a 6.3 kw radiator thermal load. The horizontal radiator is shown as configuration B in Figure D-2 due to the higher efficiency of this arrangement. Figure D-9 depicts the radiator weight for the case where absorption refrigeration is used and Figure D-8 shows the weight as a function of waste heat for the design where absorption refrigeration is not used.

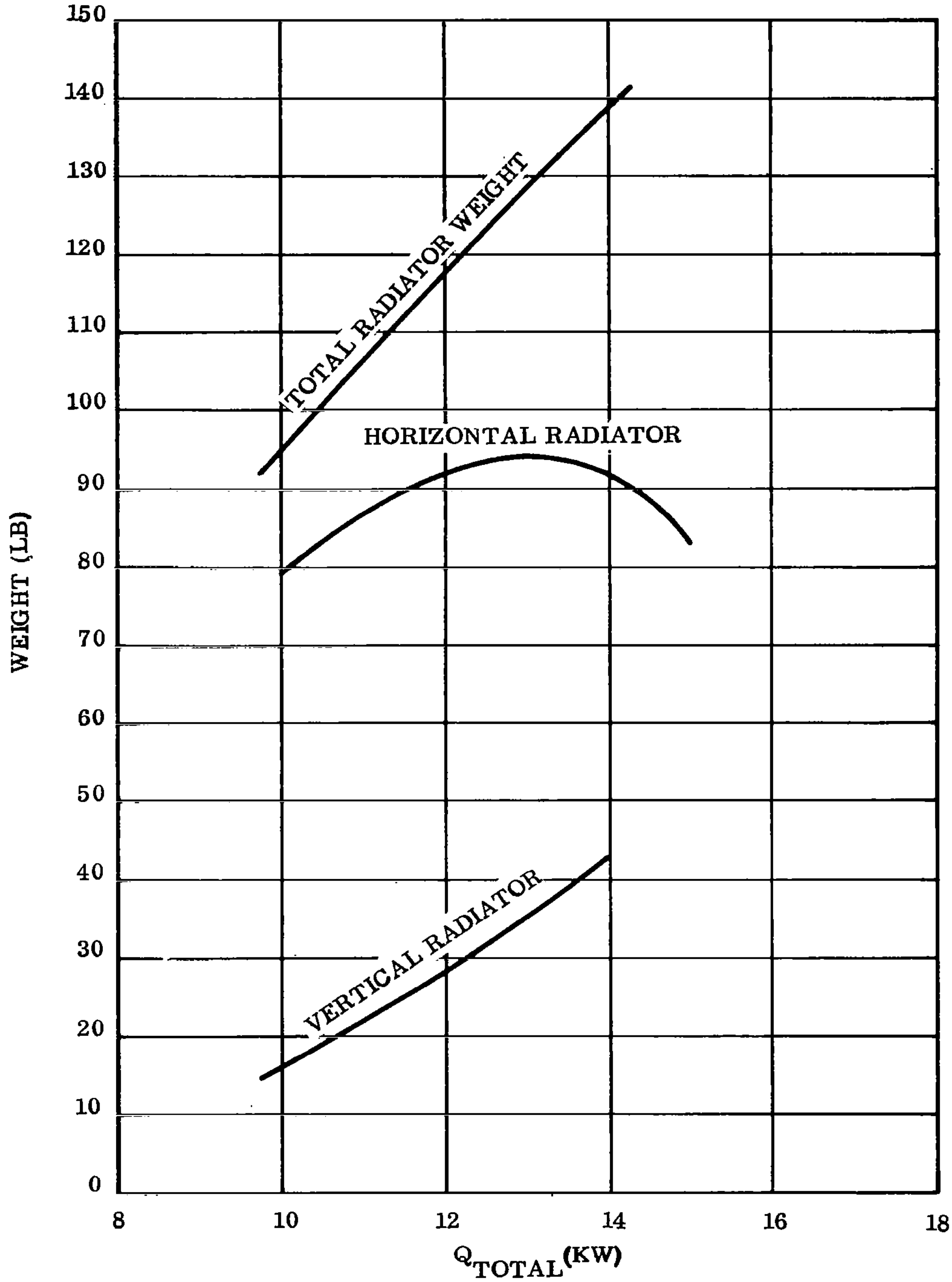


Figure D-4. Nonintegrated Brayton Cycle Radiator Design

TABLE D-3. TYPICAL HEAT EXCHANGER DATA FOR BRAYTON POWER SYSTMES

NONINTEGRATED				INTEGRATED				
Application	A	B	1	2	3	4	5	6
Heat Exchanged (KW)	8	5	1.2	.635	8.17	1.2	3.67	5.13
T_H - In (^oF)	515	203	415	375	353	415	375	250
T_H - Out (^oF)	203	76	375	353	76	375	250	76
L (ft)	.792	1.26	.42	.639	5.7	4.2	2.49	2.54
H (ft)	.26	.319	1.89	.583	.505	1.89	1.14	.466
W (ft)	1.26	1.24	.315	.250	.475	.315	.254	.50
Weight (lb)	22.4	27.5	6.3	1.39	42.0	6.3	18.9	19.2
T_C - In (^oF)	188	61	360	208	45	360	235	56
T_C - Out (^oF)	400	180	400	230	338	400	360	235
ϵ	.88	.90	.727	.132	.90	.727	.893	.90
P_{s1} (inches)	.345	.334	.915	1.25	.606	.915	.594	.579
P_{s2} (inches)	.15	.15	.15	.15	.15	.15	.15	.15
t_p (inches)	.01	.01	.01	.01	.01	.01	.01	.01
Δ P/P	.0253	.026	.0017	.0016	.048	.0017	.017	.0329

Applications:

A Vertical Radiator
B Horizontal Radiator
1 Life Support
2 Absorption Refrigeration
3 Radiators
4 Life Support
5 High Temp. Radiator
6 Low Temp. Radiator

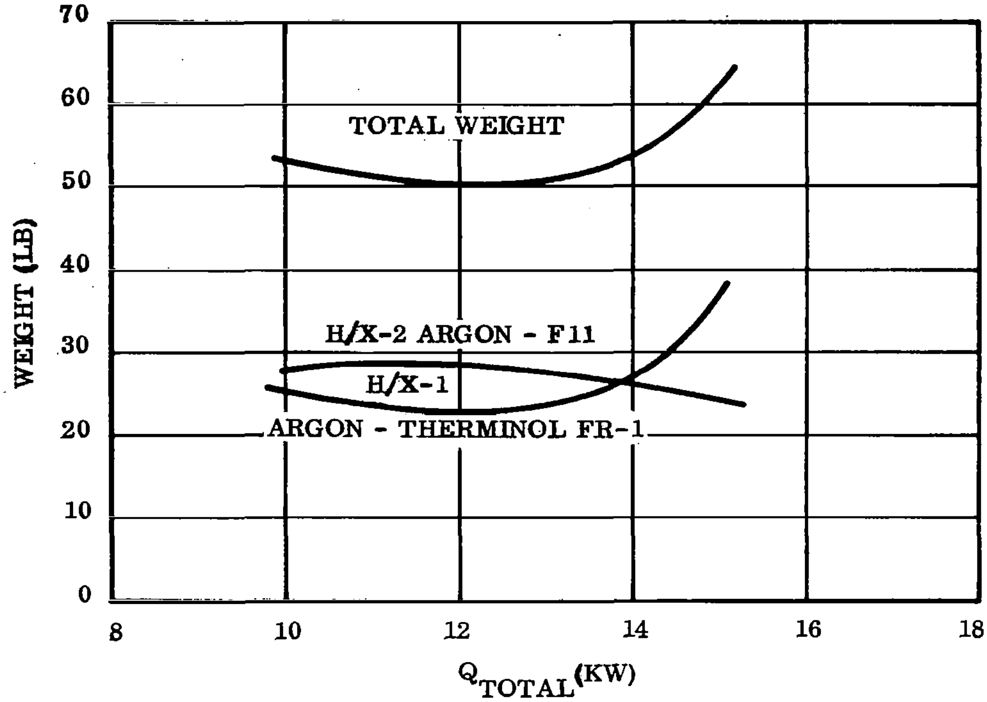


Figure D-5. Nonintegrated H/X Weight

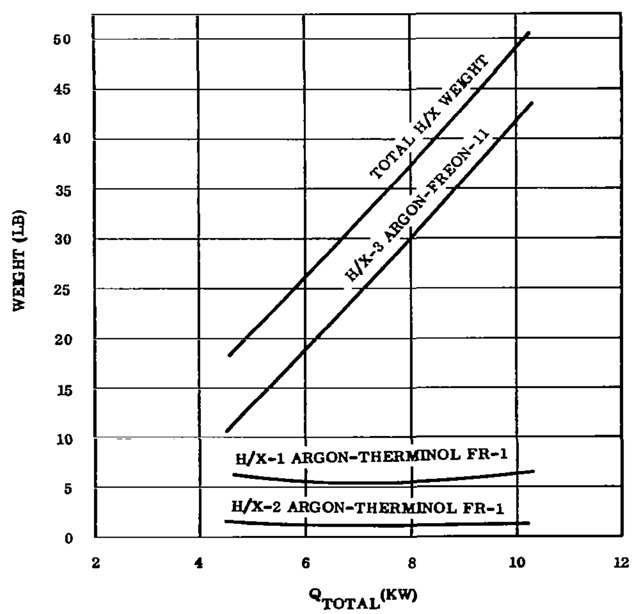


Figure D-6. Integrated Brayton Cycle Heat Exchanger Design - With Absorption Refrigeration

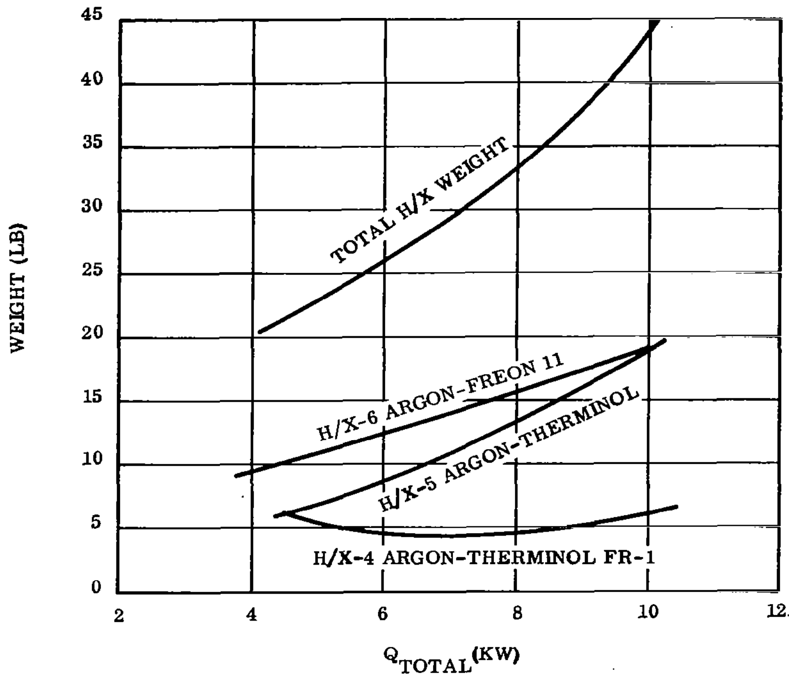


Figure D-7. Integrated Brayton Cycle Heat Exchanger Design - No Absorption Refrigeration

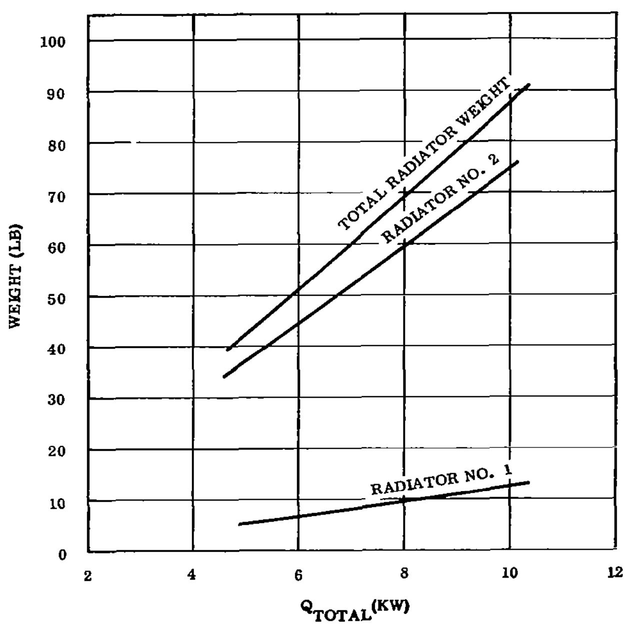


Figure D-8. Integrated Brayton Cycle Radiator Design - No Absorption Refrigeration

TABLE D-4. RADIATOR NO. 1 DESIGN PARAMETERS - INTEGRATED ISOTOPE BRAYTON SYSTEM

Q_{TOT} KWt	Q_R KWt	T_{Ri} ^oF	T_{Ro} ^oF	Header ft	Tube Length ft	Area ft^2	Weight lb
5.0	1.24	319	235	3.0	7.33	22.1	5.24
7.5	2.45	346	235	3.0	12.63	38.0	9.07
10.0	3.67	360	235	3.0	17.29	52.0	12.75

Coolant: Therminol FR-1
Radiator Material: Beryllium
Coating: $\alpha_s = 0.2$, $\epsilon = 0.9$
Configuration: vertical

TABLE D-5. RADIATOR NO. 2 DESIGN PARAMETERS - INTEGRATED ISOTOPE BRAYTON SYSTEM

Q_{TOT} KWt	Q_R KWt	T_{Ri} ^oF	T_{Ro} ^oF	Header ft	Tube Length ft^2	Area ft	Weight lb
5.0	2.56	235	56	4.0	17.6	79.6	36.9
7.5	3.85	235	56	5.0	22.0	123	57.0
10.0	5.13	235	56	6.0	23.6	141.9	74.4

Coolant: Freon 11
Radiator Material: Aluminum
Coating: $\alpha_s = 0.05$, $\epsilon = 0.84$
Configuration: Horizontal folded

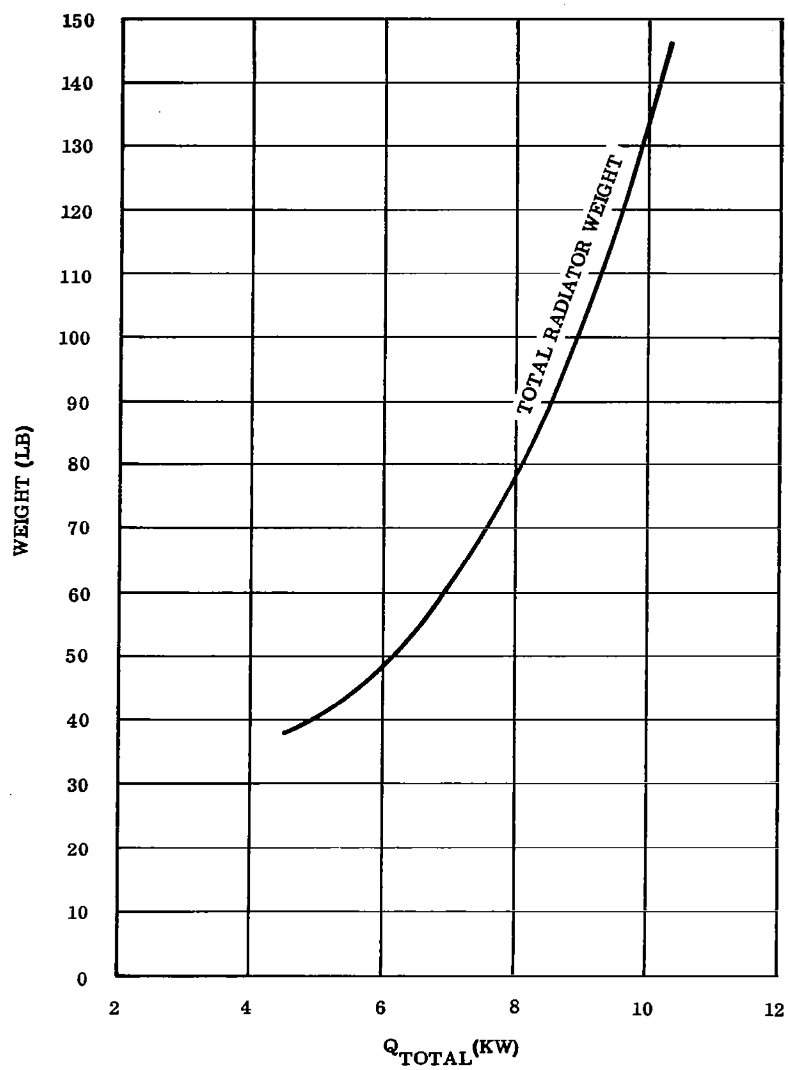


Figure D-9. Integrated Brayton Cycle Radiator Design - With Absorption Refrigeration

D.5 REFERENCES

D-1 Woods, R. W. and Erlanson, E. P., Study of Thermal Integration of Electric Power and Life Support Systems for Manned Space Stations, General Electric Company, Document Number 66SD4231, Final Report for Contract NAS 3-6478, January 26, 1966.

APPENDIX E

VAPOR FIN RADIATOR DESIGN

E.1 RADIATOR DESIGN SPECIFICATIONS

Vapor fin radiator preliminary designs have been carried out for the heat loads, argon inlet and outlet temperatures, and effective sink temperatures that are shown on Table E-I.

TABLE E-I. RADIATOR DESIGN SPECIFICATIONS

	Thermal Load	Effective Sink Temperatures		Argon Temperatures		Argon Flow
		Vertical Cylinder	Horizontal Plane	Inlet	Outlet	
Nonintegrated System	12.4 kw	$588^{\circ}R$	$380^{\circ}R$	875	536	0.277 lb/sec
Integrated System	8.6 kw	$588^{\circ}R$	$380^{\circ}R$	837	536	0.218 lb/sec

E.2 VAPOR FIN RADIATOR REQUIRED AREA CALCULATION

The radiating area requirements of the high temperature and low temperature portion of the radiator designs have been determined by the following procedure:

a. For the specified argon discharge temperature, total heat radiated by the low-temperature radiator has been calculated as a function of low temperature radiator inlet temperature, based on the assumption that the area is 144 sq ft, the maximum plane area allowed. Results of these calculations are presented in Figures E-1 and E-2 for the cases of 12.4 kw and 8.6 kw total heat load, respectively.

b. The heat load of the high-temperature radiator has been determined as the difference between the total radiator load and the load carried by the 144 sq ft low temperature radiator. The area required to radiate this load is then determined,

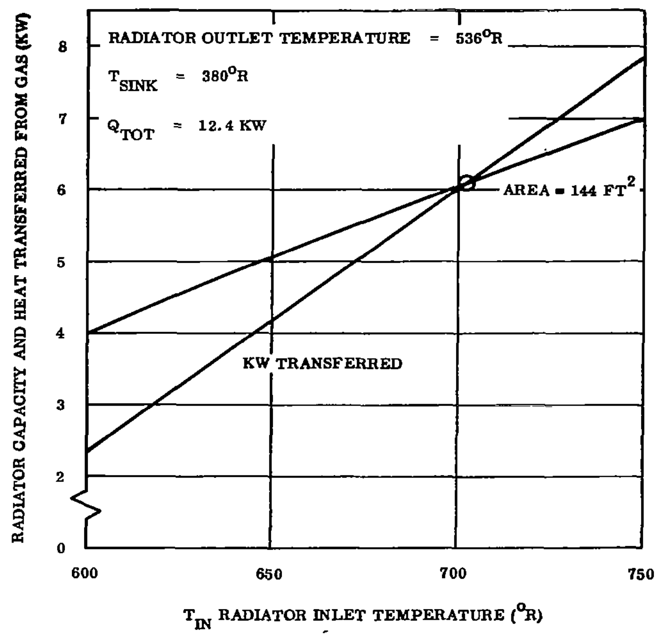


Figure E-1. Low Temperature Area Selection

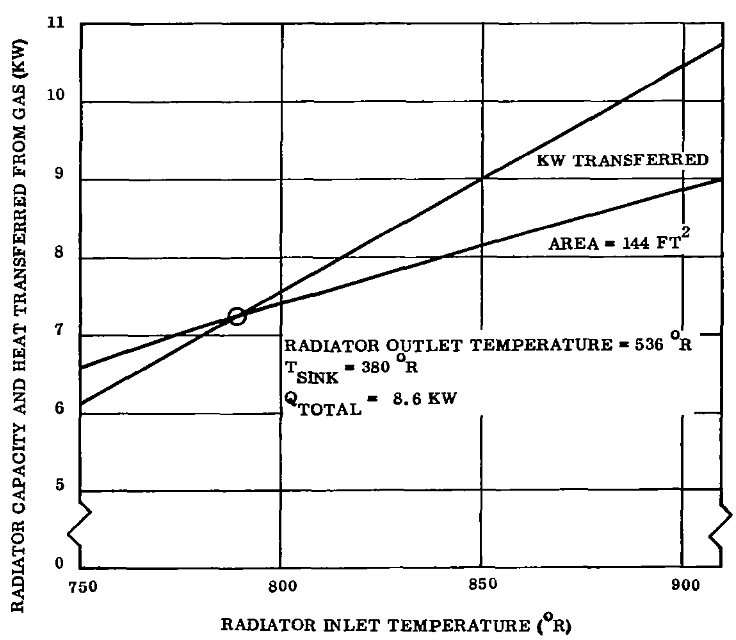


Figure E-2. Low Temperature Radiator Area Selection subject to the specified hot argon inlet temperature and the condition that the high temperature radiator outlet temperature is equal to the previously determined low temperature radiator inlet temperature.

The results for the 12.4 kw load are shown in Figure E-3, and for the 8.6 kw load in Figure E-4.

Assumptions for these calculations were as follows:

(1) A temperature difference of 20° between argon temperature and radiating surface temperature at the low temperature end of the radiator, graduating evenly to a 30° temperature difference at the high temperature end of the radiator.

(2) Effective radiating surface emissivity of 0.84.

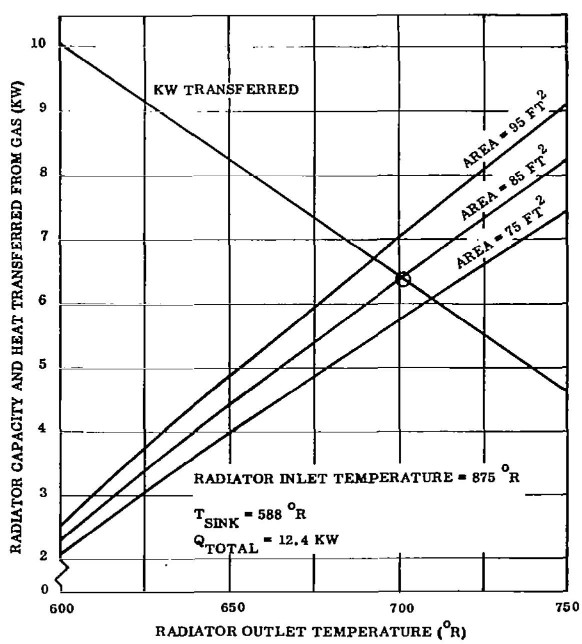
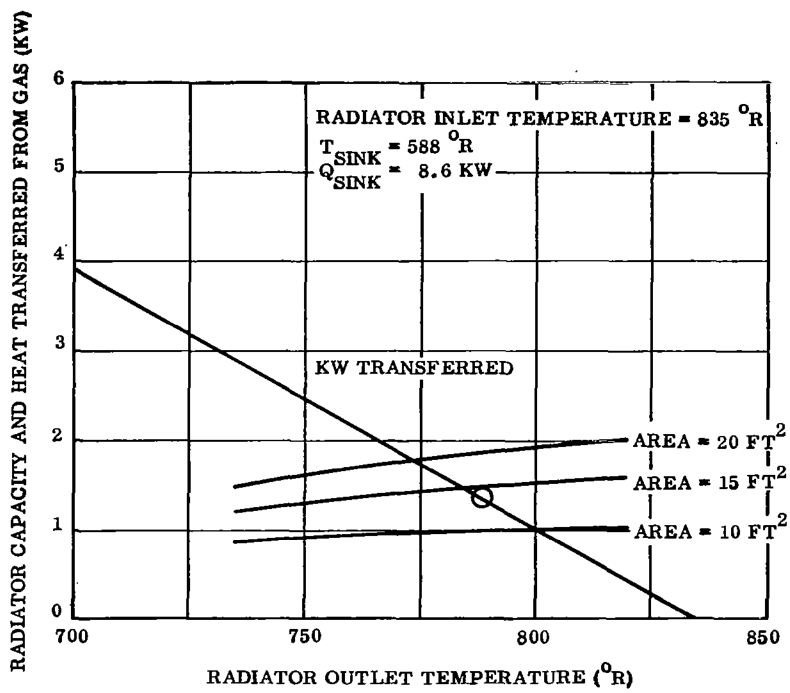


Figure E-3. High Temperature Radiator Area Selection

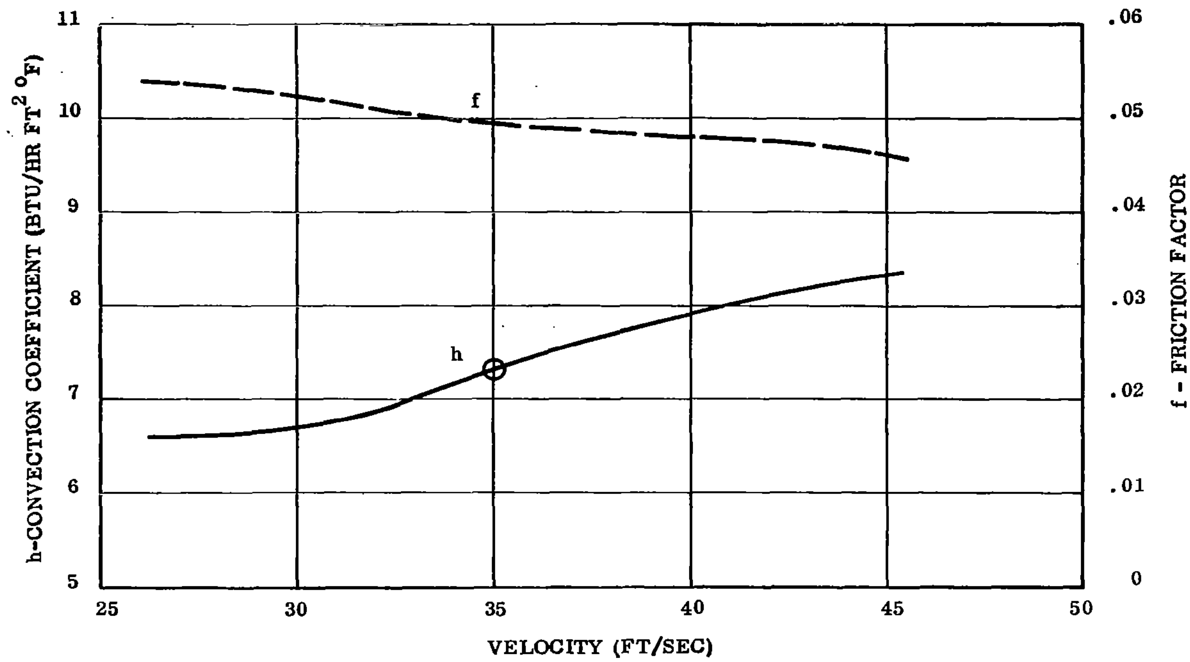


E-4. High Temperature Radiator Area Selection

E.3 ARGON DUCT DESIGN CONSIDERATIONS

Since a major feature of the vapor fin radiator design concept is the concentration of the argon flow in a single duct, one wall of which is evaporatively cooled by the attached vapor fins, it becomes necessary to provide fins inside the argon duct in order to realize convection surface area and convection heat transfer coefficients adequate to cool the argon within reasonable pressure drop limitations.

Data in Reference E-1 has been used to determine the convection coefficients and friction factors for representative fin geometries. Results of these calculations for the case of a 3-inch x 6-inch duct cross section with internal plate fin (Figure 8-10) are shown in Figure E-5. The selected design condition is indicated on this figure.



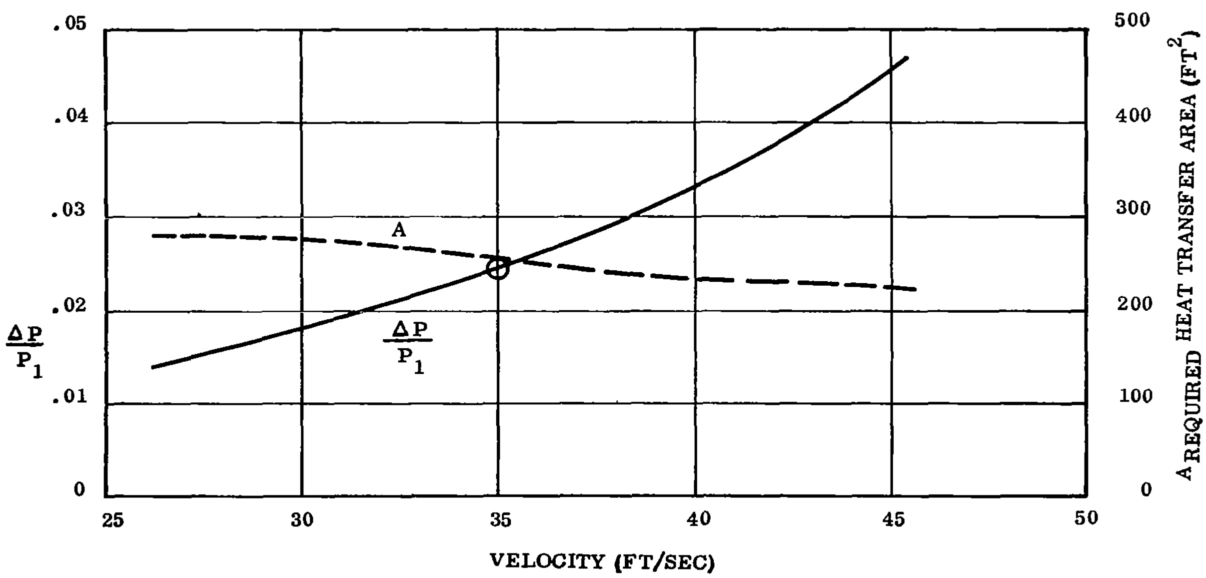


Figure E-5. Heat Transfer and Pressure Drop Characteristics of Internally Finned Argon Duct (13.5 Kw Total Heat Transfer)

E.4 VAPOR FIN INTERNAL DESIGN

Design parameters of concern in the internal operation of the vapor fin include the following:

a. Heat input surface thermal flux.

b. Vapor velocity along the fin.

c. Reflux distances.

d. Angle of reflux path to vertical

e. Working fluid properties, heat of vaporization, liquid density, surface tension, viscosity, vapor pressure at fin temperature.

f. Magnitude and duration of steady and transient accelerations to which the operating fin is subjected.

g. Vaporization surface and refluxing surface capillary structure geometry.

With water as the working fluid, experimental wick boiling data has been recorded by General Electric indicating low ΔT evaporative heat transfer at low pressure and temperature ($35^{o}F$, 0.2 psia) up to heat flux rates of 10,000 to 15,000 btu/hr/sq ft. Considerably higher fluxes can be realized at low ΔT at higher levels of temperature and pressure. The heat flux levels required by the vapor fin designs considered herein are in the range of 5,000 to 10,000 btu/hr sq ft.

The fin cross sectional area required for a vapor velocity of 100 ft/sec at representative levels of fin heat transfer is plotted for the working fluids water and Dowtherm E in Figures E-6(a) and E-6(b). These figures also indicate the fin pressure as a function of

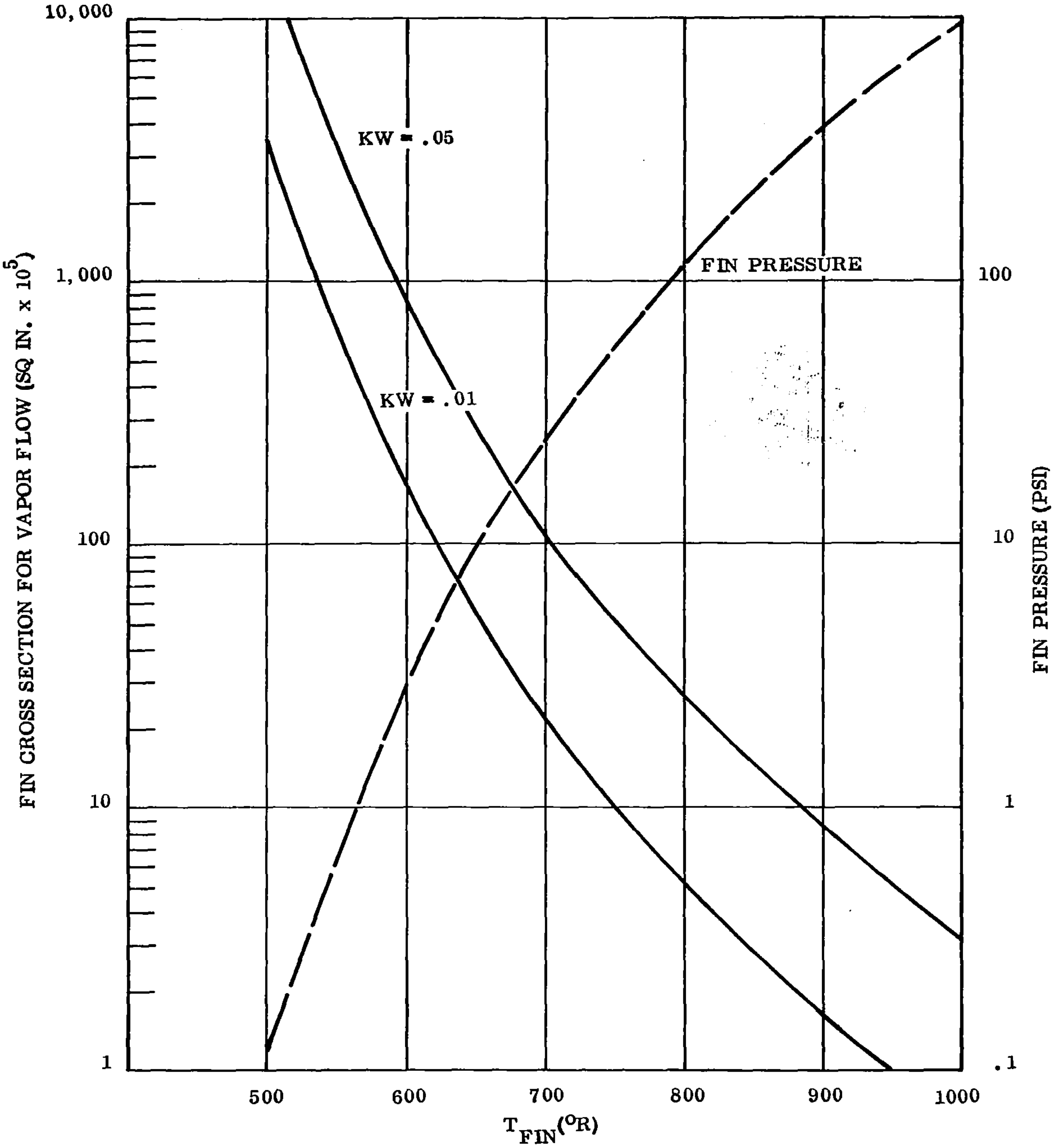


Figure E-6a. Vapor Fin Required Flow Cross Section and Working Pressure vs Fin Heat Transfer and Operating Temperature (Water Working Fluid)

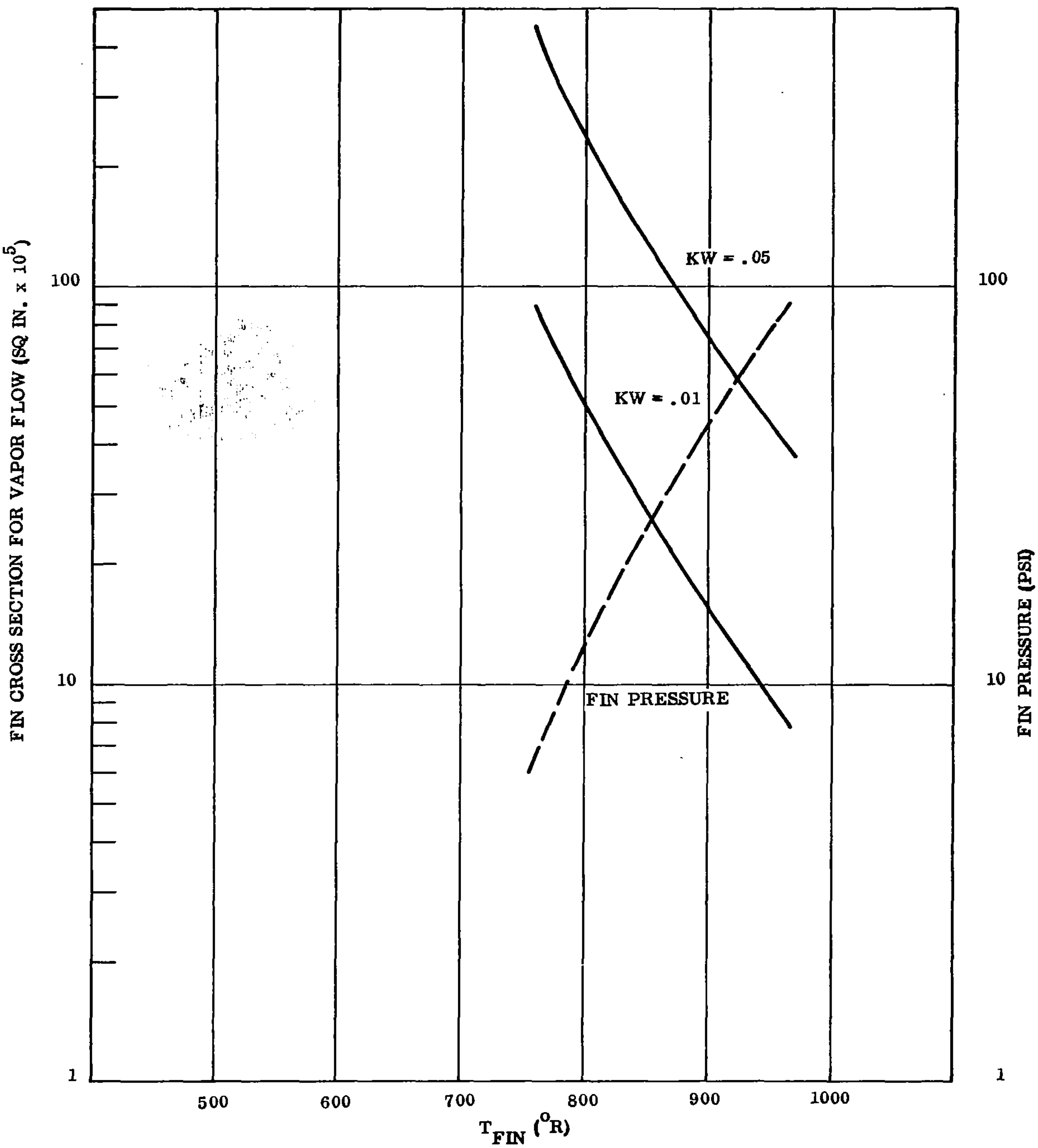


Figure E-6b. Vapor Fin Required Flow Cross Section and Working Pressure vs Fin Heat Transfer and Operating Temperature (Dowtherm E Working Fluid) temperature. In order to maintain all vapor fins at low pressure level (below 50 psi) it is advantageous to use Dowtherm E working fluid in the high temperature fins and water in low temperature fins.

Because of the low heat transfer requirements of fins required for the subject radiator designs, and also because of the low lunar gravity, it is very easily determined that the refluxing design requirements are far below those realized in 1g water experiments. By confining the reflux stream between the fin-screen insert and the fin wall, capillary forces on the refluxing liquid will substantially exceed inertia and gravity forces for all reasonably predictable shock and gravity loadings induced by jolting and tipping of the vehicle.

E.5 METEORITE PROTECTION DESIGN CONSIDERATIONS

In the development of the design concept of Figure 8-9 considerable attention has been given to the question of meteorite vulnerability and armor requirement of the vapor fins, and to the determination of required excess capacity needed to insure a specified probability of maintaining 100% radiator capacity throughout an 1800-hour mission. The design principles formulated in relation to these points can be summarized as follows:

a. Armor thickness required on the vulnerable radiating surface of the vapor tubes can be determined from curves such as those shown in Figures E-7 through E-10. These particular curves apply specifically to aluminum at 600°R and to a mission length of 1800 hours. The curves are based on the following equation relating armor thickness, vulnerable area, mission time, and materials properties:

$$ta = \left(\frac{0.448}{\gamma^{1/6}\ E^{1/3}}\right) \frac{AT^{0.249}}{-\log\ P_o}$$

where:

ta = armor thickness (in.)

γ = specific weight of vulnerable material (lb/in.^{3})

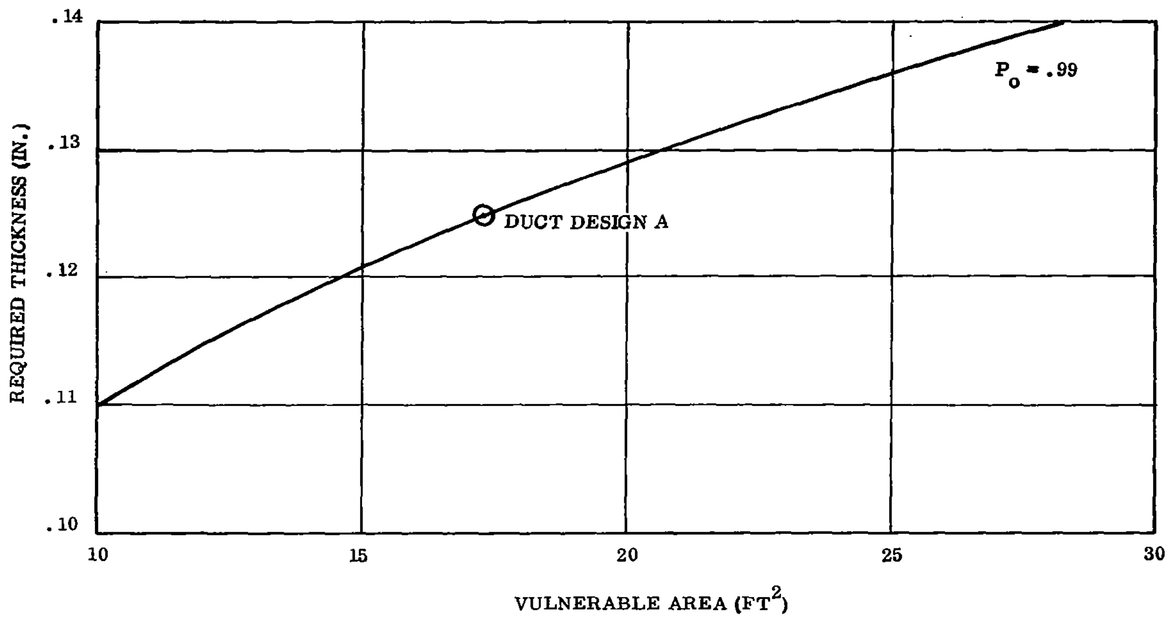


Figure E-7. Argon Duct Required Thickness vs Vulnerable Area for 0.99-1800 Hours Survival Probability (No Bumper Effect Considered)

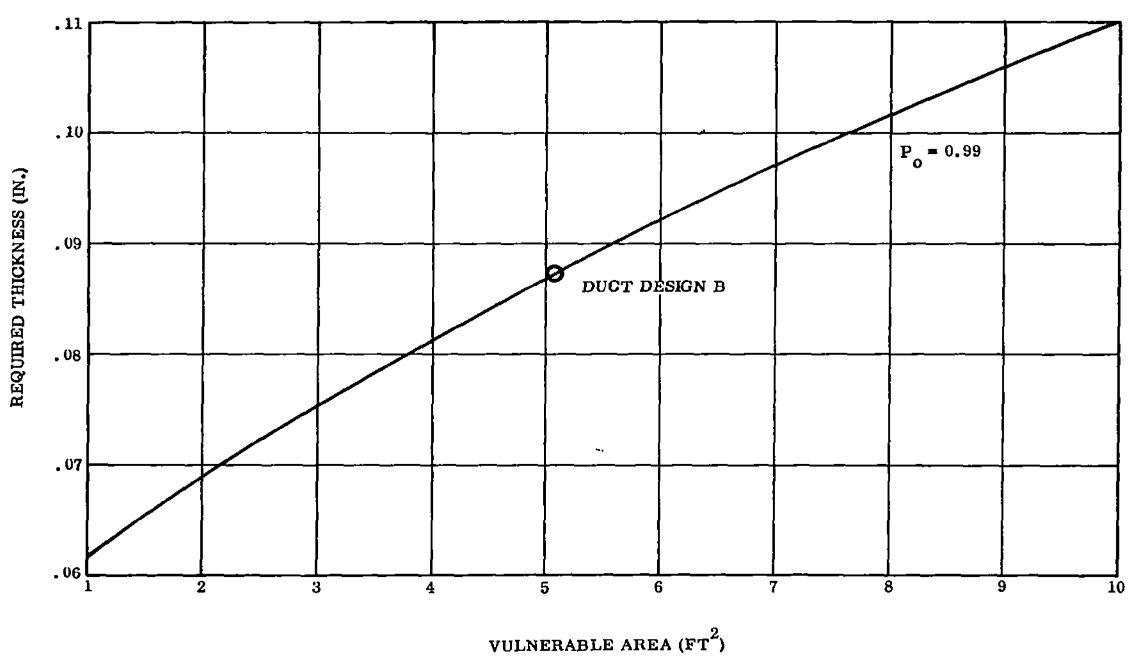


Figure E-8. Argon Duct Required Thickness vs Vulnerable Area for 0.99-1800 Hours Survival Probability (No Bumper Effect Considered)

$$\frac{tw}{ta} = \frac{2}{\pi} \tan^{-1} \left[\frac{2\,(1-t_b/t_a)}{\frac{t_b}{t_a}\left(2 - \frac{t_b}{t_a}\right)\left[\frac{4}{3} + \frac{10.4\,(s/t_a)^2}{1 + 3.6\,(s/t_a)^2}\right]} \right] - \frac{2\,(s/t_a)^2}{1 + 3.6\,(s/t_a)^2} \left[e^{-3.5\, t_b/t_a} \right] \sin 2\pi \frac{t_b}{t_a}$$

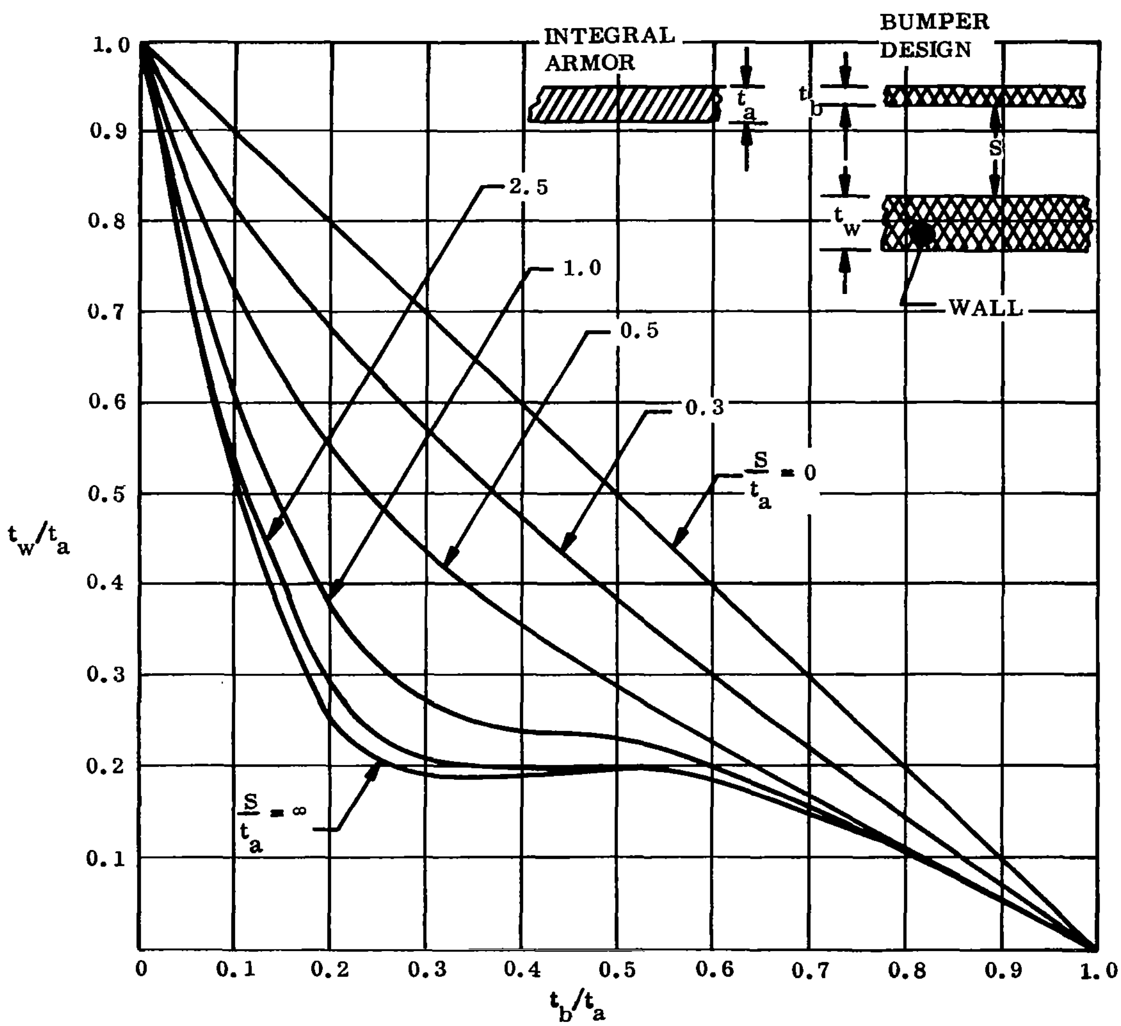


Figure E-9. Meteoroid Bumper Criterion

E = Young's modulus of vulnerable material (psi)

A = vulnerable area (ft^2)

T = time for which protection is desired (hr)

P_o = probability of no puncture

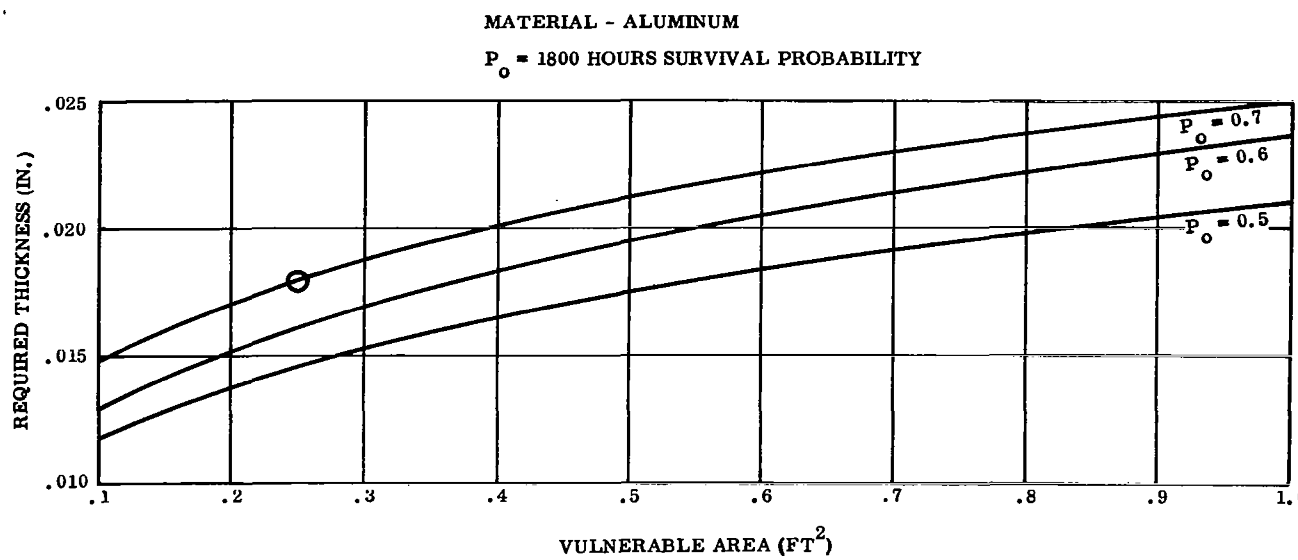


Figure E-10. Individual Vapor Fin Required Material Thickness vs. Vulnerable Area and Survival Probability

This equation has been derived from a similar equation involving sonic velocity and density rather than Young's modulus and density, which was obtained from Loeffler, Lieblein, and Clough of NASA-Lewis and Whipple at Harvard University. The value of P_o, survival probability, to be used is determined from the required overall radiator success probability (probability of completing the mission with 100% capacity), and from an assumed design number of punctured tubes. The relationship between individual vapor tube survival probability, overall radiator success probability, and the percentage of surviving vapor tubes is summarized in Figure E-11 for the case of 1000 tubes. This figure, which closely applies to all cases involving a large number of tubes (500 or more), shows that the required level of individual tube survival probability bears a straight line relationship to the assumed percentage of surviving tubes, for a fixed level of overall radiator success probability. This figure also shows that the difference required individual vapor tube survival probability corresponding to 0.90 and 0.99 overall radiator reliability is, for a fixed vapor tube survival fraction, small. As the number of tubes is increased (over 1000) this difference becomes even smaller.

b. Required vapor tube material thickness on the radiating surfaces, for a specified design level of overall radiator success probability in the range of 0.9 to 0.99, as determined from the procedure described above, can be kept small (such as 0.015 to 0.020 inches) by the following design practices:

(1) Select a large number of vapor tubes. This minimizes the vulnerable area of each tube, and also minimizes the required level of individual tube survival probability.

(2) Select a reasonably low required vapor tube survival fraction. 70% is approximately optimum.

c. A very significant feature of the design of Figure E-18 is that the vapor tubes are narrow (1/2 inch) and that the radiating surfaces of adjacent, individually sealed tubes are in thermal contact with each other. This feature has the advantage that if a given tube is punctured, thereby losing its longitudinal heat transfer capability through vapor fin heat transfer from system tube to radiating surface, it still is capable of functioning as an extended fin surface for the adjacent unpunctured vapor tubes, which now in effect have additional radiating capacity. This action, in a typical design, can compensate on the order

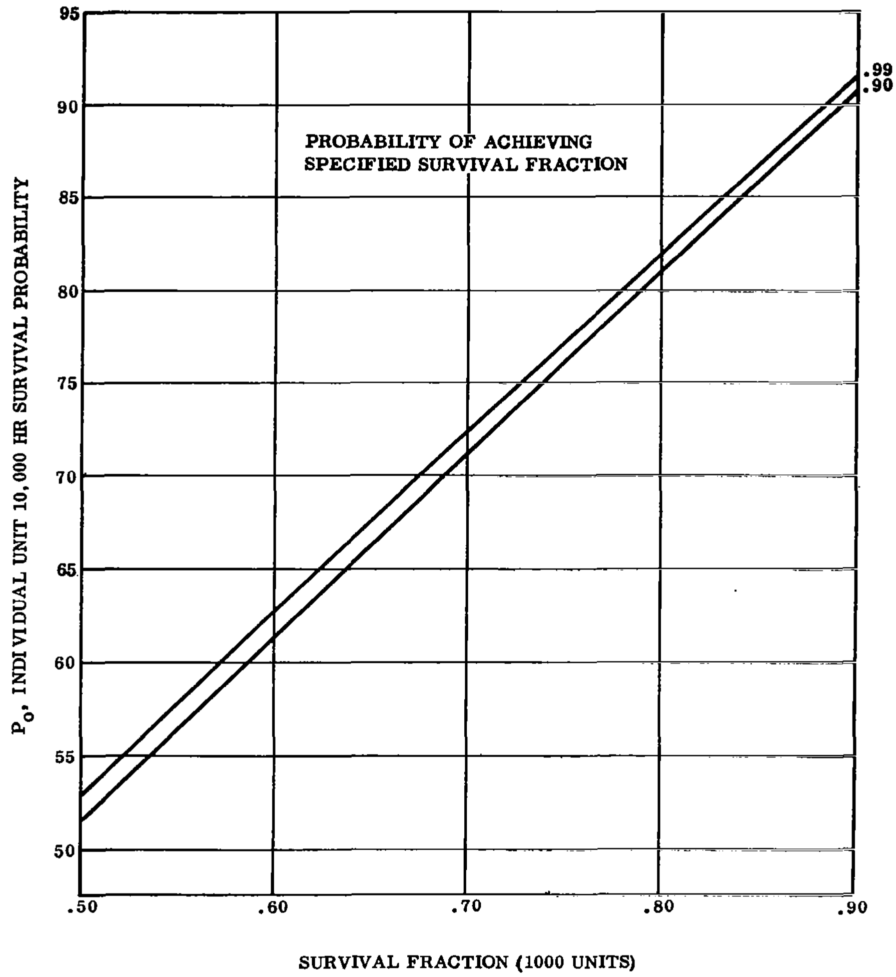


Figure E-11. Correlation of Individual Vapor Fin Survival Probability Required to Achieve Fixed Overall Radiator Success Probability vs. Vapor Tube Survival Fraction

of 95% for the loss of the punctured tube. For the case of a single punctured tube, the operating tubes on either side of the failed tube will increase their refluxing rate approximately 45% in order to support the cooling effect of the short fins (length equal to 1/2 the width of the failed tube) which have in effect become attached to one edge of these tubes. Figure E-12 correlates the start of mission excess capacity requirement with the end of mission unpunctured tube ratio. Because tube failures produce overloads on adjacent tubes, the

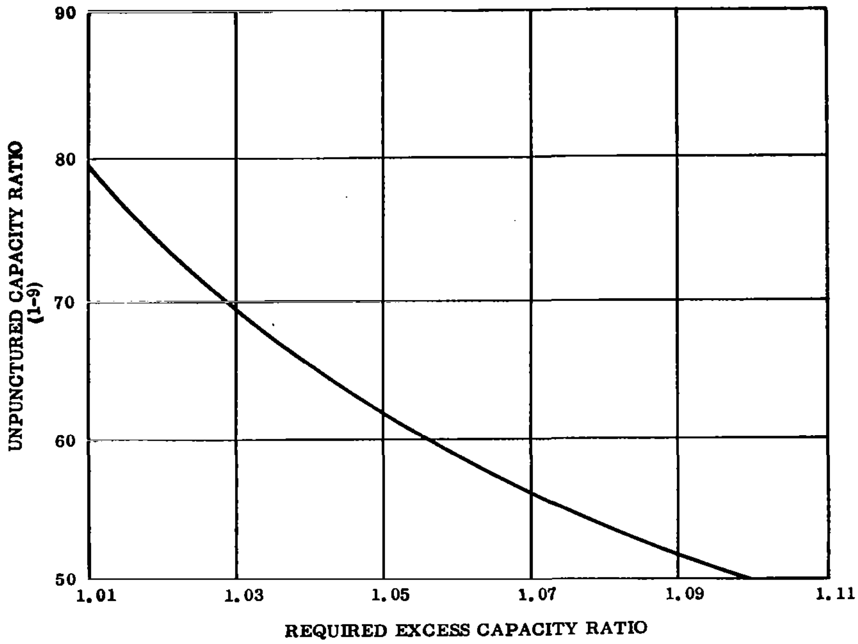


Figure E-12. Vapor Fin Radiator Required Excess Capacity to Realize Full End of Mission Capacity vs Vapor Fin Survival Fraction

normal state design levels of boiling surface heat flux and liquid refluxing rate for the vapor tubes must be conservatively selected in order to provide margin for increased vapor tube heat transfer under conditions of failure of adjacent tubes. As the number of consecutive punctured tubes involved in a failed group of tubes is increased, the net heat transfer effectiveness of the failed tubes, acting as fins on the operating tubes located on either side of the failed group of punctured tubes, decreases. This results from: (1) decreasing fin effectiveness of the contiguous failed tube surface with increasing conduction distance from the heated tubes; and (2), increasing system-tube-to-condensing-surface ΔT in the overloaded vapor tubes adjacent to the failed tube group. Of course, as this ΔT increases, there is a tendency for the overloaded tubes to pass part of the overload to adjacent working tubes.

From the foregoing discussion, it becomes evident that even the punctured tubes in a radiator design of the type shown in Figure 8-7 can have a large residual heat transfer capacity. In order to evaluate the magnitude of this effect it is necessary to determine the probable pattern of meteorite punctures in a radiator of this type. This is necessary, because, as pointed out above, a failure pattern involving isolated vapor tube punctures results in a much higher residual heat transfer capacity in the failed tube group, than does a failure pattern involving consecutive location of the punctured tubes.

A detailed probability analysis of the vapor tube failure patterns in a radiator of the Figure 8-7 type has been completed. The steps in this analysis are the following:

(1) A typical radiator design of the type shown in Figure 8-7 is assumed. The total number of vapor fin tubes is 2000. It is assumed that 400 of these have failed. This corresponds to an assumed unpunctured tube fraction of 0.80. The vapor fin tube width is 1/2 inches and the radiating surface thickness is 0.020 inches. Material is Aluminum at $650^{\circ}R$.

(2) Possible failure patterns of the following types are considered:

(a) One failed tube bracketed by two good ones
(b) Two failed tubes bracketed by two good ones
(c) Three failed tubes bracketed by two good ones
(d) Four failed tubes bracketed by two good ones
(e) Five failed tubes bracketed by two good ones

(3) The analysis determines a probable number of failures of each of these types. For each type of failure a heat transfer effectiveness of the failed group of tubes can be determined. This is essentially the fin effectiveness of the

0.020 aluminum wall over half the width of the failed tube group, but it is also subject to the assumption that the two overloaded vapor fin tubes on either side of the failed group cannot support more than three times their normal rated heat transfer.

(4) By combining the percentage of punctured radiator capacity involved in each type of failure and the fin effectiveness of the corresponding failed tube group, a probable value of overall resultant effectiveness of the total failed portion of the radiator can be determined.

(5) By combining this number with the number 0.8, which is the assumed unpunctured tube ratio, a value for required excess capacity percentage needed to insure 100% end-of-mission capacity can be determined.

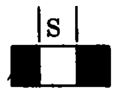
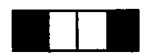
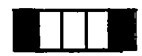
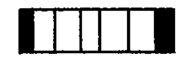
The results of this analysis for the particular case described above are summarized in Table E-2. These results indicate that single tube and double tube types of failures are, by far, the most probable. Because of this the resultant effectiveness of the failed portion of the radiator is of the order of 0.95. This means, as shown in Figure E-11, that the required excess area needed to insure 100% end-of-mission capacity is of the order of 1%, even though the total number of failed tubes is of the order of 20 to 25%. This appears to be a very attractive feature of the proposed design concept.

The bumper protection principle is employed in two places in the radiator design of Figure 8-7.

(1) The system flow tubes are enclosed within the vapor tube radiating surface. The latter surface, therefore, provides bumper protection to the vitally critical system tube surfaces.

TABLE E-2. SAMPLE CALCULATION OF REQUIRED EXCESS CAPACITY RATES FOR SPECIFIED UNPUNCTURED CAPACITY RATIO

Total Number of Vapor Tubes = 2000
Unpunctured Capacity Ratio = 0.80
S = 0.5 inches Aluminum Material 0.02 inches Thick on Radiating Surface

1	Failure Type						More Than Five Consecutive Punctures
2	Probable No. Of Failures	255	51	10	2	1	0
3	No. of Punctured Tubes Involved	255	102	30	8	5	
4	Percentage of Punctured Radiator Capacity Involved	63.75	25.5	7.5	2.0	1.25	0
5	Net Resultant Effectiveness of Punctured Tube Group	0.98	0.96	0.92	0.90	0.85	
6	Item (4) times Item (5)	62.5	24	6.9	1.8	1.06	
	Resultant Overall Effectiveness of Punctured Portion Of Radiator	0.96					
	Required Excess Capacity Ratio to Insure 100% End of Mission Capacity	$\frac{1}{0.8 + (1-0.8)\,0.96} \approx 1.01$					

□ Punctured Tube ■ Unpunctured Tube

(2) The vapor tube side wall surfaces are protected by the bumper action of the adjacent tube radiating surface and by the side walls of the adjacent tubes. In this manner the probability of incurring a double tube failure from a single meteorite impact is made much lower than the probability of a single tube failure. All analysis conducted so far has neglected, as being comparatively negligible, the probability of a double tube failure resulting from a single impact.

Hypervelocity testing has shown that the combined thickness of bumper and tube wall may be as little as half the thickness of the equivalent integral armor, depending on the spacing and relative thickness of the two. Neither theoretical nor experimental work has yet reached a sufficient level of sophistication to provide an equation relating these parameters. To meet the needs of preliminary design and digital computer analysis, a self consistent interim criterion based on computer analysis, limited test data, and a simplified phenomenological model has been devised.

Using the required integral armor thickness (t_a) as a reference quantity, wall thickness (t_w), bumper thickness (t_b), and spacing (s) may be related dimensionlessly as shown in Figure E-11. The points shown are test data and the heavy line is curve fitted to these points. Variation with (s/t_a) is established by forcing an asymptote slightly beyond the line for $s/t_a = 2.5$ and adapting the mathematical relationship to conform to findings reported by Nysmith and Summers. The asymptote provides a conservative limit on the function for large spacings since little is known of the parametric behavior in this region. In addition to its relationship to test data, this criterion follows a logical sequence from integral armor. Where the spacing is zero, the bumper and wall merge into a single member and their combined thickness should logically equal that of integral armor. Furthermore, at the extremes where either the bumper or the wall reduce to zero thickness, it is again logical for their combined thickness to reduce to the integral armor value. Figure E-9 is seen to conform to these expectations. As shown, this criterion assumes the same material for the wall and bumper. To account for different materials, equivalent thicknesses may be based on the general equation for armor thickness that follows.

The required argon duct wall thickness and the required bumper thickness, t_a is determined from the equation of Figure E-9 for an argon duct reliability of 0.99. A typical case would show

$$t_b/t_a = \frac{t_w}{t_a} - 0.25$$

E.6 OVERALL DESIGN RESULTS SUMMARY

Table E-3 summarizes the area and weight calculations for the six selected designs. These weights include heat transfer, vapor fin refluxing, and meteorite armor material, but make no provisions for special structural reinforcement which may be required. The 0.15 pound sq ft horizontal radiator laminate is also included (approximately 21 pounds).

E.7 REFERENCES

E-1. Kays, W. M., and London, A. L., "Compact Heat Exchanger" National Press, 1955.

TABLE E-3. RADIATOR DESIGN WEIGHT TABULATION

Design No.	Total Heat Rejected	T_{in} High	T_{out} High	T_{sink} High	Q High	Area High	T_{out} Low	T_{sink} Low	Q Low	Area Low	Argon Flow lb/sec	Convection Fin Area	Argon Duct Weight	Vapor Fin Weight	Total
1	12.4 kw	$875^{\circ}R$	$702^{\circ}R$	$588^{\circ}R$	6.30	83	$536^{\circ}R$	$380^{\circ}R$	6.10	144 ft^2	0.277	370 ft^2	54 lb	100 lb	154 lb
2	8.6 kw	$835^{\circ}R$	$788^{\circ}R$	$588^{\circ}R$	1.35	14	$536^{\circ}R$	$380^{\circ}R$	7.25	144 ft^2	0.218	260 ft^2	40 lb	70 lb	110 lb

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