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THERMAL INTEGRATION OF ELECTRIC POWER AND LIFE SUPPORT SYSTEMS FOR MANNED SPACE STATIONS

by R. W. Woods and E. P. Erlanson

Prepared by
GENERAL ELECTRIC COMPANY
Philadelphia, Pa.
for Lewis Research Center

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FOREWORD

The research described herein, which was conducted by the General Electric Missile and Space Division, Spacecraft Department, was performed under NASA Contract NAS 3-6478 with Mr. Lloyd W. Ream, Space Power System Division, as Technical Manager. The report was originally issued as General Electric Document Number 66SD4231.

36109

ABSTRACT

Rankine and Brayton Cycle electric power systems using Isotopic and Solar Sources are analyzed and thermally integrated with attitude control, refrigeration, life support and cabin environmental control systems. The resulting system design with the most potential for use with a six-man orbiting station is a combined life support - solar Brayton system weighing 3593 pounds. The electrical power requirements were reduced from eight kilowatts for a non-integrated system to five kilowatts for an integrated system with the use of power system waste thermal energy. The two-man lunar shelter considered had an integrated electrical power reduction of 1.2 kilowatts.

SUMMARY

This is a report on the results of a thermal integration study conducted under NASA-Lewis direction. The purposes of this study were threefold:

a. To explore the use of thermal energy from selected dynamic power systems to replace electrical power requirements in other systems in a six-man earth orbiting space station.

b. To evaluate the effect of thermal integration of space stations and choose a power system with the most potential.

c. To consider thermal integration of a two-man lunar shelter taking into account the effects of the lunar environment.

The orbiting station used in this study was the 260-inch diameter Manned Orbiting Research Laboratory (MORL) with station electrical power requirements of 8 kw average and a 12 kw peak. The two-man Lunar Shelter used was of MOLAB size, but with electrical power requirements of 5 kw.

The primary results are as follows:

a. An integrated solar Brayton system has the greatest potential for use in the six-man orbiting space station.

b. The net station electrical power required can be reduced from 8 KW_e to 5 KW_e by thermal integration.

c. Weight savings in the order of 500 pounds can be realized by using water produced by life support for attitude control.

d. Absorption refrigeration does not provide advantages for rejection of cabin heat.

e. Thermal integration of a two-man lunar shelter is practical and not materially affected by the lunar environment.

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SECTION 1
INTRODUCTION

1.1 PURPOSE

This report presents the results of an analysis and design study of the thermal integration of the systems in a six-man orbiting space station and a two-man lunar shelter. Thermal integration, the concept of using heat from the electrical power system to supply endothermic requirements in the vehicle, was investigated in a previous study. As reported in Reference 1-1, the use of heat from the power system in the closed loop life support system processes of a six-man orbiting space station, reduced electrical power requirements, total system weight, radiator area, and energy source size. This study extended the earlier study, which was limited to specific life support systems, by including cabin environmental control with absorption refrigeration, attitude control and electronic equipment cooling. The applicable concepts were also applied to a two-man lunar shelter. In addition, it was to be determined if the Brayton power system previously studied could be improved by replacing the gas tube radiator with a liquid radiator.

1.2 STUDY TOPICS AND CONSTRAINTS

Figure 1-1 illustrates the vehicle systems that were analyzed. The life support process requirements, items b1 through b6 in Figure 1-1, were unchanged from the previous study, and additional analysis was limited to consideration of the new interfaces created by the systems added to the study. Designs of non-integrated systems were established for all systems prior to the studies of thermal integrations. Comparisons of the integrated and non-integrated systems included the following factors:

a. Reduction in electrical power requirements
b. Reduction in thermal power requirements for the power systems
c. Reduction of storable Attitude Control propellants

d. Hazards and safety considerations

e. Effects of integration on the operation of the power systems, life support systems and subsystems (complexity, reliability, and component failure effects)

f. Control problems and requirements

g. Component design problems

h. System developmental problems

i. Maintainability

The solar cell-battery power system is compared with the integrated system which has the greatest potential for application to the six-man space station using the following criteria:

a. Weight

b. Electrical power requirements

c. Deployed area

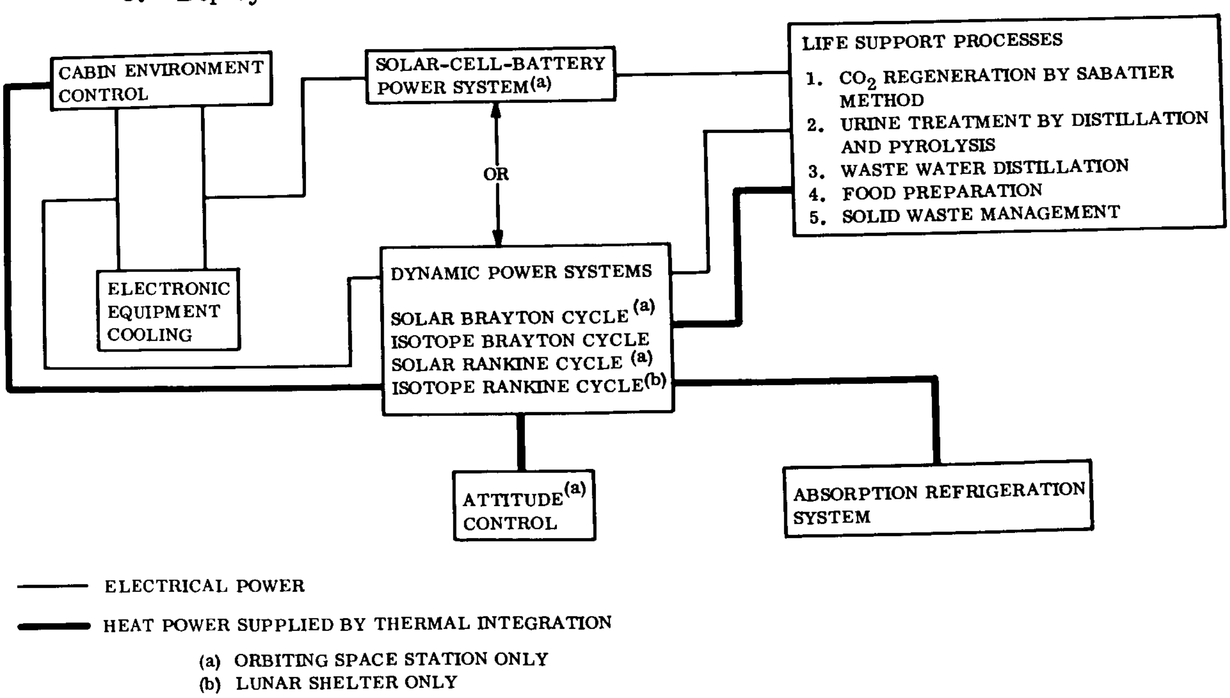


Figure 1-1. Study Topics

The effect of thermal integration of the Power and Life Support System for a two-man Lunar Shelter was determined by extrapolating results from the six-man station, while taking into account the differences in orbital and lunar environment. The lunar shelter model was based on the Molab concept using data from Reference 1-2. The vehicle was to be operable during both lunar day and night. The power systems considered were limited to Isotope Brayton and Isotope Mercury Rankine since the lunar night duration eliminated solar source power systems as likely candidates. A brief comparison of the two integrated power and life support systems is made.

1.3 REFERENCES FOR SECTION 1

1-1. Thermal Integration of Electrical Power and Life Support Systems for Manned Space Stations, Final Report for Contract NAS 3-2799, NASA Report CR-316.

1-2. Apollo Logistics Support System (ALSS) Payloads, D2-36072-3, Final Presentation Report for Contract NAS 8-11411, The Boeing Company, April 1965.

SECTION 2

SUMMARY OF RESULTS

2.1 SIGNIFICANT FINDINGS

a. The integrated life support solar Brayton power system proves to be the best integrated system for the evaluation criteria used. It compares vary favorably with the solar cell battery system, the integrated Brayton system weighing approximately 55% as much, with 90% less projected area and 63% of the net electrical power generated. Table 2-1 summarizes the evaluations.

b. Thermal integration of the power and life support systems reduces the required net electric power from 8 KW_e to approximately 5 KW_e.

c. The attitude control system can benefit through the use of excess water and power system rejection heat in reducing system weight. The reduction is 582 pounds for the Mercury Rankine system and 445 pounds for the Brayton Systems.

d. Absorption refrigeration does not provide advantages in rejecting heat for the applications considered, orbiting station or lunar shelter.

e. Thermal integration is practical for a lunar shelter. The lunar environment does not particularly affect the cycle design of either the power system or the life support system. Consideration of mobility would favor the Mercury Rankine system, since the Brayton cycle radiator is large with respect to present Molab designs.

2.2 OTHER RESULTS

2.2.1 ORBITING SPACE STATION

Results obtained for the Orbiting Space station are as follows:

a. Thermal integration of the three dynamic systems reduces the power output requirements almost identically. The principal difference is that the Brayton cycle requires about 400 watts more heat be delivered to life support than the Rankine cycle because of lower rejection temperatures. A summary of the power requirements for the different systems is given in Table 2-2.

b. The Brayton system should have some control capability. A good method is to provide some excess radiator area and bypass control of the radiator to maintain compressor inlet temperature.

c. Reduction in electric power requirements with thermal integration decreases protruding collector area for the solar systems, which in turn decreases drag forces, support complexity, stowage complexity, and positioning control requirements. Reduction in electrical requirements by thermally integrating for the isotopic systems decreases the amount of isotopic fuel required and the shield and heat exchanger weights (see Table 2-3).

d. Increasing the compressor inlet temperature for the Brayton cycle from 536°R does not result in a reduced system weight, the increased cycle heat requirements outweigh radiator savings.

e. A gas-liquid heat exchanger with liquid tube radiator combination is much lighter than a gas tube radiator for the Brayton cycle. The non-integral concept for the solar Brayton system is 1000 pounds lighter using a liquid radiator, while the integral radiator for the isotope Brayton systems is approximately 350 pounds lighter using a liquid-tube radiator.

f. The water used for attitude control would be used for one or more of the following MORL functions:

1. Fine mode control
2. Backup or standby system
3. Orbit adjust

An attitude control system using steam alone is not probable, since the amount of water available from life support can not fulfill all functions. The supply of additional water from the ground is not weight effective since the specific impulse of the fuel is higher than that of steam. The steam system would have low pressure water storage and a small steam reservoir heated by power system rejected heat.

g. The best method of rejecting cabin heat (made up of life support heat and electronic equipment heat) is to use a space radiator directly. The absorption refrigeration system is competitive in weight when the Brayton system is considered. However it requires additional power, and adds five heat exchangers to the system. In addition, gas fluid separation and necessity of a fluid density gradient in the absorber requires that artificial gravity be provided for the system to operate. The conclusion was reached that a simple radiator and pump was a better system to use, since it fully met the requirements of this application.

Non-Integrated and Integrated station power system summaries are presented in Tables 2-4, 2-5, 2-6 for information.

2.2.2 LUNAR SHELTER RESULTS

Results obtained for the Lunar Shelter are as follows:

a. The improved radiator surface coating, $\alpha_s/\epsilon = .05/.84$, defined for the lunar shelter application, results in an effective sink temperature some $30^{o}R$ less than was used for the orbiting station. The result is that the absorption refrigeration system is less favorable for use than in the orbiting case, and would weigh more on the moon than a simple radiator. Rejection of cabin heat by using vapor compression refrigeration was also studied, and found to impose a weight penalty on the power system to supply the pumping energy required.

b. The Brayton power system radiator configuration chosen is an east-west oriented vertical fin radiating from both sides with the local lunar surface modified by a specularly reflecting covering. Use of this concept is possible because the shelter does not move.

c. Thermal integration of the life support and power systems reduces the net electrical power from 5 kw to approximately 3.8 kw.

d. The integrated isotope Mercury Rankine and Life Support system weighs 2170 pounds, the integrated isotope Brayton and life support system weighs 3611 pounds. The Mercury Rankine system is superior on a weight basis. Table 2-7 presents non-integrated and integrated lunar shelter power systems characteristics.

TABLE 2-1. RESULTS OF SYSTEMS COMPARISONS

PERFORMANCE FACTOR	WEIGHT	PHOTO-VOLTAIC		SOLAR BRAYTON				ISOTOPE BRAYTON				SOLAR MERCURY RANKINE			
		NI		NI		INT		NI		INT		NI		INT	
		DOP	PI	DOP	PI	DOP	PI	DOP	PI	DOP	PI	DOP	PI	DOP	PI
Reduction in electrical power requirements	5	0	0	0	0	10	50	0	0	10	50	0	0	10	50
Reduction in thermal energy	5	0	0	0	0	10	50	0	0	9	45	0	0	8	40
Reduction of storable attitude control propellant	3	0	0	0	0	8	24	0	0	8	24	0	0	10	30
Hazards and safety considerations	5	10	50	9	45	9	45	3	15	2	10	7	35	6	30
Effects of thermal integration on operation of power and life support system	3	10	30	10	30	7	21	10	30	7	21	10	30	8	24
Control problems and requirements	3	10	30	7	21	4	12	9	27	6	18	8	24	5	15
Component problems	2	10	20	5	10	4	8	7	14	6	12	3	6	2	4
Developmental problems	2	10	20	4	8	3	6	5	10	4	8	6	12	5	10
Maintainability	3	10	30	3	9	2	6	5	15	4	12	2	6	1	3
TOTAL PERFORMANCE INDEX			180		123		222		111		200		113		206
Weight, lb		5697		3048											
Deployed Area, ft^2		2180		232*											
Electrical power requirements, KW_e		8.0		5.0											

DOP = Degree of Performance
PI = Performance Index = Weight x DOP

*Area of collector exceeding diameter of vehicle plus projected area of nonintegral radiator.

TABLE 2-2. SUMMARY OF STATION POWER REQUIREMENTS

FUNCTION	SOLAR MERCURY RANKINE		SOLAR BRAYTON		ISOTOPE BRAYTON		PHOTOVOLTAIC
	NON INTEGRATED	INTEGRATED	NON INTEGRATED	INTEGRATED	NON INTEGRATED	INTEGRATED	
Power System Net Electrical Output KW_e	8.0	4.9	8.0	5.0	8.0	5.0	8.0
Power System Waste Heat, KW_t*	57.2	42.5	27.2	19.3	27.6	19.5	-
Average Heat Requirement, KW_t	68.7	50.4	39.5	27.6	40.0	28.0	-
Power Conditioning Penalty, KW_e	0.8	0.5	0.39	0.258	0.39	0.258	1.04
Battery Charging, KW_e	0.46	0.46	0.46	0.46	0.46	0.46	11.27
Power System Pump Work, KW_e	0.36	0.36	0.032	0.025	0.19	0.10	-
Speed Control, KW_e	0.46	0.25	0.44	0.287	0.44	0.287	-
Turbine (Solar Cell) Output Power, KW	11.5	7.9	11.1	7.3	11.3	7.4	20.5
Thermal Power To Life Support, KW_t	-	3.58	-	4.01	-	4.01	-
Electrical Power For Life Support And Equipment Cooling, KW_e	5.1	2.0	5.1	2.1	5.1	2.1	5.1
Net Electric Power Available For Station Experiment Use, KW_e	2.9	2.9	2.9	2.9	2.9	2.9	2.9

*Waste heat refers to the heat removed from the working fluid. For a heat balance, bearing and other losses must be taken into account.

TABLE 2-3. SUMMARY OF STATION SYSTEMS CHARACTERISTICS

FUNCTION	SOLAR MERCURY RANKINE		SOLAR BRAYTON		ISOTOPE BRAYTON		PHOTOVOLTAIC
	NON INTEGRATED	INTEGRATED	NON INTEGRATED	INTEGRATED	NON INTEGRATED	INTEGRATED	
Power System Weight, lb	2774	2336	3797	3048	5294	4310	5694
Life Support System Weight, lb	832	990	832	990	832	990	832
Attitude Control System Weight, lb **	-	-582	-	-445	-	-445	-
Total System Weight, lb	3606	2625	4629	3593	6126	4855	6526
Total Radiator Surface Area, ft^2	800	740	1334	1033	1335	1041	616
Collector Diameter, ft	33.7	28.83	28.5	23.8	-	-	-
Gross Solar Array Area, ft^2	-	-	-	-	-	-	2180
Energy Source Area Projected Beyond MORL, ft^2	620	282	280	75	-	-	2180
Isotope Power Requirement, KW_t	-	-	-	-	40	28	-

** Negative numbers indicate the reduction in stored propellant from using a Steam thrust system.

TABLE 2-4. COMPARISON OF NON-INTEGRATED AND INTEGRATED STATION BRAYTON POWER SYSTEM CHARACTERISTICS

FUNCTION	ISOTOPE		SOLAR	
	Non-Integrated	Integrated	Non-Integrated	Integrated
Net Electric Power to Station Bus, KW_e	8.00	5.0	8.00	5.0
Generator Output, KW_e	9.51	6.09	9.32	6.02
Heat Source Size, KW_t	40.0	28.0	---	---
Collector Diameter, ft	---	---	28.5	23.8
Source Weight, lb	2640	2047	993*	711*
Turbomachinery Weight, lb	201	178	201	178
Heat Rejection System Weight, lb	595	371	668	413
Waste Heat Rejection, KW_t	27.6	19.5	27.2	19.3
Power System Weight, lb (not inc. batteries)	4294	3310	2797	2048
Battery Weight, lb	1000	1000	1000	1000
Life Support System Electric Power, KW_e	5.1	2.1	5.1	2.1
Life Support System Weight, lb	832	990	832	990

*Including absorber weight

TABLE 2-5. COMPARISON OF NON-INTEGRATED AND INTEGRATED SOLAR MERCURY RANKINE SYSTEM CHARACTERISTICS (STATION)

Function	Non Integrated	Integrated
Net Electric Power to Station Bus, KW_e	8.0	4.9
Generator Output, KW_e	9.3	6.2
Average Heat Requirement, KW_t	68.7	50.4
Collector Diameter, ft	33.7	28.83
Collector/Boiler Weight, lb	1168	854
Turbo-Machinery Weight, lb	72	56
Heat Rejection System Weight, lb (Radiator)	134	104
Waste Heat Rejection, KW_t	57.3	42.5
Power System Weight, lb (not including batteries)	1774	1336
Battery Weight, lb	1000	1000
Life Support System Electric Power KW_e	5.1	2.0
Life Support System Weight, lb	832	888

TABLE 2-6. SUMMARY OF PHOTOVOLTAIC POWER SYSTEM CHARACTERISTICS

Net Power to Station Bus	8.00
Average Electric Power to Station, KW_e	9.2
Solar Array Total Output, KW_e	20.5
Solar Array Gross Area, ft^2	2180
Power System Weight (less batteries), lb	3309
Battery Cooling Equipment Weight, lb	50
Battery Weight, lb	2385
Power System Gross Weight, lb	5694
Life Support System Electric Power, KW_e	5.1
Life Support System Weight, lb	832

TABLE 2-7. LUNAR SHELTER ISOTOPE POWER SYSTEM COMPARISON

Function	Mercury Rankine Power System		Brayton Power System	
	Non-Integrated	Integrated	Non-Integrated	Integrated
Electric Power to Station Bus, KW_e	5.00	3.79	5.00	3.81
Generator Output, KW_e	6.07	4.62	5.77	4.46
Isotope Source Size, KW_t	50.3	42.7	29.0	24.0
Isotope Source Weight, lb	1072	989	2105*	1861*
Turbomachinery Weight, lb	55	50	166	146
Heat Rejection System Weight, lb	142	121	367	305
Waste Heat Rejection, KW_t	41.8	35.3	19.5	16.1
Power System Weight, lb	1775	1612	3444	2989
Life Support System Electric Power, watts	1865	703	1865	720
Life Support System Weight, lb	491	515	491	566

*Does not include thermal control

2.3 REFERENCES FOR SECTION 2

2-1 Thermal Integration of Electrical Power and Life Support Systems for Manned Space Stations, Contract NAS 3-2799 Final Report, NASA CR-316.

SECTION 3

DESIGN REQUIREMENTS AND COMPARISONS

3.1 INTRODUCTION

A major purpose of this study was to extend the analyses and designs initiated under study NAS 3-2799 and reported in Reference 3-1. The design guidelines used in that study are, thus, principally applicable to this study. The major revisions were to change the electrical output from 6.24 KW_e to 8.46 KW_e and to add a section describing the lunar shelter portion of the study. The use of consistent guidelines allows comparison of the different power systems on a common basis.

3.2 DESIGN GUIDELINES, MANNED SPACE STATION STUDY

The study utilized the design objectives and physical restraints which are compatible with the Manned Orbital Research Laboratory (MORL) activities sponsored by NASA and being led by the Langley Research Center. This six-man space station consists of a nearly spherical cabin in a 260-inch diameter vehicle. The vehicle would be launched unmanned and would rendezvous with one or more manned vehicles. They would dock, transfer the crew and remain attached to the station. It is assumed that the vehicle will be launched with the primary power supply inoperative.

The station would function in a 250-nm orbit and would receive a resupply vehicle every 180 days. A maximum re-supply weight of 20,000 pounds is postulated. The station would be oriented towards the sun and would be expected to operate for one year. Sources of data include reports prepared by Douglas Aircraft and General Dynamics/Convair as well as NASA supplied data.

3.2.1 DESIGN GUIDELINES, DEFINITIONS

3.2.1.1 Orbit Definition

Altitude:	250 nm, circular
Inclination:	28.7 degrees
Period:	94 minutes
Time in park: (Penumbra)	36 minutes (maximum) 28.2 minutes (minimum)

The rate of precession of the orbit plane due to the oblate earth is such that the period of the variation in eclipse time is estimated to be about 60 days.

3.2.1.2 External Environment

Atmosphere:	Extended ARDC, 1959
Solar Constant:	Nominal 130 watts/ft^2
Effective Space Sink Temperature:	$400^{o}F$
Micrometeoroids:	$W (> M) = \alpha M^{-\beta}$ where: $N (> M)$ = number of particles impacting per unit time of mass M and larger $\alpha = 5.3 \times 10^{-11}$ particles/ft^2-day. $\beta = 1.34$ M = meteoroid mass, grams
Ionizing Radiation:	Negligible for this orbit (insofar as this study is concerned)

3.2.1.3 Station Configuration

The MORL configuration was used and the envelope maintained if possible (see Figure 3-1).

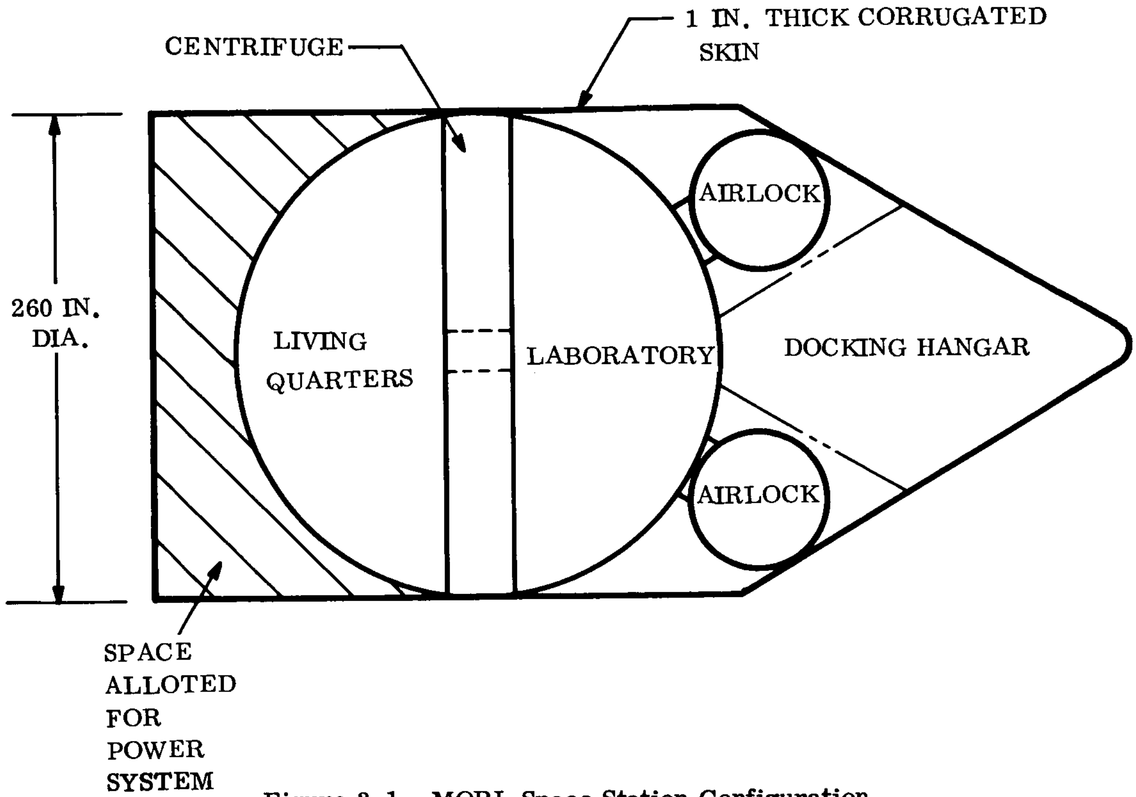


Figure 3-1. MORL Space Station Configuration

3.2.1.4 Vehicle Orientation

The vehicle is oriented with respect to the sun. Solar collector/absorber and photovoltaic performance are based on a time average misorientation of 0.1 degree with respect to the sun.

3.2.1.5 Power Requirements

Average load:	8.46 KW_e	(electrical power to station bus which includes 0.46 KW_e to charge batteries to furnish peak loads)
Minimum emergency power:	1.4 KW_e	
Peak power requirement:	12 KW_e	Up to one hour in any 24-hour time period. Time between peak demand assumed to be at least five times the duration of the peak load

Power conditioning requirements:	50% as 28V dc
	50% as 110/220V ac, 400 cps
Power conditioning penalties:	dc-to-ac, efficiency = 87%
	ac-to-dc, efficiency = 92%
	frequency change, efficiency = 91%

3.2.1.6 Crew Size and Duty Schedule

The crew consists of six men. Each astronaut is assumed to spend the following percentages of his time in the various occupations:

Sleeping	33%
Recreation	18%
Duty	47%
Inside Maintenance	1.4%
Outside Maintenance	0.2%
Rendezvous Operations	0.4%

3.2.1.7 Life Support Requirements

Cabin Atmosphere	
Total Pressure	360 mm Hg
CO_2 Partial Pressure	3.8 mm Hg
Remaining Constituents	50% O_2, 50% N_2
Relative Humidity	50%
Nominal Dry Bulb Temperature	$75 \pm 5^{o}F$
Ventilation Rate	35 cfm/man
Cabin Atmosphere Volume	3470 ft^3
Air Lock Atmosphere Volume	90 ft^3 (total)

3.2.1.8 Redundancy

Crew safety considerations are an important design factor. The power supply will be modularized to provide power in the event of a failure of a unit. For critical components, a minimum of two units capable of independent operation will be provided. Both will operate during normal conditions. The smallest size unit that will be considered will have sufficient capacity to provide 1.4 KW_e, the minimum emergency power requirement. By definition the following components are not redundant:

Mirror-absorber

Isotope Heat Source (except for cooling circuits)

Radiator (except when loss of radiator cooling will affect cooling of the Isotope heat source)

3.2.1.9 Isotope Heat Source

The isotope heat source is Pu-238. Detailed consideration of safety considerations and cooling before system start-up are beyond the scope of the study. Sufficient shielding is provided to assure that a 30 rem per year whole body dose will not be exceeded.

3.2.1.10 Other Considerations

The following guides were derived from the contract work statement.

a. The primary goal is to analyze, study and provide information on thermal integration. Optimized equipment design or advancing the state of the art for particular components is not intended.

b. Of major interest is system design and performance. It is based on (in order of preference):

1. Test data from completed programs
2. Goals for funded development programs
3. Analytical predictions

The analysis and design of components is undertaken only to the extent of determining size and weight where that information is not already available.

c. Redundancy considerations are intended to provide for crew survival. Reliability design considerations are based on sound engineering practices rather than detailed reliability analyses.

d. For the solar power systems, stowage requirements during launch, deployment after launch, and orientation are by definition second order parameters, and do not receive extensive consideration.

e. Consideration of the following items were beyond the scope of the contract:

1. Emergency power systems
2. Power distribution or conditioning
3. Vibration or stress analysis
4. Atmospheric drag
5. Isotope refueling and isotope cost
6. Effects of protruding power system components during docking maneuvers

3.3 DESIGN GUIDELINES, TWO-MAN LUNAR SHELTER

3.3.1 INTRODUCTION

The intent of this portion of the study was to extrapolate the results of the Manned Space Station thermal integration study to determine the effect of the difference in lunar and space environment on thermal integration. Two power systems were to be considered: Isotope

Brayton and Isotope Mercury Rankine. The shelter was to be sized for a two-man crew, such as found in the ALSS payload series, and designed to function during both lunar day and lunar night. The depth of the study was not to include mission analysis or definition, occupants duty cycle, or a detailed analysis of equipment locations and arrangement.

3.3.2 DESIGN GUIDELINES, DEFINITIONS

3.3.2.1 Shelter Location

The shelter location was selected to be near the Lunar Equator. This location caused the shelter to experience the extremes of lunar temperature and is representative of proposed Apollo landing sites.

3.3.2.2 External Environment

The lunar environment was based on the NASA thermal model, consisting of a surface composed of a 9mm thick layer of material A, evenly distributed over an infinite layer thickness of material B. The material properties are:

Material A:	K (thermal conductivity) = 3×10^{-6} cal/cm/sec^2
	ρ (density) = 1.8 g/cm^3
	c_p (specific heat) = 0.2 cal/cm/sec - ^oC
Material B:	K = 4×10^{-3} cal/cm/sec - ^oC
	ρ = 3 g/cm^3
	c_p = 0.2 cal/cm/sec - ^oC

Surface temperature distribution assumes that the lunar equator and its orbit about the earth are both in the place of the ecliptic.

The temperature of an element of the lunar surface while illuminated by the sun is represented by: $T = T_s \cos \frac{1}{4} \varphi \cos \frac{1}{4} \lambda$

where T_s = maximum subsolar point temperature, $700^\circ R^{+34^\circ R}_{-32^\circ R}$

ϕ = lunar latitude relative to subsolar point (0° for this application)

λ = lunar longitude relative to subsolar point

3.3.2.3 Shelter Configuration

With the requirement that the shelter be an Apollo type, the MOLAB vehicle was selected as representative of size and shape.

Shelter volume: 210 ft^3 total, 200 ft^3 free volume

Airlock volume: 80 ft^3

Shelter heat flux distribution, cabin wall heat flux and steady-state cabin temperatures are taken from Reference 3-2 (and reproduced in Figures 3-2, 3-3, and 3-4).

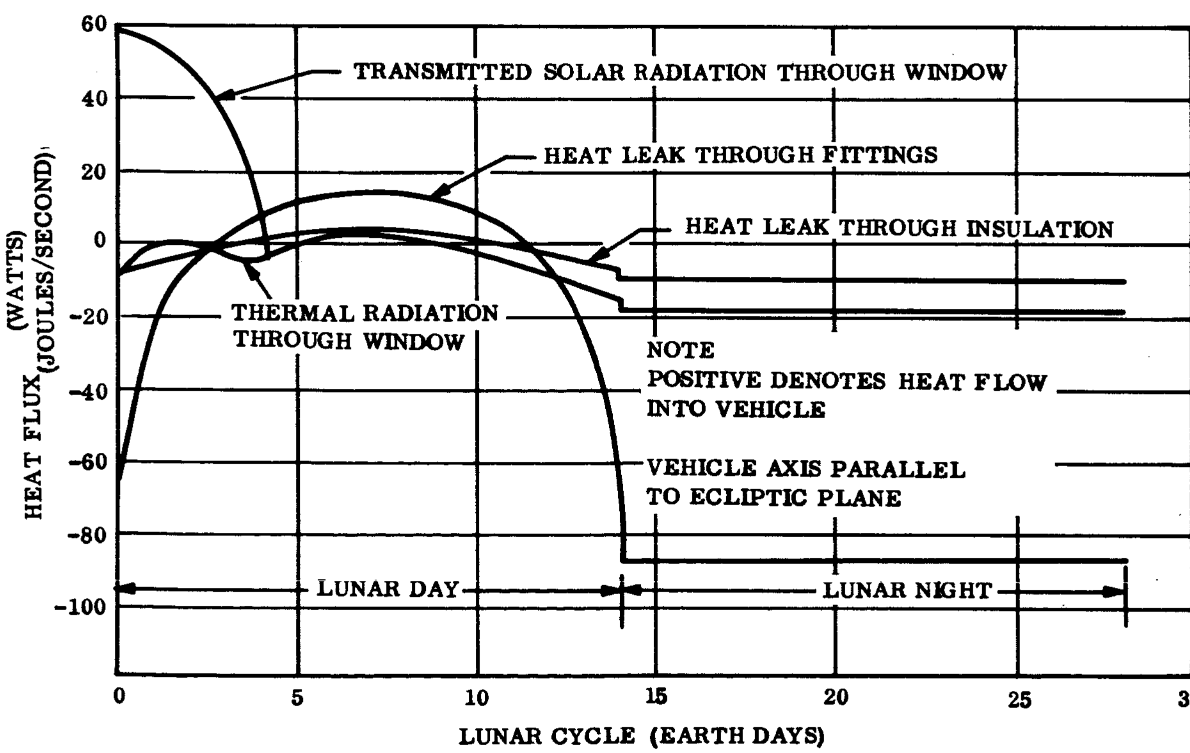


Figure 3-2. Heat Flux Distribution

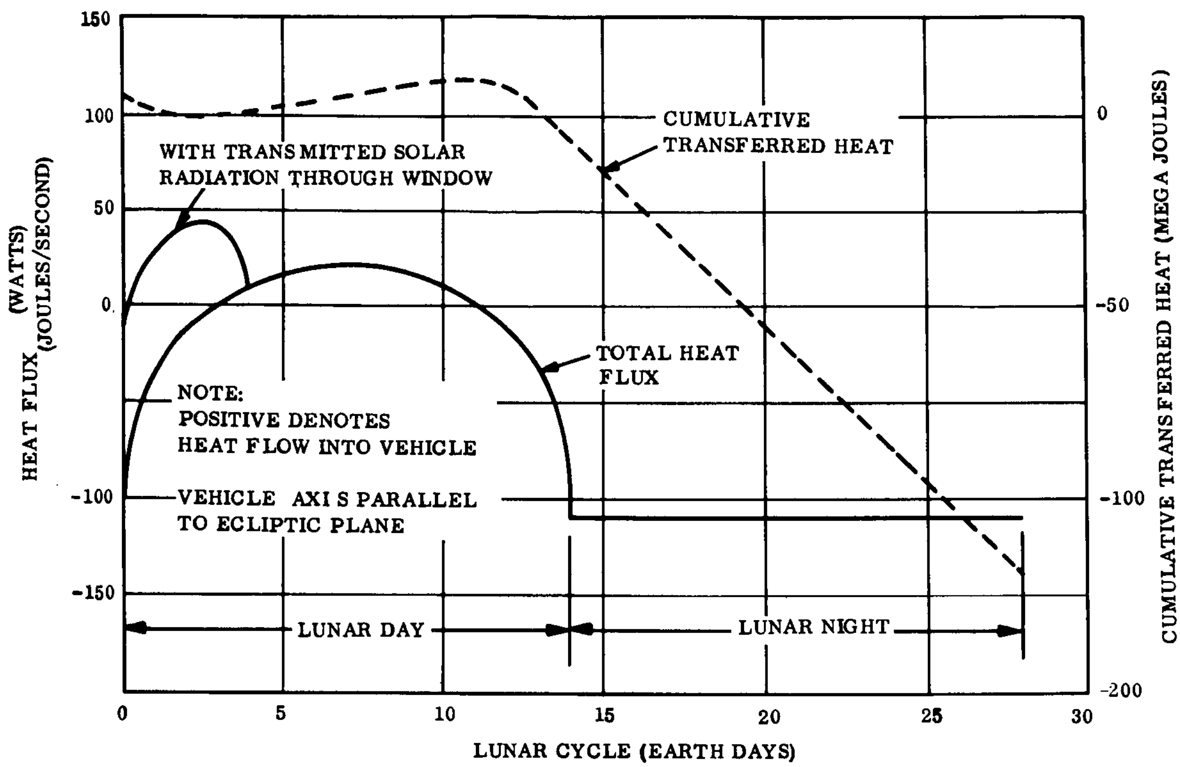


Figure 3-3. Cabin Wall Heat Flux

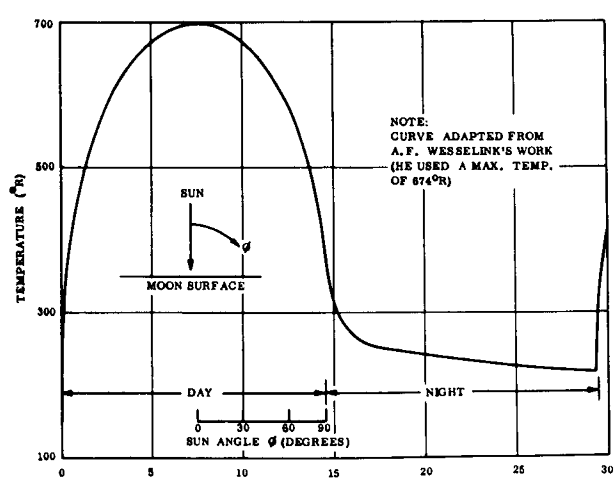


Figure 3-4. Steady-State Cabin Temperatures

3.3.2.4 Power Requirements

Average load:	5 KW_e (electrical power to the shelter bus; does not include frequency conversion penalties)
Minimum Emergency Power:	Not considered
Peak Power Requirements:	Not considered
Power Conditioning Requirements:	50% as 28V dc
	50% as 110/220V ac, 400 cps
Power Conditioning Penalties:	dc-to-ac, efficiency = 87%
	ac-to-dc, efficiency = 92%
	frequency change, efficiency = 91%

3.3.2.5 Crew Size and Duty Schedule

Two astronauts. Technical detail of the study did not include duty cycle.

3.3.2.6 Life Support Requirements

Cabin Atmosphere

Total pressure	360 mm Hg
CO_2 partial pressure	3.8 mm Hg
Remaining constituents	50% O_2, 50% N_2
Relative humidity	50%
Nominal dry bulb temperature	$75 \pm 5^{\circ}F$
Ventilation rate	As required
Cabin atmosphere volume	200 ft^3
Air lock volume	80 ft^3

3.3.2.7 Redundancy Design Philosophy

Crew safety considerations were an important design factor. The power supply was modularized to provide a minimum power level in the event of a failure of a unit. For critical components, a minimum of two units capable of independent operation will be provided. Both will operate during periods of normal occupancy.

3.3.2.8 Isotope Heat Source

a. Isotope Brayton Power System - The heat source is Pu238, with the Isotope cooled by the Brayton cycle working gas, argon.

b. Isotope Mercury Rankine Power System - The heat source is Pu238, with the Isotope heat removed with a circulating fluid such as NaK. The problems and possible methods of Isotope cooling during periods of time when the shelter is not occupied are to be identified.

Shielding requirements are to be the same as defined in Section 3.2.1.9. Detailed consideration of the Isotope package design and the problem of the Isotope going critical are beyond the scope of this study.

3.3.2.9 Occupancy

The shelter must be habitable during both lunar day and lunar night. A minimum dormant condition to protect the on-board electronic and life support equipment and to handle the Isotope cooling requirements are to be identified.

3.4 METHOD OF COMPARISON

The purpose of the evaluation procedure was to provide a method of comparing and selecting integrated space station systems for an orbiting application. The contract work statement listed the pertinent performance areas. Their relative importance, insofar as this study is concerned, was evaluated and a weight factor assigned to each. A range of 0 to 5 was used, and the performance areas with their weight factors are shown in Table 2-1. The value of the weight factor increases with increasing importance of the performance area. Performance areas outside the scope of the study are not listed. An obvious one is cost. The characteristics of each system to be evaluated were compared with each other in each performance area to establish a degree of performance. A range of 0 to 10 was used for degree of performance with the value increasing with better performance. Performance indices were formed by multiplying the degree of performance by the weight factor. The performance indices for each system were summed to obtain its figure of merit. This evaluation and comparison technique, although an orderly process for system selection, does not eliminate the need for engineering judgement. Some of the areas are qualitative in that performance cannot be reduced to a numerical value, within the scope of the study. Judgement was used to evaluate them, and to eliminate insofar as possible, the effects of overlapping performance areas.

The figures of merit in this report represent the considered judgements of the individuals participating in the study and reviewing the results. The tabulated performance indices show the detailed evaluations and comparisons and make it possible for the reader to study the selection process in detail if he so desires. The figures of merit derived in this report are relative since no absolute standard of performance is available. Comparison of these results with other evaluations must be done very carefully.

3.5 STATION SYSTEMS COMPARISON

3.5.1 INTRODUCTION

The performance factors and degrees of performance tabulated in Table 2-2 are discussed in the following paragraphs. The intent of the evaluation was to measure the non-integrated and integrated systems in common and then select one integrated system for comparison with the photovoltaic non-integrated system. The photovoltaic non-integrated system was used as the standard, and all system performances, non-integrated and integrated, were measured against that standard.

Performance Factors selected to measure the effect of thermal integration, which are not applicable to the non-integrated systems evaluation, are removed from consideration by assigning a degree of performance of zero.

3.5.2 REDUCTION IN ELECTRIC POWER REQUIREMENTS (WEIGHT FACTOR = 5)

It was considered that reduction in electric power requirements by thermal integration was a primary goal of the study, hence, a large weight factor was applied to this factor. The integrated dynamic power systems, except for minor differences in pump power, replace endothermic electric power loads equally. The small differences are not significant for ranking one system different than another. The 3 KW_e reduction in station power is a significant fraction of the total non-integrated station load of 8 KW_e.

All non-integrated systems rate a zero since they are the base from which reduction in electric power requirements is accomplished.

3.5.3 REDUCTION IN THERMAL POWER (WEIGHT FACTOR = 5)

Reduction in thermal power by thermal integration is a measure of the decrease in weight, size and cost of the thermal energy source, either Isotope or Solar. The two Brayton systems rate similarly in reducing the thermal energy required by 11.5 and 11.6 KW_t, or approximately 29%; the Solar Brayton scoring slightly better. The Mercury Rankine system was rated as having the least reduction in thermal energy (26%) although the magnitude of the change was 18.3 kw. The precent reduction rather than actual energy reduction was the governing factor. All non-integrated systems rate a degree of performance of zero.

3.5.4 REDUCTION OF STORABLE ATTITUDE CONTROL PROPELLANT (WEIGHT FACTOR = 3)

The benefits of using excess water from the life support system to provide thrust can be evaluated by the amount of fuel weight saved. The amount of excess water supplied is the same for all power systems. The temperature level of waste heat, however, is different for the Brayton and Rankine power systems, and specific impulse (energy per pound of water) is a function of fluid temperature. The specific impulse of the steam was maximized by using the highest possible waste heat temperature. The Mercury Rankine system has the highest temperature and rates the highest. The Brayton systems are of the same temperature and thus rate the same. The non-integrated systems rate a degree of performance of zero, since there was no weight saving considered.

3.5.5 HAZARDS AND SAFETY CONSIDERATIONS (WEIGHT FACTOR = 5)

This performance area represents the risks to crew safety associated with each system. Consequently a high weight factor was assigned.

Hazards include the problems that could result from failure of rotating equipment, radiation from the Isotope source, contamination of the cabin with lethal materials such as mercury vapor, and the consequences of a catastrophic battery failure. It was assumed that the subsystem controls would be designed with the capability of handling a variety of equipment failures by activating alarm circuits or starting backup equipment and that an emergency power source would be included in the station. This type of control system action was not considered in this evaluation. The Photovoltaic System was considered to offer the smallest potential hazard to the crew and station. The Photovoltaic System has only batteries which can be considered as a hazard to the crew and a catastrophic failure with no advance warning is considered very remote. The fact that the Photovoltaic System has more battery capacity than the dynamic systems does not significantly increase the risk. The isotope powered system introduced a radiation hazard not present in the solar powered systems. The risk due to mechanical failure of rotating equipment was considered essentially equal for the dynamic systems. The argon gas working fluid of the Brayton systems is considered less of a hazard than the mercury of the Mercury Rankine power system, although in the non-integrated systems, the chance of introducing the cycle working fluid into the cabin interior is very remote. The radiation hazard was considered more significant than mercury contamination.

3.5.6 EFFECTS OF THERMAL INTEGRATION ON OPERATION OF THE POWER AND LIFE SUPPORT SYSTEMS (WEIGHT FACTOR = 3)

This performance factor considers the operational interdependence imposed on the power systems and the space station systems because of thermal integration. The analysis assumes steady-state operating conditions in order to determine potential advantages. For the integrated stations the power system uses other station systems as a sink for a portion of its rejected heat. Similarly, the integrated endothermic processes depend upon waste heat from the power system for operation. Thus, neither operates independently of the other. Since there must be sufficient controls to provide for startup (and restart) the station systems will be capable of operation away from the steady-state design point and short term transients can be accommodated. Two off-nominal conditions that might exist for periods of hours would be a change in the demand for heat in the endothermic processes or a decrease in the station electrical load. The degree that these off-nominal conditions affect more than one system is a measure of the overall performance in this area.

The non-integrated systems have a minimum of interdependence and rate the highest degree of performance. A decrease in available electrical power could restrict operation of endothermic processes, or a decrease in the endothermic requirements could reduce the electrical demand on the power system in the non-integrated station. Provision for such interaction must be built into the station in any case.

For the integrated stations the Brayton systems are more sensitive to the interactions and rate lower than the Rankine systems. The Brayton cycle efficiency is decreased by any increase in compressor inlet temperature which results from a decreased endothermic process heat demand. The effects on the Rankine cycle are less because of the larger amount of heat being rejected (smaller proportion being removed) and its higher temperature. There is a large amount of waste heat at a relatively high temperature in the Rankine cycle so that decreases in electrical output would not affect the endothermic processes. In the Brayton cycle any decrease in the electrical output lowers the temperature or amount of process heat taken from the power system as it is removed from the total working fluid flow downstream of the recuperator. Thermal integration increases the complexity of startup and the control subsystem for both cycles.

3.5.7 CONTROL PROBLEMS AND REQUIREMENTS (WEIGHT FACTOR = 3)

The control subsystems must function for all operating conditions from the startup (or restart) sequence through the small transients of normal operation to emergency procedures during failure modes. The Photovoltaic System has the least stringent control requirements and, therefore, is rated as best. The battery charge regulators are the major component and do not require any elaborate, fast acting features to prevent equipment damage if a malfunction occurs. The controls for the life support and other endothermic processes are the same as for all non-integrated systems. The dynamic power systems require controls to regulate speed, pressure levels, flow-rates, etc. The solar power systems have more control requirements than the isotope systems because the conditions at the absorber change throughout orbits with shadow periods while the isotope heat source is free of transients (except the slow decay in heat rate). Both must provide overtemperature control to prevent damage if power system thermal demand decreases.

Thermal integration increases the control requirements by introducing additional equipment and equipment interactions into the station. Startup procedures will be more complicated to establish steady-state operation. The power system controls must accommodate transients resulting from changes in heat rejection as well as electrical load transients. The effect of thermal integration on control requirements was considered to be similar for the Brayton and Rankine cycles.

3.5.8 COMPONENT PROBLEMS (WEIGHT FACTOR = 3)

This area of performance is a measure of the component analysis and design effort needed to define flight hardware systems. The Photovoltaic System is superior in that a relatively small number of different components are involved through some of them, e.g., the solar array, contain a large number of identical modules and/or piece parts. All of the solar power systems do impose a solar orientation requirement on the station which adds to the scope of the component engineering task. Though the isotope heat sources will require a large amount of analysis to ensure that the radiation hazard is acceptable, they are considered superior to the solar heat sources because they are compact, massive mechanical designs rather than large area lightweight structures that have to be deployed. In addition, the solar flux control and thermal energy storage components add to the scope of the design task. The Brayton system is considered superior to the Rankine system because it involves no phase changes and the working fluid is an inert material.

Thermal integration increases the component design effort. Additional system analysis and testing will be required to develop data for the design of the thermal integration control components and to determine the additional requirements on the controls of the power systems and endothermic life support processes. Therefore, the integrated stations are rated lower than their non-integrated counterparts.

3.5.9 DEVELOPMENT PROBLEMS (WEIGHT FACTOR = 2)

Since all of the systems represent extension of the demonstrated state-of-the-art, there will be development programs and problems associated with each of them. This performance area represents the scope of the development expected to be needed to produce a flight-ready system. The weight factor is relatively low because the intent of the study is to assess the potential performance of the systems and it is not intended to heavily penalize promising systems on the basis that extensive development will be required.

The Photovoltaic System requires the least development because it is a straightforward extension of concepts, equipment, and practices that have demonstrated long-life performance in space. The life support equipment is common to all systems and, therefore, the degree of development required does not enter into a comparison of the power systems. Development of the concepts and equipment can proceed, and is proceeding, independently. It is logical to consider two elements of the power systems to evaluate their development status. Two power conversion systems, Rankine and Brayton, were studied. Because of the extensive funded development of the Mercury Rankine System in the Sunflower program, it is considered to require less development than the Brayton systems. Components for the Brayton system are being developed for evaluation so this situation may change in the near future. The large solar collectors are considered to require more development risk than the isotope heat sources. The technology of large solar collectors has been under development for a number of years, but the achievement of both optical performance and light weight in a flyable configuration remains a challenge. Isotope heat sources have been demonstrated in space in small sizes so the packaging and radiation safety problems can be solved. The problems of availability and cost do not involve development, so the isotope heat source was considered superior in this performance area.

Thermal integration increases the development required since it is a new, undemonstrated concept. However, the equipment required is conventional and of the same nature as that needed for the Life Support System heat exchangers, valves, pumps, fluid loops, and related heat transfer hardware. The thermal integration interfaces with the Power and Life Support systems can be defined and the interactions simply simulated which will allow the development of the systems to proceed independently. For this reason and because no fundamental advances in the state-of-the-art are required, the degree of performance of the thermally integrated systems in this performance area was considered only slightly lower than the non-integrated systems.

3.5.10 MAINTAINABILITY (WEIGHT FACTOR = 3)

Maintainability takes into consideration the expected requirements and relative ease of maintenance of the several systems, and complications introduced by thermal integration. Since the Life Support equipment is common to all stations, the power systems provide the difference between the various approaches. The non-integrated Photovoltaic System rates the highest in this area because of the minimum maintenance required. Solar arrays have demonstrated reliability and long-life in several space missions to date and can be repaired by replacing modules. Of the dynamic systems, the Isotope Brayton was considered to be the easiest to maintain. Restarting of the turbomachinery can easily be obtained by introducing the working argon gas into the turbines to bring the system up to speed. Single state argon represents less problems in re-start than the two-phase mercury system. The component design philosophy of the power and other systems is assumed to incorporate provision for replacement on a module basis and spares would be readily available, either from a supply carried on board or from resupply. The isotope power system is more compact than the solar powered systems and is more amenable to designing for module replacement. Repair of the argon working fluid Brayton system is considered easier than work on the Mercury system which involves draining and refilling the Mercury loop. Radiators for all systems were designed with a conservative allowance for micro-meteoroid protection and are considered to be virtually maintenance free. The equipment used for thermal integration is relatively simple and does not involve extreme temperatures or speeds. Thus, it should be both maintainable and require relatively little maintenance.

3.5.11 SUMMARY

The Total Performance Index for all the non-integrated and integrated systems is shown in Table 2-1. As can be seen, the system having the highest numerical rating, the Integrated Solar Brayton system, is the system selected to compare with the Photovoltaic non-integrated system using the three criteria listed: weight, deployed area and electrical power requirements. A direct comparison of values generated during this study is possible, and it is evident, that on the basis of this evaluation and the criteria used, the Integrated Solar Brayton System would be selected over the Photovoltaic System.

3.6 LUNAR SHELTER COMPARISONS

The evaluation and selection of lunar shelter power was not a central part of this study. However, a brief system comparison is of interest. Because of the lunar environment, the orientation of the heat rejection surfaces was restricted, and solar powered systems were not included in the analysis due to the length of the day-night cycle. Improved radiator coatings and special configurations provided a higher heat rejection per unit area for the lunar shelter than for the orbiting station. Thus, the analysis shows that the lunar environment does not play a major role in the thermal integration of a lunar shelter. Another advantage of the shelter is the environment which aids liquid vapor separation and similar processes.

The comparisons of the systems for the orbiting station showed that the differences between the integrated and non-integrated systems were significantly greater than the differences among the three integrated systems, or the differences among the four non-integrated systems. Extending these results to the lunar shelter systems indicates that thermal integration would be desirable and feasible. The weight difference between the integrated Isotope Rankine and the integrated Brayton systems is appreciable with the former weighing 1612 pounds and the latter weighing 2989 pounds. The major difference is in the weight of the isotope source where the isotope Brayton fuel capsule-to-gas heat exchangers and radiation shields are larger in size and heavier. The second difference is in the size of radiator area. The integrated Mercury Rankine system requires a 124-square foot horizontal radiator. The integrated Brayton Power system requires a 216-square foot vertical fin with 2590 square feet of the surrounding surface covered with a specular reflector, or 340 square feet of horizontal radiator. The electric power requirements of the two integrated systems are nearly alike. From these criteria, the integrated Isotope Mercury Rankine system offers advantages over the Brayton system for the lunar shelter application.

3.7 REFERENCES FOR SECTION 3

3-1. Final Report, "Thermal Integration of Electrical Power and Life Support Systems for Manned Space Stations," Contract NAS 3-2799, NASA CR-316, November, 1965.

3-2. Apollo Logistics Support Systems Payloads, The Boeing Company, Report D2-36072-3, Volume 2.

SECTION 4
LIFE SUPPORT SYSTEM

4.1 INTRODUCTION

The analysis and design of Life Support processes and equipment was accomplished in the initial Thermal Integration Study Contract. The nature of the present study is to examine the conditioning of cabin air and the methods of rejecting cabin heat, both metabolic and equipment generated. The original Life Support process equipment design remains the same in this study, a summary of the design and thermal integration carried out previously is presented in Appendix A. Included there is the CO_2 processing portion of the air management. Summaries of process equipment weight and power requirements are included in this section for the non-integrated and integrated systems in tabular form.

4.2 DESIGN REQUIREMENTS

4.2.1 OVERALL REQUIREMENTS

The basic design parameters for the cabin environment are as follows:

a. Free volume = 3,479 ft^3
b. Cabin pressure = 7 psia
c. Air composition = 50% O_2, 50% N_2
d. Relative humidity = 50% nominal
e. Cabin temperature = 75 $\pm$ 5°F
f. Recommended air flow = 580 cfm
g. Minimum recommended cabin air temperature = 65°F
h. Recommended humidity control flow = 20 cfm bypass to CO_2 removal system.

Items b, c, d and e are based on Reference 4-2, p. 72. Items f, g, and h are based on Reference 4-3, p. 99.

It is assumed that the heat transfer across the cabin walls is equal to zero, since the cabin is for the most part surrounded by life support equipment and radiators that are near cabin temperature.

4.2.2 CABIN HEAT LOAD

The amount of heat to be removed from the cabin air is composed of two elements, the amount of heat generated in the life support cycle and the amount of heat given off to the air from the electrical equipment located in the cabin. The total cabin air heat load from life support,as shown in Table 4-1, is 5464 Btu/hr. It is assumed that 80% of the heat dissipated by electrical equipment can be removed by liquid-cooled cold plates, and the remaining 20% of the heat load from the electrical equipment removed by the cabin air (Reference 4-3, p. 112). The design guidelines set the net power output of the electrical system at 8 kw. The power available for experiments is then 8 kw less the life support equipment requirements which are:

From Reference 4-1	4.566 KW_e
Air Conditioning	0.526 KW_e (calculated in this section)
Total	5.092 KW_e

TABLE 4-1. CABIN AIR HEAT LOAD (From Reference 4-1)

CO_2 Concentrator	136 Btu/hr
CO_2 Concentrator Heat Leakage	1200
Sabatier Reactor	71
Urine Water Recovery Pyrolysis	24
Wash Water	24
Food Management	235
Waste Management	24
Metabolic Heat	2780
Pumps (Cooling Liquid)	420
Miscellaneous Heat Leakage	550
TOTAL	5464 Btu/hr

Power available for electronic equipment is 2.908 kw and the heat load to the cabin air heat load is 582 watts or 1980 Btu/hr. The total air heat load is then the sum of the life support air load, the air conditioning load, and 20% of the electronic equipment heat load, that is, 9234 Btu/hr.

4.3 NON-INTEGRATED VENTILATION SYSTEM

4.3.1 DESCRIPTION

The conceptual design of the cooling and ventilation system for the Manned Orbital Research Laboratory (MORL) is based on the conventional method of using a fan to circulate air through a heat exchanger and a series of filters for purification. Figure 4-1 is a conceptual layout of a ventilation system equipment location and ducting installed in the MORL cabin.

The outlet side of the fan heat exchanger unit is divided into two main ducts feeding the living quarters and laboratory areas. Each main duct branches into four diffusers near

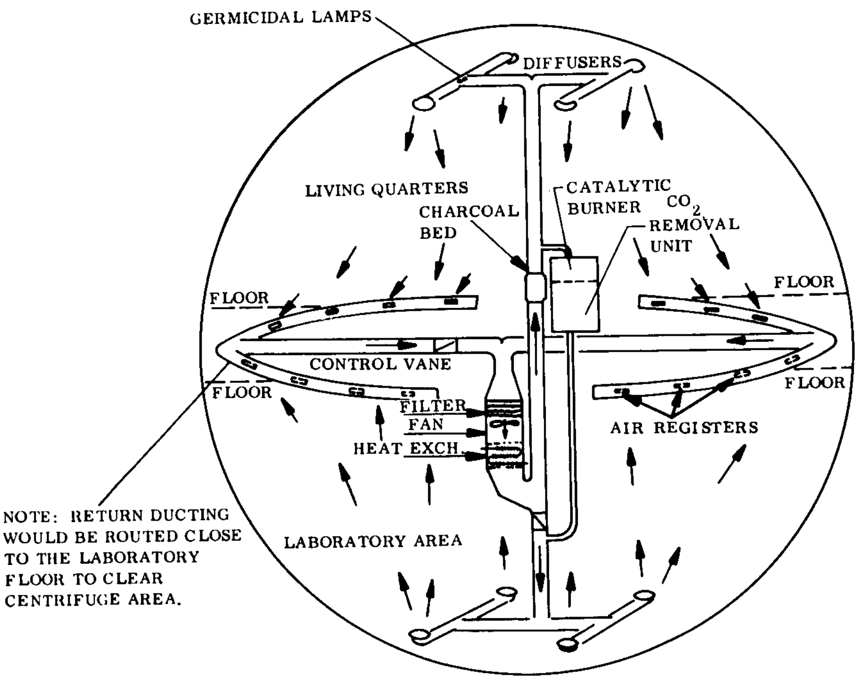


Figure 4-1. MORL Cabin Ventilation System

the ceiling of the hemispherical compartment. The air is blown downward to give some effect of gravity by flowing loose debris toward the floor where the return ducts are located.

4.3.2 DUCT SIZING AND PUMP WORK

The air flow for the MORL cabin was set at 580 cfm (Reference 4-3). This value is consistent with the cabin heat load used in these thermal integration studies.

The air velocity in the ducts has been selected at 1000 feet/minute which is a recommended value for residential applications. The size of ducting for the main branches is then calculated to be 0.29 ft^2, which corresponds to a circular duct approximately 7 inches in diameter, or a rectangular duct 4.5 inches by 10 inches. The ΔP/100 feet of length for such a duct is 0.23 inch H_2O (Reference 4-4). Assuming 43 feet of ducting for a complete air path, $\Delta P = 0.1$ inch H_2O.

ΔP due to miter tees (Reference 4-4, page 290) is that of an equivalent length of duct $L = 65D = 39.5$ feet.

Total equivalent length for six tees is 237 feet

$\Delta P = 273$ ft x $0.23/100 = 0.54$ inch H_2O

ΔP in registers = 0.1 inch H_2O

ΔP in diffusers = 0.2 inch H_2O

Total ΔP in the ducting system = $0.1 + 0.1 + 0.2 + 0.54 = 0.94$ inch H_2O. Since the above values are based on data for air at 14.7 psia, the pressure drop must be corrected for lower air density at 7 psia. $\Delta P = 0.94 \times 7/14.7 = 0.45$ inch H_2O.

The pressure drop in the heat exchanger is assumed to be 0.4 inch H_2O based on data taken from an equivalent size heat exchanger used during a four-man, 30-day simulated space flight conducted by the General Electric Company.

At least two air control vanes must be added in order to be able to balance the air flow between the living quarters and the laboratory. The ΔP for the two vanes, half open, is calculated to be 0.7 inch H_2O.

The ΔP for the filter and charcoal bed is assumed to be 0.2 inch H_2O each. The venilation system total pressure drop is then:

Ducting system	0.45 inch H_2O
Heat exchanger	0.40
Control vanes	0.70
Filter and charcoal bed	0.40
Total pressure drop	1.95 inches H_2O.

The fan power required to circulate 580 cfm at a ΔP of 1.95 inches H_2O is calculated from the air horsepower formula using an efficiency of 50%.

$$\text{fan motor HP} = \frac{\text{work}}{.5} = \frac{\text{cfm x } \Delta P}{6536 \text{ x } .5} = 0.347 \text{ hp} = 260 \text{ watts}$$

4.3.3 CABIN HEAT REMOVAL

A counter flow heat exchanger is used to remove the cabin heat from the air to the space radiator. The liquid transport used is the same glycol solution (30% glycol - 70% water) as was used in Reference 4-1. It is assumed that the heat exchanger coil is made of 3/8-inch outside diameter finned tubing with a cross sectional flow area of 0.000525 square foot and an outside surface area of 0.775 square foot/foot of length. Size and power requirements for a heat exchanger to remove the cabin heat load are based on the following calculated parameters:

a. Mass of air = 126.8 lb/volume

b. Cabin air turnover = $\frac{580 \text{ ft}^3\text{/minute}}{3470 \text{ ft}^3\text{/volume}}$ x 60 min/hr = 10 volumes/hour

c. Air $\Delta T = \frac{Q}{MC_p} = \frac{9.234 \text{ Btu/hr}}{126.8 \times 10 \text{ lb/hr} \times 0.232 \text{ Btu/lb}^o\text{F}} = 31.4^o\text{F}$

The temperature at the exit of the heat exchanger is assumed to be 55°F which is the dew point for 50% relative humidity. The entrance temperature is then 55°F + 31.4°F = 86.4°F. These temperatures are typical of test data from GE cabin simulation tests.

d. Glycol ΔT = 15°F

Lowest temperature is assumed to be 40°F

Highest temperature is assumed to be 55°F

e. Glycol flow requirement with a 15°F ΔT is:

$$M = \frac{Q}{C_p \Delta T} = \frac{9.234 \text{ Btu/hr}}{0.875 \text{ Btu/hr }^o\text{F} \times 15^o\text{F}} = 705 \text{ lb/hr}$$

f. Glycol film coefficient of heat transfer $h = \frac{0.023\, C_p\, G}{(N_{pr})^{2/3} = (N_{re})^{.2}}$

$h = 594 \text{ Btu/hr-ft}^2\ ^o\text{F}$

g. Air film coefficient of heat transfer $h = 11.6 \text{ Btu/hr-ft}^2\ ^o\text{F}$, calculated from data in Reference 4-5, page H-146.

h. Log mean temperature difference $\Delta TM = \frac{(86.4 - 55) - (55 - 40)}{\ln \frac{86.4 - 55}{55 - 40}} = 22.3^o$

i. Average overall coefficient of heat transfer $Q = UA \times \Delta TM$

$$\frac{Q}{\Delta TM} = UA = \frac{9234 \text{ Btu/hr}}{22.3^{o}F} = 413 \text{ Btu/hr } ^{o}F$$

j. To calculate heat transfer surface required:

$$\frac{1}{UA} = \frac{1}{Ah \text{ (air)}} + \frac{1}{Ah \text{ (glycol)}} \quad A \text{ (air)} = 41.7 \text{ ft}^2$$

$$\text{Feet of finned tubing} = \frac{41.7 \text{ ft}^2}{0.775 \text{ ft}^2/\text{ft}} = 54 \text{ ft}$$

k. The Reynolds number and tube size determine the glycol pressure drop per foot of tubing. To obtain pumping power, the total pressure drop is determined.

ΔP in the heat exchanger = 0.36 psi/ft x 54 ft = 19.4 psi

ΔP in the radiator (from Reference 4-1, Figure 4-20,) = 17.2 psi

ΔP in interconnecting lines and fittings, total equivalent length

= 50 ft, P = 50 x 0.36 = 18.0 psi
Total ΔP = 54.6 psi

$$\text{Glycol pump power HP} = \frac{Q\,\Delta P \times 144}{33{,}000 \times .5} = 0.088 \text{ HP} = 66 \text{ watts}$$

assuming an overall pump/motor efficiency of 0.5

4.3.4 AIR PURIFICATION

Air purification involves the removal of contaminants produced by the human body, compartment components and supplies. The oxygen recovery system discussed in Appendix A provides for an effective removal of carbon dioxide. Orders and some trace contaminants are removed by the charcoal bed. However, a catalytic burner should be used to ensure the removal by oxidation of CO, H_2, CH_4 and O_3. The catalyst considered for this purpose in this study is palladium electrically heated to 700°F. It is estimated that 150 watts of electrical power is required for the operation of this unit. Bacteria floating or suspended in the cabin air are killed by the ultraviolet energy of germicidal lamps located in the air duct. It is estimated that two standard G25T8 bulbs would be required, adding 50 watts to the electrical load. However, by the use of properly designed lamp housings with ultraviolet reflective internal coating and phosphorescent front panels, most of the ultraviolet energy could be transformed into useable light for cabin illumination.

4.3.5 NON-INTEGRATED SYSTEM SUMMARY

The total electrical power required for cabin air conditioning and ventilation is:

Fan power	= 260 watts
Glycol pump power	= 66 watts
Catalytic burner	= 150 watts
Germicidal lamp	= 50 watts
Total	= 526 watts

The total Life Support System can now be summarized, combining the processes and the ventilation system. Table 4-2 presents estimated equipment weights and electrical power requirements for a complete system.

4.4 INTEGRATED VENTILATION SYSTEM

4.4.1 DESCRIPTION

The conceptual design of an integrated cooling and ventilation system for the MORL is basically the same as that described in Section 4.3. It is assumed the cabin heat load is

TABLE 4-2. POWER REQUIREMENTS AND WEIGHTS ESTIMATE FOR NON-INTEGRATED LIFE SUPPORT SUBSYSTEMS (STATION)

System	Weight (Lbs)	Process Power (Watts)	Suppor Power (Watts)
Oxygen Recovery			
CO_2 Recovery Canisters	120	1867	40
Electrolysis Cells	100	1000	-
Pyrolyzation unit	20	130	-
CO_2 Accumulator Pump	7	-	100
Accumulator, Reactor, Condenser	26	-	-
System Cooling	100	-	70
Heat Leakage (Loss)	-	-	42
Miscellaneous Hardware	12	-	7
Urine Water Recovery	43	303	7
Wash Water Recovery	35	442	7
Food Management	19	69	-
Solid Waste Treatment	29	320	7
Heat Leakage (Loss)	-	-	95
Cooling System	106	-	60
Miscellaneous Hardware	5	-	-
Cabin Cooling (Air Conditioning and Ventilation)	210	150	376
	832	4281	811

TOTAL ELECTRICAL POWER REQUIREMENT = 5,092 watts the same as for the non-integrated system even though some variations will undoubtedly exist due to the presence of additional equipment and losses from the hot fluid lines.

4.4.2 ABSORPTION REFRIGERATION METHOD OF HEAT REJECTION

The analysis of the refrigeration cycle in Section 7 used a glycol Δt of $10^{\circ}F$ in the cabin heat rejection heat exchanger. The value used in the ventilation study was $15^{\circ}F$ (in conjunction with a space radiator).

If absorption refrigeration were used, this change would cause an increase of 50% in the glycol mass flow rate with a consequent increase in pumping power required, unless equipment tubing sizes and lengths are changed. With an increase in glycol tube diameter to 3/8 inch I.D. the total pumping power would be 55 watts, at the expense of additional glycol inventory of about one pound.

4.4.3 AIR PURIFICATION, INTEGRATED DESIGN

The only component in the cooling and ventilation system which lends itself to thermal integration with the electrical generating power system is the catalytic burner. The small amount of air that is by-passed to the catalytic bed can be heated to approximately $600^{\circ}F$ in the Rankine cycle and to approximately $400^{\circ}F$ in the Brayton cycle. These temperatures are high enough to cause the reduction of all commonly known trace contaminant gases except methane which is not affected at temperatures below $425^{\circ}F$ regardless of the catalysts that, at present, could be used. Published data on catalyst characteristics, operating temperatures, and reaction rates is sparse and presents a selection problem. The best information shows that for a cabin environment similar to that of the MORL vehicle, as much as 80% of the methane (CH_4) gas can be oxidized at $600^{\circ}F$ over a palladium catalyst. On this basis it is concluded that no additional electrical power input is required for air purification in the Rankine cycle. In the Brayton cycle, additional electric power will be required to bring the catalyst from $400^{\circ}F$ to the $600^{\circ}F$ range. A conservative estimate for the additional power needed is 50 watts. This is based on the assumption that O_3, CO, H_2, and carbon compounds are reduced with a catalyst operating at low temperatures up to 400°F while only the methane gas is reduced over a high-temperature catalyst.

4.4.4 INTEGRATED SYSTEM SUMMARY

The total integrated electrical power requirement for a thermally integrated cabin cooling and ventiliation design can be summarized as follows:

RANKINE CYCLE

Fan Power	260 watts
Glycol Pump	66
Germicidal Lamp	50
Catalytic Burner	0
Total	376 watts

BRAYTON CYCLE

Fan Power	260 watts
Glycol Pump	66
Germicidal Lamp	50
Catalytic Burner	50
Total	426 watts

Table 4-3 presents estimated equipment weights, process electrical power and process thermal power requirements for the complete six-man Life Support System. On the basis of a systems evaluation the absorption refrigeration system was discarded as a method of rejecting cabin heat to space, the ventilation system reflects this by using the non-integrated weights and pump power values.

TABLE 4-3. POWER REQUIREMENTS AND WEIGHTS ESTIMATE FOR INTEGRATED LIFE SUPPORT SUBSYSTEMS

System	Weight (Lbs)	Process Electrical Power (Watts) (Rankine/Brayton)	Process Thermal Power (Watts)
Oxygen Recovery			
CO_2 Recovery Canisters	120/222	40	1867/2360
Canister Heat Leakage			351
Electrolysis Cells	100	1000	-
Pyrolyzation Unit	20	130	-
CO_2 Accumulator Pump	7	100	-
Accumulator, Reactor, Condenser	26	-	-
System Cooling	100	70	-
Heat Leakage (Loss)	-	-	42
Miscellaneous Hardware	12	7	-
Fluid Heating System	40	93	-
Urine Water Recovery	43	67	243
Wash Water Recovery	35	7	442
Food Management	19	-	69
Solid Waste Treatment	29	7	320
Heat Leakage (Loss)	-	-	95
Cooling System	106	40	-
Miscellaneous Hardware	5	-	-
Cabin Cooling (Air Conditioning and Ventilation)	210	376/426	150/100
Fluid Heating System	16	40	-
	888/990	1977/2027	3579/4022

4.5 REFERENCES FOR SECTION 4

4-1. Hanson, K. L., Thermal Integration of Electrical Power and Life Support Systems for Manned Space Station, General Electric, MSD Final Report for Contract NAS 3-2799, NASA CR-316.

4-2. Report on a System Comparison and Selection Study of a Manned Orbital Research Laboratory, Douglas Aircraft Company Report Number 44619, Septmeber, 1963.

4-3. Report on a System Comparison and Selection Study of Manned Orbital Research Laboratory, Douglas Aircraft Company Report Number SM44607, September, 1963.

4-4. Air Conditioning and Refrigeration Data Book, The American Society of Refrigerating Engineers, New York, N.Y., 1955 Edition.

4-5. Aerospace Applied Thermodynamics Manual, Society of Automotive Engineers, Technical Division, New York, N.Y., Revised Edition, January, 1962.

4-6. Giedt, W.H., "Principles of Engineering Heat Transfer," Van Nostrand Company, Princeton, N.J., August 1957.

4-7. Technical Bulletins 1504-4, 1507-1, Mine Safety Appliance Company, Pittsburgh, Pa.

4-8. Analytical Methods for Space Vehicle Atmospheric Control Processes, ASD-TR-61-162, Part II, Wright-Patterson Air Force Base, Ohio, November, 1962.

SECTION 5
ELECTRONIC EQUIPMENT COOLING

5.1 MORL SYSTEM DESCRIPTION

A design goal for the fabrication, installation, interconnection, environmental control, electrical isolation, and maintainability of electronic and electromechanical equipment, is the development of a single, standardized package which would reduce fabrication costs, simplify installation and maintenance procedures, and increase interchangeability. While this is difficult to realize, a standardized form of installation, providing a reliable method of environmental protection and temperature control is entirely feasible. A basic installation proposed for the MORL electronic equipment (Reference 5-1) consists of a cabinet assembly housing a liquid coolant cold plate which serves as the mounting surface for equipment. These cabinets are arranged in equipment bays located on the outer periphery of the laboratory interior. The arrangement utilizes the equipment to provide additional radiation shielding for the crew. Figure 5-1 shows a typical installation for modularized packages. With the external door closed, the cabinet is sealed to protect against entry of foreign material and to partially contain noxious gases that may possibly be generated as the result of an internal failure of electronic components within the cabinet. The outside layer of modules and its cold plate are attached to the inside of the cabinet door. With the door open, all modules are easily accessible for test or replacement. The coolant lines and interconnecting wiring are grouped on one side of the cabinet to permit the unit to be rotated away from the cabin wall for easy access.

Numerous approaches are possible for the detailed design of the cold plate. The one shown would incorporate coolant tubes and provide the structural support needed during boost and other phases. The coolant lines are sized to provide heat removal capability for the assumed load. Although not described in Reference 5-1, it is assumed for the purposes of this study that variable cooling capacity to handle maximum and minimum cooling loads in a module or group of modules would be accomplished by a self-contained, thermal-bulb, gas-actuated valve which would modulate to regulate flow as a function of coolant

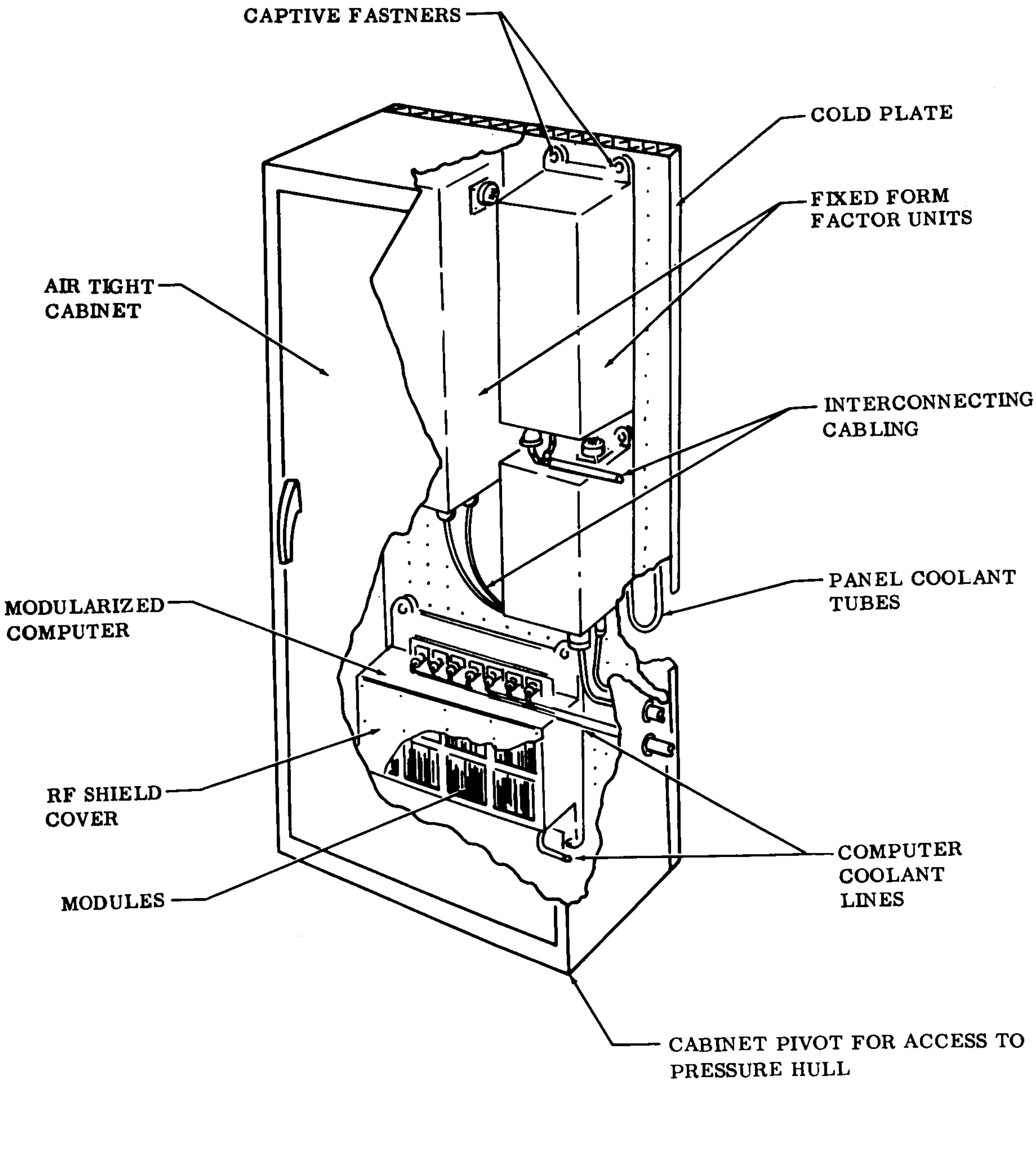


Figure 5-1. Electronic Equipment Module

temperature. For purposes of calculating the flow area of a typical cold plate, the number of flow passages and pressure drop, it is assumed that 100% efficiency of heat transfer is obtained, and that the temperature of the equipment, and the heat transferred to the cooling fluid results in a temperature rise in the cooling fluid of 15°F. Components which are most sensitive to temperature, or which are large heat generators, would be mounted near the coolant fluid entrance to the cold plate. The fluid is circulated through a radiator by a pump, with a bypass line to maintain a minimum flow through the radiator and pump loop. The fluid is distributed to the cabinets from header lines and is collected in a return header to be transported to the radiator for cooling. To determine radiator size, it is assumed that equipment cooling loop would have a separate radiator, although in the actual vehicle it could be combined with the Life Support System radiator since they operate at the same temperature. Figure 5-2 shows a schematic of the system.

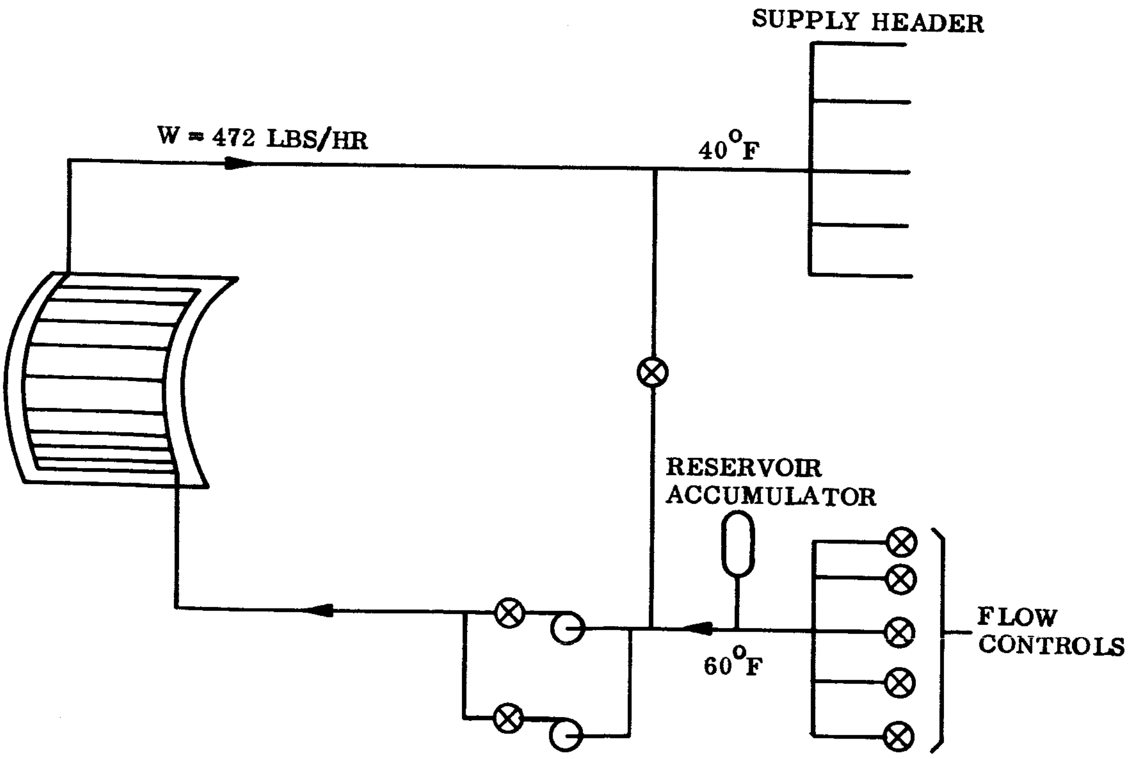


Figure 5-2. Electronic Cooling System (Non-Integrated System), Schematic

5.1.1 SUMMARY OF REQUIREMENTS

The power systems nominal power output of 8.0 KW_e provides 5.092 KW_e for the Life Support processes and the remaining 2.908 KW_e is assumed to operate the onboard electronic equipment. It is assumed that 80 percent of this load, or 2.33 KW_e must be handled by the cooling loop. The minimum coolant temperature is $40^\circ F$, space sink temperature is $400^\circ R$, and an integral radiator is assumed. The fluid is an aqueous solution of propylene glycol and 70% water by weight. A temperature rise of $15^\circ F$ in the cooling fluid is assumed.

5.2 NON-INTEGRATED SYSTEM

To determine the pumping power required for the equipment cooling loop it is assumed that the cold plates have a circular shaped flow passage with a diameter of 0.125 inch. It will also be assumed that the cold plate cooling is accomplished with 20 tubes, each 6 feet long. The tubes are arranged on the cold plates in sufficient number and with spacing to maintain the electronic components within an allowable operating temperature range.

$$Q = 2.33\ KW_e \times 3413\ \text{Btu/KW-hr} = 7960\ \text{Btu/hr}$$

$$Q = W\, c_p\, \Delta T, \quad W = \frac{Q}{c_p\, \Delta T}$$

$$c_p = 0.875$$

$$\Delta T = 15^\circ F$$

$$W = \frac{7960}{(0.875)\,(15)} = 606\ \text{lb/hr} = 10.1\ \text{lb/min} = 0.1685\ \text{lb/sec}$$

Assume 20 Flowtubes, Flowper tube $= \dfrac{0.1685}{20} = 0.00802$ lb/sec

Diameter = 0.125 in. = 0.0104 ft

$$\text{Flow area} = \frac{\pi D^2}{4} = (0.7854)\,(0.0104)^2 = 8.5 \times 10^{-5}\ ft^2$$

$$\text{Velocity} = \frac{W}{A\,\rho} = \frac{0.00802}{(8.5 \times 10^{-5})\ (65.2)} = 1.45 \text{ ft/sec}$$

$$R_e = \frac{DV\rho}{\mu} = \frac{(0.0104)\ (1.45)\ (2.025)}{6.3 \times 10^{-5}}$$

$$R_e = 486$$

$$f = \frac{64}{R_e} = \frac{64}{486} = 0.1315$$

$$\Delta P_{\substack{\text{Cold}\\ \text{Plate}}} = \frac{f\,\rho V^2 L}{2\,D}$$

$f = 0.1315$

$\rho = 2.025 = \frac{65.2}{32.2}$

$V = 1.45$ FPS

$L = 6$ ft

$D = 0.0104$ ft

$$= \frac{(.1315)\ (2.025)\ (1.45)^2\ (6)}{(2)\ (0.0104)}$$

$$= 162 \text{ lb/ft}^2 = 1.13 \text{ lb/in.}^2$$

Assume that the supply and return lines and manifolds consist of 50 feet of 3/8-inch inside diam diameter tubing.

$W = 0.1685$ lb/sec

Dia $= 0.375$ in. $= 0.03125$ ft

$$\text{Flow area} = \frac{\pi}{4} D^2 = (0.7854)\ (0.03125)^2 = 76.8 \times 10^{-5} \text{ ft}^2$$

$$\text{Velocity} = \frac{0.1685}{(65.2)\ (76.8 \times 10^{-5})} = 3.37 \text{ ft/sec}$$

$$R_e = \frac{DV\rho}{\mu} = \frac{(0.03125)\ (3.37)\ (2.025)}{6.3 \times 10^{-5}}$$

$$R_e = 3390$$

Assume Flow is turbulent

$$f = \frac{0.316}{R_e^{\ 0.25}} = \frac{0.316}{3390^{0.25}} = 0.0416$$

$$\Delta P = \frac{(0.416)\ (2.025)\ (3.37)^2\ (50)}{(2)\ (0.03125)}$$

$$= 765\ \text{lb/ft}^2 = 5.3\ \text{lb/in.}^2$$

From data presented for low temperature radiators in Reference 5-3, the electronic equipment cooling loop radiator is estimated to weigh 83 pounds, have an area of 190 ft^2 and a ΔP of 16.5 psi for a load of 7960 Btu/hr.

Total pressure drop in the cooling system is:

$$16.5 + 1.13 + 5.3 = 22.93\ \text{psi}$$

$$\text{Pump Power} = \frac{W\ \Delta P}{550\ (e)}$$

Assume a pump hydraulic/electrical efficiency of 50%

$$\text{Power} = \frac{\dfrac{10.1\ \text{lb/min}}{65.2\ \text{lb/ft}^2} \times \dfrac{1}{60}}{(550)\ (0.5)}\ (22.93)\ (144)$$

$$= 0.31\ \text{HP}$$

$$= 23\ \text{watts}$$

5.3 THERMALLY INTEGRATED SYSTEM

Thermal integration of the electronic equipment cooling loop involves the replacement of the radiator with the absorption refrigeration system evaporator heat exchanger. The absorption refrigeration system provides coolant at 45°F and assumes a fluid return temperature of 55°F. Because the average coolant temperature (50°F) is not the same as in the non-integrated system, the coolant flow rate must be increased to accomplish the same cooling. For purposes of a direct comparison, it will be assumed that the cooling loop tubing and manifolds, as well as the cold plates, do not change in size from the non-integrated system design. The weights of the cold plates, manifolding and piping can be neglected since they are constant for both systems.

5.3.1 INTEGRATED SYSTEM PRESSURE DROP AND PUMP POWER

$$Q = 7960 \text{ Btu/hr}$$

$$c_p = 0.875$$

$$\Delta T = (55^{\circ}F - 45^{\circ}F) = 10^{\circ}F$$

$$W = \frac{Q}{c_p \ \Delta T} = \frac{7960}{(0.875)\ (10)} = 910 \text{ lb/hr} = 15.2 \text{ lb/min.}$$

For 20 tubes, the Flow per tube is

$$\frac{152/60}{20} = 0.0126 \text{ lb/sec tube}$$

Tube Diameter = 0.0104 ft

Flow Area = 8.5×10^{-5} ft^2

$$\text{Velocity} = \frac{0.0126}{(8.5 \times 10^{-5})\ (65.2)} = 2.27 \text{ ft/sec}$$

$$R_e = \frac{(0.0104)\ (2.27)\ (2.025)}{6.3 \times 10^{-5}} = 760$$

$$f = \frac{64}{R_e} = \frac{64}{760} = 0.0843$$

$$\Delta P = \frac{(0.0843)\ (2.025)\ (2.27)^2\ (6)}{(2)\ (0.0104)}$$

$$\Delta P = 254\ \text{lb/ft}^2 = 1.76\ \text{lb/in.}^2$$

Manifolds and supply and return lines consist of 50 feet of 3/8-inch I.D. tubing

Flow = 0.253 lb/sec

Dia = 0.375 in. = 0.03125 ft

Flow area = 76.8×10^{-5} ft^2

$$\text{Velocity} = \frac{0.253}{(76.8 \times 10^{-5})\ (65.2)} = 5\ \text{ft/sec}$$

$$f = 0.3$$

$$R_e = \frac{(0.03125)\ (5)\ (2.025)}{6.3 \times 10^{-5}} = 5085$$

$$f = \frac{0.316}{5085^{0.25}} = 0.037$$

$$\Delta P = \frac{(0.037)\ (2.025)\ (5)^2\ (50)}{(2)\ 0.03125)}$$

$$\Delta P_{\text{Tubing}} = 1495\ \text{lb/ft}^2 = 10.4\ \text{psi}$$

Total pressure drop in the cold plates and tubing and manifolds is 10.4 + 1.76 = 11.16 psi.

$$\text{Pump Power} = \frac{\dfrac{15.2}{65.2} \times \dfrac{1}{60}}{(550)(0.5)} (11.16)(144)$$

$$\text{Pump Power} = 0.0227\ H_P = 17 \text{ watts}$$

Total pumping power required for the cooling circuit equals 17 watts plus part of the pump power required to circulate coolant through the evaporator of the absorption refrigeration system. From Section 7, the evaporator pumping power for the total coolant flow through the evaporator only is 37 watts, of which approximately half, or 18 watts, can be charged to the equipment cooling loop. Total power required to circulate the equipment coolant is then 17 watts plus 18 watts or 35 watts. From Section 7, the weight of the absorption refrigeration system is 303.5 pounds including the required system radiators. Taking a ratio on the basis of heat load, approximately 140 pounds can be charged to the equipment cooling circuit. Similarily, the electric power used in the absorption cycle, not including that which is required for coolant circulation which is already accounted for, can be prorated and the amount charged to the equipment cooling is 15 watts. Total electric power used by the absorption refrigeration system which can be charged to the electronic equipment cooling, including coolant circulation, is 50 watts.

5.4 DISCUSSION OF RESULTS

A simple and straight forward method of providing cooling capability for the thermal temperature control of electronic equipment is a fluid loop consisting of a pump, radiator and cold plates on which to mount electronic components. The radiator can be thought of as a reservoir of cold fluid from which coolant can be drawn and distributed to the various cabinets containing the equipment to be temperature controlled.

It has been assumed that the final hardware design would include a means of individual control of the temperature of components either by the design of the cold plate size and flow passage size, or by individual control valves modulating flow to a cabinet or group of components. Under extreme conditions, the radiator with no load could get quite cold with the possibility of freeze-up occurring. An analysis of this situation was considered outside the scope of this study.

The operating temperature range of the coolant fluid was selected to be compatible with the fluid temperatures used to control cabin air temperature and the humidity of the cabin air. It is probable that one radiator loop would be used for the low temperature cooling (typically in the 40°F to 60°F range) although in this study they are considered to be separate.

The cold plate design is considered representative of a workable method of equipment cooling. The gross cooling load was used in determining an overall pumping requirement and radiator size. The determination of contact resistance to heat conductance at the equipment mounting points, the method of equipment mounting, arrangement of components and variations in temperature caused by cyclic operations, were not considered.

A direct comparison of the absorption refrigeration system for equipment cooling and a liquid loop radiator yields the information given in Table 5-1.

Table 5-1. Comparison of Liquid Loop Coolant and Absorption Refrigeration Cooling

	Radiator Loop	Absorption Ref. System
Cooling Load, Btu/hr	7960	7960
Coolant Fluid	Glycol/water	Glycol/water
Coolant Flow Rate, lb/hr	606	910*
Temperature rise in cold plate, $^{\circ}$F	15	10
Pump power to circulate coolant fluid, watts	23	35*
Equipment Cooling Radiator Area ft^2	190	257*
Equipment Cooling Radiator Weight, lb	83	140*
System electrical power, watts	23	48*

* Pro-rated based on ratio of equipment coolant flow to total coolant flow through absorbtion refrigeration system evaporator.

5.5 REFERENCES FOR SECTION 5

5-1. Report on a System Comparison and Selection Study of a Manned Orbital Research Laboratory, Volume 1 Technical Summary, Appendix 12, Flight Electronics. The Douglas Aircraft Company, Inc., Report Number SM-44608.

5-2. Mark, M., Cold Plate Design for Airborne Electronic Equipment, IRE Transactions on Aeronautical and Navigational Electronics Equipment, March, 1958.

5-3. Hanson, K.L., "Thermal Integration of Electric Power and Life Support Systems for Manned Space Stations," NASA CR-316, General Electric Co., Nov. 1965 (Final Report for Contract NAS 3-2799).

5.6 LIST OF SYMBOLS

W = coolant weight flow, lb/min.

Q = rate of coolant heat transfer, Btu/hr

c_p = specific heat of coolant, Btu/lb (F)

μ = viscosity of coolant, lb sec/ft^2

f = friction factor

R_e = Reynolds number

ΔT = temperature rise of coolant, F

ΔP = pressure drop, lb/ft^2

ρ = density of coolant, lb/ft^3

V = velocity of coolant, ft/sec

D = diameter of tubing, feet

SECTION 6
ATTITUDE CONTROL

The use of waste water and waste heat from life support and power systems, respectively, to provide attitude control impulse (as steam) is to be considered as a possible improvement of the MORL vehicle. This section will first describe the MORL system, then a steam system, and finally discuss integration possibilities.

6.1 MORL SYSTEM DESCRIPTION

6.1.1 INTRODUCTION

The MORL Attitude Control System configuration and equipment selection is described in Reference 6-1. This subsection summarizes this information, at the same time modifying specific items to conform with present ground rules from Reference 6-2. The most significant changes are: the new altitude of 250 nautical miles versus 205 nautical miles of the Douglas Study (Reference 6-1), and the consideration of the effect of the different power systems on attitude control requirements. References 6-3 and 6-4 were used as aids in determining these changes.

6.1.2 CONTROL SYSTEM

The propulsion/reaction control system consists of four self-contained modules located at Station 1699 and Station 2179 at 180-degrees spacing (see Figure 6-1). These modules contain positive expulsion bipropellant storage tanks, hot gas expulsion engines, pressurization and control components. The engines provide thrusting in the following four directions at each module:

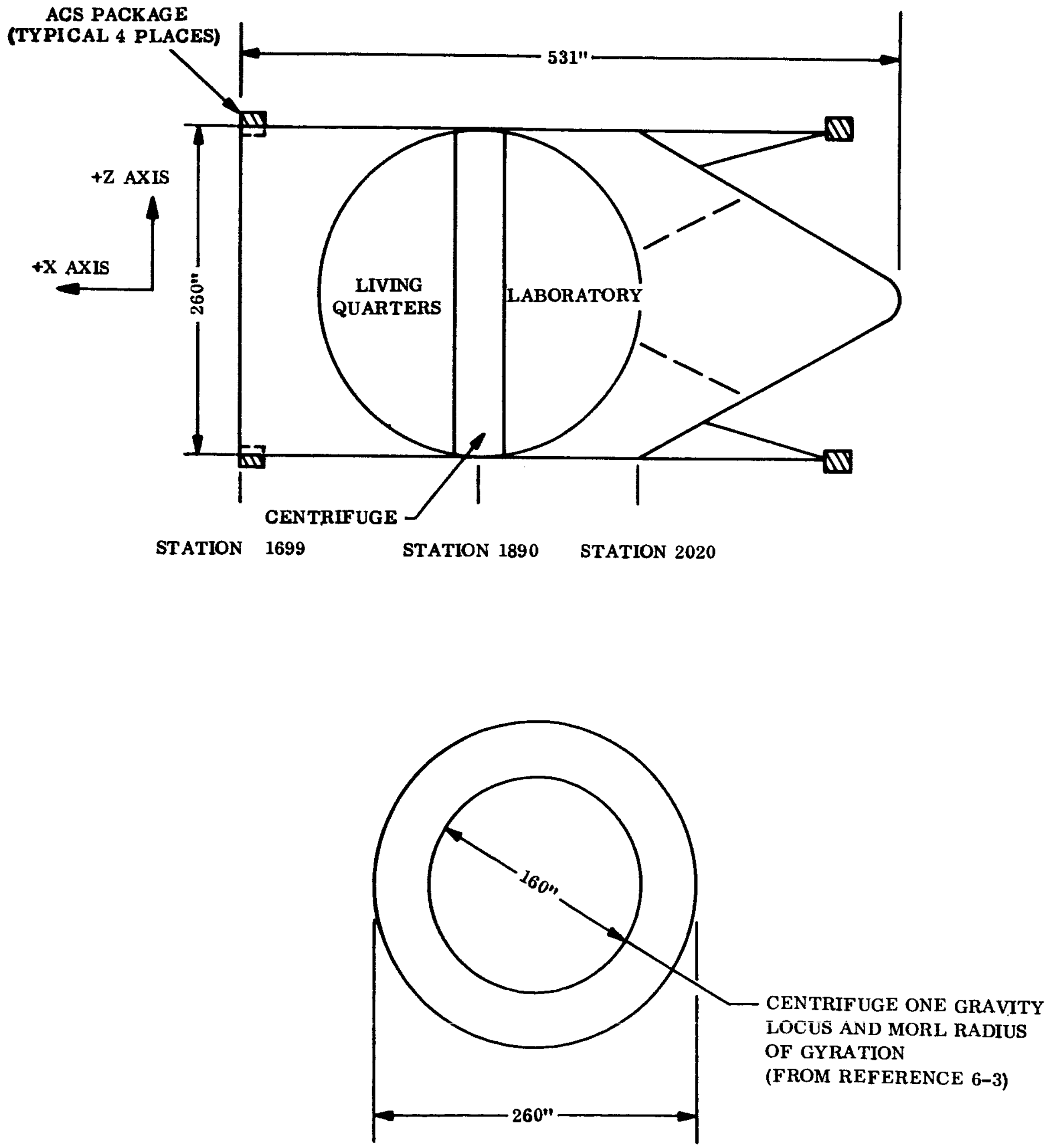


Figure 6-1. MORL Configuration

	ORBIT ADJUST	CONTROL MODE OPERATION		
		ROLL	YAW	PITCH
+X Axis	X			
+Y Axis		X	X	
		or	or	
-Y Axis		X	X	
Z Axis Away from Vehicle				X

The engines are radiation-cooled, pulse-modulated with a two-year reliability of 0.995. In the event of a single engine failure, modification of firing logic will still allow system rotational control with some performance degradation.

The propulsion modules are mounted external to the vehicle structure and are protected (except for the engines) from the radiation environment and micro-meteorite environment with a shield that also serves as powered flight heating protection.

A typical hypergolic bipropellant that could be used is N_2O_4/MMH. The available specific impulse for this combination is 300 seconds for longer period operation and 250 seconds for short pulse widths such as attitude control. The engine-propellant combination is not entirely optimized for performance; pressures and propellant ratio are chosen to enhance engine life.

6.1.3 DISTURBING FORCES

The major disturbance on an orbiting vehicle that must be contended with in providing stabilization are:

a. Atmospheric Drag - Impingement of air molecules produces a force opposite to satellite velocity.

b. Gravity Gradient - Unequal moments of inertia procude a torque as a result of the gradient of the earth's gravity field for positions other than long principal axis perpendicular, or parallel, to the gravity gradient.

c. Limit Cycle - A stabilized vehicle requires rate and position limitations just to be stabilized. For practical considerations, the vehicle would then cycle between limits, a neutral or non-propellant using position is not achieved because of other disturbances. The cycle frequency (and amount of propellant required for a mission) is directly dependent on the spacing of the limits.

d. Aerodynamic Torques - A torque is produced if the center of pressure for drag is not coincident or in line with the center of gravity. This factor is assumed small for this vehicle.

e. Maneuvering - Impulse is required to maneuver a space vehicle either in a rotational or translational sense. This area was not investigated, since maneuver requirements are not available.

f. Internal Forces - These torques are reactions to internal momentum changes such as moving astronauts, accelerating or decelerating rotating mass.

6.2 SUMMARY OF REQUIREMENTS

The required propellant weight for the various orbiting vehicle disturbances is shown for different vehicle/orientation configurations in Table 6-1. A brief description of the analysis methods for the different disturbances follows; detailed calculations are contained in Appendix B.

TABLE 6-1 NON-INTEGRATED ATTITUDE CONTROL FUEL WEIGHTS

	EARTH ORIENTED ISOTOPE (LB)	SUN ORIENTED ISOTOPE (LB)	SUN ORIENTED PHOTOVOLTAIC (LB)	SUN ORIENTED SOLAR CONCENTRATOR (LB)
Atmospheric Drag	270	463	1258	586
Gravity Gradient		1430	1430	1430
Centrifuge Roll Distrubance and Gyrotorque	117	90	90	90

TABLE 6-1 NON-INTEGRATED ATTITUDE CONTROL FUEL WEIGHTS (Cont'd)

	EARTH ORIENTED ISOTOPE (LB)	SUN ORIENTED ISOTOPE (LB)	SUN ORIENTED PHOTOVOLTAIC (LB)	SUN ORIENTED SOLAR CONCENTRATOR (LB)
2 degrees Limit Cycle	359	359	431	359
Contingency for Reserve, Crew Motion, Aero Torques, Other Unknown Users	75	250	350	270
TOTALS	821	2592	3559	2735

6.2.1 ATMOSPHERIC DRAG

Figure 6-2 shows the propellant required per year per square foot as a function of altitude. This is 0.73 lb/yr/sq ft for a 250-nautical mile circular orbit. The vehicle area is computed as the average area perpendicular to the flight path.

6.2.2 GRAVITY GRADIENT

The gravity gradient torque for a sun-oriented MORL is depicted in Figure 6-3. The calculated torque is for pitch motion alone, and neglecting products of inertia. The MORL weight was calculated from Reference 6-1 (39,3000 lb) and radius of gyration of 8, 15, and 15 feet were assumed for roll, yaw, and pitch to determine moments of inertia. Specific impulse for this application was taken as 250.

6.2.3 CENTRIFUGE MOMENTUM COMPENSATION

Reference 6-3 was used as a guideline for centrifuge properties. The operation profile includes nine-g Gemini re-entry simulation for brief periods, a total of 30 times/year.

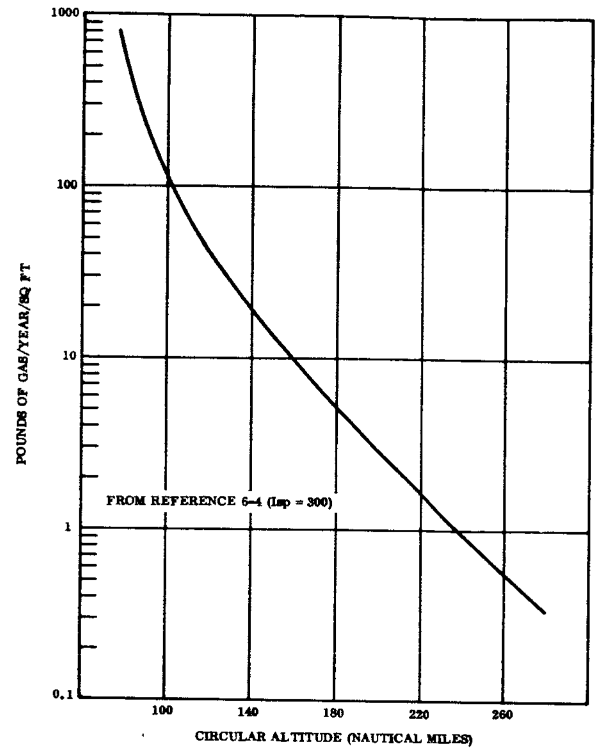


Figure 6-2. Orbit Maintenance Requirements

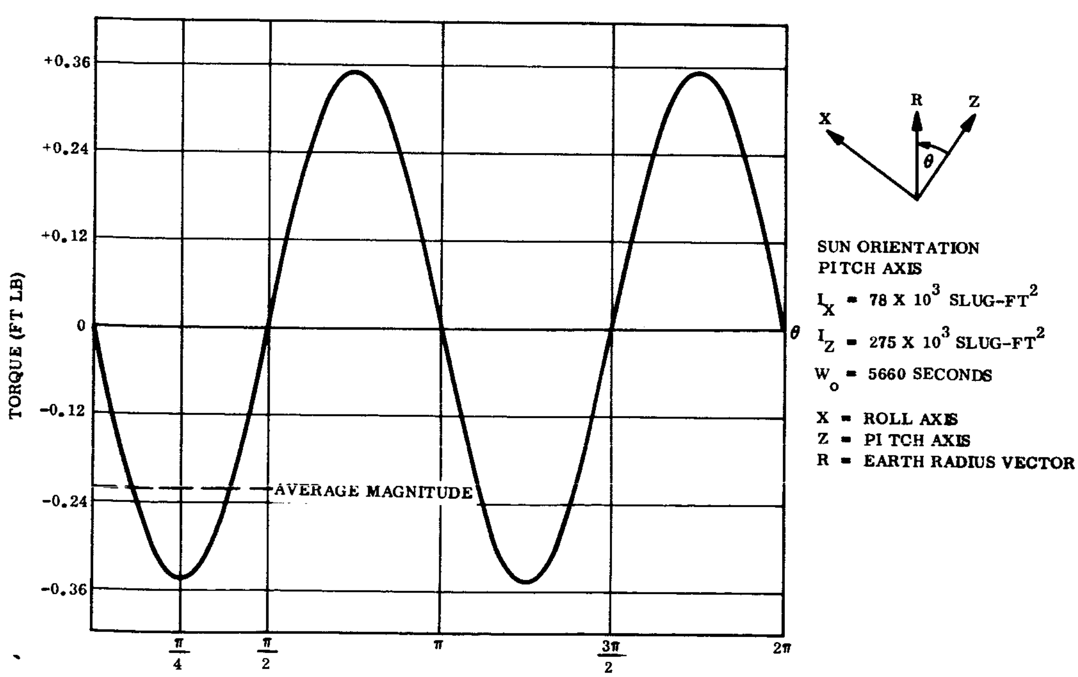


Figure 6-3. Gravity Gradient Torque

The centrifuge compensation is a fixed, one-g simulation flywheel. A 5% variation of momentum from compensation null was assumed for the one hour a day normal operation. Required impulse is then the sum of momentum deviation in the one-g cycle and the momentum deviation in the nine-g cycle.

For an earth-oriented system, additional requirements were calculated for the residual gyroscopic torques.

6.2.4 LIMIT CYCLE

The required propellant for different error limits is shown in Figure 6-4. This data is scaled from Reference 6-4, accounting for the MORL specific impulse, engine torques, and mass properties. The two-degree range was taken as a typical acceptable limit for a satellite not having a high accuracy requirement. If experiments such as astronomy are involved, then the dead band would naturally be much smaller. The 0.1-degree accuracy for the solar concentrator systems (as shown in Reference 6-2) was assumed to be mechanically compensated and not requiring additional attitude control propellant.

6.2.5 CONTINGENCY

The contingency was taken as 10% of the required propellant weight for each system. The major function is to serve as a reserve. Crew motion should have small effect on a vehicle with these inertias (except if extensive maneuvers with life-line are planned). A probable method of combating excessive errors due to crew motion is a reaction wheel, since the motions tend towards random and the net approaches zero. It has been assumed that aerodynamic torques are taken into account by vehicle design and the residual effect is quite small.

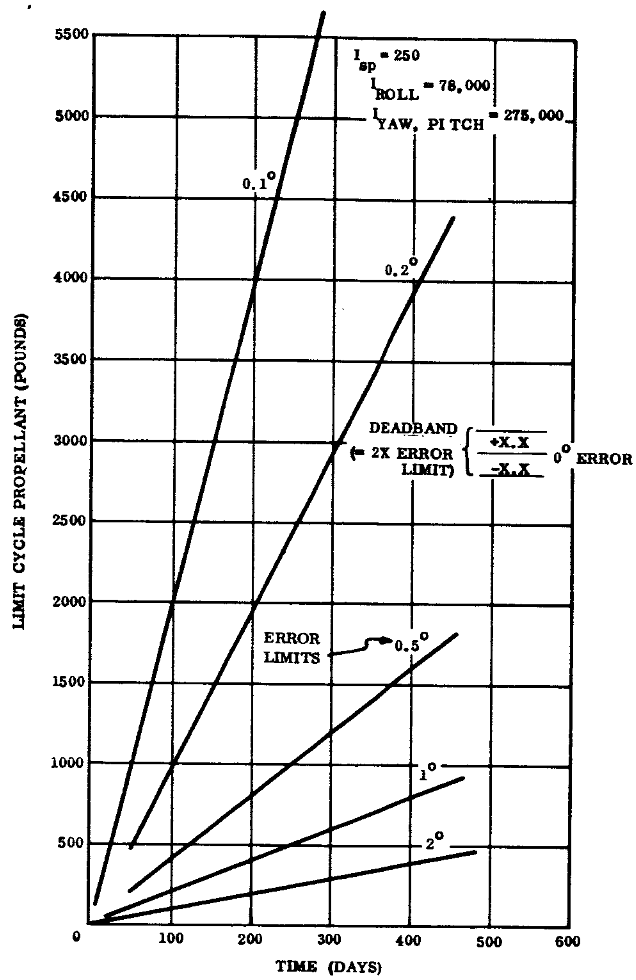


Figure 6-4. Limit Cycle Propellant Requirements

6.3 STEAM PROPULSION AND CONTROL

6.3.1 BASIC THEORY

The prime concern in evaluating mass expulsion control systems is the available specific impulse from the working fluid. Specific impulse is a direct function of fluid velocity ($I_{sp} \doteq$ velocity/g). The method of accelerating the fluid is to use a converging-diverging nozzle. The thermodynamic process is conversion of initial heat energy (enthalpy) to velocity energy, the final velocity is proportional to the square root of the enthalpy change. The nozzle process is normally considered to be near isentropic expansion. The fluid first accelerates to sonic velocity in the converging section and then accelerates further in the diverging or supersonic section (Figure 6-5). The enthalpy change is a function of the pressures and nozzle areas encountered in the process. It is generally desirable to maximize ϵ , or ratio of exit area to throat area, for highest velocities.

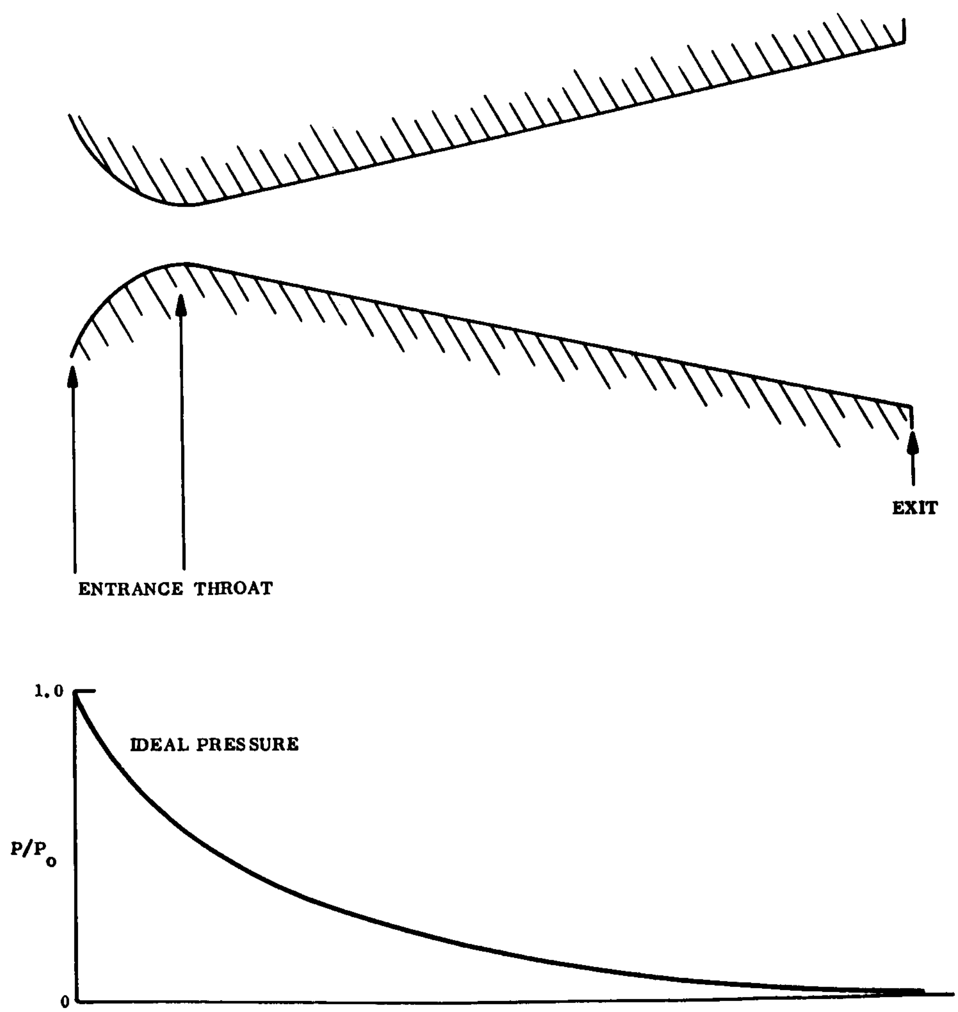


Figure 6-5. Converging-Diverging Nozzle

6.3.2 REAL NOZZLE CONSIDERATIONS

To arrive at a practical nozzle, a literature search was made for information on general nozzle information (References 6-5 through 6-10) and for steam nozzle characteristics including the wet region (References 6-8, and 6-10 through 6-19). Expansion into the wet region is specifically desirable to maximize specific impulse. Available energy is reduced from theoretical isentropic for a given final pressure by turbulence, steam line divergence and rotation, nozzle surface friction heating, gas overexpansion or exit shock (exit overpressure), and in the case of wet steam, condensation shock. The first three losses are usually handled with an efficiency factor. The number chosen for this study is 0.9, or in the normally quoted region of 0.85 to 0.95. The nozzle exit cone angle will be taken as 30 degrees, typical of real designs (References 6-5, 6-6 and 6-18).

The exit overpressure will not be a problem since it is effectively zero. The condensation shock is a phenomenon that is not well understood. Earlier research (References 6-11, 6-12 and 6-13) studied the area near saturation, a prominent effect occurring at about 94 to 96% steam known as the Wilson Line. The apparent effect was supersaturation, or all vapor existing in a region where some liquid should appear. The Wilson Line is the point where a sudden condensation occurs to the extent of the thermodynamic conditions. At the time of these studies, wet steam was not desirable, since it would damage turbine blades (principal application) through erosion. Figure 6-6 depicts the experimental energy loss data from Reference 6-13. This curve has been extrapolated for comparison purposes. Mention is made of the role of moisture particle size in energy losses. Evidently the liquid particle size is not constant, such that the shock does not necessarily occur at only one point in the nozzle, but occurs along the nozzle as expansion takes place. This appears reasonable since a near isentropic expansion process will be lowering the pressure and increasing the number of molecules in the liquid state. The simplified analysis described by Shapiro (Reference 6-10, P204) states that the condensation shock energy losses are equal to the latent heat (h_{fg}) times the percent of water vapor. Stated another way, the available energy is equal to total energy change times the percent of steam in the flow.

This theoretical loss is plotted on Figure 6-6 for comparison with the experimental data. It is evident that very close correlation exists.

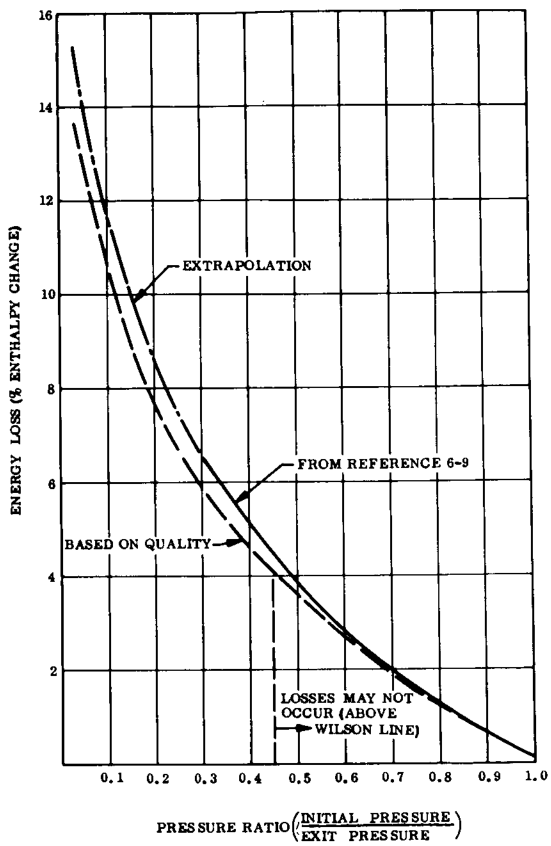


Figure 6-6. Condensation Shock, Steam Energy Loss, Saturated Steam Initial Condition

6.3.3 NOZZLE THERMODYNAMIC OPERATING POINTS

Nozzle pressure ratio dictates both nozzle sizing and specific impulse. Although obtaining maximum specific impulse is a definite goal, it was decided to limit the exit pressure so that the exit temperature was above the ice point; the exit diameter is then on the order of 7 inches for a 60-pound thrust nozzle (a possible MORL application). The exhaust pressure is 0.12 psia for all cases. Initial temperatures considered were for Mercury Rankine power system radiators (600°F) and the Brayton power system non-integrated and integrated radiator inlets (400°F, 300°F). Initial pressures were chosen so that the fluid would be a vapor at the throat. This simplifies flow calculations in that only a single-phase flow is present at the throat.

6.3.4 FLOW VELOCITY AND SPECIFIC IMPULSE

Using the data in Sections 6.3.2 and 6.3.3, velocity and specific impulse can be calculated.

$$I_{sp} = \frac{V_e}{g} = \frac{(223.7)}{g} \sqrt{(h_1 - h_e)\ (\eta)\ (x)}$$

V_e = Exit Velocity

g = Gravitational Constant

h_1 = Initial Enthalpy

h_e = Theoretical Final Enthalpy

η = 0.9, Nozzle Efficiency

x = Final Steam Quality at (Δh) (η)

The initial steam velocity is assumed to be negligible. These calculations are presented in Table 6-2. Reference 6-19 and low temperature properties from Reference 6-20 were used for steam conditions. Figure 6-7 presents the results; specific impulse as a function of temperature for the initial and exit pressures considered. It is interesting to note that initial pressures greater than 600 psia do not improve the specific impulse available. Figure 6-8 shows the relative position of the calculated specific impulses with isentropic vapor theory and with test results from Reference 6-18. It is significant to note that the test data is for very small nozzles, the largest throat diameter friction forces were found to be quite significant below this.

TABLE 6-2. SPECIFIC IMPULSE DETERMINATION

P_1 PSIA	T_1 °R	S Btu/lb-°F	h_1 Btu/lb	h_e Btu/lb	P_e PSIA	(h_1-h_e) Btu/lb	X % STEAM	V_e FT/SEC	I_{SP} lb-SEC/lb
600	1060	1.532	1290	680	0.018	549	0.72	4450	138.0
600	1060	1.532	1290	750	0.12	486	0.75	4270	132.5
100	1060	1.765	1330	880	0.12	405	0.855	4160	129.2
50	1060	1.836	1335	915	0.12	378	0.885	4090	127.0
10	1060	2.016	1335	1005	0.12	297	0.96	3770	117.0
100	860	1.652	1228	825	0.12	363	0.80	3810	118.3
50	860	1.736	1235	865	0.12	333	0.83	3720	115.5
10	860	1.917	1240	956	0.12	256	0.915	3420	106.0
50	760	1.672	1184	834	0.12	315	0.80	3550	110.4
20	760	1.781	1192	889	0.12	273	0.85	3410	105.8
10	760	1.86	1194	930	0.12	237	0.89	3250	101.0
600	1680	1.80	1640	900	0.12	666	0.903	5480	170.3
600	1480	1.723	1528	860	0.12	593	0.865	5060	157.3
2400	1680	1.625	1602	810	0.12	712	0.825	5425	168.5
2400	1480	1.542	1476	770	0.12	635	0.78	5030	156.3
50	1680	2.075	1650	1035	0.12	553	1.00	5260	163.3
50	1480	2.005	1540	1000	0.12	486	0.98	4880	151.5
2400	1900	1.86	1759	930	0.12	746	0.935	5900	183.5
1200	1680	1.712	1628	855	0.12	695	0.865	5480	170.4

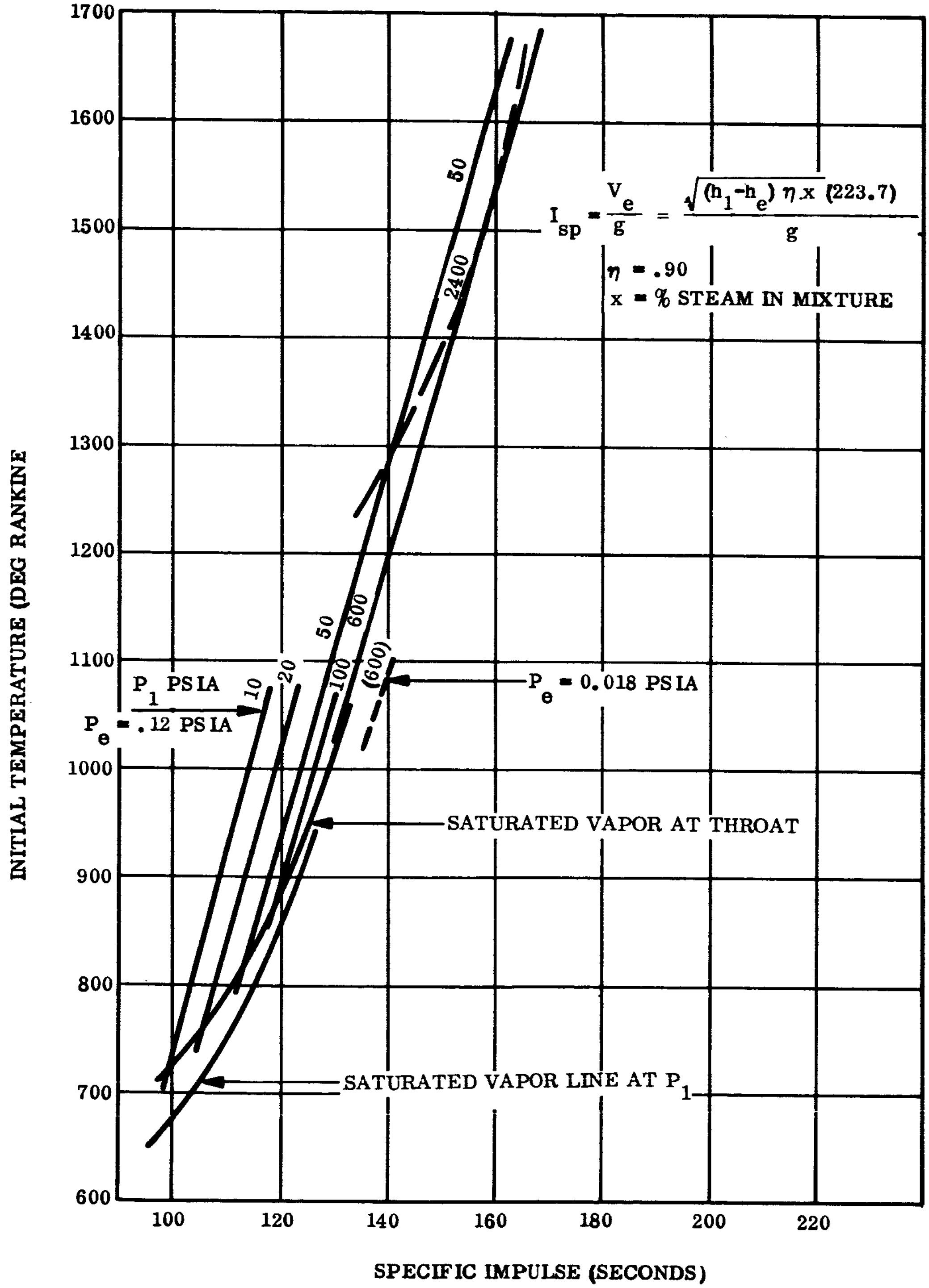


Figure 6-7. Steam Specific Impulse

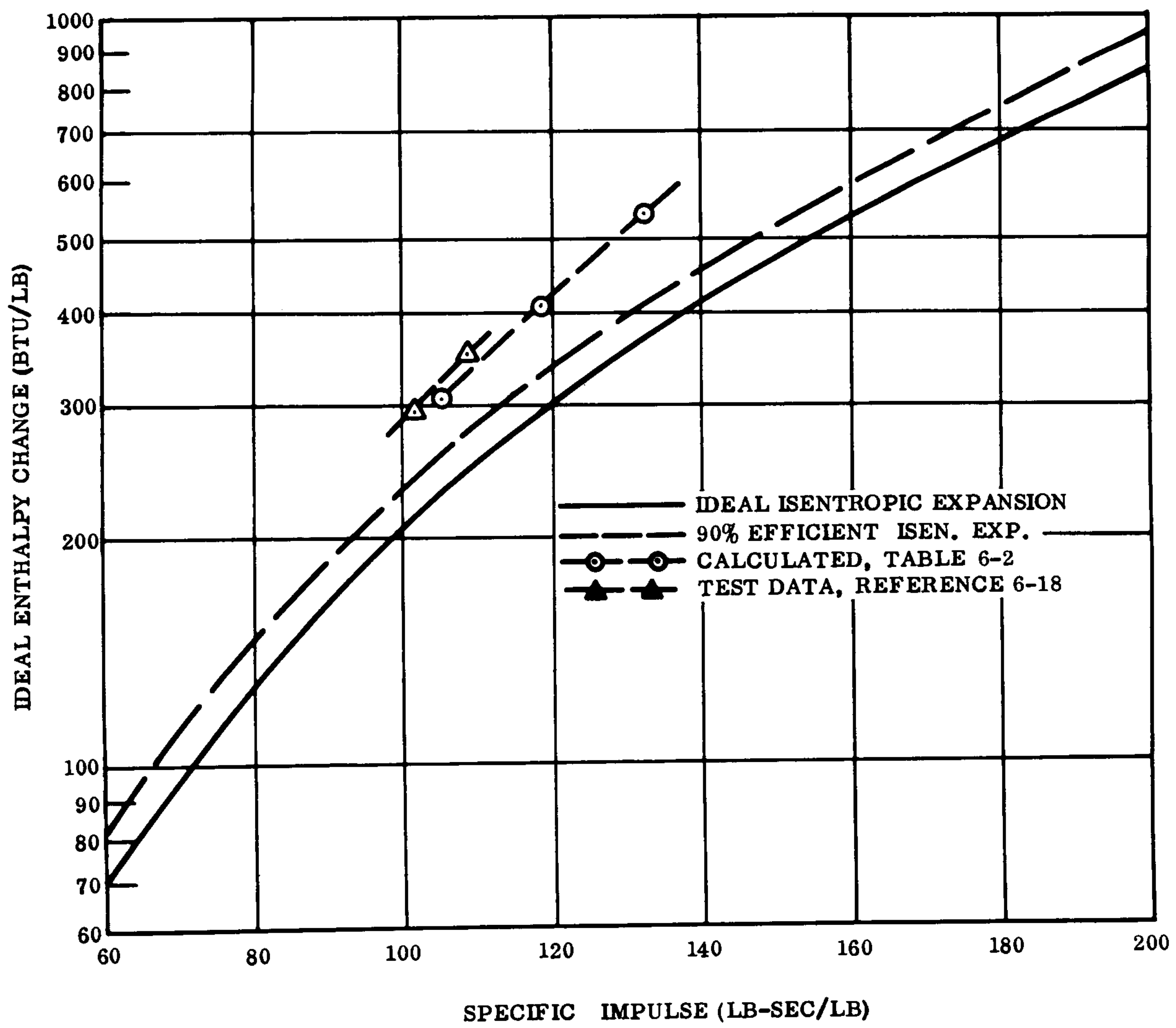


Figure 6-8. Steam Specific Impulse Comparisons

6.3.5 NOZZLE SIZE

Table 6-3 presents calculations of nozzle size for the three chosen pressure/temperature combinations and flow rates of 0.01, 0.1 and 0.5 pound per second. The results are plotted parametrically in Figure 6-9. The steam is treated as a vapor at the nozzle throat. The exhaust velocity is found by the equation in 6.3.4. The exhaust specific volume is found by using the quality from Table 6-2 and the steam tables (References 6-19 and 6-20).

6.4 MORL VEHICLE - STEAM PROPULSION INTEGRATION

6.4.1 GENERAL CONSIDERATIONS

There are several possible methods of using steam propulsion on the MORL Vehicle, ranging from providing complete control and navigation to serving as a back-up system. Table 6-3 shows possible combinations of steam/hot gas/moment gyro control. Also shown are

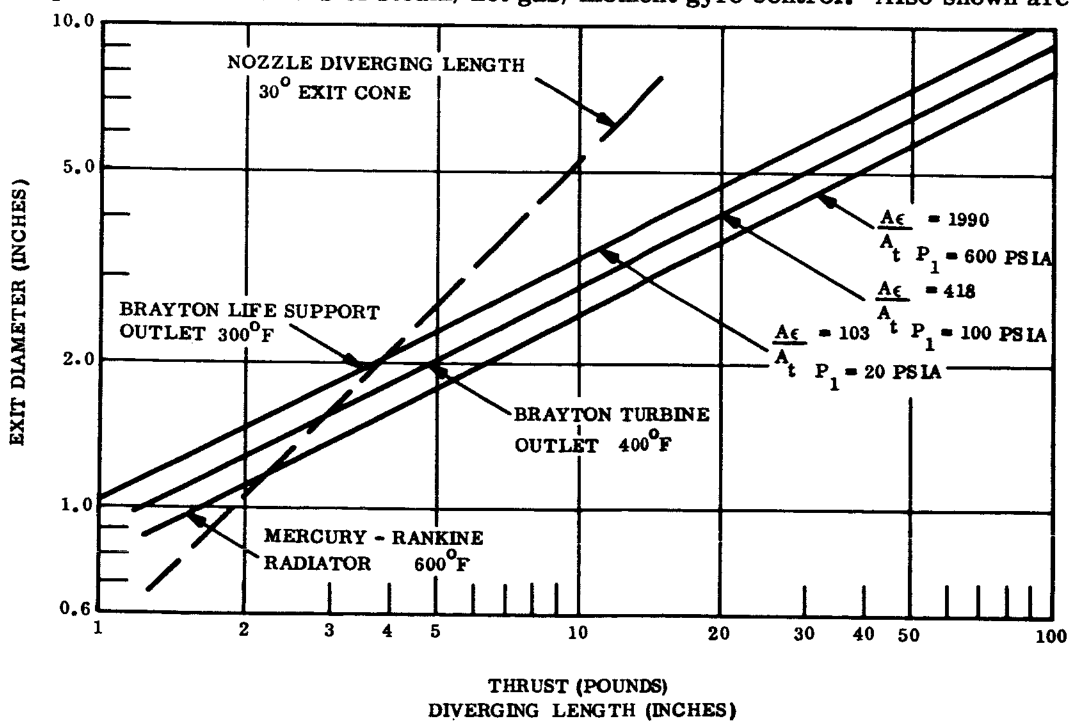


Figure 6-9. Steam Rocket Size

TABLE 6-3. NOZZLE DESIGN CALCULATIONS

P_1	T_1	h_1	h_t	Δh	P_t	γ_t	V_t	V_e	P_e	γ_e	W	thrust	A_t	D_t	A_e	D_e
psia	°R	Btu/lb	Btu/lb	Btu/lb	psia	lb/cu ft	ft/sec	ft/sec	psia	lb/cu ft	lb/sec	lb	sq ft	in.	sq ft	in.
600	1060	1290	1230	60	330	1/1.50	1730	4270	0.12	1/1830	0.01	1.25	8.6×10^{-6}	0.0198	4.3×10^{-3}	0.87
											0.10	12.5	8.6×10^{-5}	0.063	4.3×10^{-2}	2.87
											0.50	66.0	4.33×10^{-4}	0.141	0.215	6.29
100	860	1228	1177	51	55	1/7.82	1596	3810	0.12	1/1955	0.01	1.18	4.9×10^{-5}	0.0474	5.1×10^{-3}	0.967
											0.10	11.8	4.9×10^{-4}	0.150	5.1×10^{-2}	3.06
											0.50	59.0	2.45×10^{-3}	0.335	.256	6.85
20	760	1191.5	1145	46.5	11	1/35.1	1523	3410	0.12	1/2075	0.01	1.06	2.3×10^{-4}	0.103	6.1×10^{-3}	1.06
											0.10	10.6	2.3×10^{-3}	0.327	6.1×10^{-2}	3.35
											0.50	53.0	1.15×10^{-2}	0.725	0.305	7.48

application possibilities for each method. Table 6-4 shows the impulse requirements for the required controls and the possible MORL power systems. For each control requirement, propellant weights are given for specific impulses of 300/250 (N_2O_4/ MMH); 153, 132, 119 (steam). The water to be used in producing steam is supplied from the life support system. There are two different surplus rates, 0.57 and 1.0 pound per man day, or 1250 and 2190 pounds per year. The difference in the two numbers is the degree of dehydration of prepared food. The 1250-pound excess water for one year is probably the more realistic to consider.

The heat to be used in converting the water to steam will be waste heat, or energy not utilized by the power system and normally rejected. The required heat rate for the steam is quite low compared with the power available. The average rate is on the order of 180 Btu/ hr to convert all water to steam over a one year period, while the rejected heat from the power system is 50,000 to 100,000 Btu/hr. The Mercury Rankine condensing radiator temperature is 1065°R; 1060°R is taken as a reasonable temperature that the steam storage tank can maintain. The Brayton cycle temperature to the radiator is 875°R and 763°R for the non-integrated and integrated power systems in Reference 6-17. 400°F is taken as a reasonable Brayton Cycle steam storage temperature. Since the amount of heat removed is so small compared with the amount in the working fluid, it is assumed that the steam system can be placed ahead of the life support heat exchanger in the integrated case, and the 300°F source for steam generation will not be evaluated in detail. For the MORL Attitude Control engine sizing of 60 lb the vehicle accelerations will be the order of 0.5 degree/ sec^2. This seems higher than is required for vehicle control, and is probably a result of engine operating time/life constraints. The steam nozzles will operate at far lower temperatures, hence smaller thrust engines are a definite possibility. Utilization of smaller engines would lessen temperature/pressure transients upstream of the nozzle.

6.4.2 SPECIFIC INTEGRATION POSSIBILITIES AND WEIGHT ADVANTAGES

The possible applications, as limited by the ground rules of this study are boxed in Tables 6-4 and 6-5. It can be seen in Table 6-5 that there is insufficient steam impulse for the

TABLE 6-4. POSSIBLE CONTROL SYSTEM COMBINATIONS

OPTION	SYSTEM APPLICATION	N_2O_4/ MMH HOT GAS FUNCTION	STEAM FUNCTION	CONTROL MOMENT GYRO FUNCTION
1	A	-----	All attitude control and navigation	----
2	A, B, D	Attitude control Gyro impulse removal Contingency	Orbit adjust Maneuvers	Personnel motion
3	A, B	-----	Orbit adjust Impulse removal	Attitude control Personnel motion Contingency
4	A, B, D	Attitude control Maneuvers Contingency	Orbit adjust	Personnel motion
5	A, B, C, D	Attitude control Maneuvers Orbit adjust	Contingency	Personnel motion
6	A, B, C, D	Attitude control Maneuvers	Orbit adjust Contingency	Personnel motion
7	A, B, C, D	All control and navigation	Back-up control and/or fine mode control (low/variable thrust nozzles)	Personnel motion

TABLE 6-5. WEIGHT COMPARISON OF N_2O_4/ MMH AND VARIOUS STEAM STATES AS PROPELLANTS

System Application	A (Earth-oriented Isotope)				B (Sun-oriented Isotope)				C (Sun-oriented Photovoltaic)				D (Sun-oriented Solar Concentrator)			
Specific Impulse	MORL 300/ 250	153	132.5	119	MORL 300/ 250	153	132.5	119	MORL 300/ 250	153	132.5	119	MORL 300/ 250	153	132.5	119
Atmospheric Drag	270	528	613	680	463	909	1052	1167	1258	2470	2860	3170	586	1148	1330	1475
Gravity Gradient	-	-	-	-	1430	2340	2710	3000	1430				1430			
Centrifuge Roll Dist.	117	191	222	245	90	148	171	189	90				90			
2-degree Limit Cycle	359	587	680	753	359	587	680	754	431				359			
Contingency	75	130	151	168	250	398	461	511	350	572	663	734	270	441	510	565
TOTALS	821	1436	1666	1846	2592	4382	5074	5621	3559				2735			
	I_{tot} = 218600 lb-sec				I_{tot} = 671000 lb-sec				I_{tot} = 952000 lb-sec				I_{tot} = 745,000			

I_{sp} Explanation:
(a) MORL 300/250 -- typical bipropellant, N_2O_4/ MMH
(b) 153 ------------ mercury rankine, $I_{sp} = 223.7/g \sqrt{\eta\,(\Delta h)}$
(c) 132.5 ---------- mercury rankine, $I_{sp} = 223.7/g \sqrt{\eta(x)\,\Delta h}$
(d) 119 ----------- Brayton cycle, $I_{sp} = 223.7/g \sqrt{\eta(x)\,\Delta h}$

NOTE: 1250 lb/yr water nominally avail.
For one lb/man/day supply, 2190/lb/yr is avail.

power systems and orientations considered to completely replace the hot gas with a steam system. It, therefore, becomes necessary to investigate the different users to see how steam can be used best against a portion of the total. One function that is separate from the others is orbit adjust or drag compensation. The other discrete functions are back-up provision (for increased reliability) and special purpose control where different thrusting levels from the basic attitude control system might be required.

Enough extra propellant beyond the steam capability is required such that there is a decided weight penalty for an all water system. The specific impulse is enough higher such that a dual system is recommended.

Table 6-6 presents the pertinent data and results of steam/MORL hot gas integration for the three power systems considered. Integration of the steam system requires that equipment weights be estimated and added to changes in propellants/steam. The so-called contingency application is identical to the weight saving that would be realized if the steam were applied to the total attitude control job. The maximum integrated saving of 582 pounds (1250 lb water) is obtained with the Mercury Rankine power system dual contingency option. This was to be expected, since the Mercury Rankine steam has a higher specific impulse than Brayton steam and the contingency hot gas has an I_{sp} of 250 compared with 300 for orbit adjust considerations.

6.4.3 STEAM PRODUCTION METHOD

The generation of steam from excess water and waste heat can either take place in one central boiler or in separate boilers in several different locations close to nozzles. A single boiler will be discussed here, since it can be applicable to multiple locations. Steam expands considerably on vaporization, and a tank to contain all excess water, after it came from life support, in the gas state would be prohibitive in size and weight. Up to a certain point ($\sim$ 600 psi) it is advantageous to raise the steam pressure for additional specific impulse. This also means that the boiler should be kept small. A means of retaining a

TABLE 6-6. MORL VEHICLE - STEAM PROPULSION INTEGRATION

POWER SYSTEM	ISOTOPE BRAYTON				SOLAR MERCURY RANKINE				SOLAR BRAYTON			
OPTION	4	5*	6		4		5		4		5	
FUNCTION	Orbit Adjust	Cont. and Attitude Control	Orbit Adjust		Orbit Adjust		Contingency		Orbit Adjust		Contingency	
METHOD	Steam	Dual	Steam	Dual	Steam	Dual	Steam	Dual	Steam	Dual	Steam	Dual
Required Hot Gas (Before Integration)	463	250 1879	713	713	586	586	790	790	586	586	790	790
MORL Dry Weight Reduction	50	0	50	0	50		50	0	50	0	50	0
Steam System Dry Weight	100	150	150	150	100		100	100	100	100	150	150
3,43 lb/day available water Steam Fuel Weight	1167	1250	1678	1250	1330		1495	1250	1475	1250	1655	1250
Residual Hot Gas				204				108		57		214
Δ Integrated Weight	-413 and 83 lb H_2O	-445	-185	-359	-456		-495	-582	-311	-429	-285	-426
					Option 6				Option 6			
					All Steam		Dual		All Steam		Dual	
6 lb/day available water Steam Fuel Weight		2190	1678		2825		2190		3130		2190	
Δ Weight		-892	-613		-691		-996		-336		-876	
Other Attitude Control Weight			-244									
Residual N_2O_4 / MMH		1237(A/C)					280(O. A.)				350 (Cont.)	
Total Δ Weight		-892	-857		-691		-696		-336		-876	

*Note: Excess water over contingency use applied against rest of Attitude Control functions.

small and relatively light boiler while providing capability of all available water, is to have a low pressure water storage, a high pressure feed pump, and bleed water into a boiler as a mixture. The boiler is maintained at the superheat temperature by finned piping, the pipes carrying the power system working fluid to the radiator. Figure 6-10 illustrates a simplified schematic of this steam system.

Desirable conditions for locating the steam system are as follows:

a. Nozzles far from vehicle CG for maximum torque arm.
b. Boiler near power system between prime mover and radiator.
c. Nozzles near boiler to minimize line losses.
d. Boiler located such that it is protected from meteor impingement.

The best method appears to be to locate the steam system at or near the end of the vehicle where the power system is located. Figures 6-4 and 6-12 are two examples (a single

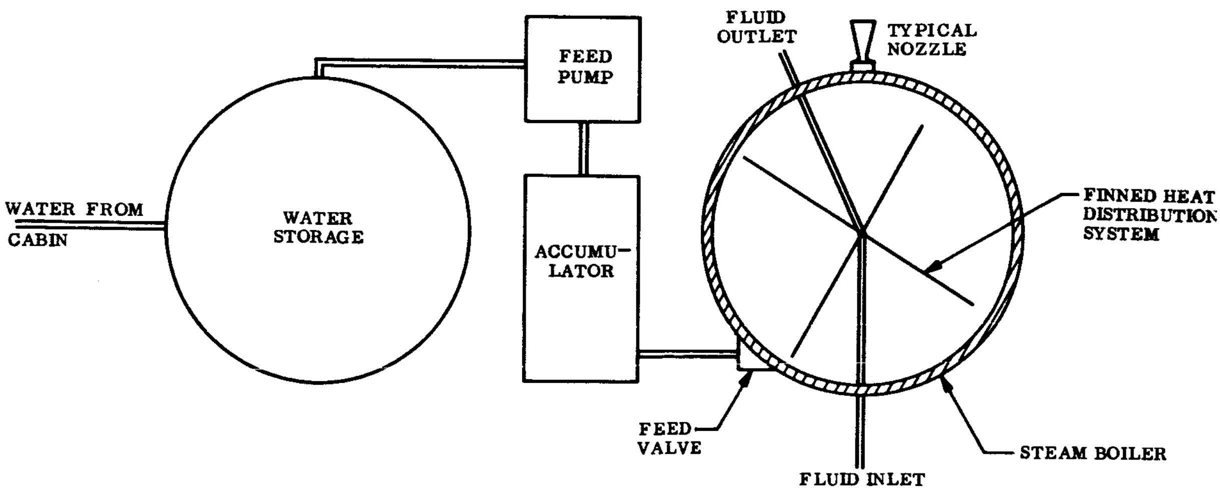


Figure 6-10. Steam System, Schematic

boiler system and a dual boiler system) of steam propulsion systems on the MORL vehicle. Note that the single boiler requires protrusion from the vehicle aft end to avoid impingement, and long tubing to the roll nozzles. The dual system requires longer working fluid piping and some method of controlling divided working fluid and water flow. Location of nozzles (as shown in Figures 6-11 and 6-12) is simplest when they are mounted directly on control valves, directly on the boiler.

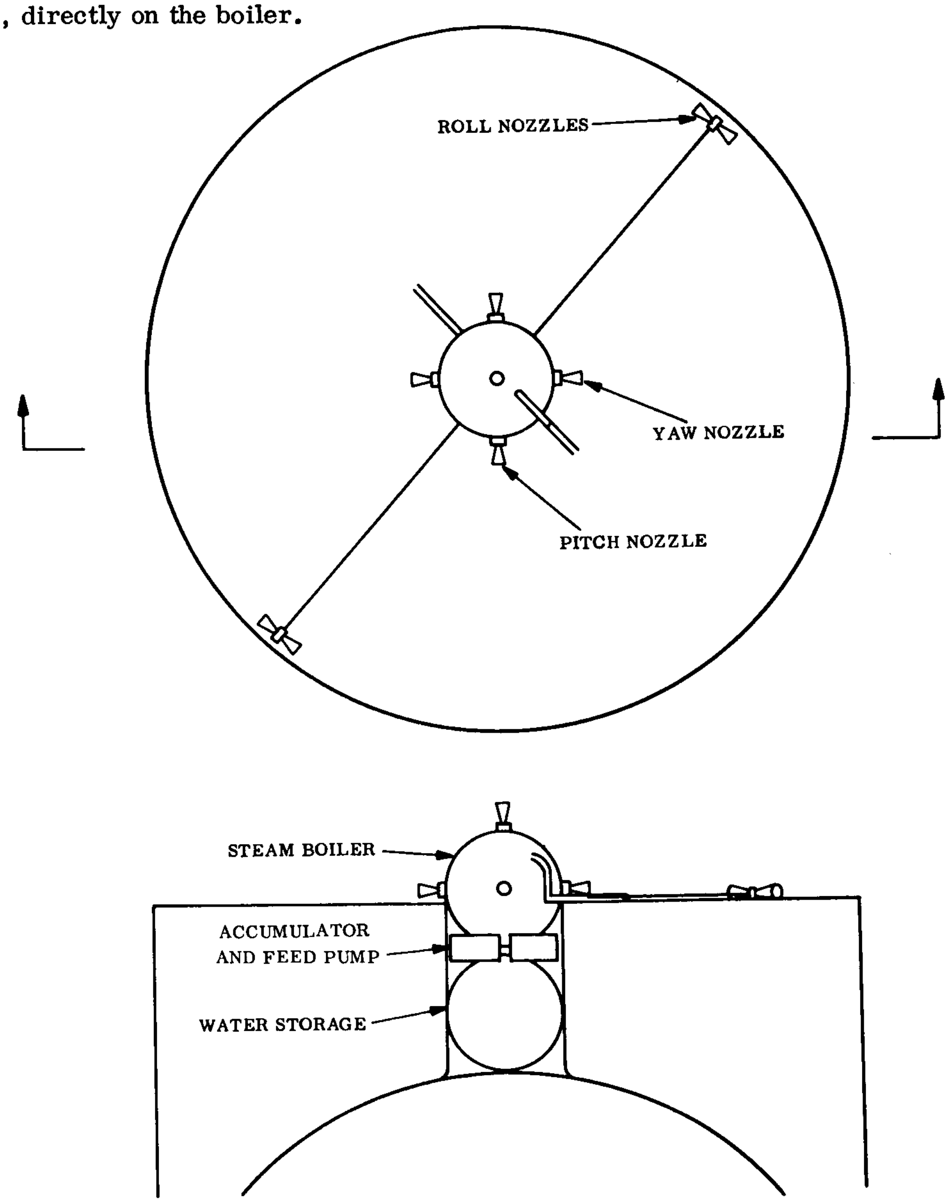


Figure 6-11. Single Boiler Concept

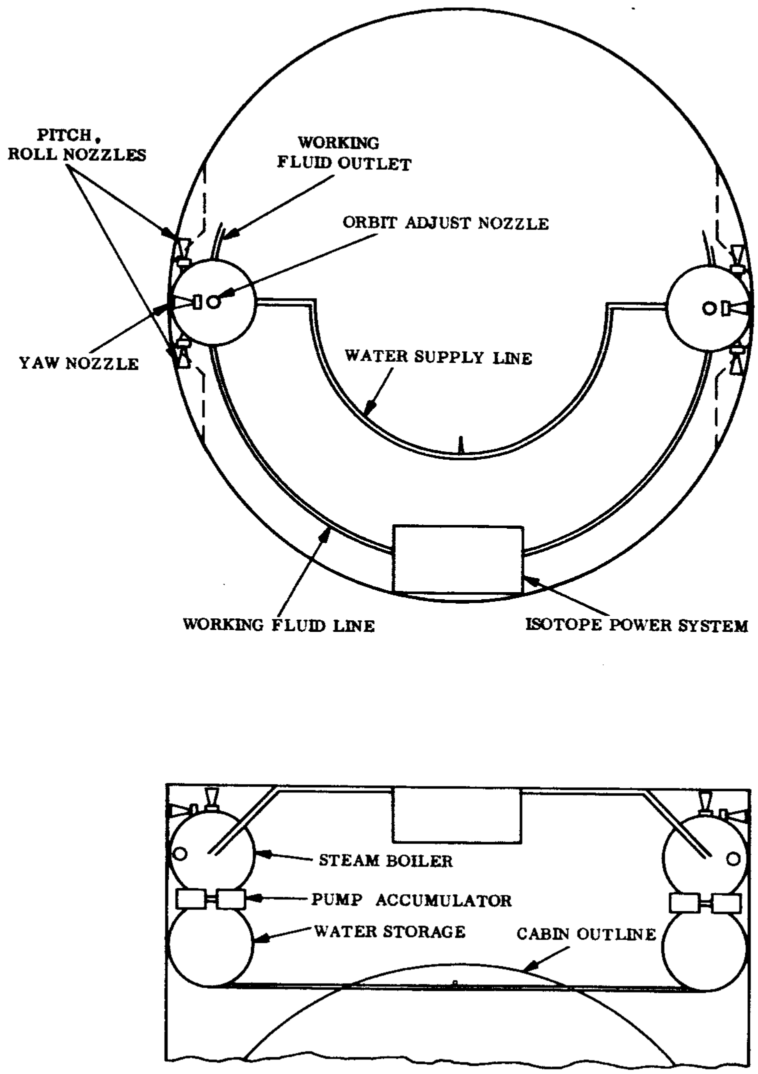


Figure 6-12. Dual Boiler Concept

For the solar concentrator power systems, provision must be made to avoid a steam exhaust impingement on the surface of the concentrator from the orbit adjust engines. Three of the possibilities are:

a. Cut holes in the concentrator.
b. Attach separate O/A boiler and nozzle system to the power system absorber.
c. Move O/A nozzles to side of vehicle.

For a sun-oriented vehicle, the thrust direction vector does not have to be on a vehicle body axis as long as the resultant force passes through the center of gravity and aligns with the orbital velocity vector at some time during the orbit. Actually, two engines are required firing at 180 degrees apart in the orbit, to maintain a circular orbit.

6.4.4 COMPONENT DESIGN CONSIDERATIONS

a. Nozzles - Desirable nozzle properties include:

1. Low surface fluid friction.
2. Thermal expansion/ strain properties rendering thermal shock insensitivity.
3. External thermal coatings such that the steady-state temperature approaches that of the steam. This will minimize boundary effects.

b. Boiler Heat Exchanger - The working fluid should be distributed in fins as it passes through the boiler, the objective being to maintain all the steam at one temperature. To facilitate heat distribution, the feed valve should admit make-up water at a tangent to the wall surface such that circulation is promoted. This heat exchanger should be conservatively designed, since it will handle primary power working fluid, and its internal location makes it difficult for on-orbit repair.

c. Pressurizing Equipment - The pressurizing pump, accumulator, and boiler feed valve provide the transfer from stored water at 70°F and 7 psia to boiler feed water at slightly higher temperature and boiler pressures (600 or 100 psia). The high pressures are somewhat desirable to enhance specific impulse (Figure 6-7), but are chiefly required for a reasonable tank size for anticipated steam reserve requirements. The accumulator is beneficial in absorbing boiler requirement transients and would provide some reserve during possible feed pump maintenance. The feed valve will allow water into the boiler on demand.

d. Boiler Tank - The tank shell is a titanium pressure vessel with an external insulator/ micrometer protector. The titanium is derated to a 60,000 psi working stress for the elevated temperature/ extended life environment. Conservative insulation properties were chosen to minimize temperature gradients in the steam and protect the interior from micrometeor penetration. The insulation is laminates of aluminum foil and fiberglass mats (from Reference 6-21). Significant tank parameters are plotted in Figure 6-13. The basic criteria is reserve impulse requirements, since make-up heat is not supplied instantaneously.

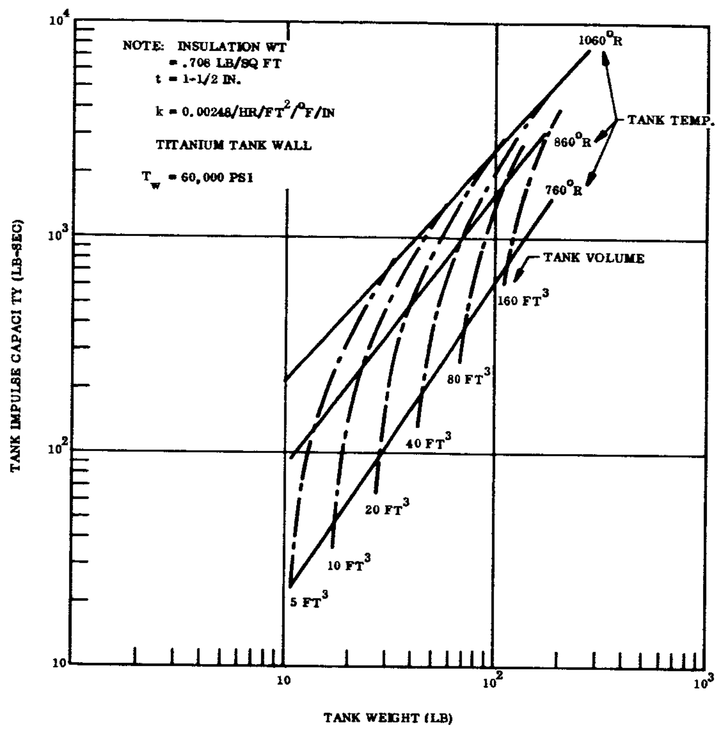


Figure 6-13. Steam Tank Parameters

e. Controls - Nozzle firings and boiler make-up water will be operated by solenoid valves. The nozzle valve requirements are zero leakage seating, area much larger than nozzle throat, and valve operation motion duration short compared with on-time. The feed valve has similar requirements, except full open position geometry can be a sharp edged orifice, and the size will be much smaller. Nozzle control will come from the same logic that operates the MORL propulsion engines, with a switch providing choice of hot gas or steam system operation. The feed valve requires pressure sensing in the boiler. Make-up fluid is provided at the rated pressure in pulses. The differential pressure will be sufficient such that the fluid will attain some velocity and become thoroughly mixed with the steam in the boiler. The flow rate will be limited to minimize boiler temperature transients and resultant pressure transients. It may be desirable to have a feed-back circuit where feed flow is governed by nozzle on-times. The feed pump operation will be governed by the amount of water in the accumulator.

f. Water Tank - The water tank requirements are:

1. Maintain 70°F water temperature.

2. For contingency case, have full year water containment capability.

3. Keep water in tank in contact with boiler feed pump intake.

The last requirement indicates that a bladder mechanism will be required to conteract the zero-g condition.

6.5 DISCUSSION

6.5.1 INTEGRATED RESULTS

The MORL dry weight reduction is an estimate from Reference 6-1. The steam system weight additions are estimated from data presented in Table 6-7 and Figure 6-13. The Brayton system is considered to have a larger boiler due to its lower operating temperature.

TABLE 6-7. WEIGHT OF STEAM SYSTEM CONCEPTS

	SINGLE SYSTEM	DOUBLE SYSTEM
Nozzles	13.5	12
Valves	4.5	4
Water Tank	6	6
Pump	15	15
Accumulator	10	10
Piping, Structure	15	15
	64	62
Tanks and Insulation	See Figure 6-13	

6.5.2 COMPONENT DEVELOPMENT AND RELIABILITY

Little, if any, development work is envisioned as being required for the steam system. The possible investigation area would be material or space environment nozzle/valve transient interactions; ground simulation should be straightforward. Boiler parameters are well known for power generation systems. Except for the internal heat exchanger and bladder, all parts are readily accessible for space repair. The steam system is simple, and has high reliability of operation.

6.5.3 NOZZLE SIZES AND THRUST LEVELS

The 60-pound thrust nozzles described in Reference 6-1 for use on MORL result in rotational accelerations in the order of 0.5 deg/sec^2. This acceleration is far in excess of known disturbance accelerations. Possible high torques may occur during docking or manned extra-vehicular activities, but they would be of an impact or jolt nature, with little total energy. It seems reasonable to assume that steam engines with a 10-pound thrust would fall in line with required vehicle acceleration control and, at the same time, simplify nozzle valve operation transients. The nozzle exit diameter (see Figure 6-9) would be in the order of three inches and the length in the order of six inches.

6.5.4 SPECIFIC IMPULSE CALCULATIONS - ASSUMED NOZZLE EFFICIENCY

The nozzle efficiency used in this section may appear somewhat conservative, in view of the inclusion of the quality factor separately. It is felt that an allowance should be made for the actual nozzle application, considering thermal losses, tank pressure/temperature losses, valve losses, operating transients, etc. Rather than attempt to detail the contribution of each area, an efficiency was assumed. Note the correlation with test data in Figure 6-8.

6.5.5 USE OF POWER SYSTEM SOURCE HEAT

A possible area of further exploration is to use the power system source for heat to produce steam. The temperature would be 1700 to 1900° Rankine, and the resultant specific impulse (see Table 6-1 and Figure 6-7) would be 170 to 183 lb-sec/lb. The heat rate requirement would be in the order of 0.1 kw thermal or 10 lb system weight penalty. The additional boiler requirements would be in the order of 100 pounds, while the fuel weight savings would be 300 to 500 pounds. This combination would be sufficient to supply all attitude control requirements for an earth oriented isotope Brayton system with a net weight saving of about 1000 pounds.

6.6 REFERENCES for SECTION 6

6-1 Report on a System Comparison and Selection Study of a Manned Orbital Research Laboratory, Douglas Report SM-44612, Appendix 16, Propulsion and Reaction Control, September 1963, (submitted on NASA Contract NAS1-2974).

6-2 Thermal Integration of Electric Power and Life Support Systems for Manned Space Stations, NASA CR-316, General Electric Company, November 1965, (Final Report for Contract NAS 3-2799).

6-3 Report on a System Comparison and Selection Study of a Manned Orbital Research Laboratory, Douglas Report SM-44612, Appendix B, Artificial Gravity Provisions, September 1963.

6-4 MOL Stabilization Disturbance Effects and Boundaries, General Electric, MSD, Internal Memo 4-35, February 2, 1965.

6-5 Barrene, Marcel; Jaumote, Andre; deVeubeke, Baudouin Fraeijs; Vandenkerchove, Jean. Rocket Propulsion. Elsevier Publishing Company, 1960.

6-6 Zucrow, M. J. Aircraft and Missile Propulsion, Volume I "Thermodynamics of Fluid Flow and Application to Propulsion Engines," Purdue University, John Wiley and Sons, Inc., 1958.

6-7 Sutton, George P. Rocket Propulsion Elements. John Wiley & Sons, Inc., Third Edition, 1963.

6-8 Hawkins, George A. Thermodynamics. John Wiley & Sons, Inc., Second Edition, 1951.

6-9 Streeter, Victor L. Fluid Mechanics. McGraw-Hill Book Company, Inc., Second Edition, 1958.

6-10 Shapiro, Ascher H. The Dynamics and Thermodynamics of Compressible Fluid Flow, Volume I, Ronald Press Company, 1953.

6-11 Yellot, John I., Jr. Supersaturated Steam, ASME Transactions, Volume 56, Pages 411-430, 1934.

6-12 Rettaliata, J. T. Undercooling in Steam Nozzles, ASME Transactions, Volume 58, Pages 599-606, 1936.

6-13 Yellott, J. I. and Holland C. K. The Condensation of Flowing Steam Part I - Condensation in Diverging Nozzles, ASME Transactions, Volume 59, Pages 171-183, 1937.

REFERENCES (CONT'D)

6-14 Lottes, P. A. Expansion Losses in Two Phase Flow. Nuclear Science and Engineering, Volume 9, Page 26, May 1961.

6-15 James, R. Steam Water Critical Flow Properties, Proceedings of IME, Volume 176, No. 26, 1962.

6-16 Neusen, Keneth Fred. Optimizing of Flow Parameters for the Expansion of Very Low Quality Steam. University of California, Lawrence Radiation Laboratory, January 1962.

6-17 Levy, Sol. Prediction of Two-Phase Critical Flow Rate. Journal of Heat Transfer, ASME Transactions, Pages 53-58, February 1965.

6-18 Tinling, Bruce E. Measured Steady State Performance of Water Vapor Jets for Use in Space Vehicle Attitude Control Systems. NASA-AMES Research Center, May 1962.

6-19 Keenan, J. H. and Keyes, F. G. Thermodynamic Properties of Steam. John Wiley and Sons, Inc., 1936.

6-20 ASHRAE Guide and Data Book, 1961, Fundamentals and Equipment.

6-21 Riede, P. M. and Wang, D. Characteristics and Applications of Some Superinsulations. The Linde Company, 1959.

SECTION 7
ABSORPTION REFRIGERATION

7.1 INTRODUCTION

The Absorption Refrigeration System is a method of moving or pumping heat energy from a desired cooling surface to a higher temperature heat rejection surface. The process that operates in the absorption cycle is the absorbing of lower temperature cooling fluid (usually a gas) by a higher temperature absorbent. Heat energy is used to operate the cycle, driving the cooling fluid off by boiling. Figure 7-1 is a schematic of an absorption refrigeration cycle.

The amount of energy required to operate absorption cycle is much higher than for vapor compression cycles; however, heat energy used in the absorption process is far more economical to use than electrical energy used in compression pumping in many cases.

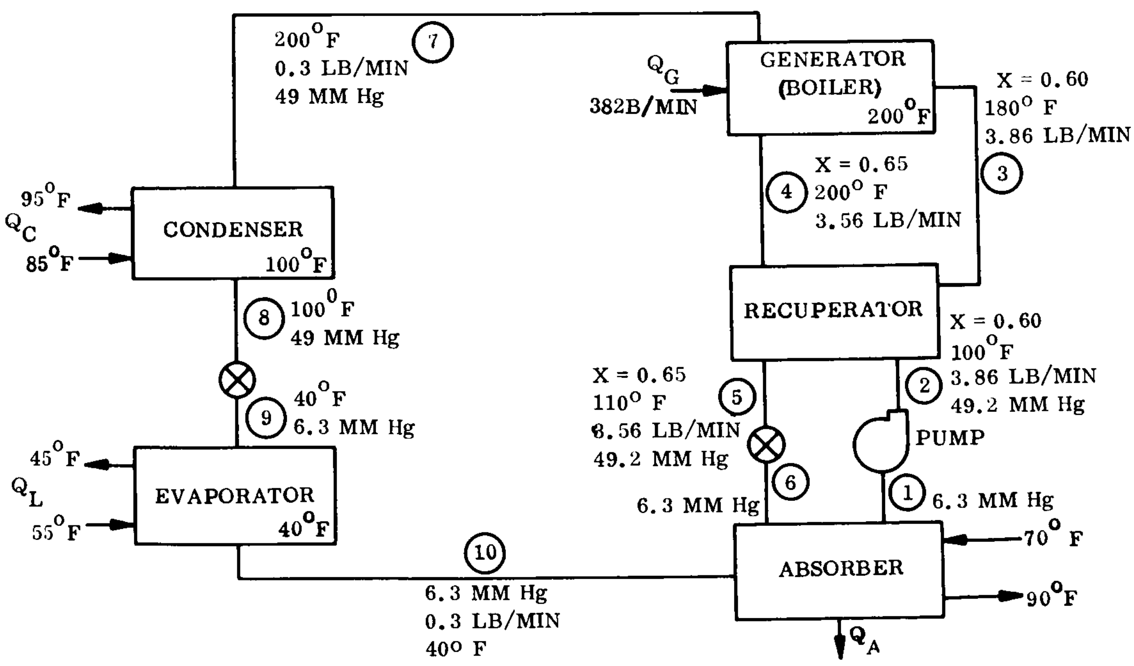


Figure 7-1. Absorption Cycle, Schematic Diagram

The application under consideration in this study is an area where heat energy is readily available from the dynamic power systems, while additional electric power requirements would impose a system weight penalty.

There are several fluid combinations possible for use in absorption refrigeration. The two combinations most widely used are ammonia-water and lithium bromide-water. The lithium bromide-water systems have the highest coefficient of performance (COP) of the absorption systems, but are restricted to above 32°F as an evaporator temperature. For colder temperatures, ammonia-water or one of the Freon combinations with dimethyl ether of tetraethylene glycol (see Reference 7-1) would be used. The temperature requirements of this study dictate the use of lithium bromide-water. Table 7-1 shows the refrigeration system requirements. The following subsections contain cycle analysis, equipment design, integration with MORL Vehicle and a discussion.

TABLE 7-1. ABSORPTION CYCLE REQUIREMENTS

Evaporater Temperature	40°F
Heat Transfer Fluid	30% Ethylene Glycol, 70% Water
Heat Rejection Method	Space Radiator (400°R sink temperature)
Heat Source	Power System Waste Heat
	a. Brayton cycle liquid loop
	b. Mercury Rankine cycle condensing radiator
Capacity	17,176 Btu/hr
	Size for 18,000 Btu/hr (1.5 tons of ref.)
Compared System	40°F Space Radiator

7.2 CYCLE DESCRIPTION AND ANALYSIS

7.2.1 CYCLE DESCRIPTION

The basic system consists of four cycle processes which can be evisioned as four components; evaporator, absorber, generator, and condenser (see Figure 7-1).

The function or cycle process of each is as follows:

a. Evaporator - Provide cooling to external heat load. Saturated water absorbs heat at constant temperature until the saturated vapor line is reached. Some superheat is possible, but not of principal benefit to the cycle.

b. Absorber - Saturated steam is absorbed by a lithium bromide-water solution. The vapor is typically much colder than the solution, but absorption is an exothermic process and considerable heat is rejected at the solution temperature. In real systems the solution is sprayed into the entering vapor to increase the absorption surface area. The variation of specific gravity with percent of water vapor in solution (see Figure 7-2) allows separation such that the portion with the highest percentage of captured water is pumped to the generator, while the portion with the lowest percentage of captured water is pumped to the spray heads.

   The preceding two processes are on the low pressure side of the system, a pump between the absorber and the generator raises solution pressure for generator and condenser operation.

b. Generator - High temperature heat energy is supplied to drive-off captured water as steam. The resulting liquid is returned to the absorber through a reducing valve. A significant reduction in cycle heat is possible with a heat exchanger or recuperator between the liquid solution leaving the generator and the solution entering the generator.

d. Condenser - The steam is condensed by cooling coils or direct radiation until the saturated water condition is reached. Some sub-cooling is possible, utilizing vapor out of the evaporator in a heat exchanger. The water passes through a reducing valve after leaving the condenser and enters the evaporator as low temperature, very wet steam.

7.2.2 CYCLE ANALYSIS

An ideal cycle analysis of a lithium bromide-water system can be made using equilibrium diagrams (Figures 7-2 and 7-3A and B) and the steam tables (Reference 7-2). A typical cycle is as follows, the cycle is identical to the example used in Reference 7-3.

Evaporator Temperature = 40°F

Condenser and Absorber Temperature = 100°F

Generator Temperature = 200°F

TABLE 7-2. TYPICAL ABSORPTION REFRIGERATION CYCLE DATA

Cycle Point	Pressure Mn Hg	Temp. °F	Quality % LiBr	Enthalpy BTU/lb	Flow lb/Min
1	6.3	100	0.60	-70	3.86
2	49.2	100	0.60	-70	3.86
3	49.2	180	0.60	-35	3.86
4	49.2	200	0.65	-27	3.56
5	49.2	110	0.65	-65	3.56
6	6.3	110	0.65	-65	3.56
7	49.2	200	0	1150.4	0.297
8	49.2	100	0	68	0.297
9	6.3	40	0	68	0.297
10	6.3	40	0	1079.3	0.297

HEAT BALANCE

	Heat Input - Btu/min	Heat Rejected - Btu/min
Q_c		321
Q_L	300	
Q_g		358
Q_g	381.9	
	381.9	379

$$\text{Coefficient of Performance} = \frac{Q_L}{Q_G} = \frac{300}{381.9} = 0.785$$

The following conditions are assumed; all cycle point numbers refer to the identified point in Figure 7-1 and the thermodynamic properties in Table 7-2.

1. The generator liquid input (3) is at saturated liquid condition.
2. Heat Exchanger effectiveness - 0.9 maximum.
3. The liquid leaving the generator (4) is at saturated liquid condition.
4. The liquid leaving the condenser (8) is at saturated condition.
5. The vapor leaving the evaporator (10) is at saturated condition.
6. Cycle pump work is negligible (1) $\longrightarrow$ (2).
7. The reducing valves operate at constant enthalpy ($h_5=h_6$).
8. Point (1) is sub-cooled.
9. Operation can not extend past the crystallization line (see Figure 7-2).

The first step is determination of points (8) and (10). The pressures and enthalpies are determined by the saturation conditions. The required flow (m_9) is found by $\dfrac{Q_L}{(h_{10}-h_9)}$.

Since the system requirement is for 18,000 Btu/hr, $Q_L = 300$ and $M_9 = 0.297$ lb/min. The state point for (4) can be found with the known pressure (49 mn Hg) and temperature (200°F) on Figure 7-3; h = -27 Btu/lb, x = 0.65. Point (3) can be found in the same manner; h = 35 Btu/lb, x = 0.60. The flow rate can be found by:

$$m_4 = m_{10} \frac{(x_3 - x_{10})}{(x_4 - x_3)} \quad = \quad 3.56 \text{ Btu/min.}$$

and

$$m_3 = m_{10} + m_4 \quad = \quad 3.86 \text{ Btu/min.}$$

Point (1) is determined from $x_1 = x_3$ and t = 100°F, h = -70 Btu/lb. Point (2) is identical to point one except for the 49.2 mn Hg pressure. Point (5) is determined by $h_5 = h_4 - \frac{m_3}{m_5}$ $(h_3 - h_2)$, or conservation of energy in the recuperator. Point (6) is identical to (5) except for the 6.3 mn Hg pressure. Point (7) is superheated steam at 49.1 mn Hg and 200°F. The enthalpy of 1151 Btu/lb is found in Reference 7-2, steam tables .

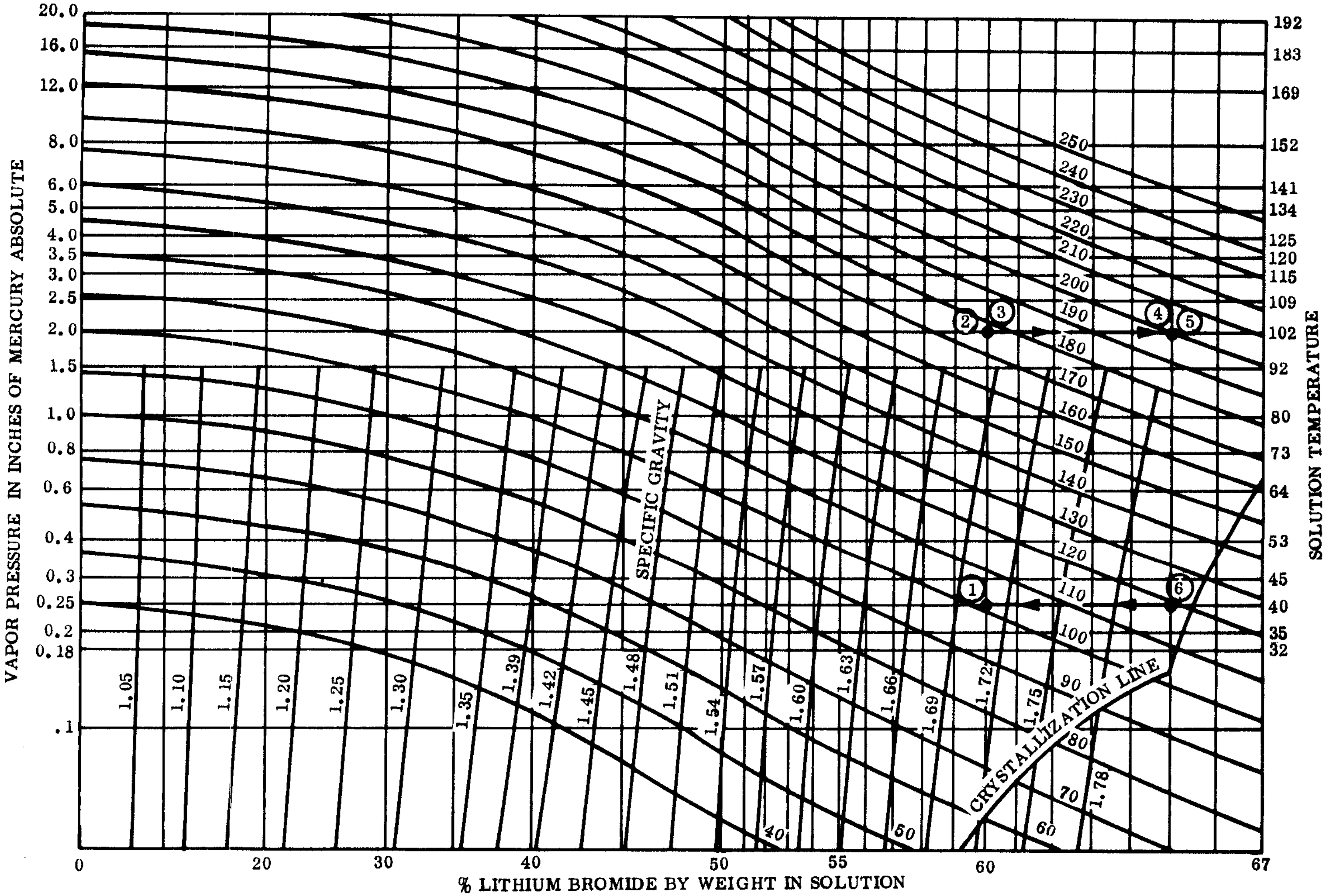


Figure 7-2. Equilibrium Diagram for Lithium Bromide

The thermodynamic properties and fluid flow rates are now known and an energy balance can be computed.

$$Q_L = 300 \text{ Btu/min. (given)}$$
$$Q_g = m_4h_4 + m_7h_7 - m_3h_3$$
$$Q_a = m_6h_6 + m_{10}h_{10} - m_1h_1$$
$$Q_c = m_8 (h_7 - h_8)$$

The results are tabulated in Table 7-2. The balance is in agreement to ~0.5%, better than the accuracy of Figure 7-3. The coefficient of performance is $\frac{Q_L}{Q_G} = \frac{300}{382} = 0.785$. It is of interest to look at the cycle and attempt an optimization or at least define typical design points for various conditions. Ideally, comparison of Carnot Efficiencies defined as $\left(\frac{T_L (T_G - T_A)}{T_G (T_A - T_L)}\right)$, should indicate the relative merit of different operating points. The trends for the temperature are increased efficiency with increasing generator temperature, decreasing absorber temperature, and increasing load temperature. From a practical standpoint, however, the possibility of crystallization at high LiBr concentrations severely limits optimization. Note that the example cycle (in Figures 7-2 and 7-3) comes very near the limit line. The problem becomes evident when an attempt is made to lower condenser and absorber temperatures in any effort to increase the COP. Due to the existence of the crystallized area the generator temperature must also be decreased. The net result (see Table 7-3) is an increase of COP to 0.8. In order to evaluate effect, another case was conducted with evaporator temperature raised to 60°F. The result, as shown in Table 7-3 is an increase of COP to 0.815.

Another consideration is the cycle breadth or the quality spread achieved between x_3 and x_4 by the generator.

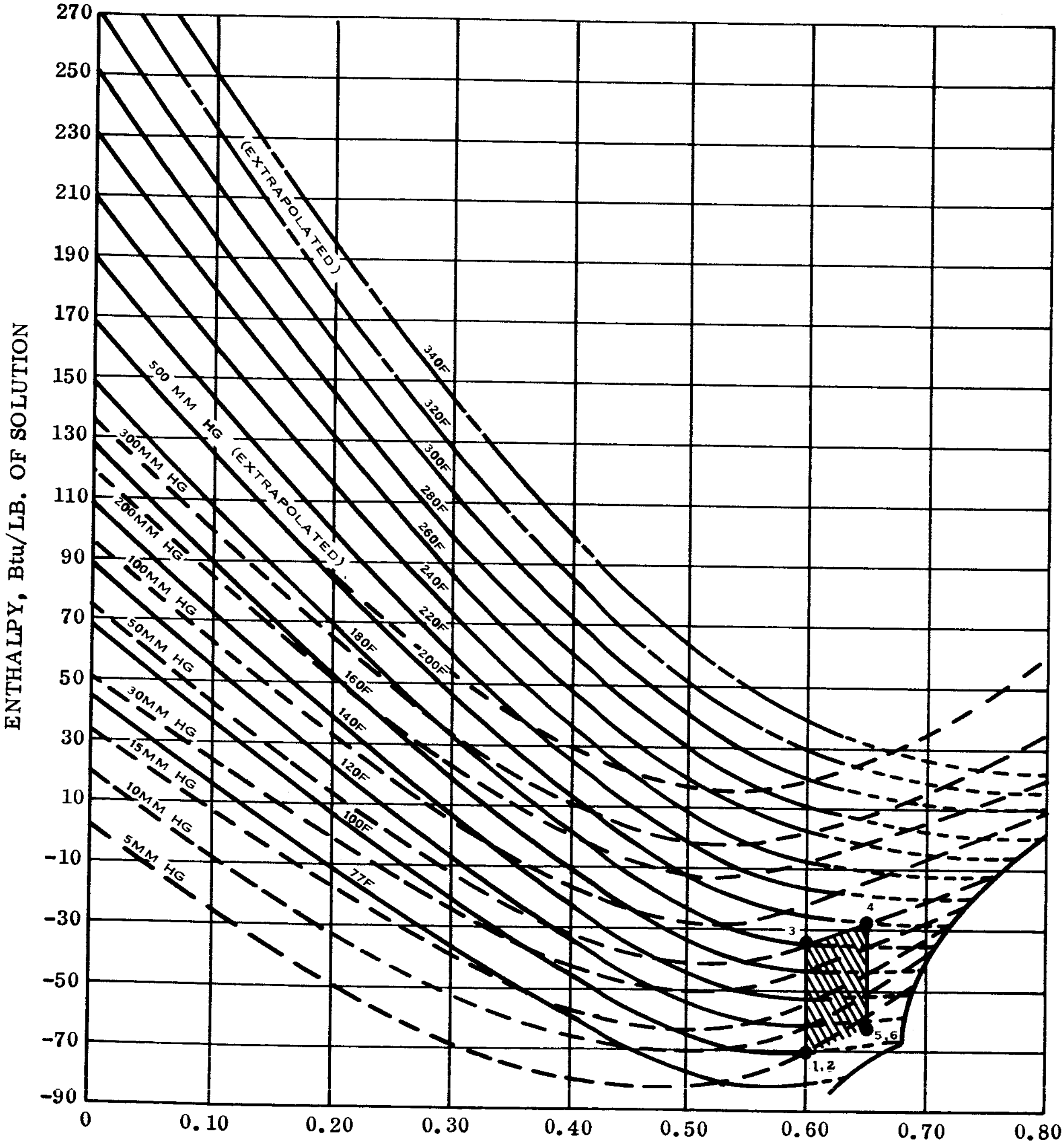


Figure 7-3. Equilibrium Diagram for Lithium Bromide and Water Solution (Sheet 1 of 2)

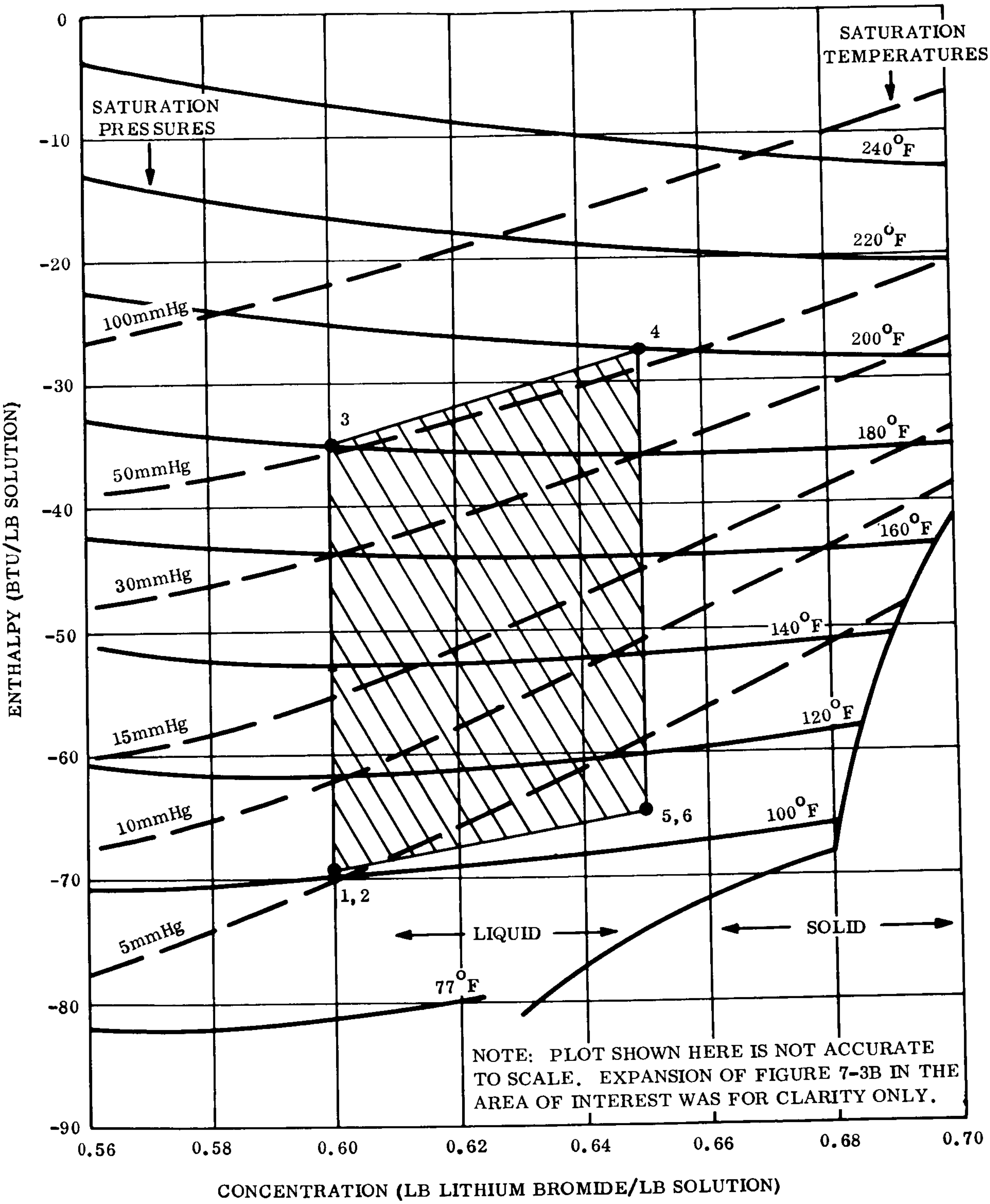


Figure 7-3. Equilibrium Diagram for Lithium Bromide Water Solution (Sheet 2 of 2)

TABLE 7-3. OTHER LITHIUM BROMIDE - WATER ABSORPTION CYCLE EXAMPLES

State Point	Pressure Mn Hg	Temp. °F	Quality % LiBr	Enthalpy Btu/lb	Flow lb/Min
CASE 1					
1	6.3	90	0.565	-74	2.044
2	36.1	90	0.565	-74	2.044
3	36.1	150	0.565	-46	2.044
4	36.1	190	0.66	-32	1.75
5	36.1	112	0.66	-62.4	1.75
6	6.3	112	0.66	-62.4	1.75
7	36.1	200	0	1146	0.294
8	36.1	90	0	58	0.294
9	6.3	40	0	58	0.294
10	6.3	40	0	1079.3	0.294

Let $T_{A,C}$ = 90°F

T_G must be = 190°F

T_E = 40°F

	Heat in Btu/min	Heat out Btu/min
Q_L	300	
Q_A		360
Q_C		320
Q_G	375	
	375	380

COP = 0.8

State Point	Pressure Mn Hg	Temp. °F	Quality % LiBr	Enthalpy Btu/lb	Flow lb/Min
CASE 2					
1	13.5	100	0.545	-64	1.824
2	49.2	100	0.545	-64	1.824
3	49.2	160	0.545	-41	1.824
4	49.2	200	0.65	-28	1.53
5	49.2	130	0.65	-55.3	1.53
6	13.5	130	0.65	-55.3	1.53
7	49.2	200	0	1150.4	0.294
8	49.2	100	0	68	0.294
9	13.5	60	0	68	0.294
10	13.5	60	0	1087.4	0.294

Let $T_{A,C}$ = 100°F

Let T_G = 200°F

Let T_E = 60°F

	Heat in Btu/min	Heat out Btu/min
Q_L	300	
Q_A		352
Q_C		318
Q_G	370	
	370	370

COP = 0.81

As the breadth increases, recuperator effectiveness decreases; however, the amount of fluid pumped decreases with increasing breadth, offsetting the recuperator efficiency loss. The constraint on increasing the breadth is decreasing absorber-condenser heat rejection temperatures for a given evaporator temperature (see Figure 7-2). An increase in absorber temperature quickly diminishes the breadth. At 120°F, the flow to a 40°F evaporator goes to infinity (see Figure 7-2).

The 40°F evaporator temperature requirement limits the absorber temperature to less than 120°F. For higher rejection temperatures a dual cycle is next examined. Figure 7-4 shows a dual (cascade) cycle schematic. The cycle point thermodynamic properties and heat balance are shown in Table 7-4. Since it is desirable to maximize the cycle COP, the cycle points are chosen such that the condenser rejection of the higher temperature loop supplies the generator heat for the lower temperature loop. The low temperature condenser and absorber heat supply the high temperature load. For a 200°F absorber and condenser rejection temperatures the COP is 0.32. As a comparison, calculations were made for a cascade ammonia-water system (from Reference 7-4). Two difficulties presented themselves. First was lack of information for high temperature water-ammonia properties, second was the inability to obtain high condenser temperatures at the high pressure side. Table 7-5 presents cycle thermodynamic data for high pressure absorber and condenser temperatures of 200°F and 140°F, respectively. The computed COP is 0.214, or about 1/3 less than the Lithium Bromide-Water Cycle (which has a better, 200°F condenser temperature). The cycle schematic appears in Figure 7-5. Note that this is a true cascade cycle, having only one condenser and evaporator, while the LiBr cycle has two condensers and evaporators. The coefficient of performance for dual or cascade cycles appears to be about one-half that of the single cycle.

7.3 EQUIPMENT DESIGN

To evaluate the usefulness of the absorption refrigeration process it is necessary to make preliminary design calculations of the equipment required; to obtain weight, power require-

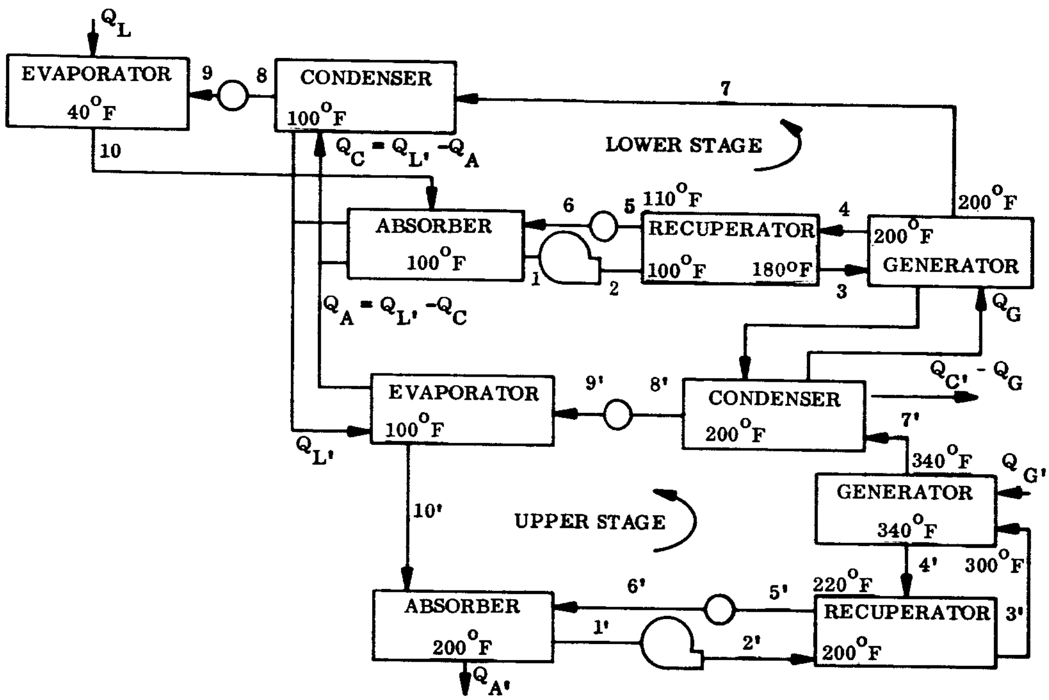


Figure 7-4. Dual LiBr-$H_2$0 Absorption Cycle

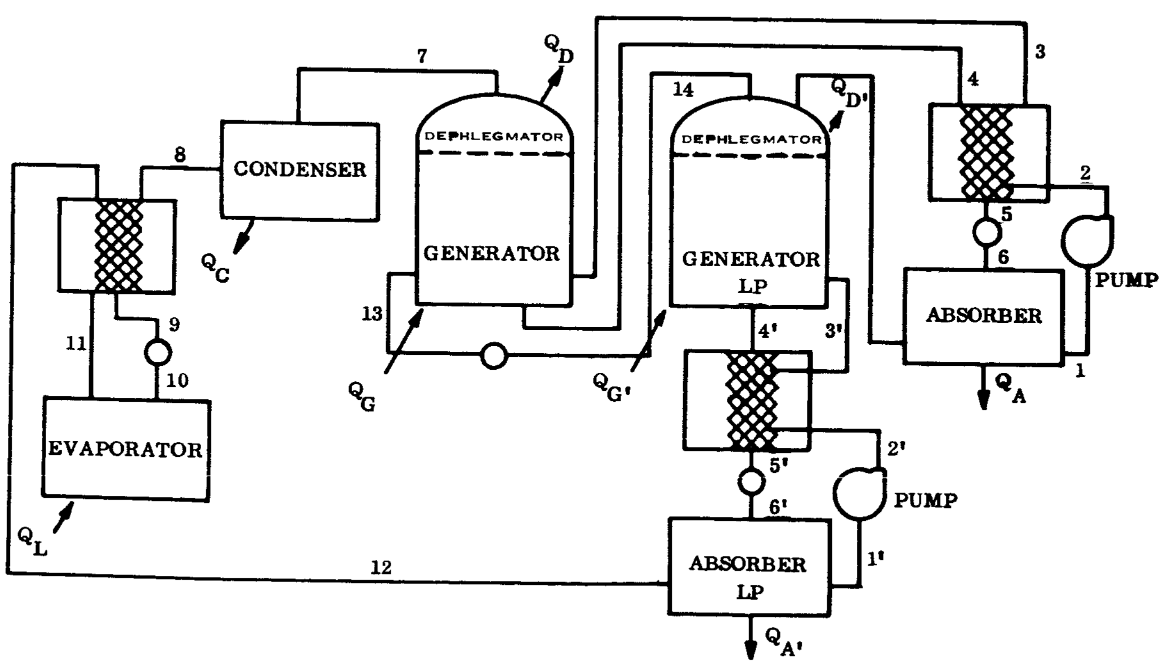


Figure 7-5. Ammonia-Water Cascade Absorption Cycle

TABLE 7-4. DUAL (CASCADE) LITHIUM BROMIDE-WATER ABSORPTION PROCESS THERMODYNAMIC DATA

State Point	p	t	x	h	m
1	6.3	100	0.60	-70	3.86
2	49.2	100	0.60	-70	3.86
3	49.2	180	0.60	-35	3.86
4	49.2	200	0.65	-27	3.56
5	49.2	110	0.65	-65	3.56
6	6.3	110	0.65	-65	3.56
7	49.2	200	0	1150.4	0.297
8	49.2	100	0	68	0.297
9	6.3	40	0	68	0.297
10	6.3	40	0	1079.3	0.297
1'	49.2	200	0.605	-26	7.48
2'	596	200	0.605	-26	7.48
3'	596	300	0.605	19	7.48
4'	596	340	0.67	30	6.75
5'	596	220	0.67	-19.9	6.75
6'	49.2	220	0.67	-19.9	6.75
7'	596	340	0	1212.3	0.725
8'	596	200	0	168	0.725
9'	49.2	100	0	168	0.725
10'	49.2	100	0	1104.7	0.725

HEAT BALANCE

	In Btu/min	Out Btu/min	
Q_C		321	
Q_L	300		
Q_A		358	
Q_G	381.9		Net
$Q_{C'}$		756	374
$Q_{L'}$	679		
$Q_{A'}$		863	
$Q_{G'}$	939.0		
	1618	1619	

$$COP = \frac{300}{939} = 0.32$$

TABLE 7-5. AMMONIA-WATER CASCADE ABSORPTION CYCLE THERMODYNAMIC DATA

State Point	p	t	x	n	m	Condition
1	200	200	0.408	110.0	2.536	Saturated
2	500	200	0.408	111.1	2.536	
3	500	260	0.408	179	2.536	Saturated
4	500	340	0.235	271	1.93	Saturated
5	500	255	0.235	182	1.93	
6	200		0.235	182	1.93	
7	500	220	0.965	715	0.6	Saturated
8	500	140	0.965	190	0.6	Saturated
9	500	82	0.965	120	0.6	
10	50	26	0.965	120	0.6	
11	50	40	0.965	620	0.6	
12	50	140	0.965	690	0.6	Saturated
13	200	340	0.235	271	0.006	
1'	50	200	0.150	135	7.13	Saturated
2'	200	200	0.150	135.5	7.13	
3'	200	305	0.150	245	7.13	Saturated
4'	200	340	0.075	295	6.53	Saturated
5'	200	225	0.075	175.4	6.53	
6'	50		0.075	175.4	6.53	
7'	200	200	0.958	711	0.606	Saturated

ENERGY BALANCE

Component	Heat in Btu/min	Heat out Btu/min
Absorber		503
Pump	3	
Generator	595	
Dephlegmator		150
Condenser		315
Evaporator	300	
Absorber LP		584
Pump LP	4	
Generator LP	807	
Dephlegmator LP		183
	1709	1735

$$\text{COP} = \frac{300}{1402} = 0.214$$

ments, and identify problem areas. Table 7-6 summarizes the design result. A sketch of the assembly is depicted in Figure 7-6. The total weight including fluid inventory is 43 pounds. The power required for pumping, including the coolant radiators pressure losses is 69 watts. Detailed calculations for the following components can be found in Appendix B.

TABLE 7-6. ABSORPTION REFRIGERATION DESIGN RESULTS

Unit	Weight (lb)	Coolant t_{in} (°F)	Coolant t_{out} (°F)	Coolant t_{eff} (°F)	Pump Power (Watts)	U
Evaporator	9.2	55	45		36.7	100
Condenser	4.8	85	95	90	17.5*	200
Generator	8.8	238	208		3	300
Absorber	12.8	70	90	80	11 *	200
Heat Exchanger	3.5	200	110		0.2	200
Pump, Cycle and Recirculating	0.5				0.8	
	39.6				69	
H_2O Inventory	1.0					
Support Structure	2.4					
System Weight	43.0 lb.				*Assumed 3-psi drop in radiator	

7.3.1 EVAPORATOR DESIGN SUMMARY

The evaporator consists of a cone outer wall (water vapor retainer) with stacked concentric circle tube-fin arrangements carrying the ethylene glycol-water mixture (30%-70%) to be cooled. The water vapor enters at 40°F as low quality steam and exists as saturated steam. The ethylene glycol-water enters the heat exchanger at 55°F and leaves at 45°F.

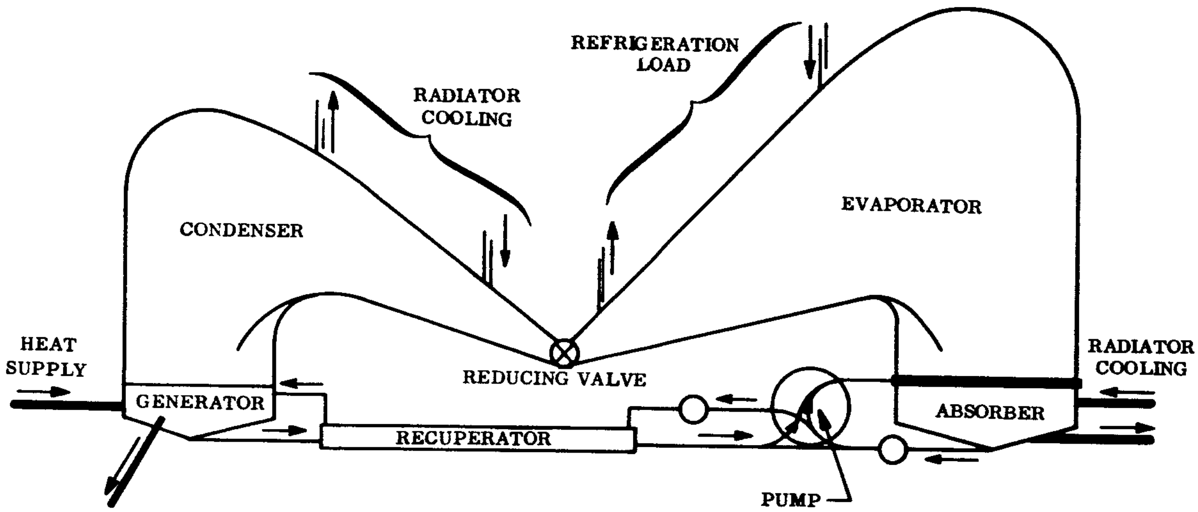


Figure 7-6. Absorption Refrigeration System

The module weights are:

Heat Exchanger	3.4 lb
Ethylene Glycol-Water	1.8
Conical Container	2.2
Container Insulation	0.3
Support Structure	0.5
Pump and Motor	1.0
	9.2 lb

The pump power (assuming η motor-pump = 0.50) is 36.7 watts.

7.3.2 CONDENSER DESIGN SUMMARY

The condenser consists of a cone outer wall (water vapor container) with stacked concentric circle tube-fin arrangements carrying the cooling fluid. The water is condensed from 100°F steam to 100°F saturated water. The cooling fluid enters at 85°F and leaves at 95°F. The associated radiator is discussed in Section 7.4. The condenser parts weights are:

Heat Exchanger		1.7 lb
Ethylene-Glycol-Water Inventory		0.9
Conical Container		1.1
Container Insulation		0.2
Support Structure		0.2
Pump and Motor		0.7
	Total	4.8 lb

The pump power required (assuming η motor-pump = 0.50, and ΔP of the radiator is 3 psi) is 17.5 watts.

7.3.3 GENERATOR DESIGN SUMMARY

The generator is a cylindrical-sided, tapered-bottom bowl with concentric rings of fin-tube heating coils. An interconnecting flue is attached to the top and connects to the condenser entrance. The lithium bromide-water solution is maintained at 200°F by the heating coils, which supply heat from the power system heat rejection loop. The part weights are:

Container		0.4 lb
Container insulation		0.3
Heat Exchanger		1.1
Heat Fluid Inventory		0.9
Lithium bromide-water Inventory		5.1
Insulated Flue		1.0
	Total	8.8 lb

The pump power required to move the heating fluid through the coils is three watts.

7.3.4 ABSORBER DESIGN SUMMARY

The absorber, like the generator, is a cylindrical-sided, tapered-bottom bowl. Cooling coils consisting of concentric rings of fin-tube arrangements remove the heat of absorption to maintain the temperature at 100°F. The top is attached to the evaporator exhaust by a flue. A recirculating pump and spray heads around the absorber periphery (above the liquid level) increase the absorption surface area of the lithium bromide-water solution in absorbing water vapor exhausting from the evaporator. The parts weights are:

Absorber Container		0.3
Heat Exchanger		0.9
Ethylene Glycol-Water Inventory		2.4
Lithium Bromide-Water Inventory		7.7
Insulated Flue		1.5
	Total	12.8 lb

The pump power required to circulate the cooling fluid is 10.9 watts.

7.3.5 RECUPERATOR DESIGN SUMMARY

The recuperator is of counterflow design with the returning refrigerant from the generator flowing in longitudinal tubes. The external container is a 3/4-inch by 0.13-inch rectangle containing the refrigerant traveling from the absorber to the generator. The total length is 20 feet folded into five 4-foot sections. The temperature rise in the cooler fluid is 100°F, while the temperature decrease in the warmer fluid is 200 to 110°F. The unit weights are:

Fin-Tube Heat Exchanger		0.61 lb
Refrigerant		1.2
External Container		0.9
Insulation		0.8
	Total	3.5 lb

Pump power required is 0.15 watt.

7.3.6 PRESSURIZING AND RECIRCULATING PUMP POWER

The system requires a pump to raise the fluid out of the absorber from 6.3 mn Hg pressure to 49.2 mn Hg. The recirculation in the absorber can be accomplished with the same pumping pressures. Assuming that twice the fluid is recirculated as is sent to the generator, the resulting pump power is 0.8 watt.

7.4 ABSORPTION REFRIGERATION INTEGRATION WITH MORL VEHICLE

The MORL Vehicle application of the absorption refrigeration system is to provide heat rejection for equipment cooling and cabin cooling fluid loops. The two loops are mixed and supply Ethylene Glycol-Water (30%-70%) at 55°F to the absorption refrigeration evaporator. The evaporator removes 17,176 Btu/hr of heat energy returning the fluid at 45°F. This energy is added to that provided by the power system at the refrigeration boiler (generator), and the total heat is rejected in space radiators, which remove heat from the absorber and condenser. Since the generator heat is not supplied at the same temperature for the two power systems, thermal integration will differ to an extent.

7.4.1 BRAYTON POWER SYSTEM INTEGRATION

The wast heat in the liquid radiator loop is supplied to the Absorption Refrigeration (AR) equipment at 325°F and leaves at 204°F. The AR Generator temperature was taken as 200°F or just below the liquid temperature leaving the generator. The waste heat available is, therefore, just sufficient to drive the full size absorption refrigeration system. The radiator parameters of interest for the AR radiators and the waste heat radiators are presented in Table 7-7. The resulting total weight for the absorption system and its heat rejecting radiators is 303.5 pounds. The radiator area is 579 feet.

TABLE 7-7. RADIATOR DATA FOR ABSORPTION REFRIGERATION AND POWER SYSTEMS

	Waste Heat Radiator	Condenser Radiator	Absorber Radiator
BRAYTON CYCLE			
T_{in}, °F	200	95	90
T_{out}, °F	38	85	70
T_{eff}, °F	120	90	80
η_{fin}	0.75	0.75	0.75
ϵ	0.9	0.9	0.9
Q/A, Btu/hr-ft^2	102	76.9	69.3
w, lb/ft^2	0.472	0.45	0.45
Q, Btu/hr	28,400	19,200	22,800
A, Ft^2	316	250	329
W, lb	149	112.5	148
MERCURY RANKINE CYCLE			
T_{in}, °F	605	95	90
T_{out}, °F	605	85	70
T_{eff}, °F	605	90	80
η fin	0.545	0.75	0.75
ϵ	0.9	0.9	0.9
Q/A, Btu/hr-ft^2	1070	76.9	69.3
w, lb/ft^2	0.78	0.45	0.45
Q, Btu/hr	108,400	19,200	22,800
A, Ft^2	101	250	329
W, lb	79	112.5	148

7.4.2 MERCURY RANKINE POWER SYSTEM INTEGRATION

Provision must be made for reducing the 600°F Rankine condenser temperature to near the generator boiling level. Due to the crystallization constraints in this particular case, a higher generator temperature would only reduce the COP. The simplest method would be to provide a simple loop in the condensing radiator that would heat generator circulating fluid only to some lower temperature, such as 250°F. This might add up to one pound to the total AR System weight. The Mercury Rankine wast heat supply is five times the requirements of the absorption refrigeration system. Therefore, there is capability for increasing the refrigeration load should real MORL missions dictate it.

The radiator parameters of interest are presented in Table 7-7. The AR radiators are unchanged from the Brayton Cycle. The waste heat radiator is typical for the Mercury Rankine application.

7.4.3 LOW TEMPERATURE RADIATOR

As an alternate method of rejecting heat from the MORL vehicle, a simple low temperature radiator can be used, since its outlet temperature is about 505°R, well above the sink temperature of 400°F. Table 7-8 presents data for a typical low temperature radiator, and compares the results with the absorption refrigeration (AR) systems. It is shown that the AR is lighter than the low temperature radiator (LTR) by 20 lb for the Brayton cycle. In considering the radiator calculations this is a small number, and does not represent a clear-cut weight victory for the absorption refrigeration system. The LTR is 100 lb lighter than the AR using the integrated Mercury Rankine Power System. The large increase of the cabin radiator from 404 square feet to 579 square feet lowers the reduction of the power system radiation from 123 square feet to 101 square feet. The increased AR complexity strongly favors having a low temperature radiator for the MORL application.

TABLE 7-8. ABSORPTION REFRIGERATION, LOW TEMPERATURE RADIATOR COMPARISON

Low Temperature Radiator Parameters	
T_{in}, °F	55°F
T_{out}, °F	45°F
T_{eff}, °F	48°F
η fin	0.7
ϵ	0.9
Q/A, Btu/ft^2	44.6
w, lb/ft^2	0.444
Q, Btu/hr	18,000
A, ft^2	404
W, lb	180

Comparison of Heat Rejection	Brayton Cycle		Mercury Rankine Cycle	
	Absorption Refrigeration	Low Temp. Radiator	Absorption Refrigeration	Low Temp. Radiator
Weight for equipment and cabin cooling, lb	303.5	180	304.5	180
Weight for waste heat rejection, lb	149	293	79	96.5
Total Weight, lb	452.5	473	383.5	276.5
Radiator area, cabin, ft^2	579	404	579	404
Radiator area, waste heat, ft^2	316	514	101	123.6
Total Area, ft^2	895	918	680	527.6

7.4.4 MECHANICAL INTEGRATION CONSIDERATIONS

The equipment designed in Section 7.3 did not consider operation under zero-gravity conditions. For integration with the MORL Vehicle as defined by this study, some means of gravity simulation must be used in order for the absorption refrigeration equipment to operate. The generator and absorber require gravity for differing specific gravity separation while the condenser requires gravity to collect the condensed water. A solution to these problems is to rotate the system. A conceptual sketch of the rotating system is presented by Figure 7-7. The system should be well balanced to minimize bearing torques. The most important part in design is the rotating seals at the top and the bottom for the fluid lines. The power required to rotate the system depends directly on the seal forces required to eliminate any leakage.

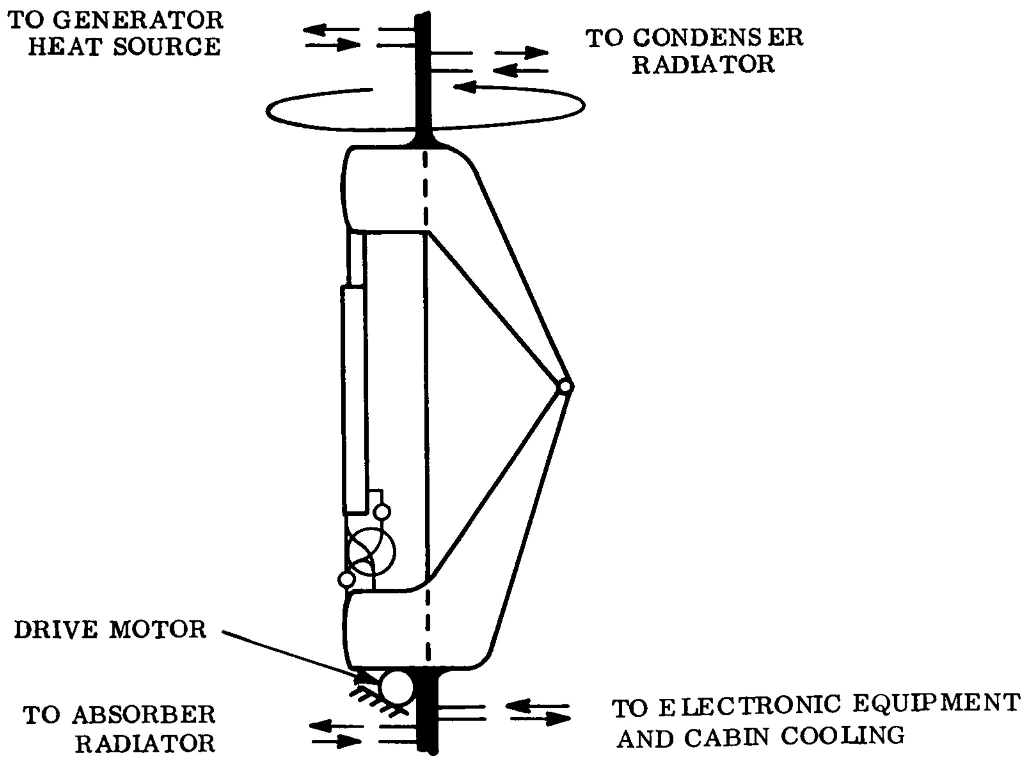


Figure 7-7. Rotating Absorption Refrigeration System

7.5 DISCUSSION

7.5.1 ABSORPTION CYCLE

Some additional thermodynamic improvement can be made to the absorption cycle by adding a heat exchanger to cool the condenser liquid with gas leaving the evaporator. The net benefit is at best about 2% increase in COP, so it was not considered important in the analysis.

Another concept that could have applications is to cascade a heat pump and absorption cycle for increased rejection temperature elevation. This could be used where a refrigeration requirement exists and modest amounts of was heat energy and electrical power are available.

The Freon combinations described in Reference 7-1 do allow the possibility of operation in a zero gravity field. Differing electrical properties with refrigerant-absorbent con- concentrations allows substitution of an electrical field for a gravity gield. However, the maximum theoretical COP found was 0.5, leaving the cycle too inefficient to supply necessary cooling in the Brayton cycle case, and also resulting in higher system weights than the low temperature radiator. Clearly, a higher COP is necessary for the Freon Dielectric process to compete for the MORL Vehicle application.

7.5.2 EQUIPMENT DESIGN

The conical-cylindrical aluminum evaporator, condenser and connecting flues were treated as simple monocoque structures in the design calculations. For real structures, a probable approach is to modify this to circumferential corrugations, eliminating buckling problems.

The tube-fin design used in the heat exchanger loops throughout the AR system makes use of very thin (0.010 and 0.020 inch thick) aluminum which is difficult to weld or braze. A simple fabrication method is to pressure fuse two flat sheets together with a foreign material in the desired tube run. The tubes are then pressure-formed by applying pressure at the ends until local yielding occurs. This process is in use at present for making heat exchanger elements.

For an absorption refrigeration system that would not have to contend with the zero-gravity problem, a condensing radiator might offer a method of reducing weight and pumping power for the refrigeration system condenser. The fin efficiency could be higher, and the effective temperature would become the condensing temperature. The pump work required for the ethylene glycol-water would be eliminated.

7.5.3 SYSTEM CONSIDERATIONS

Areas where an absorption refrigeration system could be of benefit include:

a. Areas with higher sink temperatures such as Lunar Surface; Venus, Mercury, and Solar Probes.

b. Where radiator area is limited and some method of raising rejection temperature is required.

c. Where refrigeration temperatures approaching sink temperatures (or below) are required.

In terms of weight, where the waste heat is available, the absorption refrigeration system compares favorably with vapor compression refrigeration methods. In terms of cost, the AR system use may be cheaper than requiring additional isotopic heat for vapor compression power. It is possible that cascade vapor compression-absorption refrigeration combinations would find use where refrigeration to rejection temperature differences are very large.

7.6 REFERENCES FOR SECTION 7

7-1 Studies of Fluorinated Refrigerants for Absorption Refrigeration, FREON Technical Bulletin RT-28, FREON Products Division, E. I. duPont de Nemours & Co.

7-2 Keenan, Joseph H., and Keyes, Frederick G "Thermodynamic Properties of Steam." Massachusetts Institute of Technology, John Wiley & Sons, Inc., 1956.

7-3 Threlkeld, James L.,"Thermal Environmental Engineering." University of Minnesota, Prentice-Hall Inc., 1962.

7-4 Bosnjakovic, Dr. Fran, Director, Institute of Technical Thermodynamics, Stuttgart. Technical Thermodynamics (Translated by Perry L. Blackshear, Jr.), University of Minnesota, Holt, Rinehart and Winston, 1956.

7-5 Streeter, Victor L., "Fluid Mechanics." University of Michigan, McGraw-Hill Book Company, Inc., Second Edition, 1958.

7-6 Hanson, K. L., "Thermal Integration of Electrical Power and Life Support Systems for Manned Space Stations." General Electric Company, MSD, (Final Report for NASA Contract NAS 3-2799). NASA CR-316.

7-7 ASHRAE Guide and Data Book, 1961, Fundamentals and Equipment.

7-8 Brown, Aubrey I., and Marco,Salvatore M., "Introduction to Heat Transfer." Ohio State University, McGraw-Hill Book Company, Inc., Third Edition, 1958.

7-9 McAdams, William H.,"Heat Transmission." Massachusetts Institute of Technology, McGraw-Hill Book Company, Inc., Second Edition, 1942.

7-10 Hutchinson, Francis W.,"Design of Refrigeration Systems for Air Conditioning." University of California, The Industrial Press, 1963.

SECTION 8

BRAYTON CYCLE POWER SYSTEMS

8.1 INTRODUCTION

The Brayton Cycle power systems convert isotope or solar heat energy to electric power by compressing Argon, heating it, and then allowing it to expand through a turbine which drives an alternator. Addition of a regenerator and a space radiator to the system allows closed-cycle operation.

The initial thermal integration study examined and analyzed Brayton system components, isotope and solar sources, radiators, and heat exchangers. The intention of this section is to resize the power system for an increased power level and to consider the effects of using a liquid tube radiator, and variation of compressor inlet temperature (radiator temperature) on system weight. The change in system size is based on the data assembled in the initial study. The two new topics require more analysis.

The following sections will first describe two non-integrated systems, then consider the liquid-loop radiator and rejection temperature variation, and finally, describe resulting integrated power systems.

8.2 NON-INTEGRATED SYSTEMS

8.2.1 DESCRIPTION

The analysis of the Brayton cycle power systems is based on the use of two sources of heat. These are an Isotope heat source using Pu_{238} and a Solar Collector/Absorber. The major components, which were scaled in size, weight and performance from the design used in Reference 8-1, are:

a. Pu_{238} Isotope Heat Source, heat exchanger and shield with source/shield temperature control; Solar Collector/Absorber, Collector/Absorber deployment mechanism and absorber temperature control

b. Turbine driven gas generator

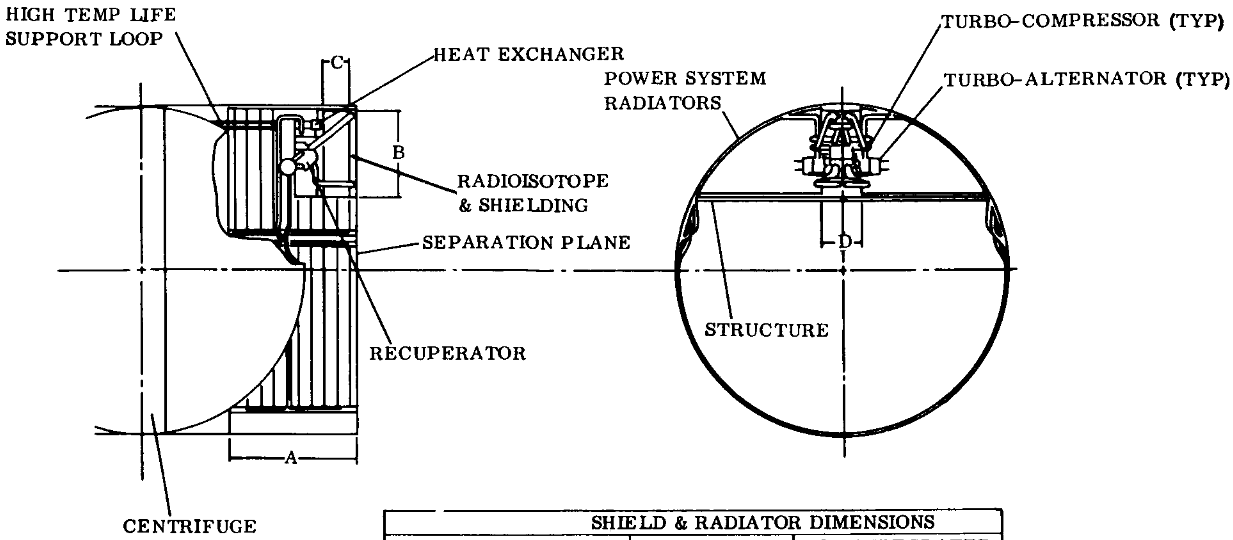
c. Turbine driven alternator

d. Recuperator

e. Argon radiator

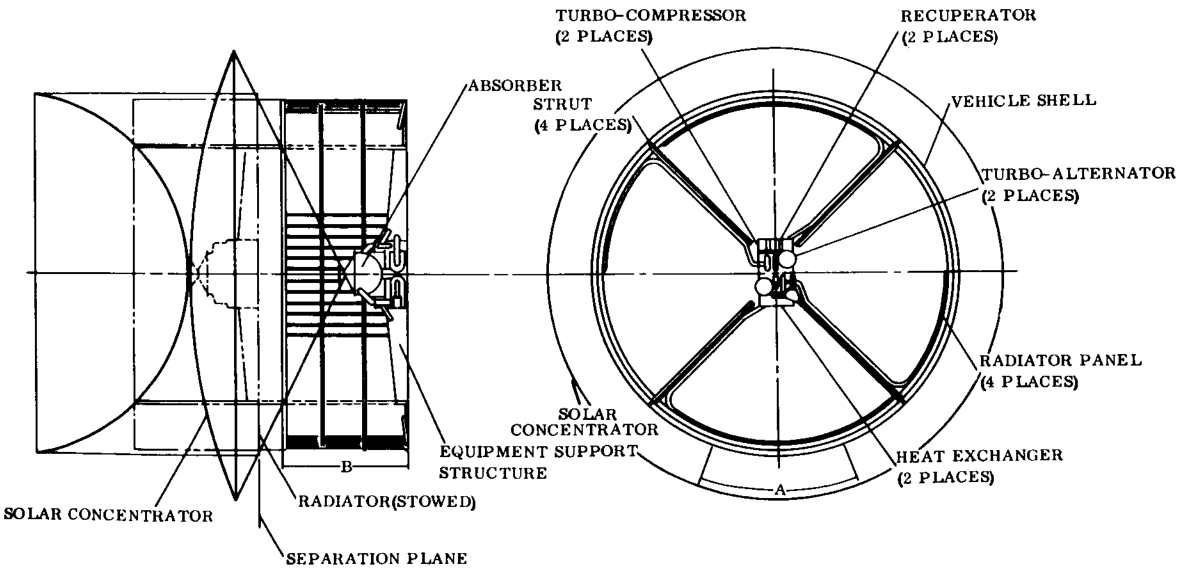
The arrangement of the turbomachinery, heat sources and waste heat radiators is the same as was used in Reference 8-1. This is shown in Figures 8-1 and 8-2. Figure 8-3 shows the system schematic and Figure 8-4 shows the temperature/ entropy diagram for the Brayton cycle. Cycle performance is based on the following conditions:

Compressor inlet	536°R	6.0 psia
Compressor outlet	800	13.8
Heat source inlet	1470	13.66
Compressor turbine outlet	1950	12.97
Power turbine inlet	1678	8.10
Power turbine outlet	1545	6.37
Radiator (heat exchanger) inlet	875	6.32
Turbine work/ lb argon		17.4 Btu/ lb
Heat added/ lb		61.0 Btu/ lb
Heat rejected/ lb argon		42.5 Btu/ lb
Turbo-alternator turbine efficiency		0.85
Turbo compressor efficiency		0.81
Turbo compressor turbine efficiency		0.84
Recuperator effectiveness		0.90



SHIELD & RADIATOR DIMENSIONS		
DIMENSION	INTEGRATED	NON-INTEGRATED
A. (HEADER LENGTH)	60 IN.	60 IN.
B. (SHIELD LENGTH)	68.6 IN.	69.6 IN.
C. (SHIELD HEIGHT)	26.24 IN.	26.65 IN.
D. (SHIELD WIDTH)	34.35 IN.	38.6 IN.

Figure 8-1. Power System Component Layout, Isotope Brayton



DIMENSION	INTEGRATED	NON-INTEGRATED
A. (RADIATOR HEADER LENGTH)	120.0 IN.	192 IN.
B. (RADIATOR TUBE LENGTH)	86.4 IN.	133.2 IN.
COLLECTOR DIAMETER	23.8 FT.	28.8 FT.

Figure 8-2. Solar Brayton Power System Layout

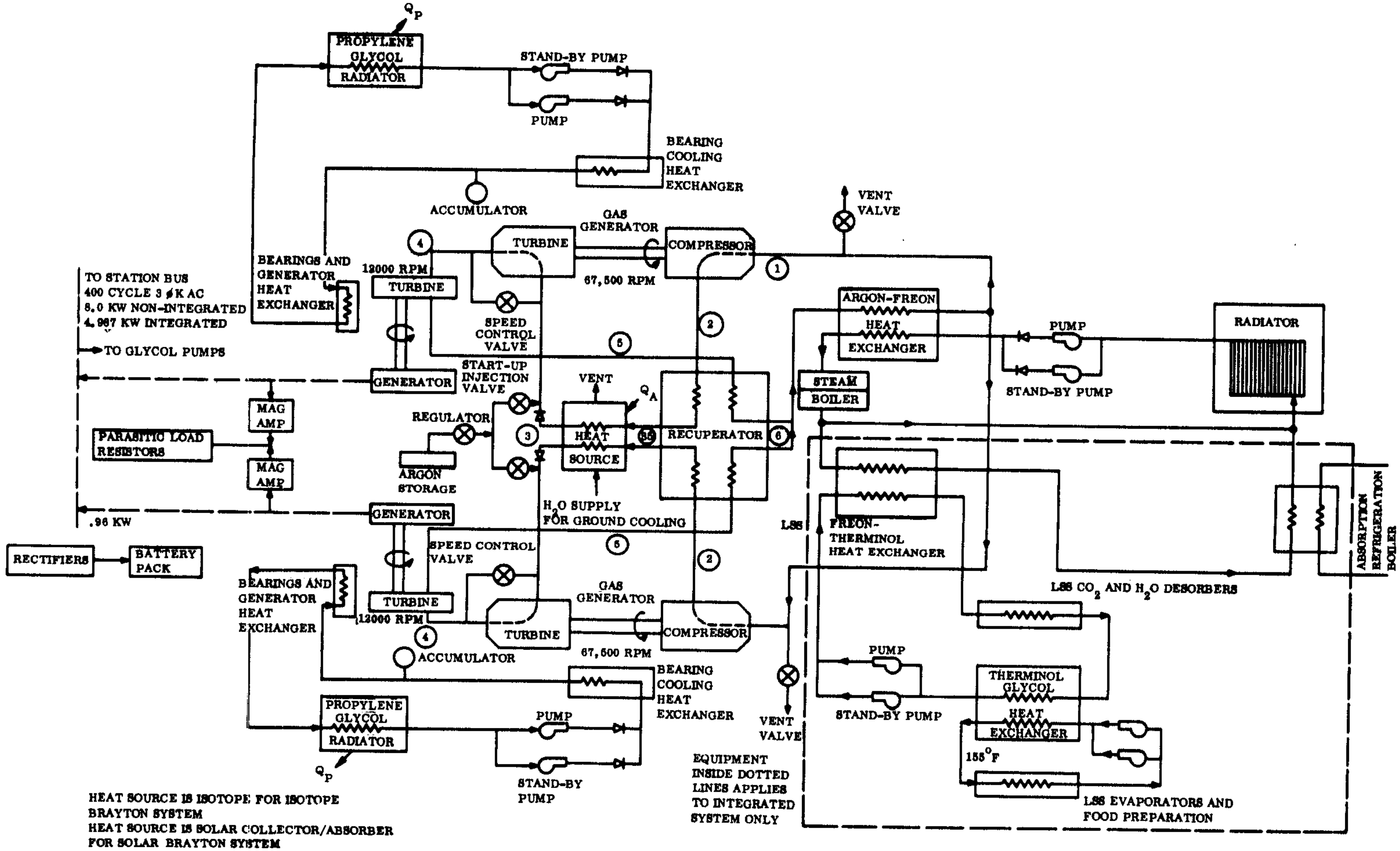


Figure 8-3. Brayton Cycle Schematic

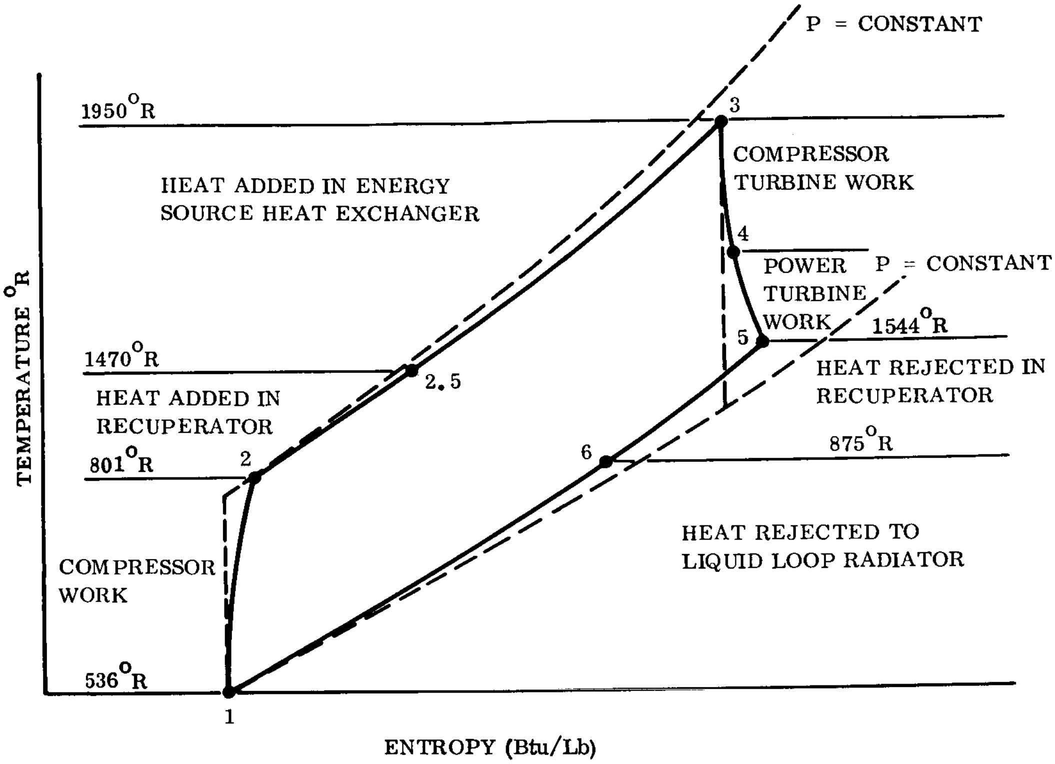


Figure 8-4. Brayton Cycle Operating Points

8.2.2 COMPONENT DESIGN

8.2.2.1 Isotope Package

The heat source power requirements were determined from the cycle heat balance shown in Table 8-1. For the non-integrated power system, a heat source of 40 KW_t is required. The package design is similar to the design used in Reference 8-1. In that design, 30 KW_t was furnished in a 30-tube heat exchanger with each tube being 2.23 inches in diameter and 45 inches long. For this design, the same size tube is used with the number being increased to provide 40 KW_t. The weight is scaled directly, $(\frac{40}{30})$ x 730 lb = 973 lb. Lithium hybride is used as the shield material. From Figure 6-13 of Reference 8-1 the shield thickness is determined to be 12.6 inches. The shield volume was calculated to be 50,382 cubic inches. With a reinforced lithium hybride weight of 57.1 lb/ft^3 the shield weight is calculated to be 1667 pounds. The total weight of the shield, heat exchangers and heaters is 2640 pounds. The heat exchanger and shield arrangement used to arrive at weight estimate is shown in Figure 8-5.

TABLE 8-1. NON-INTEGRATED BRAYTON CYCLE ENERGY BALANCE

Item	Isotope Brayton	Solar Brayton
Electrical power to station bus, KW_e	8.0	8.0
Battery charging and power conversion, KW_e	0.85	0.85
Speed Control (5% of load), KW_e	0.44	0.44
Coolant and Liquid Loop radiator pump power, KW_e	0.190	0.032
Generator output, KW_e	9.51	9.322
Generator Efficiency, neglecting bearing losses, KW	0.87	0.87
Net shaft power to generator, KW_t	10.94	10.75
Gas bearing losses, turbo-generator, KW_t	0.4	0.4
Required Turbine Power, KW_t	11.34	11.15
(Btu/sec)	(10.72)	(10.575)
Specific Turbine Work, Btu/lb	17.4	17.4
Argon flow required, lb/sec	0.616	0.6075
Energy added, Btu/lb	61.0	61.0
Energy rejected, Btu/lb	42.5	42.5
Compressor bearing losses, KW	0.8	0.8
Heat added by source, Btu/sec	37.55	37.1
(KW_t)	(39.6)	(39.1)
Assumed thermal losses, KW_t	0.4	0.4
Heat supplied by source, KW_t	40.0	39.5
Heat rejected by radiator, Btu/sec	26.16	25.82
(KW_t)	(27.6)	(27.2)

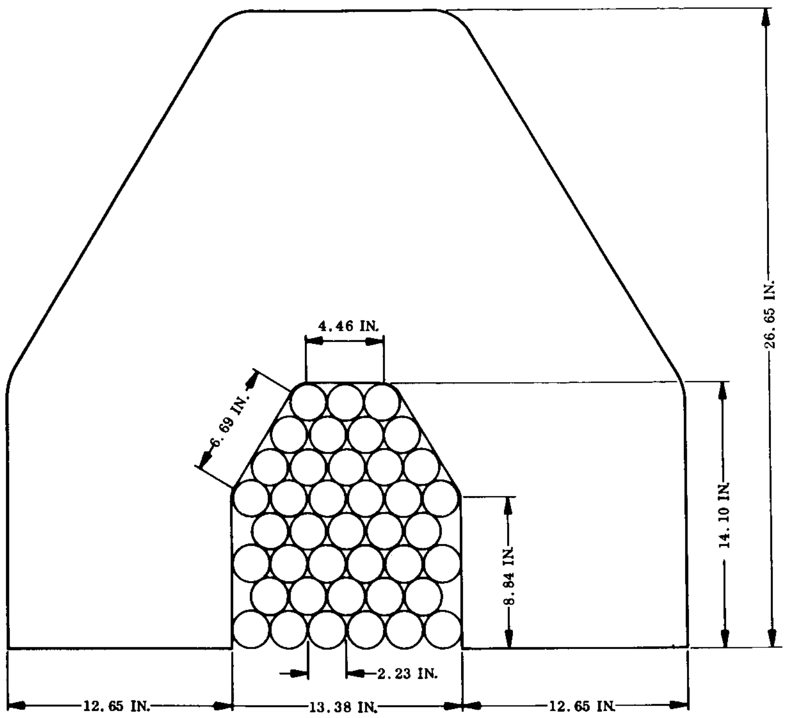


Figure 8-5. Forty-KW_t Isotope Heat Source

8.2.2.2 Solar Collector/Absorber

From the energy balance shown in Table 8-1, the heat source for the Solar Brayton non-integrated power system must furnish 39.5 KW_t net. In sizing the collector and absorber, the night time portion of the orbit must be considered and the equipment sized to collect and store the heat needed by the power system during that portion of the orbit.

Sizing the solar collector:

$$Q_A = (39.5\ KW_t)\ \frac{(96 \text{ minutes orbit})}{(58 \text{ minutes of light})} = 65.4\ KW_t \text{ heat input required}$$

during the daylight portion of the orbit. Power required from the collector is the heat required divided by collector efficiency (from Reference 8-1): $\frac{65.4\ KW}{0.79} = 82.8\ KW_t$.

Collector area needed $= \frac{82,800 \text{ watts}}{130 \text{ watts/ft}^2} = 637 \text{ ft}^2$. Assuming a collector weight of one lb/ft^2 as was done in Reference 8-1, the collector weight = 637 pounds. Collector diameter is calculated to be 28.5 feet. The absorber weight was scaled directly from Figure 6-11 of Reference 8-1 and is plotted in Figure 8-6. The absorber weight was determined to be 356 pounds.

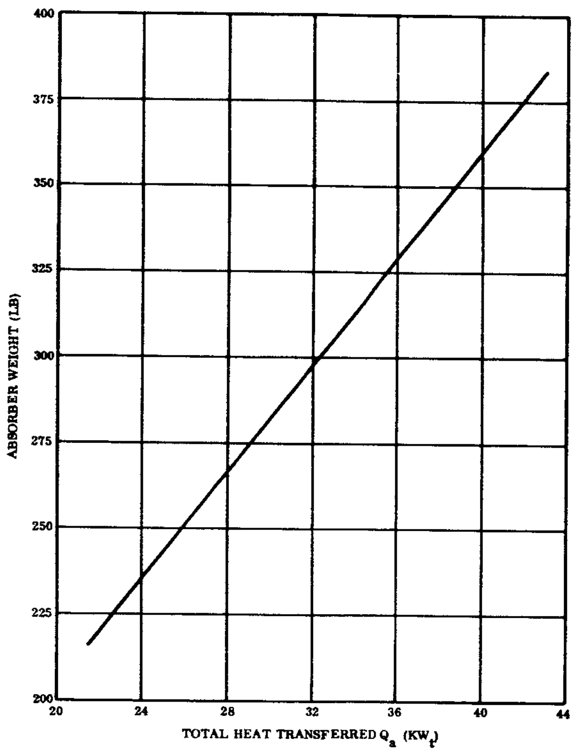


Figure 8-6. Absorber Weight

8.2.2.3 Turbo Machinery

Electrical power is provided by two identical sets of turbomachinery operating in parallel. The output of one set of turbomachinery was required by the study guidelines to be capable of producing at least the minimum power requirement of 1.4 KW. The gas generator or turbo compressor operates at 67,500 rpm and is identical in design to the unit described in detail in Reference 8-1. The 12,000-rpm homopolar four-pole turbo alternators are also identical in design to the units described in Reference 8-1. Speed control is accomplished by 5% electrical energy dissipation in a resistor bank. Speed control of the turbo compressor is accomplished by a bypass control valve to vary argon flow. The turbo machinery weight was determined by directly sealing from the equipment weight of Reference 8-1. The weight of the two turbo compressors was determined to be 77 pounds, and the weight of the two turbo alternators was determined to be 124 pounds.

8.2.2.4 Recuperator

The recuperator weight was scaled from the weight determined in Reference 8-1. A plot of recuperator weight versus power system size is shown in Figure 8-7. For this non-integrated system the recuperator weight was estimated to be 295 pounds.

8.2.2.5 Radiators

Radiator design was assumed to be the same as in the initial study. Linear extrapolations were made to find new sizings. The weight and area of the radiators are presented

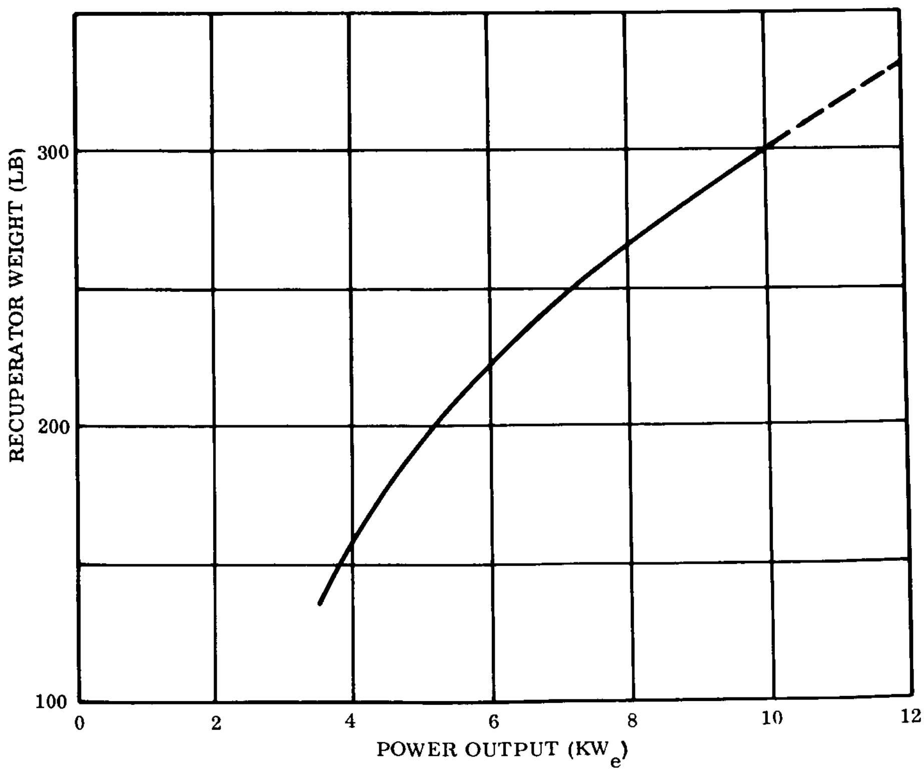


Figure 8-7. Recuperator Weight

in Figure 8-8. The non-integral radiator, used with the solar Brayton system, weighs 1670 pounds. Its general configuration is shown in Figure 8-2 and 8-17. The integral radiator, used with the isotope Brayton system, weighs 1400 pounds. It is constructed as a part of the MORL external structure, as shown in Figure 8-1.

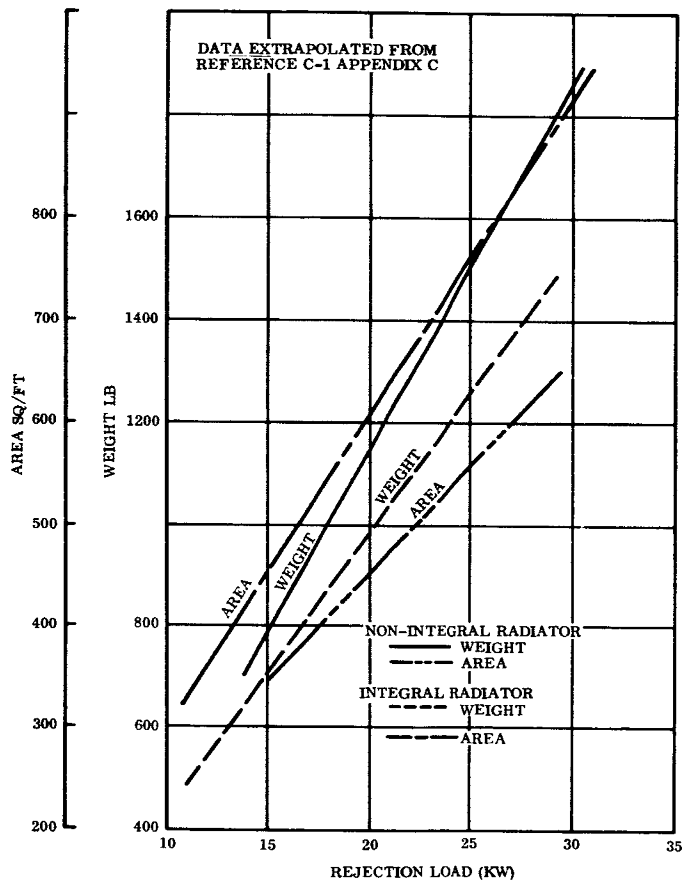


Figure 8-8. Brayton Cycle Radiators

8.2.3 NON-INTEGRATED BRAYTON SYSTEM SUMMARY

The non-integrated system component weights are listed in Table 8-2. The Isotope Brayton system weighs 6099 pounds, the Solar Brayton System weighs 4799 pounds. If, however, the liquid loop radiator system described in the next section (8.3) is used, a significant reduction in weight occurs. With the liquid loop, the Isotope Brayton System weighs 5388 pounds, and the Solar System 3797 pounds.

TABLE 8-2. NON-INTEGRATED BRAYTON SYSTEMS WEIGHT SUMMARY

Item		Isotope Brayton (lb)	Solar Brayton (lb)
Collector		-	637
Absorber		-	356
Isotope Heat Source		2640	-
Turbo-compressor (2)		77	77
Turbo-alternator (2)		124	124
Recuperator		295	295
Radiator		1400	1670
Vehicle skin structure saved by using an integral radiator		-127	-
Speed control		150	150
Support structure and piping		540	490
Batteries		1000	1000
	TOTAL	6099	4799
Remove gas radiators		-1400	-1670
Add cooling loop (heat exchanger, pumps, controls and liquid radiator) from 8.3		595	668
	TOTAL	5294	3797

8.3 LIQUID LOOP RADIATOR

8.3.1 BACKGROUND

The engine waste heat rejection system investigated in this study consists of a gas to liquid heat exchanger in combination with a liquid radiator. The goal of this part of the study is to obtain a minimum weight combination of the heat exchanger and radiator. It was anticipated that this arrangement would weigh less than a single gas radiator, the amount of heat to be rejected being equal. Figure 8-9 shows a schematic of the heat exchanger and radiator combination.

The liquid radiator will allow much smaller diameter tubing and lower film drops than a gas radiator. For fixed pump work, a reduction in tube inner diameter lowers the armor protection required, and lower film temperature drops increase effective radiating temperatures which in turn reduces radiator area. The weight savings realized with the liquid loop radiator will be of benefit up to the point where the added heat exchanger weight offsets these savings.

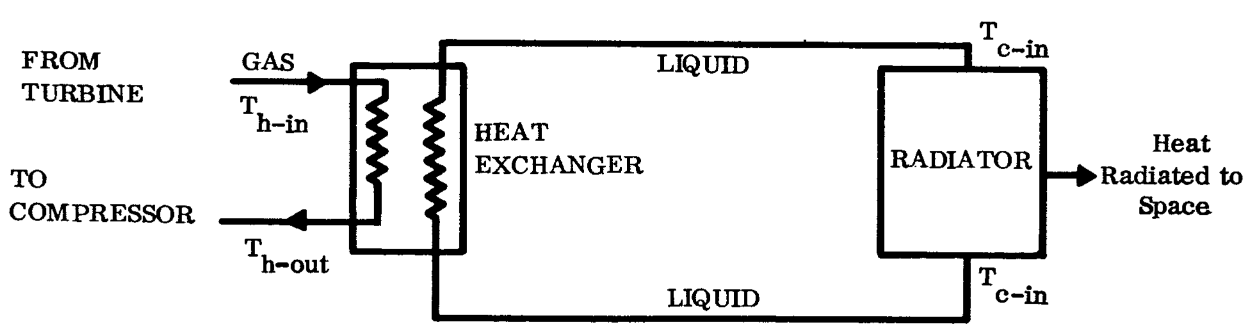


Figure 8-9. Non-Integrated Liquid Loop Schematic

8.3.2 COOLANT FLUID SELECTION

The choice of a heat transfer fluid for application in the power cycles under study can be made on the basis of several criteria:

a. Saturation pressure, freezing point

b. Toxicity

c. Materials compatibility

d. Low pump power requirement for a fixed coolant temperature rise

e. Low pump power requirement for a fixed film temperature drop

f. High flash point

g. High Reynolds number in order to operate in the turbulent region

Unfortunately, a survey of potential heat transfer fluids did not produce any fluid which had all of the desired characteristics. Most of the fluids considered were ruled out on the basis of their flash point being incompatible with the required operating temperature. The comparison of various potential heat transfer fluids on the basis of low pressure drop for a fixed coolant temperature rise and for a fixed film temperature rise is shown in Figures 8-10, 8-11, and 8-12.

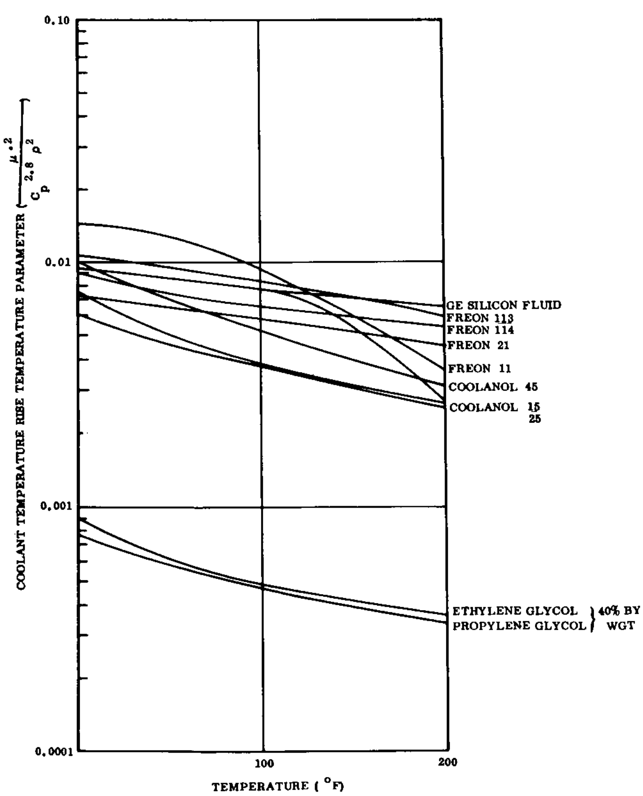


Figure 8-10. Coolant Temperature Rise Parameter Vs Temperature

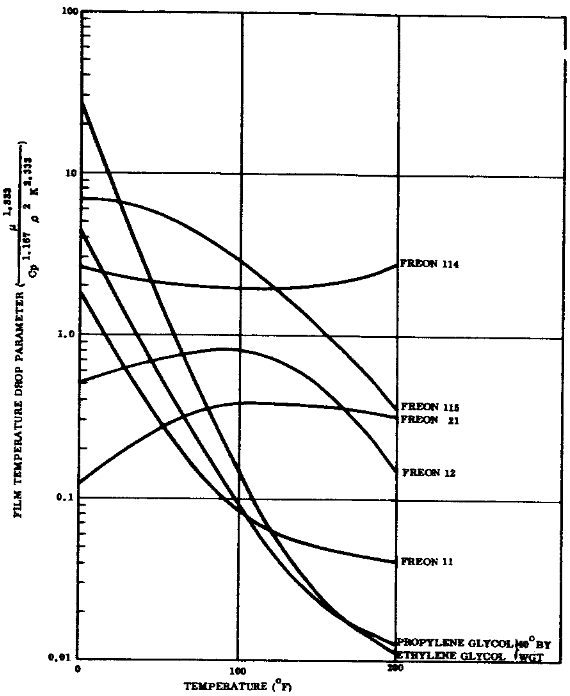


Figure 8-11. Film Temperature Drop Parameter Vs Temperature

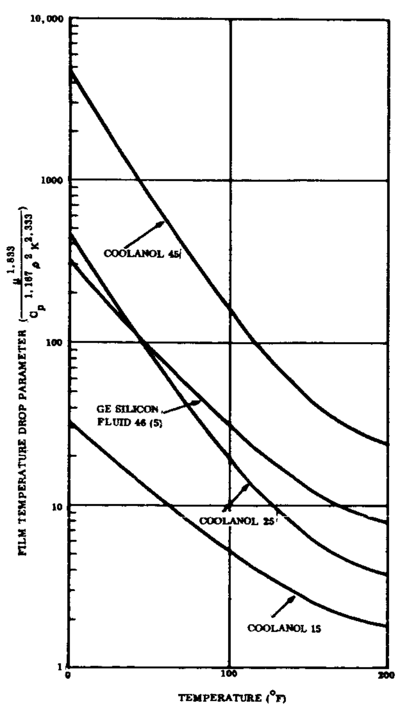


Figure 8-12. Film Temperature Drop Parameter Vs Temperature

A brief development of these criteria is given below:

The ideal pump work for an incompressible fluid is simply:

$$P = \frac{W \Delta p}{\rho} \tag{8-1}$$

where Q = heat transfer rate

W = Fluid weight flow rate = $\frac{Q}{C_p \Delta T_c}$

p = Pressure drop

ρ = fluid density

C_p = Fluid specific heat

T_c = Fluid temperature rise

The pressure drop can be described by the equation:

$$\Delta P = \frac{fL}{D} \frac{W^2}{g_c \rho A^2} \tag{8-2}$$

where f = Friction factor = $\frac{0.046}{R_e^{.2}}$ (smooth tube relation)

L = Flow length

D = Equivalent diameter of flow passage

A = Flow area

g_c = Gravitational constant

R_e = Reynolds number = $\frac{DW}{A\mu}$

μ = Viscosity

k = Thermal conductivity

Combining the above equations we obtain:

$$K_1 \left(\frac{P}{L}\right) \left[\frac{D^{1.2} A^{1.8}}{Q^{2.8}} \; gc \; \Delta \; Tc^{2.8} \right] = \frac{\mu^{.2}}{C_p^{2.8} \rho^2} \tag{8-3}$$

where K_1 = Proportionality constant

The right hand side of equation (8-3) is plotted in Figure 8-9.

For a fixed value of coolant temperature rise, geometry, and fixed beat input the larger the value of the right hand side the larger the pump-power required. For a non-metallic fluid in turbulant flow, the film coefficient can be written as:

$$N_u = CR_e^{.8} P_r^{1/3} \tag{8-4}$$

where:

N_u = Nusselt's number = $\frac{hD}{k}$

P_r = Prandtl's number = $\frac{C_p \mu}{k}$

h = convective film coefficient

C = proportionality constant

and

$$Q = h(\pi DL) \Delta T_f \tag{8-5}$$

where:

ΔT_f = film temperature rise.

If equations (8-4) and (8-5) are inserted in equation (8-3), the resulting equation is:

$$K_2 \left(\frac{P}{L}\right) \frac{gc\ D^{3.5}\ L\ A}{Q^{3.5}} \Delta T_f^{3.5} \frac{\mu^{1.833}}{C_p^{1.167} \rho^2 k^{2.333}} \tag{8-6}$$

where:

K_2 = proportionality constant

The right hand side of equation (8-6) is a measure of the pump work required for a fixed film temperature rise, geometry and heat input. This is plotted for various potential heat transfer coolants as a function of temperature in Figures 8-11 and 8-12. On the basis of minimizing pump work for a fixed coolant temperature rise, the glycols appear to be the very best choice. However, their low flash point precludes their use in this particular application. The next best choice is either Coolonol 15 or 25. On the basis of minimizing pump work for a fixed film drop the glycols are still the best choice with the Freons being the next best choice.

A major factor in minimizing both the radiator weight and the heat exchanger weight is in minimizing the film temperature drop. In the case of the heat exchanger the low film drop is reflected in smaller heat transfer area by virtue of a higher film coefficient. In the case of the radiator, the low film drop provides a higher effective radiator temperature. From the point of view of keeping the lowest possible vapor pressure, and operating in the regime below the fluid critical temperature, both Freon 11 and 113 are the best choices and, hence, were selected as the secondary heat transfer fluids in this study. The selection of a particular fluid for a given design will depend on its specific operation parameters.

8.3.3 HEAT EXCHANGER DESIGN

The heat exchanger selected is a plate-fin counter flow type design because of its attractive performance characteristics. Figure 8-13 illustrates the counter flow arrangement.

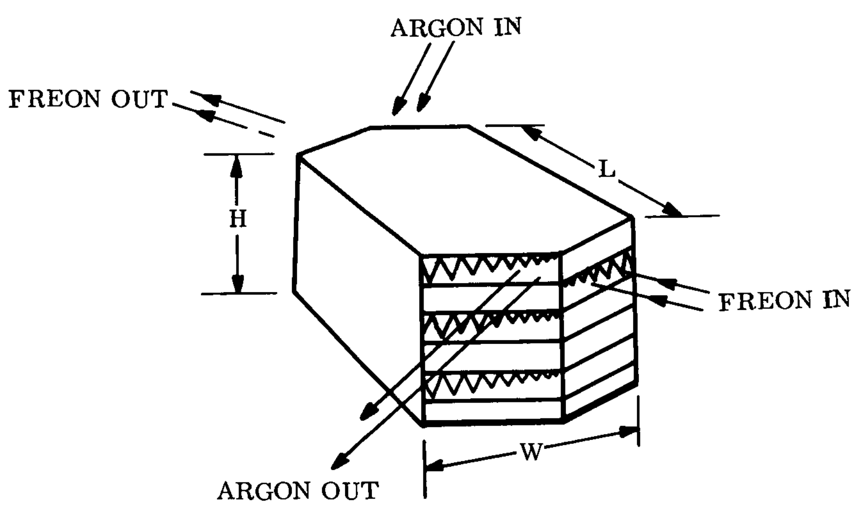


Figure 8-13. Counter Flow Heat Exchanger

A typical cross-sectional view of the plate-fin configuration is shown in Figure 8-14.

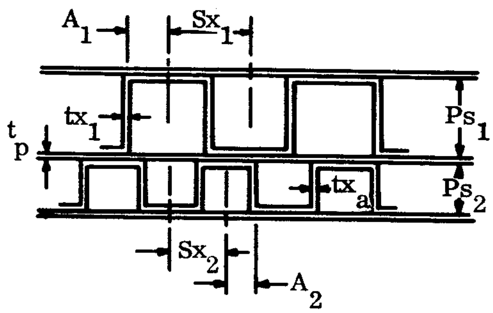


Figure 8-14. Typical Plate-Fin Configuration

The following characteristics of the plate fin arrangement were fixed:

$Sx_1/Ps_1 = Sx_2/Ps_2 = 0.5$

$A_1/Ps_1 = A_2/Ps_2 = 0.25$

$Ps_2 = 0.1$ inch

$t_p = 0.006$ inch

$tx_1 = tx_2 = 0.01$ inch

$W = 2$ ft

Pressure drop in gas side, $\Delta P = 0.32$ psi (consistent with work done in Contract NAS 3-2799).

Liquid side pump penalty is 417 lb/ KW_e

One gas to one liquid passage, alternating.

Core material is nickel.

With these parameters selected, the weight optimization of the heat exchanger was obtained by means of a computer program. Properties of gas, fluid, and core material as a function of temperature were used. The length (L_1) was varied to meet the heat exchanger effectiveness selected for a given set of gas inlet and outlet temperatures. Then theg as side plate spacing, PS_2 was adjusted in order to get the prescribed gas side pressure drop; L was then re-adjusted to meet the effectiveness. A core weight, and total heat exchanger weight including the liquid side pump penalty weight were then calculated. New weights were found by varying the gas side plate spacing, adjusting H and L to meet the effectiveness and pressure drop. This procedure was repeated with each new weight being compared with the previous one found in order to arrive at a minimum heat exchanger weight. Figures 8-15 and 8-16 show the characteristic curves of Colburn Modulus (j) and friction factor (f) for plate fin cores versus Reynolds number.

8.34 RADIATOR DESIGN

The design criteria for the radiators as well as the method of obtaining their minimum weights are identical to the methods described in Reference C-1. Appendix C. The radiator tube configurations analyzed are illustrated in Figure 8-17.

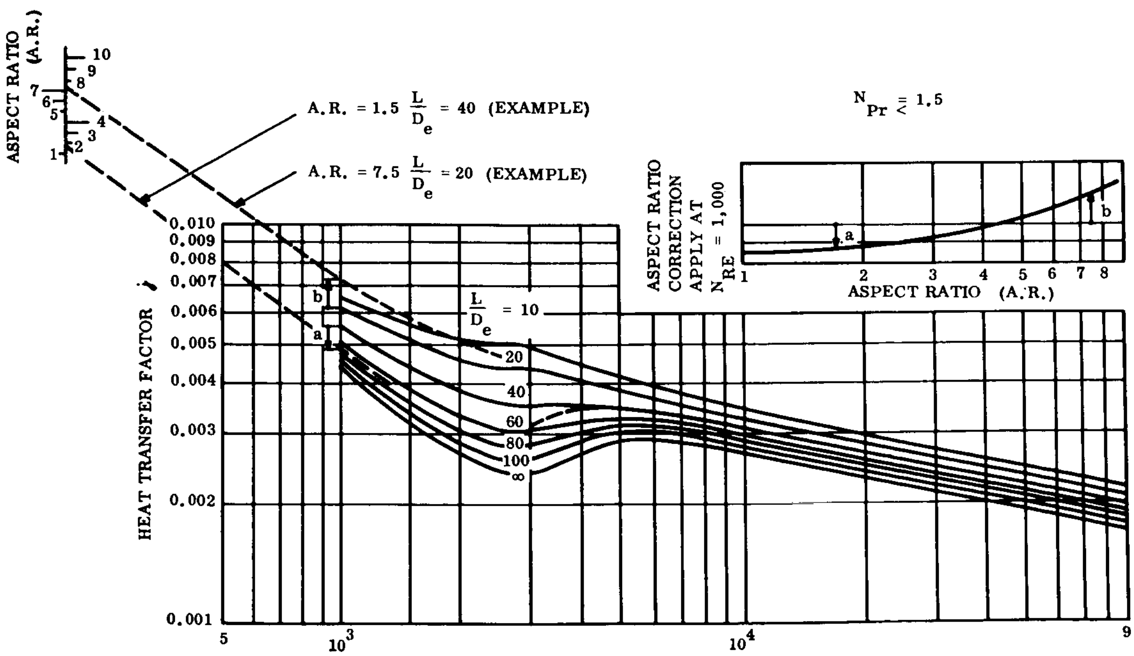


Figure 8-15. Gas Flow in Plain Plate Fin Surfaces, Heat Transfer Factor Vs Reynolds Number

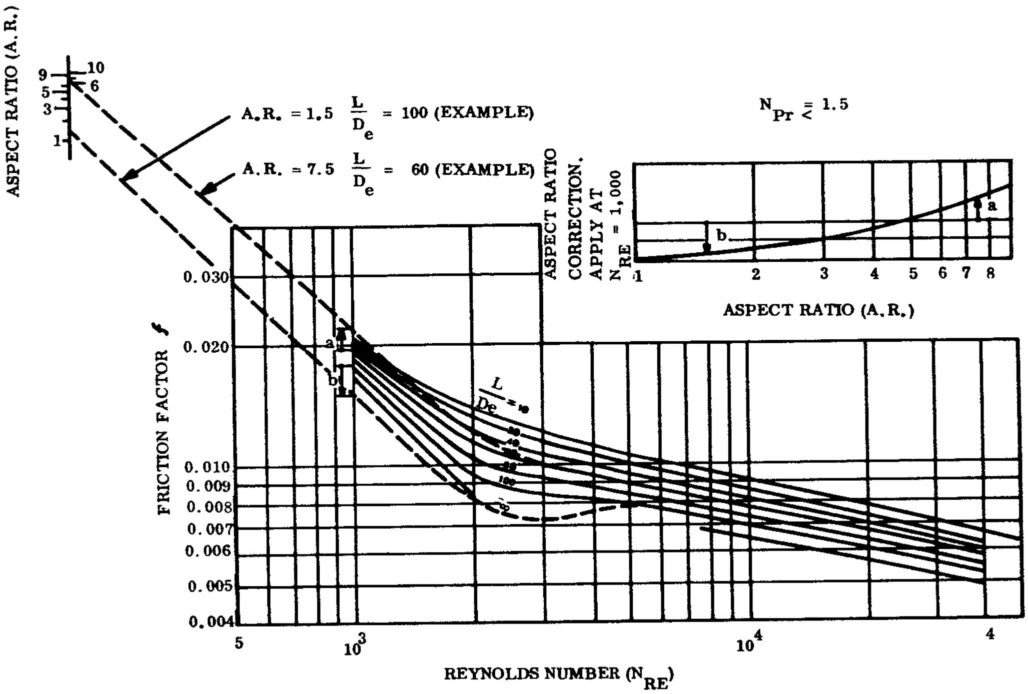


Figure 8-16. Gas Flow in Plain Plate Fin Surfaces, Friction Factor Vs Reynolds Number

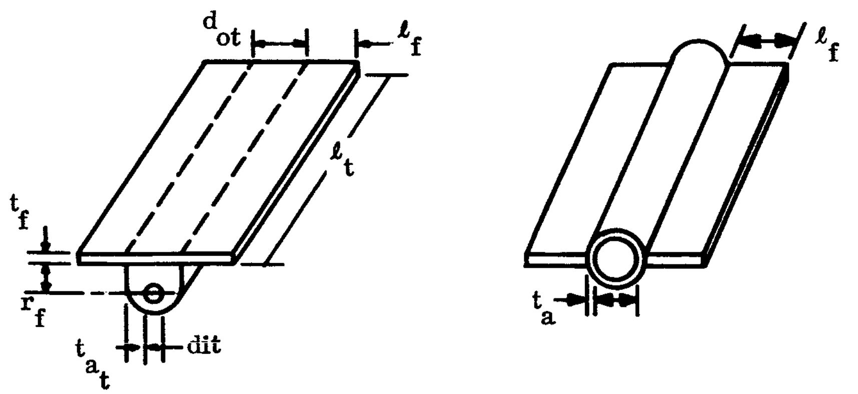


Figure 8-17. Tube Configurations

The envelope of the radiators, consisting of tubes and headers, were made to conform to a cylindrical shape having maximum dimensions of 250 inches in diameter and 120 inches in length.

8.3.5 RADIATOR-HEAT EXCHANGER OPTIMIZATION

8.3.5.1 Non-Integrated Systems Optimization

The radiator and heat exchanger were combined to optimize the weight. The two variables used were heat exchanger effectiveness (ϵ) and the ratio of hot gas to cold fluid capacity rate (C_p/C_c, where $C_h = W_h \times C_{ph}$, or gas flow times specific heat and $C_c = W_c \times C_{pc}$, or liquid flow times specific heat). These variables determine temperature points, defining radiator and heat exchanger temperatures, the gas inlet/outlet temperatures being fixed at 415°F and 76°F. A restriction of the effectiveness was to consider 0.9 as a maximum practical number.

All non-integral structure radiators were conceived as having four panels connected in parallel. Headers were taken to run circumferentially along the periphery of the cylindrical configuration; the tubes were chosen to run parallel to the cylinder axis. Figure 8-18 illustrates this four panel arrangement. All integral structure radiators consisted of two panels connected in parallel. Headers run parallel to the cylindrical axis, and tubes are located on the cylinder periphery, running normal to the axis.

For a given gas outlet temperature and a C_h/C_c, the lightest heat exchangers were associated with the lower effectiveness values, as could be expected. On the other hand, the radiator weight increased with lower values of C_h/C_c and effectiveness (ϵ). This trend in radiator weight is caused by the reduction in liquid inlet and/or outlet temperatures with lower values of both C_h/C_c and ϵ. The lower these temperatures are, the lower the radiator average temperature. This in turn forces an increase in the area required to reject a given heat load to space. The radiator weight penalty resulting from lowering

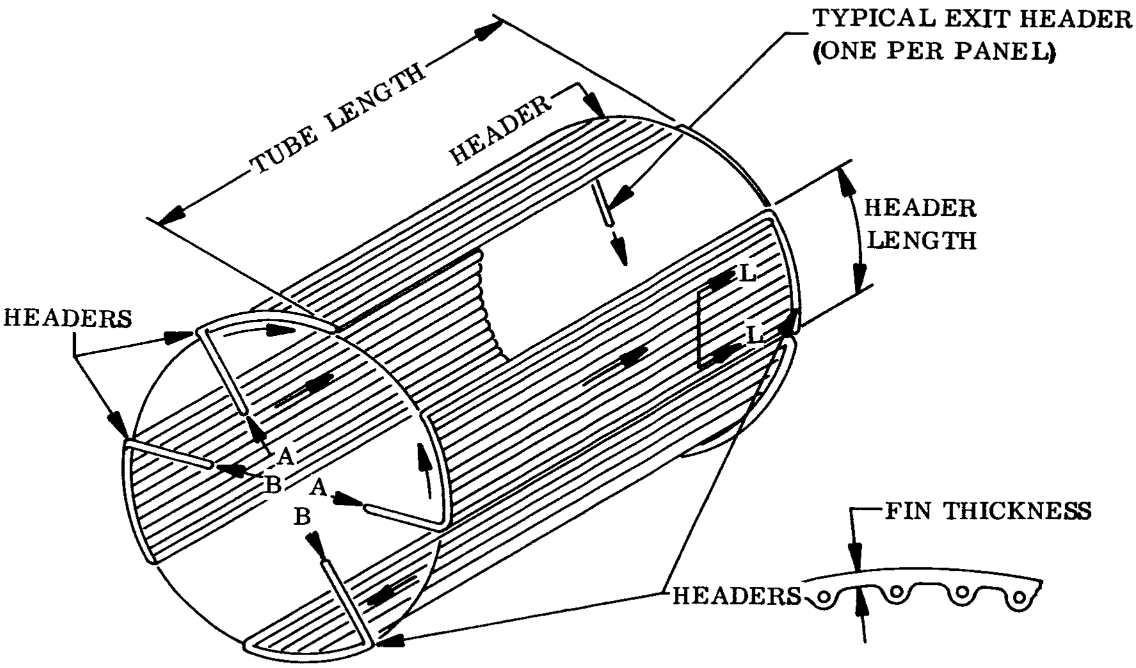


Figure 8-18. Four-Panel Radiator Configuration for Non-Integral Design

the heat exchanger effectiveness more than offsets potential weight savings in the heat exchanger. As a result of the overriding weight influence of the radiator the lightest heat exchanger and radiator combinations were found to exhibit large values of both C_h/C_c and effectiveness. Considering the system parametrically, heat exchanger weights were found to be a linear function of energy rejected over a wide heat output. Figure 8-19 illustrates this relationship for a given heat exchanger condition. Radiator weights can only be truly considered to be a linear function of heat rejected over a narrow range of load, approximately $\pm$ 15% of a given load, due to the micrometeroid survival criteria.

The chosen radiator heat exchanger combination is detailed in Table 8-3. The resulting combined weights of 595 pounds for the Isotope Brayton heat rejection system and 668 pounds for the Solar Brayton heat rejection system are far lighter than their gas radiator counterparts (see Table 8-2).

TABLE 8-3. NON-INTEGRATED BRAYTON POWER SYSTEMS HEAT REJECTION SYSTEM CHARACTERISTICS

Item	Isotope Brayton Power System Integral Radiator	Solar Brayton Power System Non-Integral Radiator
Argon-to-Liquid Heat Exchanger		
Q_r, KW_e	27.6	27.2
T_{in}, °F	415	415
T_{out}, °F	76	76
ΔT °F	377	377
Weight, lb	176	176
Length, ft	3.8	3.8
Width, ft	2.0	2.0
Height, ft	0.425	0.425
Radiator		
T_{in}, °F	377	377
T_{out}, °F	38	38
ΔT, °F	339	339
ΔP, psi	17.9	1.3
Panels	2	4
Radiator weight, lb	413	491.1
Pumping penalty weight, lb	6.7	0.45
Area, ft^2	719	718
Header length, ft	10	16
Number tubes	11 (circumferential)	14
Total system weight, lb	595	668

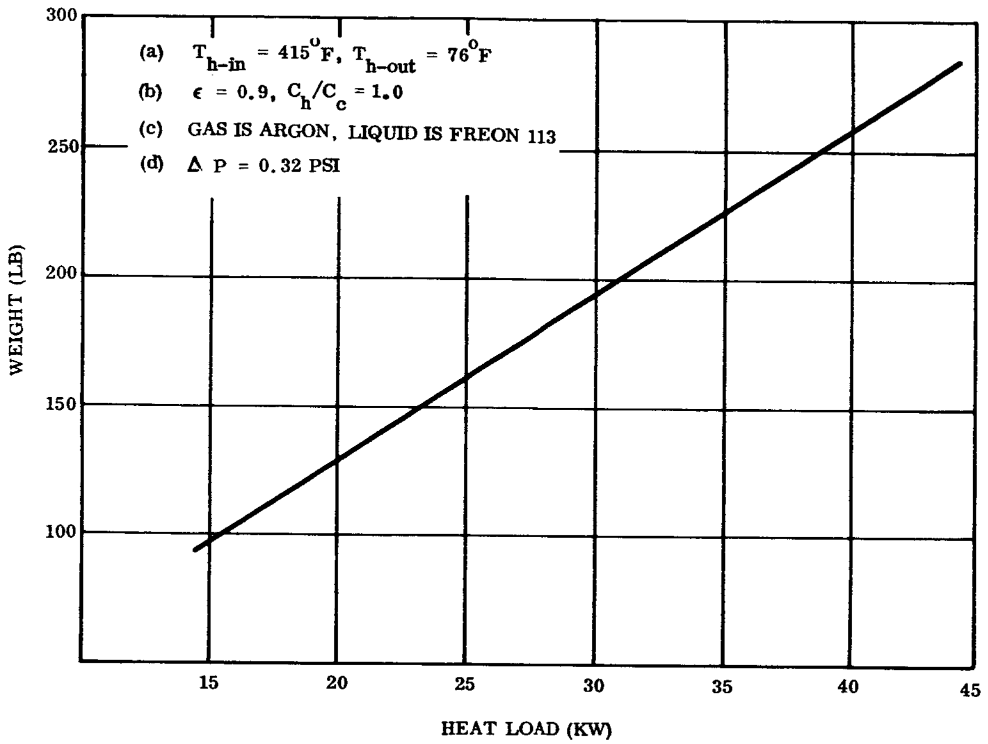


Figure 8-19. Heat Exchanger Weight

8.3.5.2 Integrated System Optimization

The liquid loop for the integrated systems differs from that of the non-integrated systems in that a life support loop consumes some of the heat contained in the transport fluid, reducing the radiator heat load. The tube and header configurations used corresponds to those selected for the non-integrated systems.

Since the desired inlet temperature of the liquid at the life support heat exchanger is 400°F, Freon 113 was substituted for Freon 11 in order to increase the useful temperature range of the liquid. Physical properties of both Freons are similar enough to readily allow a heat exchanger weight to be found from previous calculations due to the linear relationship between heat rejected and weight.

The radiator outlet temperature was selected at 38°F, which from earlier analysis imposed minimum weight penalties for the radiator. Knowing the load dissipated at the life support heat exchanger (4.01 kw) the radiator liquid inlet temperature was found to be 325°F.

The characteristics of the near optimum weight radiators for the integral and non-integral structures were obtained by means of a computer program. Parameters used were fin thickness, header length, number of tubes and tube inside diameter. The results are presented in Table 8-8. A discussion of these results is presented in Section 8.5.

8.4 CYCLE REJECTION TEMPERATURE VARIATION EFFECTS

An analysis was made of the Brayton gas cycle parameters to determine the effect on cycle performance of an increase in exhaust gas temperature. This change would have the effect of increasing the performance of the waste heat radiator, but at the expense of decreased cycle efficiency.

Since maximum cycle temperature is limited to 1950°R, the cycle performance is reduced when the compressor inlet temperature is raised. Shown in Figure 8-20 is the variation in cycle thermal efficiency with compressor inlet temperature and compressor pressure ratio. The thermal efficiency is defined as the shaft power output divided by the heat input to the system, or $\eta = W/Q$. From this plot the heat addition requirements can be determined for any desired shaft output.

Shown in Figure 8-21 is the gross shaft power output per unit flow rate of argon gas. This plot is used to determine the flow rate required for any desired shaft power output. Shown in Figure 8-22 is the heat rejection required as a function of compressor inlet temperature and compressor ratio. This plot is directly related to cycle efficiency since $Q_{R/W} = (1/\eta) - 1$, and indicates the heat rejection capacity required for various cycle conditions. Figures 8-20, 8-21 and 8-22 are based on the following cycle assumptions:

Table 8-4 lists the significant parameters for the three temperature cases considered. The combined weight of the isotope heat source and heat rejection system as a function of compressor inlet temperature is plotted in Figure 8-23. The radiator and heat exchanger weights are shown in Table 8-5.

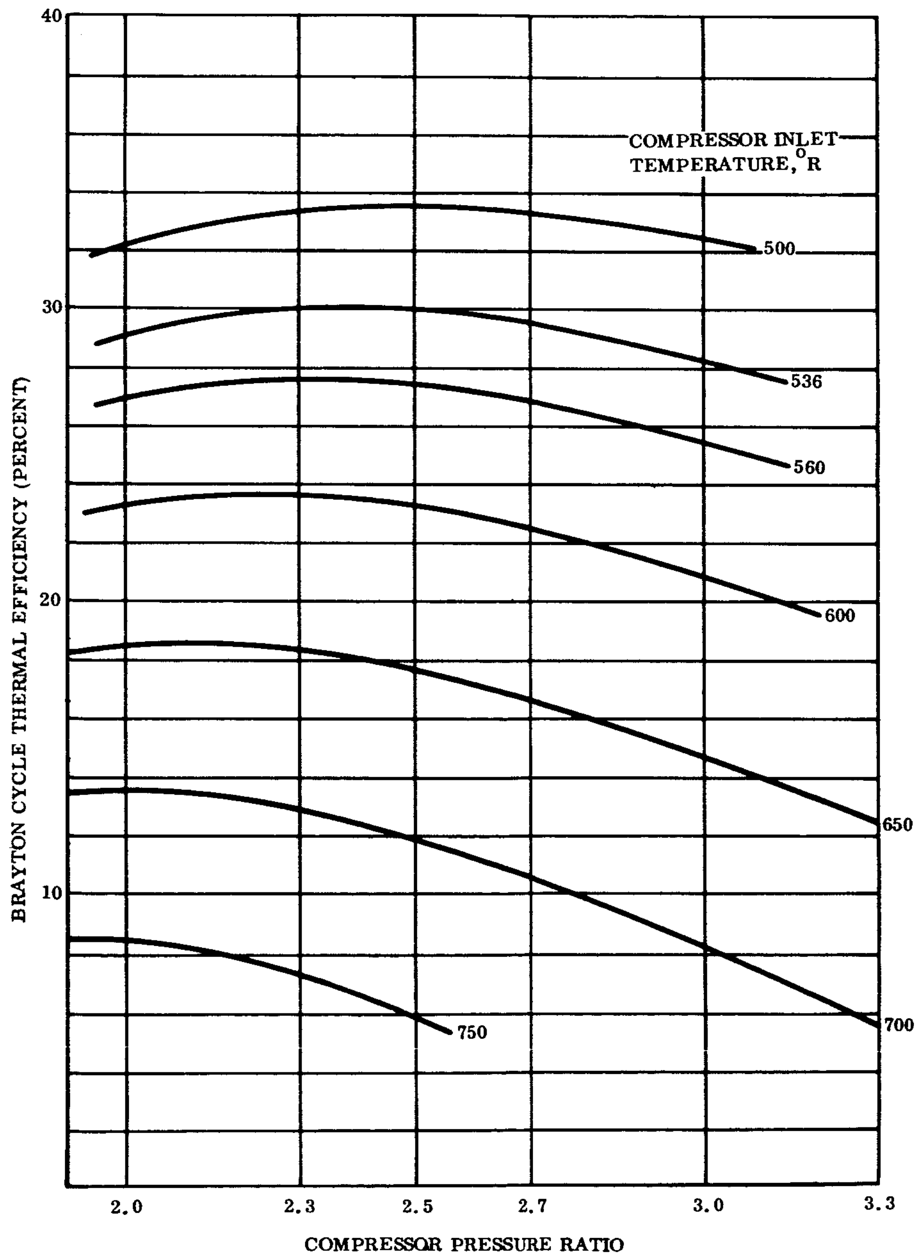


Figure 8-20. Variation in Cycle Thermal Efficiency with Compressor Inlet Temperature and Pressure Ratio

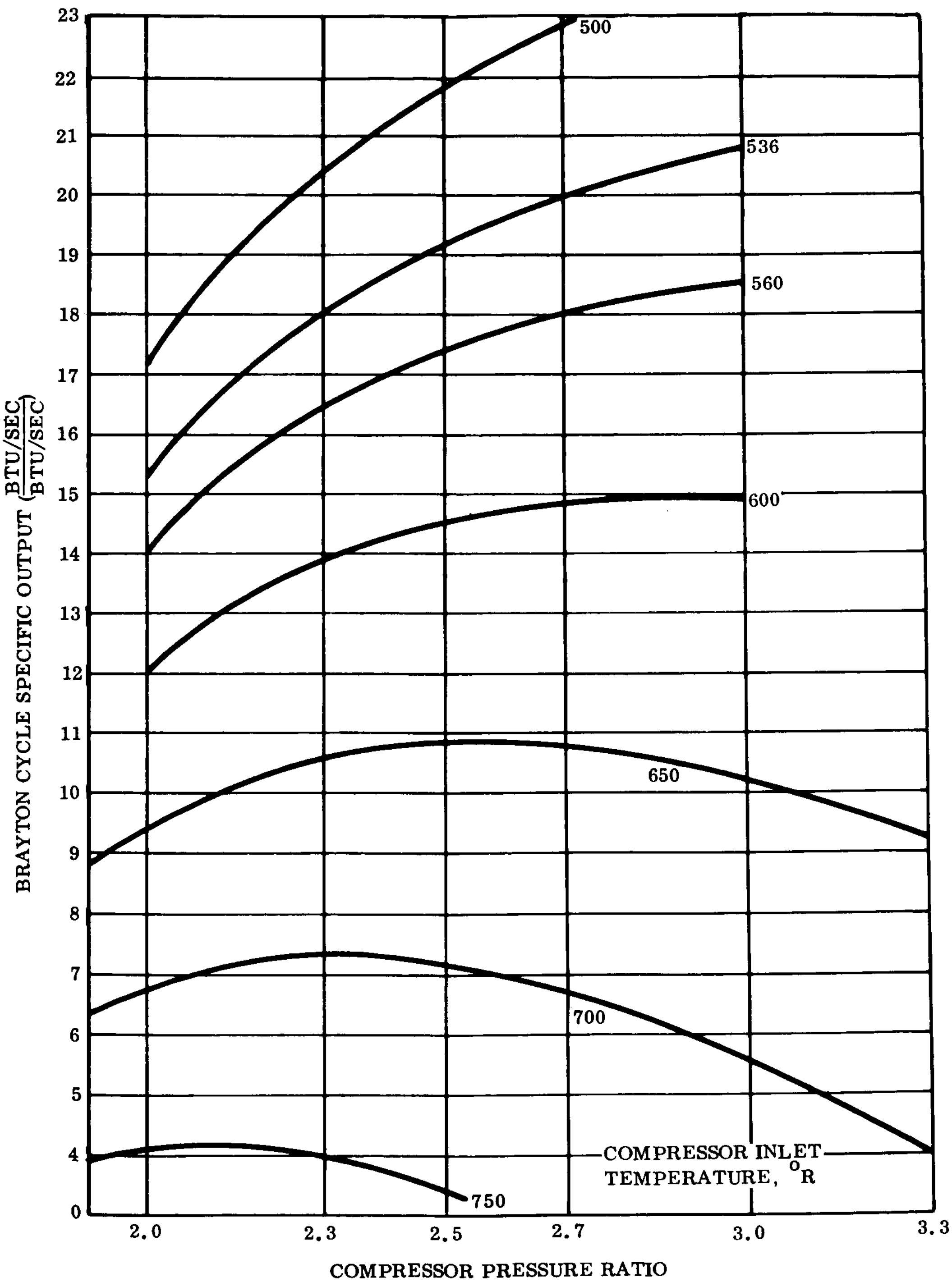


Figure 8-21. Gross Shaft Power Output per Unit Flow Rate of Argon Gas

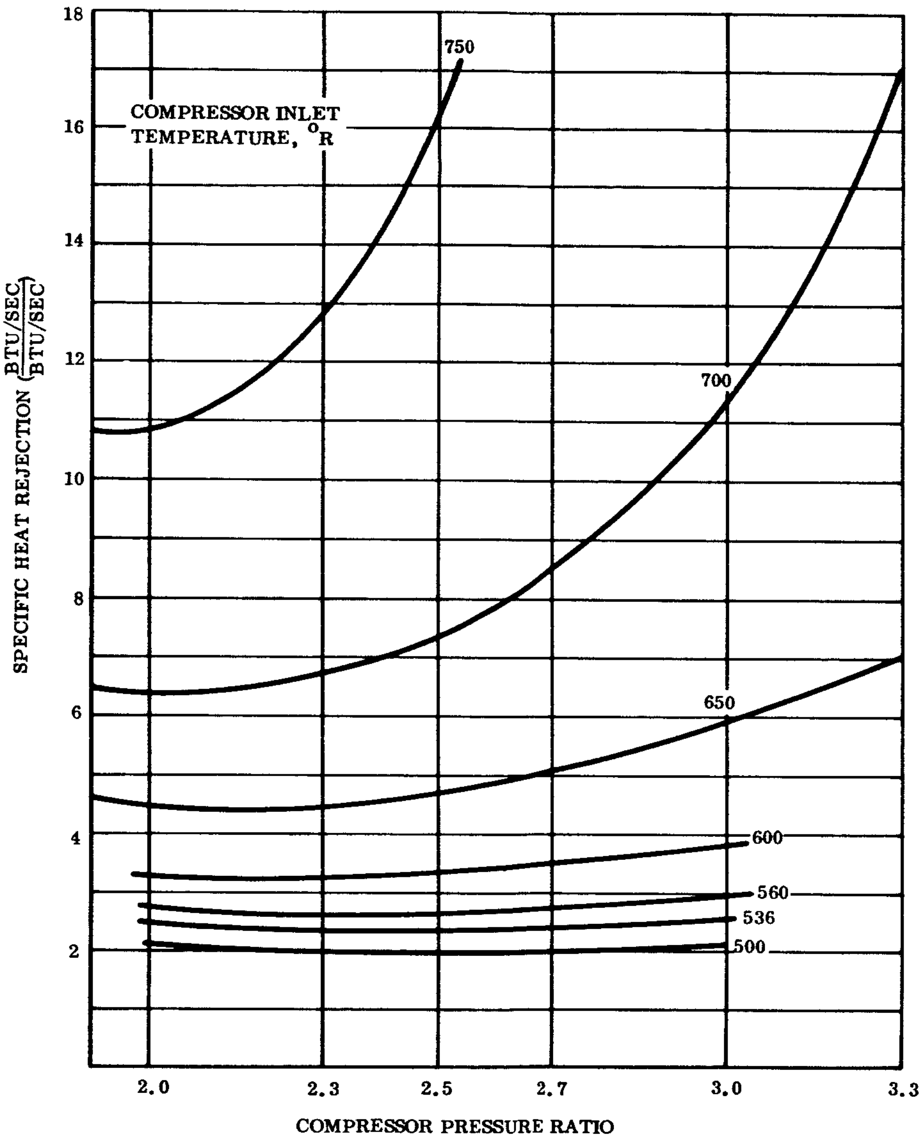


Figure 8-22. Heat Rejection Required as a Function of Compressor Inlet Temperature and Ratio

Compressor Efficiency	0.80
Compressor Turbine Efficiency	0.83
Power Turbine Efficiency	0.86
Loss Pressure Ratio	0.884
Recuperator Effectiveness	0.90

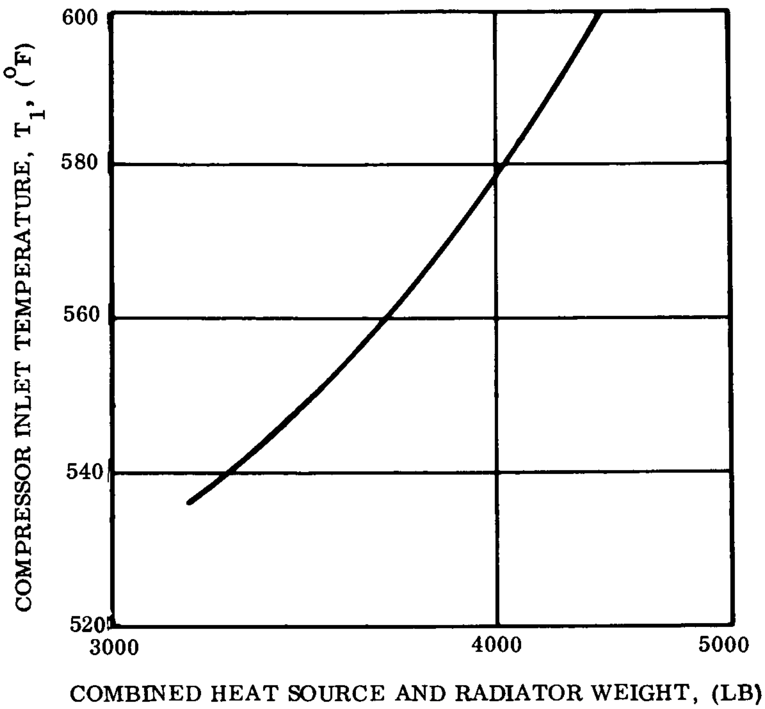


Figure 8-23. Combined Weight of The Isotope Heat Source and Heat Rejection System as a Function of Compressor Inlet Temperature

TABLE 8-4. EFFECT OF INCREASE IN HEAT REJECTION TEMPERATURE ON BRAYTON CYCLE PERFORMANCE

	T_1 = 536	T_1 = 560	T_1 = 600
Compressor Ratio P_2/P_1	2.3	2.07	1.72
Cycle Efficiency	0.3	0.273	0.22
Ideal Specific Work w/w Btu/lb	17.4	14.45	9.5
Weight Flow, lb/sec	0.605	0.727	1.106
Isotope Source Weight, lb	2611	2825	3485
Solar Collector Weight, lb	627	680	837
Heat Rejected, Q_r, KW_t	27.1	30.2	39.9
Integral Radiator Weight, lb	588	537	761
Non Integral Radiator Weight, lb	668	761	881
Heat Source Size, Q_A, KW_t	38.9	42.1	51.9

TABLE 8-5. BRAYTON CYCLE HEAT EXCHANGER AND RADIATOR CHARACTERISTICS FOR NON-INTEGRATED SYSTEMS

INTEGRAL STRUCTURE

HEAT EXCHANGER										RADIATOR									
Q_r	T_{h-in}	T_{h-out}	L	W	H	E	C_h/C_c	Core Wt	H/X Wt	T_{c-in}	T_{c-out}	ΔP	Wt	A	H	Lt	NT	d_{it}	Total Wt
KW	°F	°F	ft	ft	ft			lb	lb	°F	°F	psi	lb	ft^2	ft	ft		in	lb
27.1	415	76	3.8	2.0	0.426	0.9	1.0	124	176	377	38	17.9	412	719	10	35.9	11	0.103	588
30.2	415	110	3.6	2.0	0.493	0.9	1.0	137	218	381	76	18.6	374	615	10	30.6	12	0.103	592
39.9	415	140	3.5	2.0	0.436	0.8	1.0	104	162	346	71	43.1	599	872	10	43.5	12	0.103	761

NON-INTEGRAL STRUCTURE

HEAT EXCHANGER										RADIATOR									
27.1	415	76	3.8	2.0	0.426	0.9	1.0	124	176	377	38	1.3	492	718	16	11.1	14	0.102	668
30.2	415	110	3.6	2.0	0.493	0.9	1.0	137	218	381	76	2.05	471	702	16	10.8	14	0.102	689
39.9	415	140	3.5	2.0	0.436	0.8	1.0	104	162	346	71	1.5	719	714	16	11.0	22	0.102	881

The best heat rejection temperature of the cases considered to minimize weight for a given power output is 536°R (76 F). It is possible that a further reduction in temperature might achieve a lighter weight, however, the 400°R sink temperature becomes a larger factor as the radiator fluid outlet is decreased. If sink temperatures significantly lower than 400°R can be achieved, then a re-optimization would be in order.

8.5 INTEGRATED BRAYTON SYSTEMS

8.5.1 INTRODUCTION

For the Brayton power cycle with a liquid loop radiator, the most convenient place in the cycle to obtain waste heat without affecting the cycle is in the radiator loop.

In the previous thermal integration study the Brayton power systems were designed using an argon gas radiator. For the integrated design it was necessary to insert an argon-to-therminol heat exchanger in the gas stream between the recuperator and the radiator. The effect on the system was to lower the temperature of argon at the radiator inlet and to reduce the heat load on the radiator by the amount of heat given up to the therminol. In the therminol loop was a therminol-to-glycol/water heat exchanger which was sized to provide the low temperature heat to the Life Support processes.

8.5.2 DESCRIPTION

Thermal integration of this power system has a similar effect on the power system in that the amount of heat to be rejected by the radiator is reduced by the amount of heat given up to the different thermal energy users. The temperature of the radiator fluid entering the radiator is also reduced so that the radiator radiates a reduced amount of heat, but at a lower average temperature.

The electrical power delivered to the station bus is reduced by the amount of energy furnished as heat to the Life Support System and is increased by the electrical power required by that system in excess of the non-integrated system requirements.

The turbo-machinery, controls, and all other components of the Brayton Cycle Power systems in the integrated design, provide the same function as in the non-integrated design. Two sets of identical turbo-machinery are used, each sized to furnish one-half of the electrical load. Start up, emergency and shut down procedures are the same as described in Reference 8-1.

The alternator output of the integrated systems is near enough the alternator output of 6.57 KW_e for the non-integrated Brayton power systems of Reference 8-1 to allow the use of the same cycle efficiencies in determining the design parameters for the 6.092 KW_e integrated power system sized for this study. The energy balance for both the Isotope and Solar Brayton integrated power systems is shown in Table 8-6. The source size requirement for the Solar Brayton system is slightly smaller than the Isotope Brayton source size requirement because a smaller amount of pumping power is required.

Table 8-7 is a weight summary of the two Brayton systems. The flow schematic for the liquid loop radiator system was shown on the power system schematic, Figure 8-3, and is shown in larger detail in Figure 8-24. Table 8-8 gives the characteristics of the heat exchanger/radiator for the heat rejection loop.

8.5.3 COMPONENT DESIGN

a. <u>Isotope Heat Source Design</u> - The Isotope Heat Source package design and layout is based on a 28-tube (fuel capsule) requirement to provide 28 KW_t. The arrangement of the fuel capsules used to determine shield weight is shown in Figure 8-25. The Isotope/Argon heat exchanger weight was determined by taking a ratio of the number of tubes in this study to the previous study, 28/30 x 730 lb = 680 lb. In Reference 8-1 the lithium hydride shield thickness was determined to be 11.6 inches. The shield volume was calculated to be 49,530 cubic inches, weighing 1,367 pounds. The total weight of the heat source package is 2047 pounds.

TABLE 8-6. BRAYTON POWER SYSTEMS INTEGRATED ENERGY BALANCE

Item		Isotope Brayton	Solar Brayton
Non-integrated electric power to station bus, KW_e		8.0	8.0
Non-integrated electric power to:			
Life Support System KW_e	4.566		
Cabin Air Cooling, KW_e	0.525		
Electronic Equipment Cooling, KW_e	0.023	5.115	5.115
Non-integrated electric power to other loads, KW_e		2.885	2.885
Integrated Electric Power to:			
Life Support System, KW_e	1.601		
Cabin Air Cooling, KW_e	0.415		
Absorption Refrigeration	.069		
Electronic Equipment Cooling, KW_e	0.017	2.102	2.102
*Station Electrical Load, KW_e		4.987	4.987
Battery charging and AC-DC conversion losses, KW_e		0.718	0.718
Speed Control (5% of load), KW_e		0.287	0.287
Bearing Coolant and Radiator Loop Pump Power, KW_e		0.100	0.025
Generator output, KW_e		6.092	6.017
Generator efficiency		0.86	0.86
Net Shaft power to generator, KW_t		7.08	7.08
Gas bearing losses, turbo-generator, KW_e		0.30	0.30
Required turbine power, KW_t	(Btu/sec)	7.38 (6.98)	7.38 (6.91)
Specific turbine work, Btu/lb		15.97	15.97
Argon flow required, lb/sec		0.437	0.433
Energy added, Btu/lb		59.76	59.76
Energy rejected, Btu/lb		42.2	42.2
Compressor bearing losses, KW		0.8	0.8
Heat added by source, Btu/sec	(KW_t)	(27.7)	(27.7)
Assumed thermal losses, KW_t		0.3	0.3
Heat supplied by source, KW_t		28.0	27.6
Heat rejected to radiator loop, KW_t		19.5	19.3

*Actual station load is decreased by 50 watts from this level with Absorption Refrigeration not being used.

TABLE 8-7. INTEGRATED BRAYTON SYSTEMS WEIGHT SUMMARY

Item	Isotope Brayton	Solar Brayton
Collector	-	446
Absorber	-	265
Isotope Heat Source	2047	-
Turbo-compressor (2)	68	68
Turbo-alternator (2)	110	110
Recuperator	225	225
Cooling Loop (Heat exchanger, pump, radiator)	371	413
Life Support System Heat Exchanger (Freon 113 to Therminol)	3	3
Skin structure saved by general radiator	-32	
Speed Control	114	114
Support structure and piping	404	404
Batteries	1000	1000
TOTAL	4310	3048

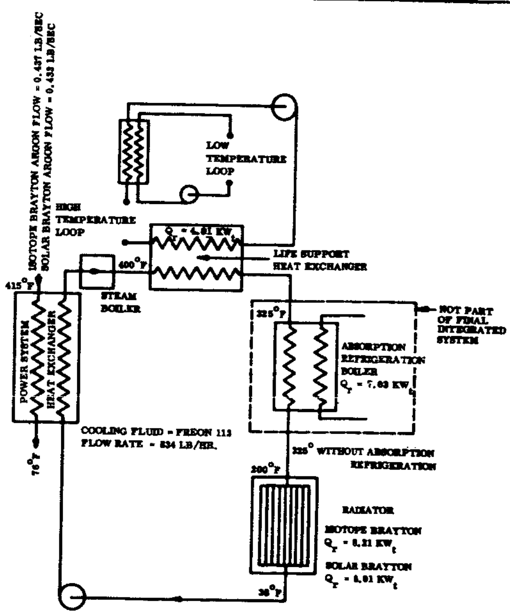
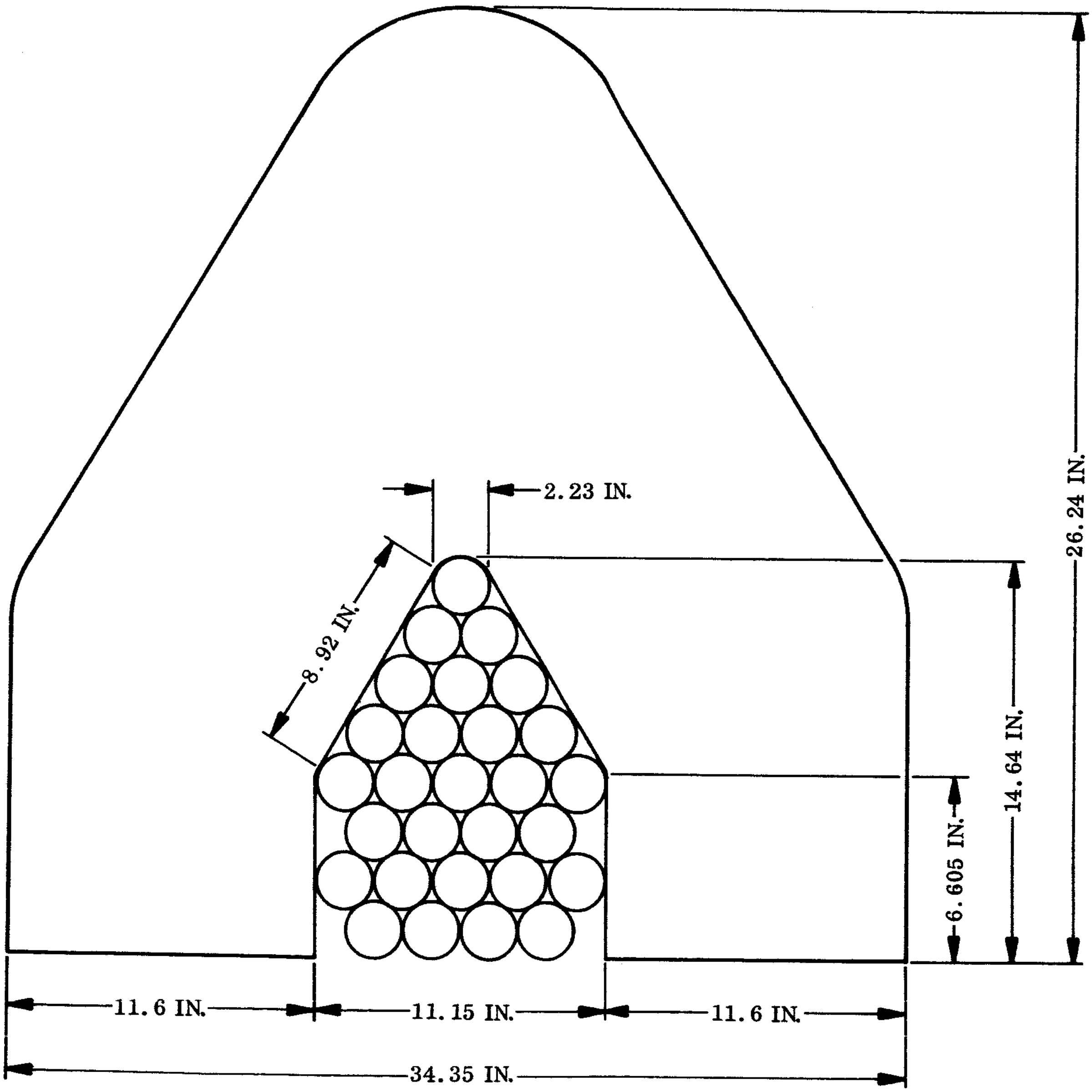


Figure 8-24. Integrated Brayton Power Systems, Heat Rejection System

TABLE 8-8. INTEGRATED BRAYTON POWER SYSTEMS
HEAT REJECTION SYSTEM CHARACTERISTICS

Item	Isotope Brayton Power System Integral Radiator	Solar Brayton Power System Non-Integral Radiator
Argon-to-Freon 113 Heat Exchanger		
Q_r, KW_t	19.5	19.3
T_{in} (argon), °F	415	415
T_{out} (argon), °F	76	76
T, °F	339	339
Weight, lb	125	125
Life Support System Heat Exchanger		
Q_r, KW_t	4.01	4.01
T_{in} (Freon 113), °F	400	400
T_{out} (Freon 113), °F	325	325
Weight, lb	2.8	2.8
Absorption Refrigeration Boiler		
Q_r, KW_t	7.3	7.3
T_{in} (Freon 113) °F	325	325
T_{out} (Freon 113) °F	200	200

Radiator	Without Absorp. Refr.	With Absorp. Refr.	Without Absorp. Refr.	With Absorp. Refr.
Q_r, KW_t	15.5	8.2	15.3	8.0
T_{in} (Freon 113)°F	325	200	325	200
T_{out} (Freon 113)°F	38	38	38	38
Δ P, psia		18.9		0.72
Weight, lb	246	149	288	161
Header length, ft		5		10
Tube length, ft		31.5		7.2
Number panels		2		4
Number tubes/panel,		6		9
Total System Weight (Heat Exchanger & Radiator)	371 lb	274	413	286



SHIELD LENGTH = 68.6 IN.
SHIELD VOLUME = 49,530 $IN.^3$
SHIELD WEIGHT = 1,367 LB

Figure 8-25. Integrated Isotope Brayton System Heat Source

b. Solar Collector/Absorber Design - From the energy balance shown in Table 8-8, the energy to be furnished by the absorber can be determined and the solar collector sized. The absorber must furnish 27.6 KW_t on a continuous basis. The heat input required during the daylight portion of the orbit is Q_a = 27.6 KW_t x $\frac{96 \text{ minutes/orbit}}{58 \text{ minutes of light}}$ = 45.7 KW_t. Power to the absorber equals Q_a/collector efficiency = $\frac{45.7 \text{ KWt}}{0.79}$ = 57.9 KW_t. The collector area needed = $\frac{57.900 \text{ watts}}{130 \text{ watts/ft}^2}$ = 446 ft^2. The collector diameter can be calculated to be 23.8 feet, and at one pound/feet square, the weight of the collector is 446 pounds. From Figure 8-6, the absorber weight can be determined to be 265 pounds. Total weight of the collector/absorber is 711 pounds.

c. Turbo-machinery - The integrated systems electrical output of 6.092 KW_e was near enough the non-integrated electrical output of 6.57 KW_e for the non-integrated machinery of Reference 8-1 to use the parameters determined for that study directly. Two turbo-alternators in parallel, powered by two gas generators are used to provide the electrical power, and one unit alone can easily furnish the minimum electrical power requirement. Two turbo-compressors were determined to weigh 68 pounds, and two turbo-alternators were determined to weigh 110 pounds. One hundred fourteen pounds is allocated for speed control.

d. Recuperator - The recuperator recovers or transfers heat from the high pressure side of the cycle to the low pressure side of the cycle. From Figure 8-7, the recuperator weight is 225 pounds.

e. Liquid Loop Radiator - As in the non-integrated power systems, the argon is cooled in a heat exchanger which in turn is cooled by a liquid loop radiator. This liquid loop lends itself to thermal integration by making a hot fluid available without affecting the design conditions of the gas side of the heat exchanger. Figure 8-24 is a schematic of the fluid loop. The Life Support system extracts

4.01 KW_1 from the cooling fluid, reducing the temperature from 400°F to 325°F. It is assumed that the fluid line passes through a boiler, maintaining the steam thrust system at approximately 400°F. The amount of heat extracted on a continuous basis is small enough that it can be neglected for all practical purposes in this analysis. The radiator loop fluid arrives at the Absorption Refrigeration system boiler at 325°F where 7.3 KW_t is extracted, cooling the fluid down to 200°F. In systems comparison and evaluation it was decided that the use of excess thermal energy for powering Absorption Refrigeration cycle did not offer enough benefit to be included in the Brayton Integrated systems. By removing the Absorption Refrigeration system boiler heat input from consideration in the liquid loop radiator design for the integrated system, the Brayton radiator size (area and weight) would increase, and the electrical power required for the now non-integrated cabin cooling and equipment cooling loops decrease 50 watts from the value shown in Table 8-8.

The equipment sizes and weights calculated in this section were for the higher station power requirements (4.987 KW_e). However, the final Station Systems selected were at the lower power requirement (4.937 KW_e) without Absorption Refrigeration. The difference in size is only 1%, so recalculations were not made.

f. System Control Requirements - Full realization of the benefits of thermal integration for actual hardware may require further analysis of power system and life support equipment transient operation. The present analysis dealt with steady-state systems, assuming life support processes to be operating at a continuous rate. The most sensitive integrated systems are those using the Brayton cycle, since the cycle efficiency is so dependent on compressor inlet temperature. Twenty percent of the rejected heat is removed from the heat rejection loop by the life support heat exchanger, and the remaining heat is rejected in a radiator. For real systems, both the life support heat removal and the radiator sink temperature would fluctuate to some extent, forcing Brayton cycle output variations unless the heat rejection system is oversized and/or certain controls are introduced.

The best area for system control appears to be at the radiator. By slightly oversizing the radiator and providing partial radiator bypass capability, provision is made for off-nominal control in either direction. The most direct control point is compressor inlet temperature maintenance. Since two machinery sets are used, there should be two separate control systems with cross over capability.

8.5.4 DISCUSSION OF THERMALLY INTEGRATED SYSTEMS

From the analysis made in Reference 8-1, it was determined that the Brayton cycle power systems could best furnish waste heat at 400°F. To obtain heat at any other (higher) temperature would require a corresponding increase in size of the heat source and a change in cycle efficiency. It was determined that the Life Support subsystems could utilize the 400°F heat with only a small increase in the size of components and a small increase in the amount of heat used due to the longer time of operation for a process to be completed at the lower temperature.

While the Absorption Refrigeration system can utilize waste thermal energy at a temperature lower than 400°F, the net effect or improvement in the performance and weight of the total integrated system did not warrant it being included in the final analysis and evaluation of the integrated systems.

In addition of the liquid loop radiator significantly affects the power system weight, making it lighter than the gas radiator loop. In addition, the liquid loop makes integration easier because the hot liquid transfers heat more readily than gas.

Start-up of the integrated system can present a problem different from that of starting the non-integrated system. In the latter system, the turbo-machinery can be brought up to operating speed by inserting the charge of argon working gas into the turbines. The The system is not dependent on anything other than the radiator loop for waste heat rejection. In the integrated system, however, the start-up could possibly be required to be coordinated with start-up of the endothermic processes which effectively serve as part of the radiator loop or waste heat rejection system.

The isotope source has the requirement to be cooled continuously. It would be possible to utilize a radiator loop if the isotope source were designed for liquid cooling, and sufficient power to drive the circulating pumps was provided from an auxiliary source such as a RTG unit. Use of liquid in the primary cooling loop, similar to the NaK loop used with the Isotope Stirling power system discussed in Reference 8-1 should result in a beneficial reduction in power system weight, simplify start-up, and provide a relatively simple method of cooling the isotope source.

8.6 REFERENCES FOR SECTION 8

8-1 Hanson, K. L., Thermal Integration of Electric Power and Life Support Systems for Manned Space Stations, General Electric Co., MSD Final Report for Contract NAS 3-2799, NASA CR-316.

SECTION 9

SOLAR MERCURY RANKINE POWER SYSTEMS

9.1 INTRODUCTION

The Mercury Rankine power system is a solar energy power conversion system using a mercury vapor turbine to drive a permament magnet alternator, producing ac power at 2000 cps.

The major system components are:

a. A parabolic solar energy concentrator.

b. An energy absorber (mercury boiler) with lithium hydride contained between two concentric hemispherical shells, a single boiler tube spiraled within the lithium hydride.

c. A one shaft sealed, turbine-alternator-pump unit of the TRW Sunflower type.

d. A condenser-radiator integral with the vehicle external structure.

The equipment design characteristics, sizing, weight, etc. are for the most part scaled from Section 7 of Reference 9-1. The principal differences are the use of updated mercury properties (Reference 9-2) and the TRW Sunflower turbine efficiencies (Reference 9-3). The result is a more efficient system. The following paragraphs discuss detailed characteristics; a summary is presented in Tables 9-1 and 9-2 for the non-integrated systems and Tables 9-3 and 9-4 for the integrated systems.

9.2 NON-INTEGRATED POWER SYSTEM

Data is presented for two turbo-alternator systems, one system having two units on-line, the second system having one turbo-alternator unit on-line and the other on standby.

TABLE 9-1. SOLAR MERCURY RANKINE HEAT BALANCE NON-INTEGRATED SYSTEM

	ONE PLUS ONE TURBINE	TWO TURBINES
Net Power System Output	8.0 kw	8.0 kw
AC/AC Conversion	0.4 kw	0.4 kw
Battery Charging	0.46	0.46
AC/DC Conversion	0.4 kw	0.4 kw
Generator Output	9.26 kw	9.26 kw
Speed Control 5%	0.46	0.46
Alternator Efficiency	90%	87.5%
Pump Work	0.18	0.36
Alternator Losses	1.09	1.37
Turbine Power Required	10.99	11.47
Turbine Efficiency	64%	55%
Turbine Heat Requirements	626/Min	652 Btu/Min
Theoretical Heat	973 Btu/Min	1180 Btu/Min
Theoretical Energy Available	43.2 Btu/lb.	43.2 Btu/lb.
Flow Rate	22.5 lb/Min	27.3 lb/Min
Heat Work/lb	27.7	23.9
Heat Rejected	115 Btu/lb. 45.7 kw	119.2 Btu/lb 57.3 kw
Boiler Heat Required	56.6 kw	68.7 kw
Collector η = 0.816, Collector Flux Required	69.5 kw	84.2 kw
Shadow Factor = 0.375, Shadow Storage	26 kw	31.6 kw
TOTAL	95.5 kw	115.8 kw
Reflector Intercept Area = $\frac{\text{kw}}{\text{130 watts/sq. ft.}}$	735 ft^2	893 ft^2
Reflector Diameter	30.6 ft.	33.7 ft.

TABLE 9-2. WEIGHT SUMMARY SOLAR MERCURY RANKINE NON-INTEGRATED SYSTEM

	ONE PLUS ONE TURBINE	TWO TURBINES
Solar Collector @ 1 lb/sq ft	735	893
Boiler Absorber	225	275
Turbo-Alternator	48	72
Radiator	110	134
Mercury Inventory	38	38
Speed Control	55	70
Auxiliary Starter	120	150
Structural, Piping, Fittings	142	142
Second Turbo Alternator	48	
Second Mercury Inventory	38	
Additional Structure	50	
Dynamic System TOTAL	1,609	1,774
Battery System	1,000	1,000
	2,609	2,744

9.2.1 NET OUTPUT - EIGHT KW

The gross generator output is 9.66 kw of which 460 watts is allotted for speed control and 410 watts is required for battery maintenance. The ac/ac converter and the ac/dc converter each require 400 watts, such that the net continuous power to the bus is 8 kw.

9.2.2 HEAT BALANCE (SEE TABLE 9-1)

Based on Reference 9-1, pump power was assumed independent of system size. The rationale is that pump efficiency will increase as output increases for the operating region considered.

The alternator losses of 1.37 kw and 1.06 kw for the two systems are based on efficiencies of 87.5% and 90% as determined from Figure 7-1 of Reference 9-1.

Turbine requirements follow directly from the sum of generator and pump requirements. Division of the energy requirements by the turbine efficiency (See Figure 9-1) yields isentropic energy available. A Mollier diagram was constructed from the data in Reference 9-2 for cycle analysis. The Sunflower turbine entrance and exit pressures and temperatures were used as given. The energy available per pound is 43.2 Btu, the final quality of mercury being 79.5%. Flow can then be calculated, using theoretical energy required and theoretical energy per pound available. The actual energy required divided by the flow gives actual Btu/lb; turbine exit enthalpy and quality can then be calculated. The two-turbine exit quality is 95% and the one-turbine exit quality is 92%.

Heat rejection occurs in a primary radiator and in a subcooling radiator. For cycle analysis purposes they can be considered as one. The radiator inlet enthalpy = 160.4 Btu/lb. for the two-turbine case, and 156.5 Btu/lb. for the one-turbine case. Primary radiator exit (Vapor quality = 1%) is at 43.2 Btu/ lb. both cases. The net subcooling is 2.1 Btu/lb. both cases. Total heat rejected is then 119.3 and 115.4 Btu/lb., respectively.

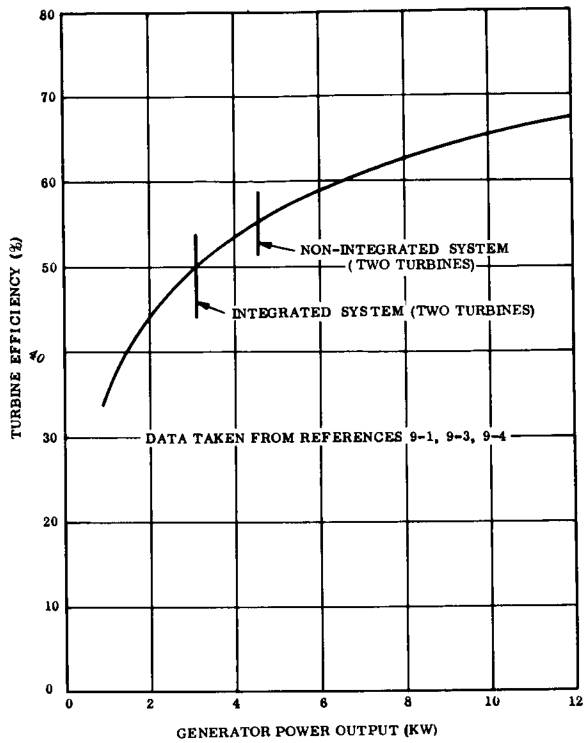


Figure 9-1. Mercury Rankine Turbine Efficiency

9.2.3 POWER SYSTEM COMPONENTS

The turbo-alternator unit weight was taken from Figure 7-3 of Reference 9-1. The turbine efficiencies, as shown in Figure 9-1, are higher for outputs under 7 kw than the previously used Figure 7-2 of Reference 9-1. This was accomplished by using the 51% efficiency for the TRW Sunflower 3.5 kw system in References 9-3 and 9-4.

Figure 7-5, Reference 9-1, was extrapolated to arrive at radiator area and weight. This data is presented in Figure 9-2. Redesign of the radiator may find a more optimum point than extrapolation, however, the net saving in this case would be quite small. The fact that the subcooling radiator would be at a lower temperature was ignored and all the heat to be rejected was considered the same. The subcool heat is less than 2% of the total, so the resulting effect is about 1/2% on radiator design.

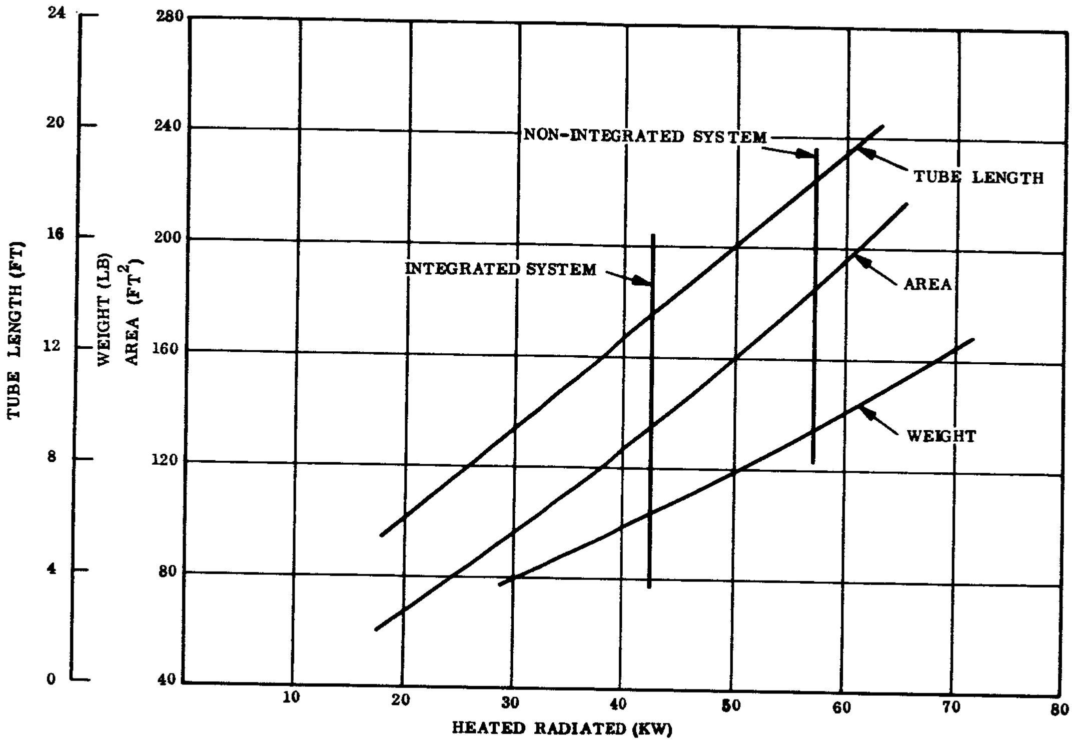


Figure 9-2. Radiator Parameters

Reference 9-3 has several boiler weights for systems with different power ratings. This data is plotted in Figure 9-3. However, the Sunflower (Reference 9-3) specifications included a 72-minute shadow duration. The ground rules for this study specify a 36-minute shadow, thus cutting the lithium hydride weight almost in half. The resulting hydride weights are 106.7 pounds and 128.7 pounds (including 10% contingency) for the one plus one and two-turbine systems, respectively. The saving or change from Reference 9-3 data is 87.3 and 105.3 pounds. The resulting absorber weight curve compares with the absorber design in section 5 of Reference 9-1 for the Stirling cycle, taking into account the difference between NAK and Mercury inventories.

Reference 9-1, Appendix B, provides the collector efficiency of 81.6%. The average boiler power was then divided by this efficiency to determine flux requirements. The shade-sun ratio multiplied by the flux required yields additional sun side flux to supply shade-side power. The total flux required then determines the collector intercept area,

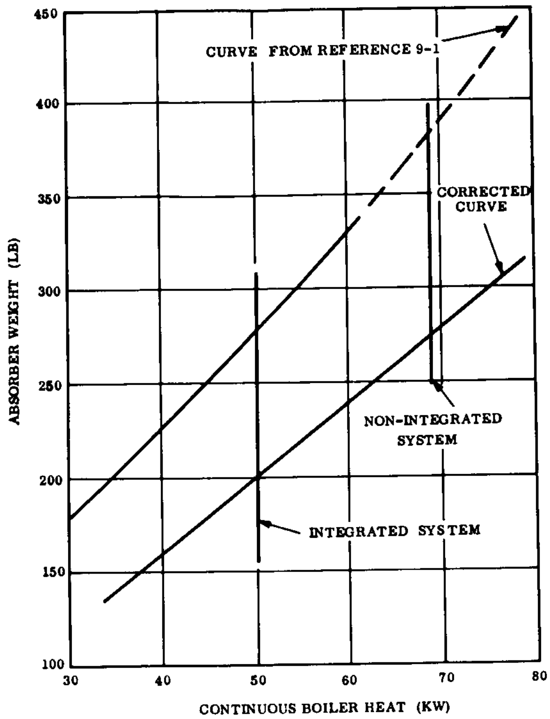


Figure 9-3. Absorber Weight Versus Average Absorber Heat

and its diameter can be determined. The total flux requirements are 95.5 kw and 115.8 kw for the one-and two-turbine systems. The resulting diameters are 30.6 feet and 33.7 feet. The weights of each are 735 and 893 pounds (at one pound/square foot projected).

The remaining component weights, the speed control, auxiliary starter, mercury inventory, structure, piping, and fittings were read from Figures 7-8, 7-9, 7-10, 7-11 in Reference 9-1.

9.2.4 DISCUSSION

The two-turbine system efficiency is 12.25%. The cycle efficiency equals $\frac{11.4\text{ kw}}{68.7\text{ kw}}$ or 16.6%. This compares with Reference 9-1 results of 10.4% and $\frac{8.3}{60} = 13.8\%$. The primary radiator energy available is 192,500 Btu/hr at initial mercury vapor quality 95%, final quality 1%, temperature 605°F, pressure 7 psia. Figure 9-4 shows a simplified cycle of the two-turbine system. Use of the vapor mixture for other uses such as integration with other subsystems will not penalize Rankine radiator sizing efficiency if the heat is removed at the primary radiator stage, since the temperature remains constant during the phase change.

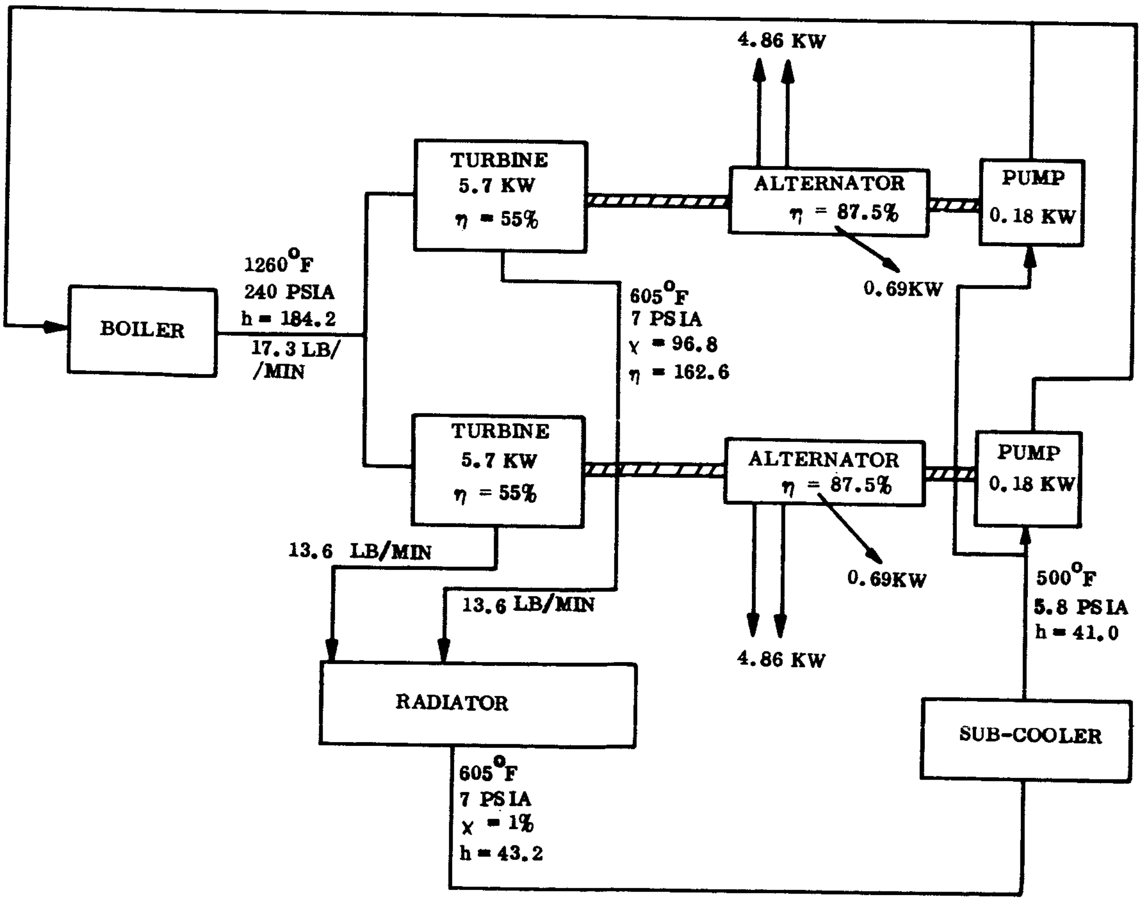


Figure 9-4. Non-Integrated Mercury Rankine Cycle Schematic Diagram

The solar collector would have to be folded during forward flight for 4.4 feet all around in one case and six feet all around in the other to fit within the vehicle external diameter of 21.7 feet.

The two studied alternatives, two turbines on-line versus one on-line, another on standby, can be compared for relative effectiveness. The one plus one system has a weight saving of 165 pounds. Further, its solar collector is three feet smaller in diameter, making it easier to stow, and saving an additional 50 pounds/year in orbit maintenance requirements. Balancing factors are second turbine start up and the slightly wetter turbine exit quality.

Another consideration would be to pay the 165 pounds for an extra radiator and speed control. This would then have the primary advantage of peak load flexibility (up to 16.4 kw vs 8 kw) without recourse to batteries by running two turbines at once. A secondary advantage would be increased system survival probability. As long as the per orbit average power does not exceed the 8 kw_e net, then the 16.4 kw peak is available without increase in the weight/size of the collector or boiler. It is expected that real missions for an orbiting laboratory would definitely have intermittent high power requirements. A weight summary of the non-integrated system is shown on Table 9-2.

9.3 INTEGRATED SYSTEM

Data is presented for two configurations, one with two turbo-alternators sharing the load, the other with a single turbo-alternator carrying all the load and a second unit on standby.

9.3.1 INTEGRATED POWER OUTPUT

Integration with life support and equipment cooling yields a resultant 4.937 kw station bus power requirement. The battery charging, speed control, and conversion power increments bring the total alternator output to 6.19 kw continuous (see Table 9-3). This compares with 9.71 kw in the non-integrated system.

TABLE 9-3. INTEGRATED MERCURY RANKINE

	Power System Sizing	
	1 + 1 Turbine System	2 Turbine System
*Net Output, KW_e	4.937	4.937 (Net power = 5.4 kw)
AC/AC Conversion KW_e	0.246	0.246
Battery Charging KW_e	0.46	0.46
AC/DC Conversion KW_e	0.153	0.253
	6.192	6.192
Alternator Efficiency %	89	85
Pump Work KW_e	0.18	0.36
Alternator Losses KW_e	0.77	1.09
Speed Control KW_e	0.25	0.25
Turbine Power, KW_t	7.39	7.89
Turbine Efficiency %	60.7	51.5
Turbine Heat Required Btu/min	420	449
Theoretical Heat, Btu/min	992	875
Theoretical Energy Available	43.2 Btu/lb	43.2 Btu/lb
Flow Rate lb/min	16.0	20.3
Heat Work, Btu/lb	26.2	22.2
Quality out of turbine %	93	96.3
Heat Rejected Btu/lb	114.8	118.8
Heat Rejected, KW_t	32.3	42.5
Boiler Heat Required, KW_t	39.7	50.4
Collector Flux Required (Eff = 0.816)	48.6	61.6
Shadow Factor = 0.375, shadow storage	18.4	23.1
Total Heat	66.9 KW_t	84.7 KW_t
Collector Area, ft^2	514	652
Collector Diameter, ft	25.6	28.83
System Efficiency, $\frac{\text{Req'd flux}}{\text{Net output}}$, % (included collector eff.)	10.2	8.0

* Actual station load is decreased by 50 watts from this level with absorption refrigeration not being used.

9.3.2 HEAT BALANCE

The heat balance calculations are listed in Table 9-3. The cycle parameters of temperature and pressure have been maintained from Sunflower Data identical to Section 9.2. Efficiency data for the alternator was taken from Figure 7.1 of Reference 9-1, efficiency of the turbine was taken from Figure 9-1.

As in Section 9.2, and in the original study, a constant pump power was used. Note that this is power per pump, and the redundancy reliability concept results in two pump-alternator-turbine packages.

The alternator losses are calculated from efficiency plot, Figure 7-1 of Reference 9-1. The Sunflower Mercury Rankine cycle (Reference 9-4) application showed retrieval of the alternator losses (heat) by the working fluid. For the purposes of the original study and this study, the energy gain at the alternator is assumed to be equal to the energy losses (through radiation to space, etc.) for the boiler, tubing, pump, etc.

Figure 9-1 shows turbine efficiency as a function of output power. The two-turbine system alternator output is 3.09 kw per unit, turbine efficiency is 50%. Using the same cycle points as before, the turbine work per pound is 21.5 Btu, the flow is 10.1 lb/min/turbine and the exit quality is 97.8%.

The heat is rejected in two radiators, a condensing unit, and a fluid subcooling unit. The amount per pound is the difference between the turbine exhaust enthalpy and the end-point enthalpy of 43.2 Btu/lb., which is 119.4 Btu/lb for the two-turbine system. Figure 9-5 is a simplified schematic of the Integrated Mercury Ranking System.

9.3.3 POWER SYSTEM COMPONENTS

Component weights and sources are listed in Table 9-4. In general, the components are scaled versions of Reference 9-1 as are the non-integrated components. Two versions are shown, one turbine on-line with one on standby, and two turbines on-line continuously.

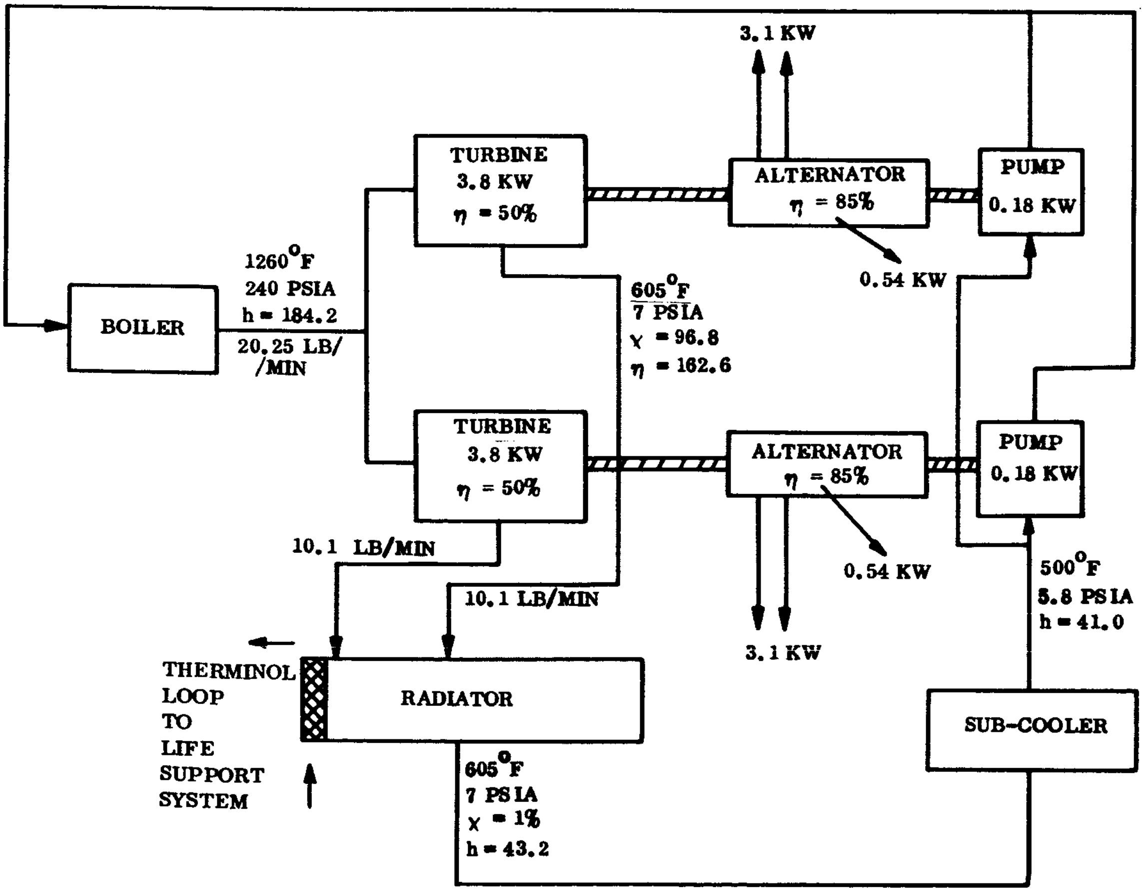


Figure 9-5. Integrated Mercury Rankine Cycle Schematic Diagram

TABLE 9-4. INTEGRATED MERCURY RANKINE WEIGHT SUMMARY

	1 + 1	2
Solar Collector @ 1 lb/ft^2	514	652
Boiler-Absorber (Figure 9-3)	157	202
Turbo-Alternator (Figure 7-3, Reference 9-1)	38	56
Radiator (Figure 9-2)	83	104
Mercury Inventory (Figure 7-10, Reference 9-1)	24	28
Speed Control (Figure 7-8, Reference 9-1)	39	56
Auxiliary Starter (Figure 7-9, Reference 9-1)	82	126
Structure (Figure 7-11, Reference 9-1)	103	120
Second Alternator	38	
Second Mercury Inventory	24	
Additional Structure	82	
	1184	1344
Batteries	1000	1000
	2184	2344
Absorption Refrigeration Radiator Saving	-17	-17
Life Support Radiator Saving	-8	-8
	2159	2319

The integrated radiator size is reduced from the non-integrated case from 184 square feet down to 136 square feet (two-turbine case). Absorption refrigeration integration reduces the power system radiator size further to 101 square feet. Original one-turbine system radiator area was 147 square feet. Power system size reduction changes radiator requirements to 105 square feet, and integration reduces the size to 84 square feet.

The solar collector is reduced in size from 735 square feet to 514 square feet for the one-turbine system; however, the diameter at 25.6 feet is still above the MORL vehicle diameter, and the outer two feet all around will have to be folded inboard during powered flight.

9.3.4 DISCUSSION

The discussion of the one-plus-one versus the two-turbine power system approaches that appeared for the non-integrated system (Section 9.2.4) is equally valid for the integrated case. Use of the one-plus-one turbine configuration permits weight savings, or increased system flexibility, or both.

The integrated two turbine system efficiency, equals $\frac{\text{net energy out}}{\text{Boiler Heat}} = \frac{5.4}{50.24} =$ 10.75%. The Rankine cycle efficiency is 7.64/50.24 = 15.2%. The cycle efficiency decline from the non-integrated case is to be expected since the turbine is smaller. The integrated weight saving is 2774 pounds - 2319 pounds = 455 pounds or 16.5% of the non-integrated system for the two-turbine configuration. The chief saving is in the collector weight.

The equipment sizes and weights calculated in this section were for the higher station power requirements (4.937 KW_e). However, the final station systems selected were at the lower power requirement (4.887 KW_e) without absorption refrigeration. The difference in size is only in the order of one percent so resizing of equipment was not made.

9.4 REFERENCES FOR SECTION 9

9-1 Thermal Integration of Electrical Power and Life Support Systems for Manned Space Stations, NASA CR-316, General Electric Company, March 31, 1965 (Final Report for NASA Contract 3-2799).

9-2 Thermodynamic Properties of Alkali Metal Vapors and Mercury, R60FPD358-A, General Electric Company, Second Revision, 1960.

9-3 Status Review of Sunflower PIC-SOL209/4, Section B-1, TRW, Inc., 24 September 1963.

9-4 Sunflower Status and Application Considerations Bulletin 311-MRD-3, TRW, Inc., November, 1963.

SECTION 10
SOLAR CELL/BATTERY POWER SYSTEM

10.1 INTRODUCTION

The Photovoltaic Battery Power System consists of solar-electric energy converting solar cells, power conditioning equipment, a nickel-cadmium battery set for energy storage, and connecting wire cabling.

The Solar Cell Supply and Battery Storage Power system is used as the standard for comparison with the various dynamic power systems. The reason that this combination is used as the standard is that it was the primary system considered in the MORL application.

10.2 SOLAR ARRAY MODULE DESCRIPTION

The information contained here has been extracted from Reference 10-1, Section 8.1.1. The solar array contains N/P silicon solar cells with an 11% efficiency at 85°F and direct solar radiation. The following area/efficiency losses are assumed:

a.	Soldering and manufacturing processes	.97	
b.	Cover glass/filter attenuation	.92	
c.	Erosion, degradation, cell failures	.95	
d.	Lumped measurement uncertainties	.96	
e.	Temperature degradation	1.00 - .0026	per °F above 85°

The estimated temperature is 124°F, factor is 89.7%. The solar constant is 130 watts/square foot for input power to the solar cells. The output is determined by multiplying the input by the loss factors: 130 (.11) (.97) (.92) (.95) (.96) (.897) = 10.45 watts/square foot of cell area. The gross array area includes non-active structure, etc. The packing factor of area effectiveness is .90, consisting of .99 module packing and .91 module distribution on panel.

The panel with cells weighs 1.164 pounds/square foot of active cell area (Reference 10-1). The cell modules weigh 0.764 pound/square foot and the supporting structure 0.4 pound/ square foot of active cell area.

10.3 BATTERY DESCRIPTION

The batteries consist of nickel-cadmium cells charged with a two-step method. Nickel-cadmium was chosen as the best choice considering charge-discharge cycles, life, and weight. To maintain battery capacity over the use duration, overcharging is required for oxygen recombination. The overcharge amount is 25%, the maximum overcharge rate is 1/6 rated capacity. The direct recharge can be accomplished at 1/3 rated capacity rate. This two-step method reduces total battery requirements considerably over a one-step system. However, a control system to sense 100% charge and control the overcharge will be required. The specific battery energy capacity is 11 watt-hours/pound. This includes case, connections, mounting brackets.

10.4 SYSTEM ANALYSIS

Figure 10-1 is a block diagram of the power system with associated loads and efficiencies. In order to provide power with voltage regulation equivalent to the dynamic systems, a voltage regulator has been included in the system.

10.4.1 POWER SYSTEM SIZE

The continuous power supplied during the light periods is determined from the required average power of eight kw, peak output of 12 kw for one hour in 24, the battery energy charge requirements, and the many losses which occur (the system components being less than 100% efficient). The method of handling the peak load is to distribute it uniformly over the 24-hour period, thus creating an average of 8.16 kw. The power is split between the ac and dc sides; dc voltage regulation efficiency = 90%, ac inverter efficiency = 87%. Resultant average power upstream is 9.23 kw.

$$\frac{\text{AC Output}}{\text{Inverter}} = \frac{4.08}{0.9} = 4.54$$

$$\frac{\text{DC Output}}{\text{Regulation}} = \frac{4.08}{0.87} = 4.69$$

Total 9.23 kw

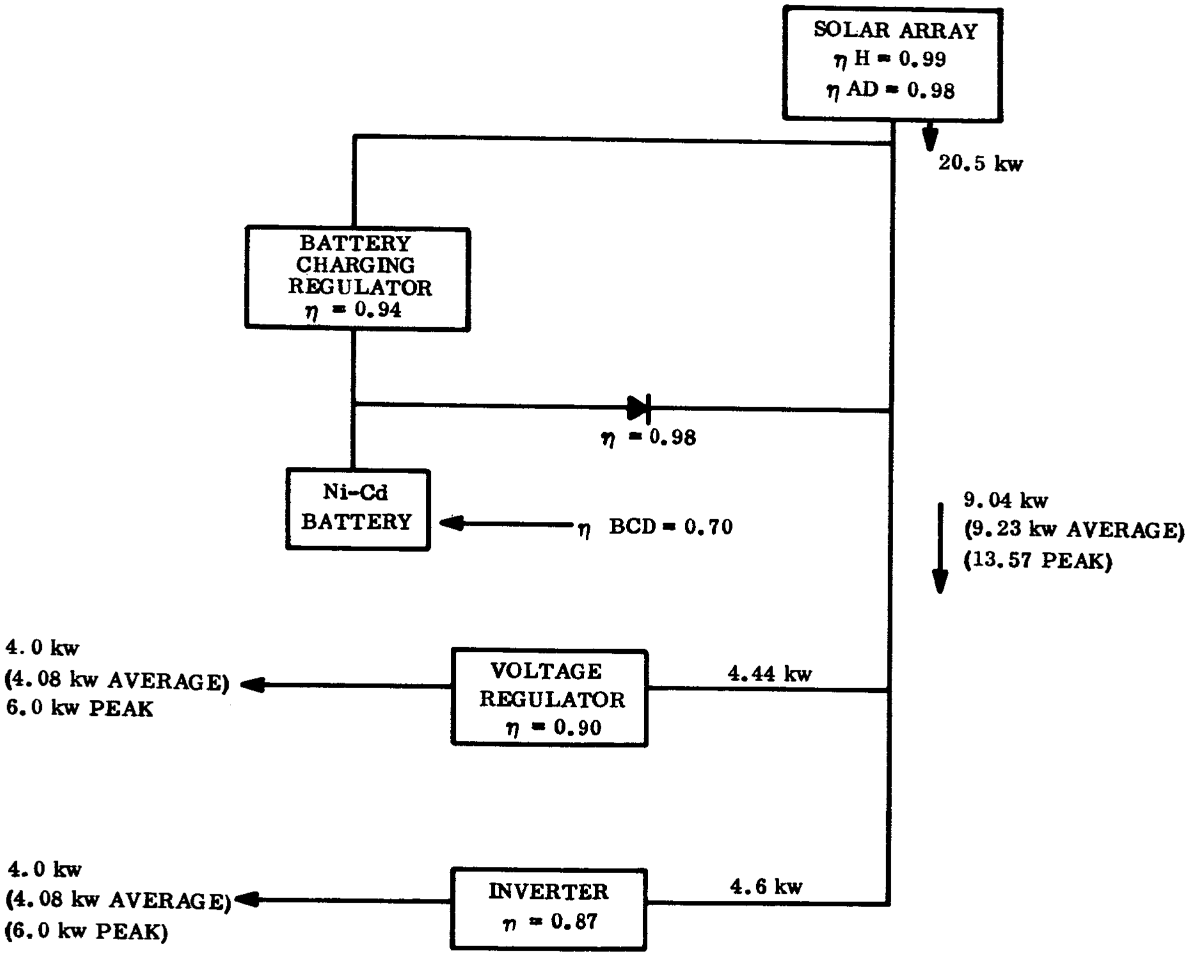


Figure 10-1. Photovoltaic-Battery Power System, Block Diagram

10.4.2 BATTERY ANALYSIS

The required battery power is given by the equation (Reference 10-1, Section 8.2.1):

$$\text{Battery Power} = \frac{T_D}{TL\ \eta\epsilon} \frac{1}{UF} PC$$

Where

T_D = Dark Time = 36 minutes

TL = Light Time = 58 minutes

η_ϵ = Battery Power Efficiency, Includes

η	Harness	=	0.99
η	Array Diodes	=	0.98
η	BCR	=	0.94 (Battery Charge Regulator)
η	BCD	=	0.70 (Battery Charger-Discharge Efficiency) Including Overcharge
η	ID	=	0.98 Isolation Diode

$\eta\epsilon = 0.626$

$\frac{1}{UF}$ is the two-step charge factor = 1.20

P_c = 9.23 kw Continuous Power

Then system input power can be determined:

$$\text{Power} = \frac{36}{58} \frac{1}{0.626} (1.20)\ (9.23) = 10.97 \text{ kw}$$

The battery size can be found if the η AD, η H, and η BCR are removed, since they are upstream of the battery.

Battery Power Input = 10.97 (0.94) (0.99) (0.98)
= 10.02 kw

Average Charge: Voltage = 39.8 volts
Current = 252 amperes

Since the charge rate is one-third the one hour rating current:

Battery Rating = 252 (3) = 756 ampere-hours

Battery Energy Capacity = Ampere Hours (volts) = Watt-Hrs

Battery Energy Capacity = 756 (34.7) = 26,200 Watt-Hrs
(where 34.7 volts is average discharge level)

Specific Energy/Lb = 11 w-hr/pound

$$\text{Battery Weight} = \frac{\text{Capacity}}{11} = \frac{26,200}{11} = 2385 \text{ lb}$$

$$\text{Depth of Discharge} = \frac{9.04 \ \frac{36}{60}}{26,200} = 20.7\% \quad \text{For Nominal Load}$$

$$\text{Depth of Discharge} = \frac{13.57 \ \frac{36}{60}}{26,200} = 31.1\%$$

for peak load occurring on the dark side. These are less than those described in Reference 10-1 for similar profiles. The information presented there is included here for comparison

DISCHARGE DEPTH		NO. OF CYCLES IN YEAR	LIMIT FOR NI-CD DESIGN
Reference 10-1	0.23	5600	8800
	0.36	610 (worst case)	5500
This study	0.21	5600	9500
	0.31	610 (worst case)	6700

It is clear that the batteries should have sufficient life for a one-year mission.

10.4.3 SOLAR ARRAY SIZE

The required battery system input power is 10.97 kw (from the preceding section). The direct power requirement is:

$$\frac{\text{Power Continuous}}{\eta \text{ Harness} \ \ \eta \text{ Array Diode}} = \frac{9.23}{(0.99)\ (0.98)} = 9.53 \text{ kw}$$

The total power required from the array is then 10.97 + 9.53 = 20.5 kw. The active area is:

$$\frac{\text{Total Power}}{\text{Power/Sq Ft}} = \frac{20,500}{10.45} = 1962 \text{ Sq Ft}$$

The gross area is $\frac{1960}{0.9} = 2180$ Sq Ft

Array Weight = (1960) (1.164) = 2285 lb

10.4.4 POWER CONDITIONING EQUIPMENT

The following power conditioning equipment is required for system operation:

a. Voltage Regulator - regulation shall be $\pm .5$ volts, efficiency is 90% and the specific weight is 15 lb/kw (from Reference 10-1).

b. Battery Charge Regulator - a pulse-width modulated device with efficiency of 94% and a specific weight of 4 lb/kw (from Reference 10-1).

c. Ampere-Hour Meter - used to control charge regulator depending on battery charged level, 5 lb total weight estimated (from Reference 10-1).

10.4.5 POWER HARNESS

The wire cabling that takes the electrical energy from the solar array and distributes it to other parts of the power system is estimated to weigh 13 lb/kw (from Reference 10-1).

10.5 SOLAR ARRAY STORAGE AND DEPLOYMENT

The solar array is stored in eight sections around the vehicle perimeter between stations 1690 and 1795 (see Figure 10-2). The section panels and the sections are held in place during powered flight and otherwise prior to use by support bearings, structure, and clamps. The array storage and deployment design was changed from that of Reference 10-1 since that design did not allow any more array area without diminishing panel thickness.

NOTE: CALLOUTS ARE LOCATED ON FIGURE 10-3.
TRIANGLE INDICATORS IDENTIFY DEPLOYMENT
SEQUENCE AS DESCRIBED IN SECTION 10.5

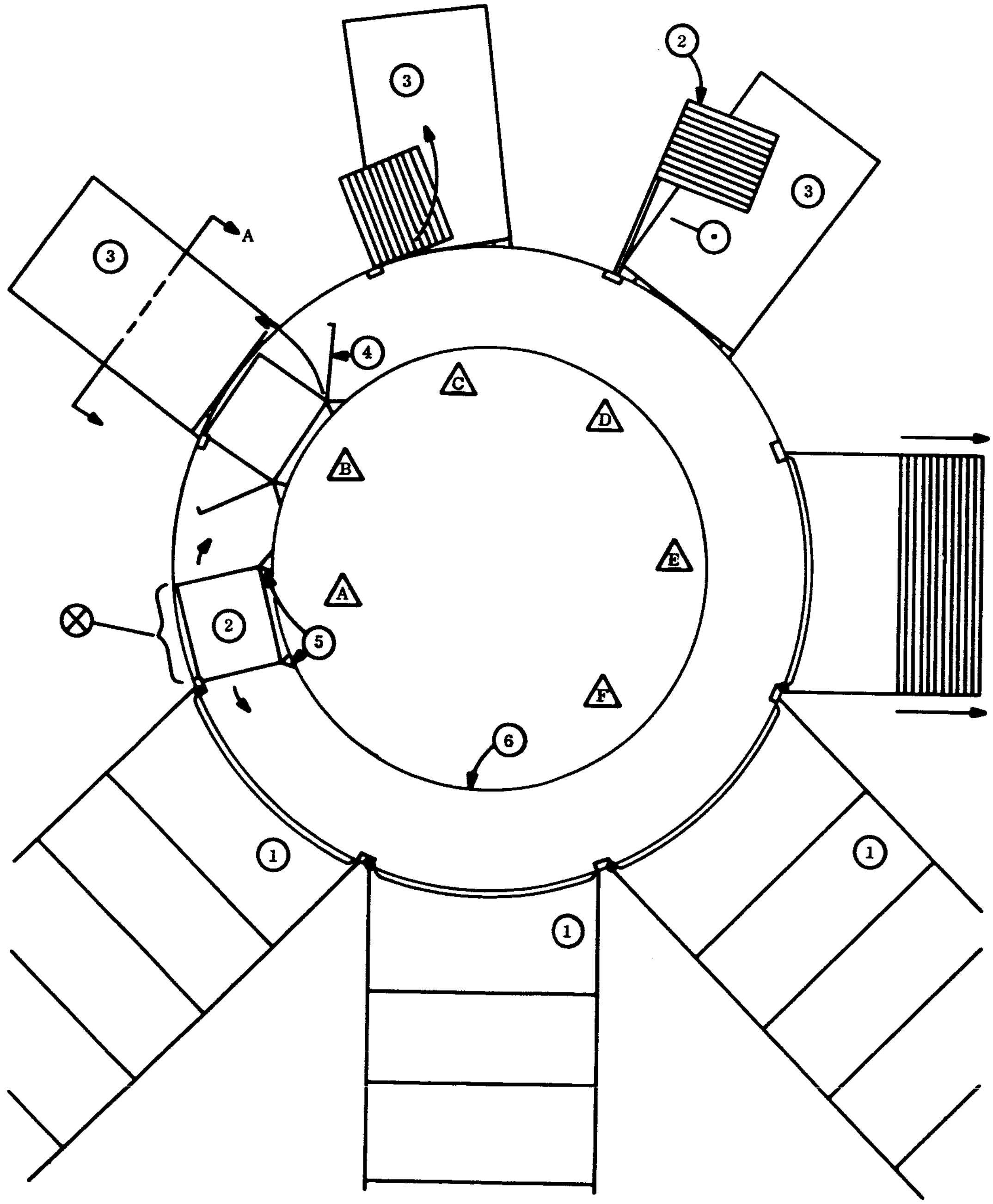


Figure 10-2. Solar Array Deployment, View From Station 1699 Looking Towards Cabin

An advantage of the design presented here is a roll moment of inertia lower by 2×10^4 slug-feet^2 and pitch and yaw moment lower by 2×10^4 slug-feet^2.

The array stored volume is 570 cubic feet. The associated structure, clamps, bearings, hinges, etc. occupy, in effect, another 80 cubic feet of space. An additional 400 cubic feet must be kept clear for motion during deployment. The deployment sequence is as follows (see Figure 10-2):

a. Fire squibs allowing skin section to rotate outward, and panel clamps to rotate free(forced by torsion springs) (see Figure 10-3).

b. Command relays to free section from rear supports, section will rotate about skin area bearings to outside of vehicle.

c. Command relay to free outer nine panels, they will rotate about outer edge of inboard panel.

d. Free forward bearing support on inboard panel and rotate array section such that inboard panel faces away from cabin; latch in place.

e. Extend panels to deployed position. The deployed configuration will look as depicted in Figure 10-3.

10.6 SYSTEM SUMMARY

Table 10-1 presents the weight breakdown for the power system. The pounds per net kw is $\frac{5697}{8.16} = 698$ lb/kw. This compares with the system in Reference 10-1 which was $\frac{4323}{5.87} = 737$ lb/kw. The net power to total power factor is $\frac{8.16}{20.5} = 0.398$. This compares with the system in Reference 10-1 which was $\frac{5.87}{15.54} = 0.378$.

10.7 REFERENCES FOR SECTION 10

10-1 Hanson, K. L., "Thermal Integration of Electrical Power and Life Support Systems for Manned Space Stations," General Electric Company, MSD, NASA Report CR-316 (Final Report for NASA Contract 3-2799).

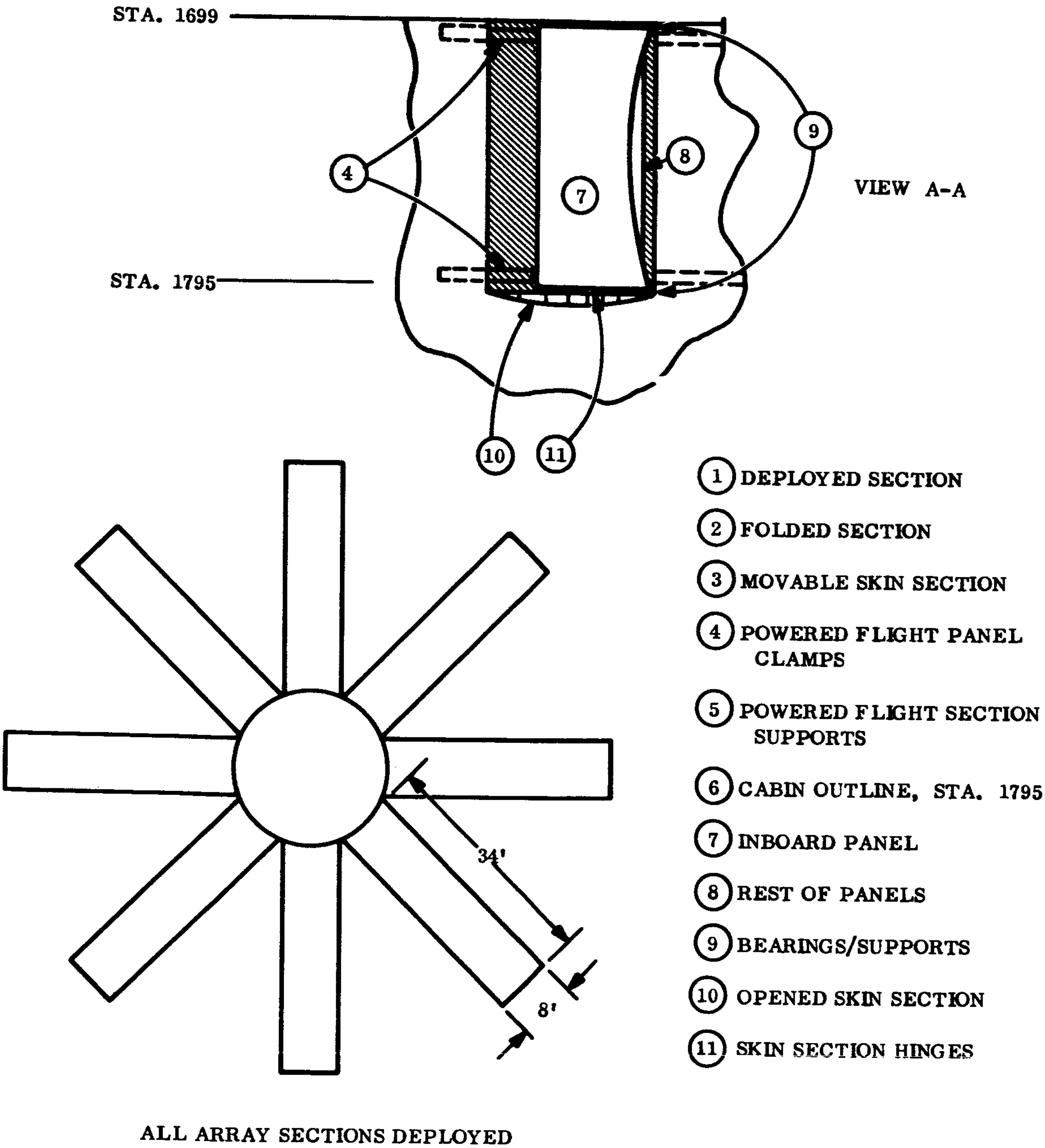


Figure 10-3. Array Deployment Details, Deployed Configuration

TABLE 10-1. WEIGHT SUMMARY

COMPONENT	WEIGHT (LB)
Battery	2385
Solar Array	2285
Cells - 1500 Structure - 785	
Voltage Regulator	67
Battery Charge Regulator	45
Ampere Hour Meter	5
Supply Harness	267
Battery Colling Equipment	40
Additional Structure (Array Support, Deployment Mechanism, Panel Tie-Downs)	600
TOTAL	5694

SECTION 11

TWO-MAN LUNAR SHELTER

11.1 INTRODUCTION

The benefits of thermal integration of electric power systems with the other systems found on a manned space station have been identified from the several station systems studied, with one system being selected for comparison with a Photovoltaic power system. It follows that other space missions, in manned lunar missions, offer a potential for reducing the size of dynamic electric power plants by thermally integrating the power systems. Analyses and configuration estimates for the thermal integration of two power systems with a life support system for a two-man lunar shelter is discussed in this section.

11.2 SHELTER DESCRIPTION

For the purpose of evaluating the potential advantages of thermal integration it is not necessary to perform a shelter design. To utilize existing information and previous studies to the fullest possible extent, the Roving Vehicle MOLAB has been used as a reference vehicle size. The characteristics of the MOLAB are presented in detail in References 11-1 and 11-2.

The shelter configuration selected for the study was a horizontal cylinder similar in size and shape to the MOLAB. A cabin volume of 210 Feet^3, with a free volume of 200 Feet^3 was selected. An airlock volume of 80 Feet^3 was assumed. Shelter heat flux distribution, cabin wall heat flux and steady-state cabin temperatures, reproduced from Reference 11-1 are shown in Figures 11-1, 11-2 and 11-3. Life Support equipment, sized by extrapolation for the Two-Man Shelter from the MORL configuration will be assumed to be located either inside or outside the cabin, depending on the equipment requirements to survive the Lunar Environment. Internal arrangement of equipment is not considered important to the assessment of the effect of lunar thermal environment on thermal integration.

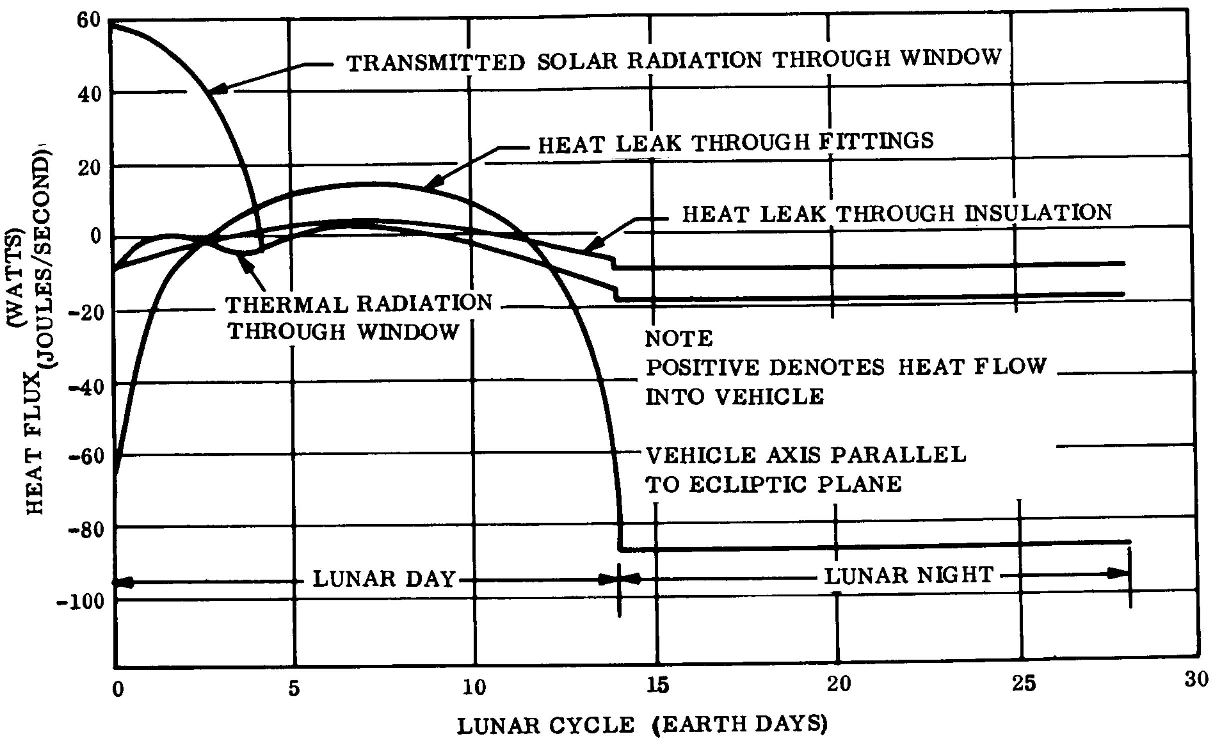


Figure 11-1. Heat Flux Distribution

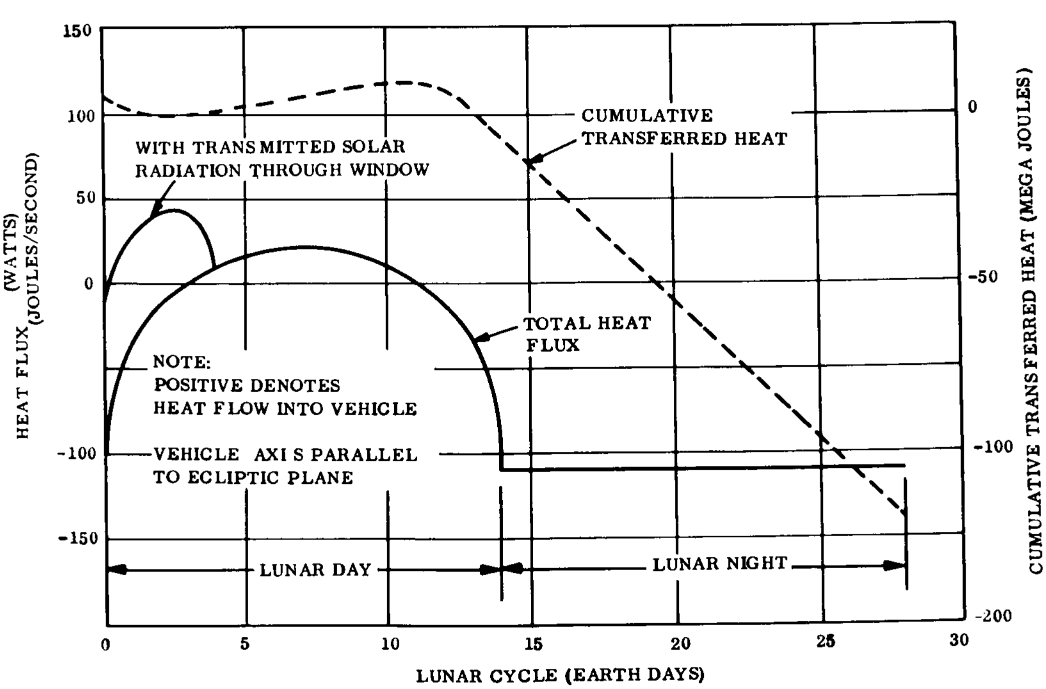


Figure 11-2. Cabin Wall Heat Flux

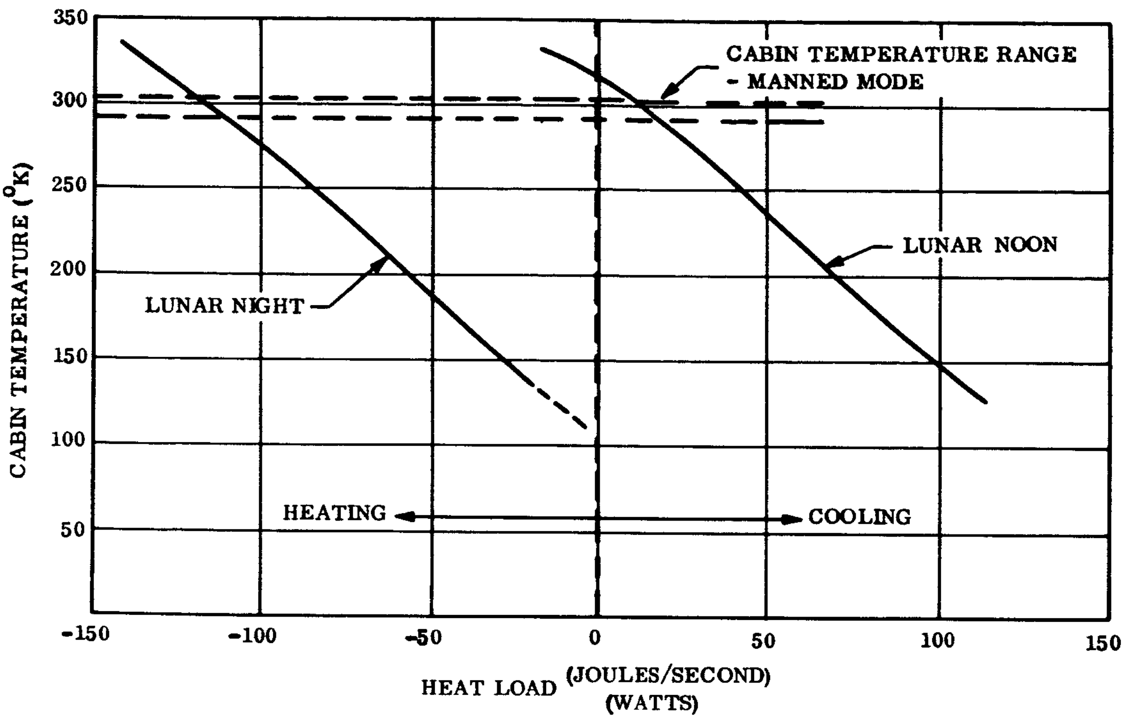


Figure 11-3. Steady-State Cabin Temperatures

11.3 LUNAR THERMAL ENVIRONMENT

It is necessary to determine and define for this study the lunar thermal environment used to evaluate the effect of that environment on thermal integration. This section is intended to present information which is considered to be most representative of accepted information on the thermal characteristics of the lunar surface.

11.3.1 SURFACE THERMAL ENVIRONMENT

The NASA thermal model consists of a layer of material A 9 mm thick, evenly distributed over an infinite layer of material B. Material properties are:

Material A — K (thermal conductivity) = 3×10^{-6} cal/cm/sec/^{o}C

ρ (density) = 1.8 g/cm^{3}

C_p (specific heat) = 0.2 cal/g/^{o}C (at 25^{o}C)

Material B $\qquad K = 4 \times 10^{-3}\ cal/cm/sec/^{\circ}C$

$\rho = 3\ g/cm^3$

$C_p = 0.2\ cal/g/^{\circ}C$ (at $25^{\circ}C$)

The significance of these values is that the lunar surface temperature may be locally affected by thermal interaction with a vehicle, behaving like an adiabatic surface.

11.3.2 TEMPERATURE-TIME HISTORY OF THE LUNAR SURFACE

If it is assumed that the lunar equator and its orbit about the earth are both in the plane of the ecliptic, then the temperature of an element of the lunar surface while illuminated by the sun can be reasonably represented by:

$$T = T_s \cos^{1/4} \phi \cos^{1/4} \lambda$$

where: T_s = maximum subsolar point temperature, $700^{\circ}R \pm \overset{34^{\circ}F}{32^{\circ}F}$ (see Section 11.3.3).

ϕ = Lunar latitude relative to subsolar point (0° for this application).

λ = Lunar longitude relative to subsolar point

Figure 11-4 shows the time-temperature relationship of a point near the center of the moon disc for one lunation. This relationship as developed by Wesselink, seems to be agreed upon by most students of the moon.

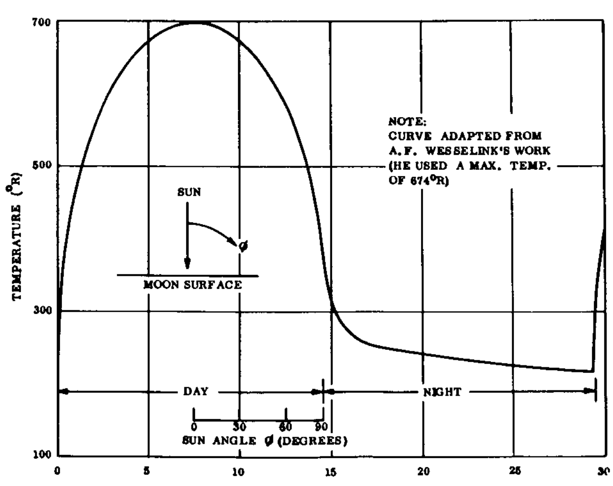


Figure 11-4. Estimated Moon Surface Temperature History for an Area Near the Center of the Disc

11.3.3 TEMPERATURE EXTREMES OF THE MOON'S SURFACE

The range of values used are those recorded by Pettit and Nicholson (Reference 11-3) in their various experiments during normal lunar phases as well as during eclipses. These are:

Maximum Temperature (subsolar point)	Minimum Temperature (Lunar night)
$700^{\circ}R \begin{smallmatrix} +34^{\circ}R \\ -32^{\circ}R \end{smallmatrix}$	$216^{\circ}R$ Tolerances have not been included because of lack of data

Maximum Temperature: $700^{\circ}R$ is a theoretical value based on an albedo of the moon (0.073), on an emissivity of the moon (0.99), and on the average solar constant. $734^{\circ}R$ was measured by Pettit and Nicholson in 1930. $668^{\circ}R$ was measured by Pettit and Nicholson just before the lunar eclipse of October 27, 1939.

Minimum Temperature: $216^{\circ}R$ was measured by Pettit and Nicholson.

11.3.4 SPHERICAL ALBEDO

The ratio between the light of the sun scattered in all directions by a hemisphere to the total light falling on the hemisphere is reported as follows:

Russell reports	0.073
Rougier reports	0.073
Piddington and Minnett reports	0.100

from the minimum temperature reported by Pettit and Nicholson ($668^{\circ}R$) and an emissivity of 0.99. It is, therefore, assumed that a reasonable value for spherical albedo is $0.073 \begin{smallmatrix} +0.16 \\ -0.0 \end{smallmatrix}$.

11.3.5 LOCAL ALBEDO

The ratio of brightness of a diffusing surface to the brightness of an absolutely white surface placed normal to the rays of the sun is defined as local albedo. The local albedo applies to a small area of the moon which can be considered plane; the average of the local albedos over the entire surface of the moon corresponds to the spherical albedo in Section 11.3.4, above. Sytinskaya (Reference 11-4) reports the following:

Variation in Local Albedo

Maria	0.066	each represents average of several measurements
Paludes	0.092	
Continents	0.105	
Craters	0.112	
Rays	0.131	

Smallest value measured was 0.051

Highest value measured was 0.176

It is assumed that $0.10 \begin{smallmatrix} +0.13 \\ -0.05 \end{smallmatrix}$ is a reasonable value for albedo affecting a vehicle landed on the moon. The range above includes the value of 0.23 which results from radiation heat balance calculations for the moon (see Section 11.3.6).

11.3.6 MOON EMISSIVITY

The value assumed for the moon emissivity is $0.9 \begin{smallmatrix} +0.09 \\ -0.13 \end{smallmatrix}$. The range has been set with some discretion since no data was found on this subject. The upper limit was derived from radiation balance with Pettit and Nicholson's minimum subsolar temperature of 668°R (albedo was then 0.23). The lower limit was derived from their minimum temperature of 734°R (albedo used was 0.1, from Piddington and Minnett).

11.3.7 SURFACE OF THE MOON

The following discussion is extracted from Reference 11-5. The jagged peaks and precipitous lunar landscape that was once believed to exist on the Moon has given way to a landscape in favor of very gentle slopes, not exceeding about 10 degrees. The basis for this change in concept is the recent work on photographic shadow analysis of lunar features, which indicates that the larger surface formations do indeed have gentle slopes, despite their rough appearance through the telescope. Unfortunately, this information has been applied to all lunar surface features, resulting in very unconservative specifications for local lunar conditions. It is the purpose of this discussion to indicate that the smaller and more frequently encountered surface features of the Moon may have relatively steep slopes, and that slopes much greater than 15 degrees may be encountered. This argument is based primarily on lunar craters, since these form a continuous family with features that may be extrapolated below the limit of telescopic resolution. Whether or not other surface features contribute to small scale roughness has not been resolved, and as a result the total roughness of the lunar surface may exceed that contributed by craters alone, as seems to be the case from Ranger VII data. Figure 11-5A depicts the relationship between crater diameter and slope in degrees, the data being plotted from Baldwin (Reference 11-6). The curve shown is for a diameter less than 5000 feet, which is the limit of telescopic resolution. Notice that Ranger VII data shows 30-degree slopes at 100-foot diameters and 45-degree slopes at 10-foot diameters.

Baldwin has plotted crater depth versus crater diameter for lunar craters as well as some terrestrial craters and explosion pits. The results of these plots are shown in Figure 11-5B. The data have been extrapolated for diameters less than 5000 feet (terrestrial telescopic resolution) and form the basis for the argument that steep slopes do exist on the moon. Here again, plotting of Ranger VII data shows slopes as steep and shapes in general agreement with those depicted by Baldwin.

Also shown in Figures 11-5A and 11-5B are depth and diameter ratios and slopes taken from sources which disagree with Baldwin's data. Note, however, that these estimates

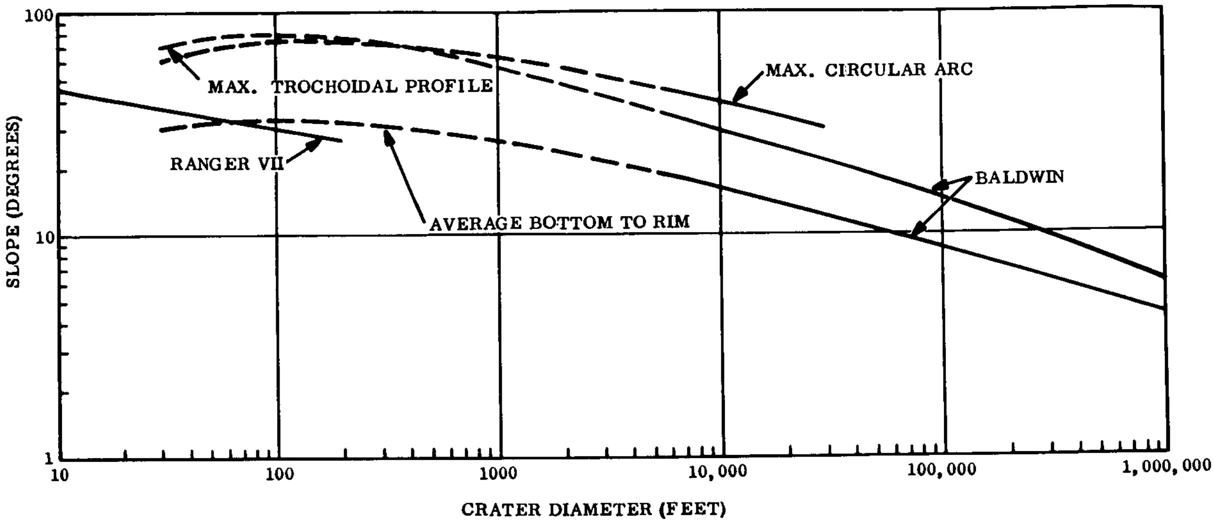


Figure 11-5A. Slopes in the Lunar Craters

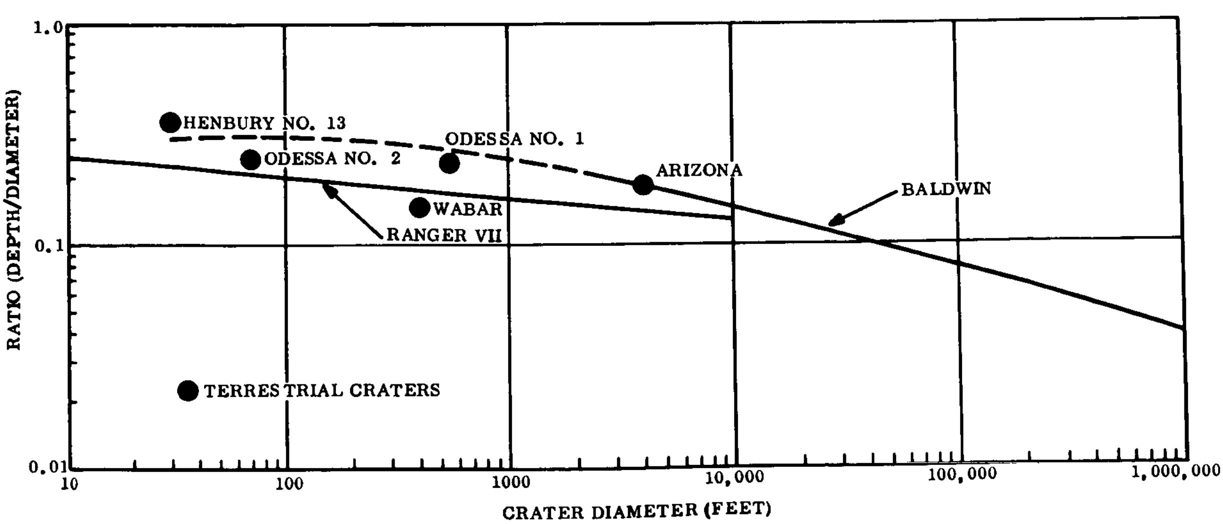


Figure 11-5B. Shapes of Lunar Craters

result in slopes higher than Baldwin's. Also note that had the curves been extrapolated back to the region of small crater sizes without using terrestrial craters and explosion pits as a guide, the result would undoubtedly have been estimates of slopes higher than Baldwin's. Thus, on all counts, Baldwin's estimates are low.

AVERAGE LUNAR CRATER DATA

Crater Diameter	100 ft	1000 ft	10,000 ft
Average Slope	30 deg	24 deg	16 deg
Maximum Slope	70 deg	46 deg	28 deg

It is interesting to note, and most significant to observe, that Ranger VII data agree well with Baldwin's data for small crater diameters, thus Baldwin must be recognized for having performed some early work of high quality and validity.

11.4 LUNAR HEAT REJECTION

11.4.1 INTRODUCTION

A significant aspect of the lunar environment affecting the power system and life support systems in their rejection of waste heat is the wide variations in heat sinks between lunar day and lunar night. This imposes thermal control problems for lunar base heat rejection that are far less severe for an orbiting vehicle. Another consideration is elevated sink temperatures during the lunar day. The combined effects of the hot surface and impinging solar flux must be circumvented to reject waste heat. Elevating the radiator effective temperature with a heat pump type system can benefit heat rejection at the expense of additional power. In the same manner, an absorption refrigeration system is of possible use. These methods will be explored in Sections 11.5, 11.6 and 11.10. This section considers the radiator configuration and control loop possibilities.

11.4.2 RADIATOR DESIGN

The radiator design problem is simply to maximize the amount of heat to be rejected per unit weight. For the purposes of this section, the unit weight will be considered equivalent to unit area, thus direct comparison of different heat rejection systems can be made in terms of Q/A (Btu/hour/square foot).

Some of the methods available for obtaining lowered sink temperatures include: (a) local lunar surface modification, (b) variable radiator flow paths and rates, (c) radiator orientation, and (d) low solar absorptance radiator surface coatings. Optimization might result in a combination of two or more of these methods.

Figures 11-6 through 11-11 indicate the ideal heat rejection rate per unit area which may be expected during the lunar day for a range of radiator effective mean temperatures for several radiator configurations. The ideal heat rejection rates have been presented rather than just radiator sink temperatures for two reasons: (a) Q/A values are more useful in sizing radiator requirements for given mean effective temperatures; and (b) when the moon is not considered as an infinite source and sink in the local area immediately adjacent to a vehicle, as done in Figures 11-8 through 11-11, sink temperatures are a function of the radiator effective temperature. The condition where the sun is directly overhead is emphasized in the analysis since this is the worst case position for heat rejection in most cases.

For practical radiator designs, both radiator effective temperature and fin efficiency must be considered. The effective temperature can often be estimated as the mean of inlet and outlet temperatures if the difference is not large and if the outlet temperature is much higher than the sink temperature. The effectiveness of a radiator then accounts for the inlet-to-outlet temperature gradient. Fin efficiency arises when the radiator is designed as tubes spaced on a thin metal sheet, a configuration used to optimize weight. The fin efficiency accounts for the temperature gradient from a tube (where the working fluid is located) to the center line of the thin sheet. Radiators considered for the MORL application generally have a weight optimized fin efficiency of 0.7 to 0.85 depending on the effective temperature. The radiator thermal equation is then:

$$Q/A = \eta \; F_{\epsilon} \; F_{A} \left(T^{4}_{1_{EFF}} - T^{4}_{2_{EFF}} \right)$$

where:

F_{ϵ}	=	emissivity factor
F_A	=	area factor
T	=	Stephan-Boltzmann constant
η	=	fin efficiency
$T_{1\ EFF}$	=	radiator effective temperature
$T_{2\ EFF}$	=	sink effective temperature

To use the plots of heat rejection for real radiators, the radiator effective temperature must be computed. The equation for this is as follows:

$$T^{4}_{effective} = T_S^{\,4} + \frac{4\,T_S^{\,3}\,(T_H - T_L)}{\log\left[\left(\frac{T_L+T_S}{T_L-T_S}\right)\left(\frac{T_H-T_S}{T_H+T_S}\right)\right] + 2\tan^{-1}\left(\frac{T_L}{T_S}\right) - 2\tan^{-1}\left(\frac{T_H}{T_S}\right)}$$

T_S	=	sink temperature
T_L	=	radiator tube temperature at exit
T_H	=	radiator tube temperature at entrance

11.4.2.1 <u>Horizontal Radiators</u>

The simplest radiator configuration is a flat plate, oriented parallel with the lunar surface. Since the lunar shelter is stationary, it is assumed that the shelter location can be chosen such that there is no viewing of hills, crater rims, etc., by the space directed side of the radiator. The radiator space side must be coated to minimize the solar absorptance. Two coating values are used, $\alpha s/\epsilon = 0.25/0.9$ (Figure 11-6) and $\alpha s/\epsilon = 0.05/0.84$ (Figure 11-7). The first is reasonable value for state of the art paints, the second is apparently a quartz reflector sandwich that has recently been developed (Reference 11-7). By comparing Figures 11-6 and 11-7 it is easy to see that the 0.05/0.84 coating is far the superior heat dissipator for the sun overhead condition on an area basis.

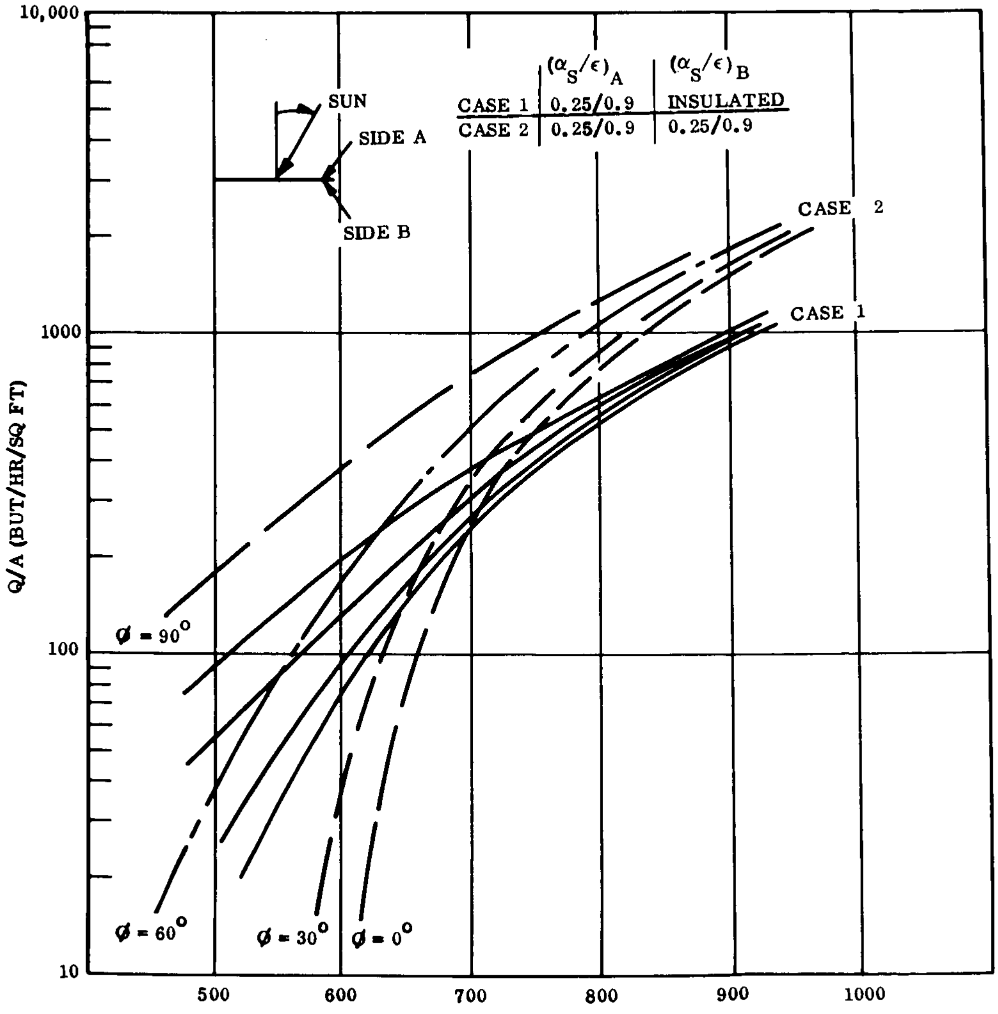


Figure 11-6. Mean Heat Rejection Capability of Flat Plate Horizontal Radiator (Parallel with Moon Surface)

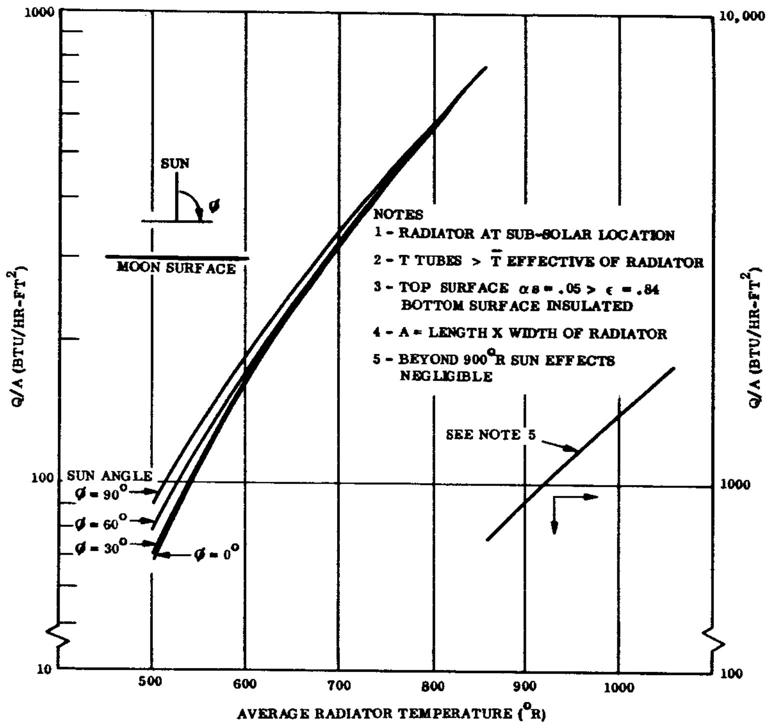


Figure 11-7. Mean Heat Rejection Capability of Flat Rate Radiator Horizontal to Moon Surface as a Function of Effective Mean Temperature

11.4.2.2 <u>Cylindrical Radiators</u>

Another concept is to have a cylindrical radiator, with its axis perpendicular to the lunar surface. An application easily visualized is to make this radiator integral with the cabin wall. Two kinds are analyzed, one with a coating of 0.25/0.9, another with a coating of 0.25/0.9 and locally modifying the lunar surface with a 0.25/0.9 coated rug. Figure 11-8 depicts the heat rejection for the cylinder alone. Figure 11-9 shows the heat dissipation capability using the lunar rug.

A simplifying assumption that was used is that the heat rejected represents a circumferential mean. For actual radiators, this would mean many circumferential tubes with a high circulation, or re-analysis on a zonal basis. An advantage of this concept is its relative independence of sun position for performance (see Figure 11-9) easing the control problem from lunar day to night.

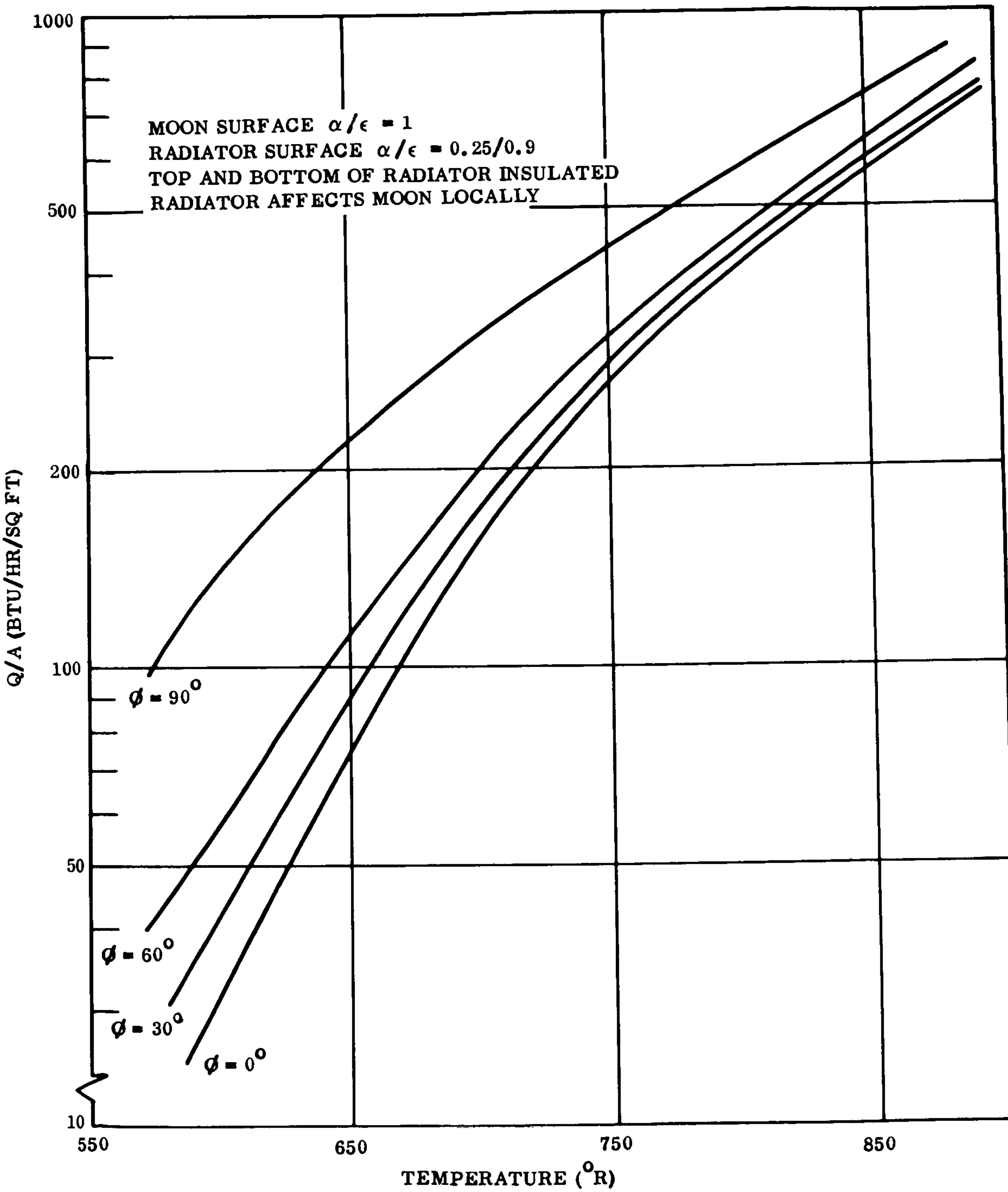


Figure 11-8. Mean Heat Rejection Capability of a Cylindrical Radiator on the Moon (℄ Cylinder Perpendicular to Moon)

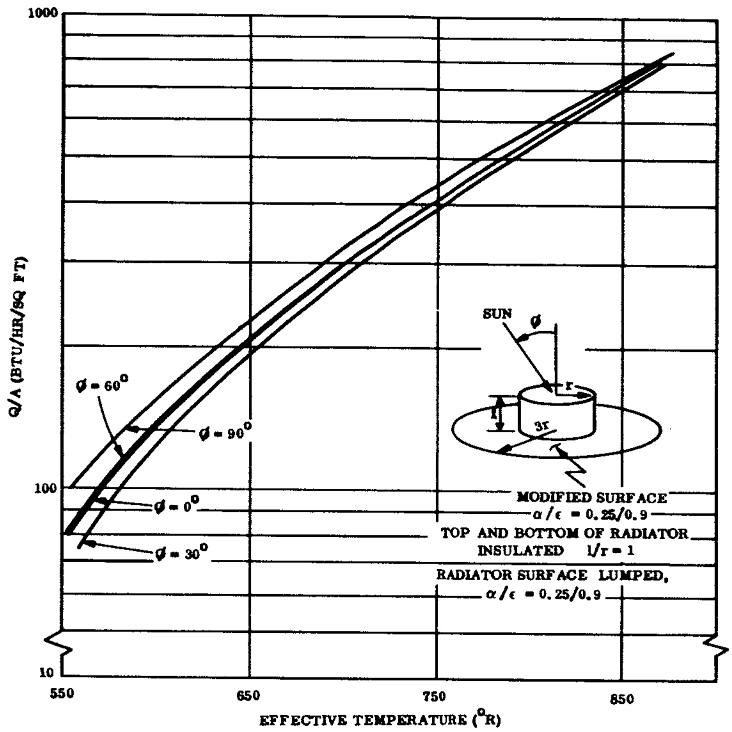


Figure 11-9. Mean Heat Rejection Capability of a Cylindrical Radiator on the Moon (Radiator ℄ Perpendicular to Moon), Moon Surface Modified Locally

11.4.2.3 Vertical Fin Radiators

The third concept is to use a flat vertical fin, oriented East-West on the lunar surface to virtually eliminate incident solar energy on the radiator. One design with a locally modified lunar surface is shown in Figure 11-10. The surfaces used were $\alpha s/\epsilon = 0.05/0.84$. Another possibility is to use a low absorptivity and emissivity rug $\alpha s/\epsilon = 0.06/0.06$ (Reference 11-8) as shown in Figure 11-11. For comparison purposes it is assumed that the shelter will be located near the lunar equator, then both sides of the fin can be utilized as radiating surfaces, doubling the amount of heat that can be radiated per unit area (it is assumed that there are rugs on both sides also).

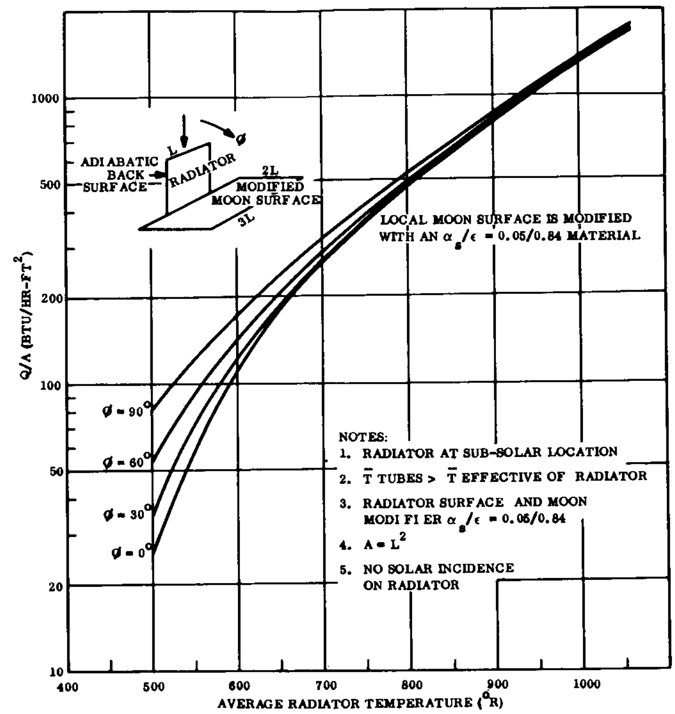


Figure 11-10. Mean Heat Rejection Capability of Flat Plate Radiator Normal to Moon as a Function of Effective Mean Temperature and Sun Angle

11.4.2.4 Radiator Selection

The heat rejection capabilities for the various radiator designs are summarized for the lunar noon condition in Figure 11-11. The best heat rejector is then the horizontal radiator with $\alpha s/\epsilon = 0.085/0.84$ for effective temperatures 520°R and below (T_s for this case = 370°R). Above 520°R effective radiator temperature, the best dissipator is the vertical fin with $\alpha s/\epsilon = 0.06/0.06$ rug. Note that the Q/A for this fin is an average for the configuration as shown in Figure 11-10. There is a variable distribution in the vertical direction, the point at the knee of the fin at the rug has the greatest heat rejection per unit area (smallest lunar view factor) while the point furthest from the lunar surface has the least heat rejection.

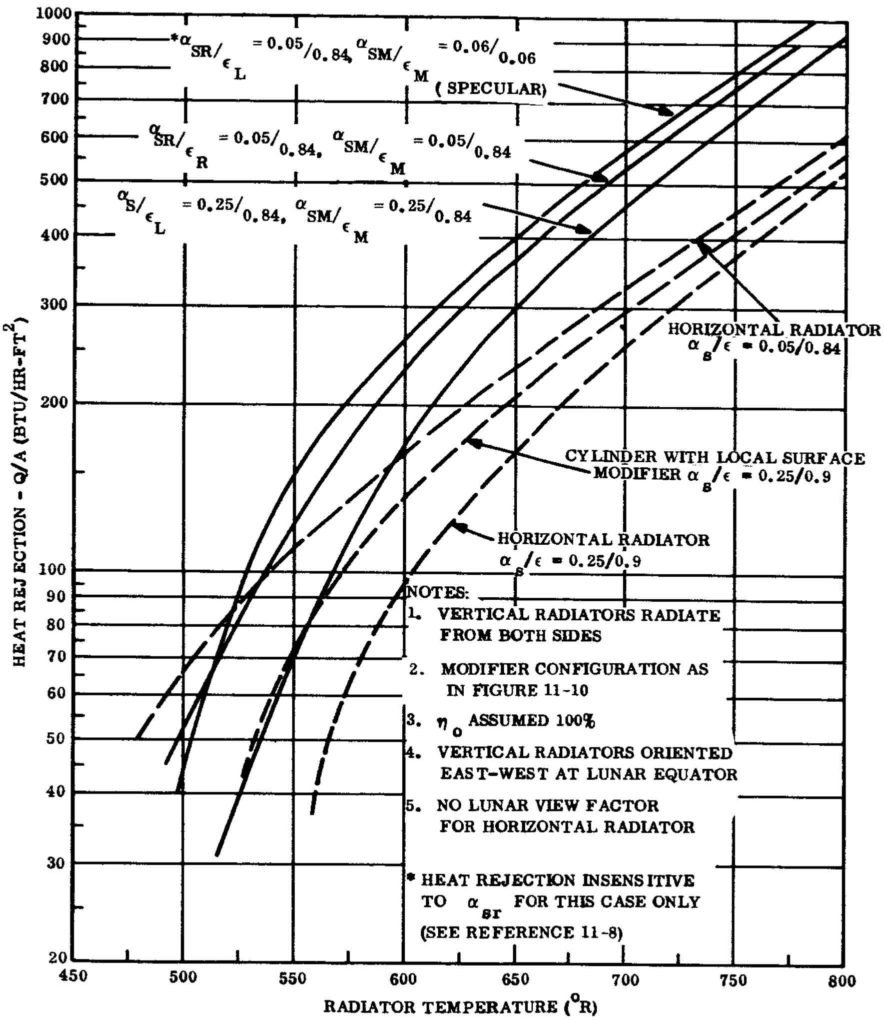
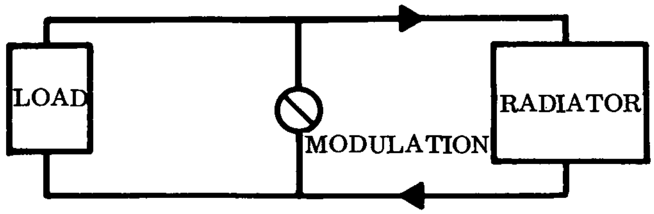


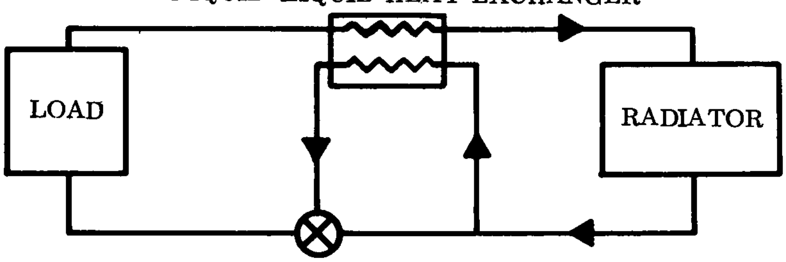
Figure 11-11. Comparison of Radiator Performance

11.4.3 THERMAL CONTROL

A second thermal control problem resulting from the lunar environment extremes is caused by the wide fluctuations in radiator sink temperature. Therefore, a radiator sized to allow adequate heat rejection during the lunar day may drop to an excessively low temperature at night. Figure 11-12 illustrates two radiator thermal control concepts.



A. RADIATOR BYPASS CONFIGURATION



B. REGENERATOR CONFIGURATION

Figure 11-12. Radiator Thermal Control Concepts

The first is a bypass arrangement in which the thermal transport fluid is diverted around the radiator to maintain a constant temperature for the total flow. Since the radiator inlet temperature is constant, a greater temperature drop exists across the radiator under cold conditions when part of the fluid bypasses the radiator. Lower radiation discharge temperatures can either cause the liquid to freeze or may create high viscosities. On the moon, assuming the radiator is sized for hot environmental conditions, when elevated sink temperatures prevail, an expression for the minimum discharge temperature during lunar night ($\approx 0^{\circ}$ R sink temperature) is presented below for the bypass configuration.

The heat rejection from a radiator surface A can be written as:

$$Q_R = \dot{w}\, c_p\, \Delta T = \epsilon\, A\, \varkappa\, \sigma \left(\overline{T}^4 - T_s^4 \right) \tag{11-1}$$

Where:

- $\dot{w}\, c_p$ = thermal capacity of fluid
- ϵ = tube and fin emittance
- $\varkappa$ = fin effectiveness
- $\overline{T}$ = mean radiator tube surface temperature
- T_s = sink temperature, maximum
- ΔT = temperature drop across radiator
- σ = Stefan Boltzmann constant

If T_1 and T_2 are the inlet and outlet temperatures of the radiator, an approximate expression for $\overline{T}$ is $T_s + \dfrac{T_1 - T_2}{\ln\left(\dfrac{T_1 - T_s}{T_2 - T_s}\right)}$. When $T_s = 0^\circ R$, $\overline{T}^4$ may be exactly given by $\dfrac{3\, T_1^3\, T_2^3}{T_1^2 + T_1 T_2 + T_2^2}$.

Since $T_1^2 + T_2^2 \approx 2 T_1 T_2$, $\overline{T}^4 \approx T_1^2 T_2^2$ for $T_s = 0$. Inserting both $\overline{T}$ expressions in Equation (11-1) and combining the resulting equations for Q_R (assumed constant for night and day) yields T_{2min} during lunar night for the bypass control scheme.

$$T_{2min} = \left[\frac{\left(T_s + \dfrac{T_1 - T_2}{\ln \dfrac{T_1 - T_s}{T_2 - T_s}} \right)^4}{T_1^2} - T_s^4 \right]^{1/2} \tag{11-2}$$

The second thermal control concept involves the use of a liquid-liquid heat exchanger to allow the radiator inlet temperature to be depressed without adversely affecting the load temperature conditions. Regeneration is modulated by means of a control valve. With this arrangement ΔT across the radiator remains constant since liquid flow stays at full capacity. The regenerative concept, although it creates a higher power requirement than the the bypass method, usually can accommodate greater variations in both vehicle payload and dissipation thermal environment. The T_{2min} for the regenerative concept, using $\overline{T}$ expressions indicated above is given by:

$$T_{2min} = \left[\frac{\Delta T^2}{4} + \sqrt{T_s + \frac{\Delta T}{\ln\left(\frac{T_1 - T_s}{T_1 - T_s - \Delta T}\right)}} \right]^{1/2} - \frac{\Delta T}{2} \qquad (11\text{-}3)$$

To show the difference in T_{2min} for both concepts, the following numerical example is presented: If $T_1 = 800^{o}R$, $T_2 = 560^{o}R$, and $T_s = 525^{o}R$, all at a point in time when thermal environmental conditions are the greatest, the T_{2min} (lunar night) for the bypass and regenerative concepts are $403^{o}R$ and $542^{o}R$, respectively.

A reduction in the regeneration needs may be obtained by totally bypassing radiator tubes, therefore, decreasing the radiator effectiveness. Unless conduction paths are adequate, non-operating tubes may reach temperatures low enough to freeze the liquid upon being allowed to return into those tubes.

11.4.4 DISCUSSION

The radiator designs were compared solely on the basis of heat rejection capability per unit area. Another consideration that should be made is relative weight of the different designs. The horizontal radiator, if assumed similar to other low temperature radiators used for the orbiting space station, weigh 0.47 pound/square foot. However, the surface modification to obtain $\alpha s/e = 0.05/0.84$ includes a layer of quartz. This is estimated to weight an additional 0.46 pound/square foot (based on a 1/32-inch layer) for a total weight of 0.93 pound/square foot. This increase suggests that fin-tube configurations would be revised from the radiator configuration to optimize total radiator weight. The vertical radiator fin is not burdened with a coating requirement; however, it requires a lunar surface modifier to operate effectively. This modifier might be several layers of aluminized or other metallized mylar to provide the necessary surface characteristics. It is estimated that the total weight of vertical fin and rug would make the vertical fin radiator weigh approximately the same per square foot as the horizontal radiator.

The cylindrical radiator affords the possibility of integration with cabin structure. This can decrease the radiator weight, making this a likely choice as a design for some lunar applications. It retains the requirement for a surface rug to be effective.

The radiators that are used in the rest of the shelter study are the 0.05/0.84 horizontal radiator for effective temperatures below $520^{o}R$. and the vertical fin and lunar rug for temperatures greater than $520^{o}R$. The specific weight used is 0.93 pound/square foot. The reasons for this choice of radiators are that they afford the maximum Q/A potential (see Figure 11-11) and sufficient information is not available to make a weight tradeoff in this study with the cylindrical concept.

11.5 HEAT PUMP

One of the major problems associated with the operation of a manned vehicle on the lunar surface lies in the fact that both the power system and the Life Support system radiators must operate during the lunar day. The potential reduction in radiating capacity is especially severe in the cabin cooling and equipment cooling low-temperature radiators from their capacity during the lunar night. These radiators operate at an average fluid temperature of approximately $40^{o}F$. Three approaches to the solution of this problem are possible: (a) optimization of radiator geometry; (b) optimized radiator geometry combined with local lunar surface modification; and (c) optimized radiator geometry combined with local lunar surface modification combined with a heat pump. Radiator configurations with and without local surface modification are discussed in Section 11.4. A limited look at some of the characteristics of heat pumps and their attendant advantages and disadvantages is in order.

By a heat pump specific reference is made to a vapor cooling cycle. Figures 11-13 and 11-14 show the ideal and actual vapor cycle plotted on the basis of: (a) temperature-entropy diagram; and (b) pressure-enthalpy diagram. The basic process is as follows:

a. Isothermal evaporation of the working fluid to a superheated vapor.

b. Compression of the saturated vapor (ideal cycle uses an isentropic compression).

c. Almost isothermal condensation of vapor in a radiator.

d. Almost isenthalpic expansion of the vapor to give a two-phase mixture entering into the evaporator.

The evaporator heat load per pound of fluid is simply:

$$Q_L = h_1 - h_2$$

The total compressor work is:

$$W_t = \frac{h_2 - h_1}{\eta_c \, \eta_m}$$

where:

η_c = compression efficiency (0.65)

η_m = motor efficiency (0.8)

The radiator heat load is:

$$Q_R = \frac{(h_1 - h_4)\,\eta_c \, \eta_m + (h_2 - h_1)}{\eta_c \, \eta_m}$$

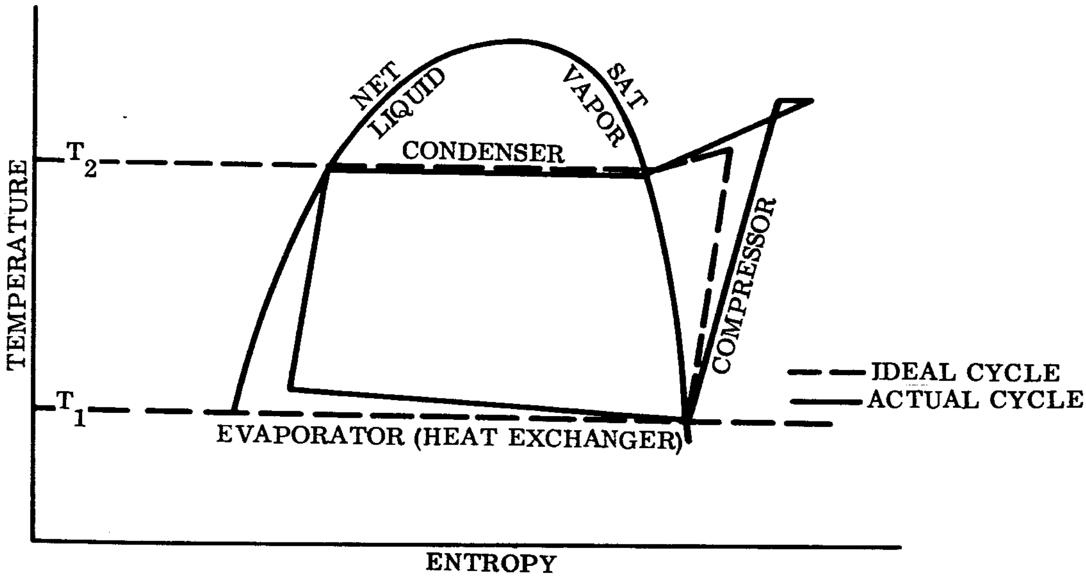


Figure 11-13. Temperature - Entropy Diagram

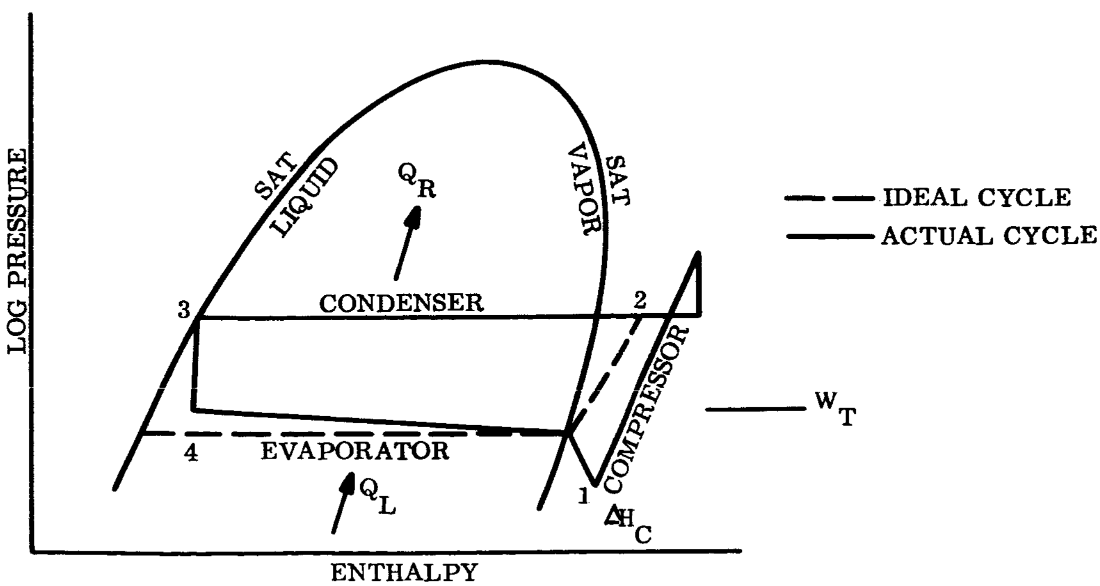


Figure 11-14. Pressure-Enthalpy Diagram

Finally the ratios of total compressor work to the refrigeration heat load and the ratio of the radiator heat load to the refrigeration heat load are:

$$\frac{W_T}{Q_L} = \frac{h_2 - h_1}{\eta_c \eta_m (h_i - b_4)} \quad \text{and} \quad \frac{Q_R}{Q_L} = \frac{(h_1 - h_4)\, \eta_c \eta_m + (h_2 - h_1)}{\eta_c \eta_m (h_1 - h_4)}$$

These two expressions are plotted for Freon 11 as a working fluid in Figures 11-15 and 11-16.

The choice of the weight optimized vapor cycle parameters depends largely on two system parameters: (a) the radiator area weight penalty; and (b) the pump power work penalty. The development of the weight optimized parameters is based on a study reported in Reference (11-9). A redevelopment of the basic equation is given below:

The compression work is: $$W_T = \frac{Q_L (h_2 - h_1)}{(h_1 - h_4)}$$

The radiator heat load is: $$Q_R = \frac{Q_2 (h_2 - h_3)}{(h_1 - h_4)}$$

Finally the system weight (W_s) can be simplified and represented by:

$$W_s = PW_t + \frac{R\, Q_R}{\epsilon\, \eta_o\, \sigma\, (T_R^4 - T_s^4)}$$

where

P = compression work weight penalty

R = radiator weight per unit area

T_R = average radiator temperature, assumed = T_2

T_s = effective sink temperature

η_o = overall radiator efficiency

σ = Boltzmann's constant

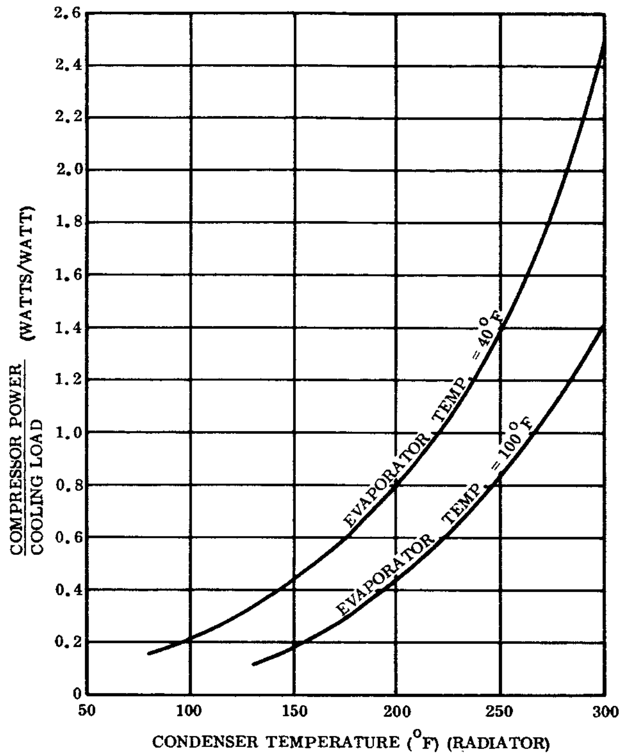


Figure 11-15. Heat Pump Condenser Power Versus Condenser Temperature

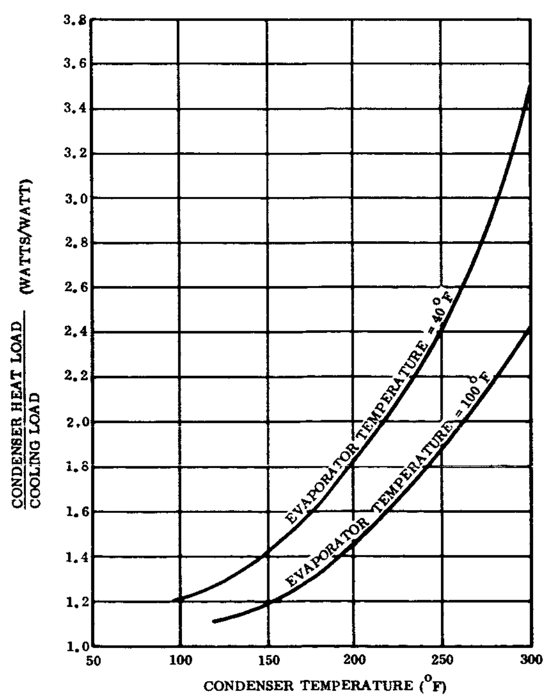


Figure 11-16. Condenser Heat Load Versus Condenser Temperature

Omitted from this expression are the weights of the compressor and the connecting piping with its associated pressure loss. This will tend to predict a higher optimum condenser temperature than would otherwise be predicted. Offsetting this fact is the assumption made in setting the effective radiator temperature equal to the condensing temperature which tends to predict a lower optimum condensing temperature. The system coefficient of performance (C_p) is by definition:

$$C_p = \frac{Q_L}{W_T}$$

Using this definition the system weight can be rewritten as:

$$\frac{W_S}{Q_L} = \frac{P}{C_p} + \frac{R\left(1 + \frac{1}{C_p}\right)}{\epsilon\,\eta_o\,\sigma\left(T_2^{\,4} - T_s^{\,4}\right)}$$

Differentiating the above expression with respect to the condenser temperature, and setting the resultant expression equal to zero results in the following expression:

$$\epsilon\,\eta_o\,\sigma\left(T_2^{\,4} - T_s^{\,4}\right) = \frac{R}{2P} + \sqrt{\left(\frac{R}{2P}\right)^2 - 4\,\epsilon\,\eta_o\,\sigma\,T_2^{\,3}\,\frac{R}{P}\;P\;\frac{\left(C_p^{\,2} + C_p\right)}{d\,C_p/d\,T_2}}$$

Noting that commonly:

$$\frac{4\,\epsilon\,\eta_o\,\sigma\,T^3\,\frac{R}{P}\left(C_p^{\,2} + C_p\right)}{d\,C_p/d\,T_2} \gg \frac{R}{2P} \text{ and } \left(\frac{R}{2P}\right)^2$$

The condition for a minimum weight heat pump can be further simplified as:

$$\epsilon\,\eta_o\,\sigma\left(T_2^{\,4} - T_s^{\,4}\right) = \sqrt{4\,\epsilon\,\eta_o\,\sigma\,T_2^{\,3}\,\frac{R}{P}\left[\frac{\left(C_p^{\,2} + C_p\right)}{-\,d\,C_p/d\,T_2}\right]}$$

Figure 11-17 shows a plot of the computed coefficient of performance versus condenser temperature. A plot of the optimum condenser temperature (T_2) versus the effective sink temperature is given in Figure 11-18 for Freon 11 as the working fluid, with several combinations of evaporator temperature, R and P. Figure 11-19 shows the weight penalty associated with heat pump use on the moon.

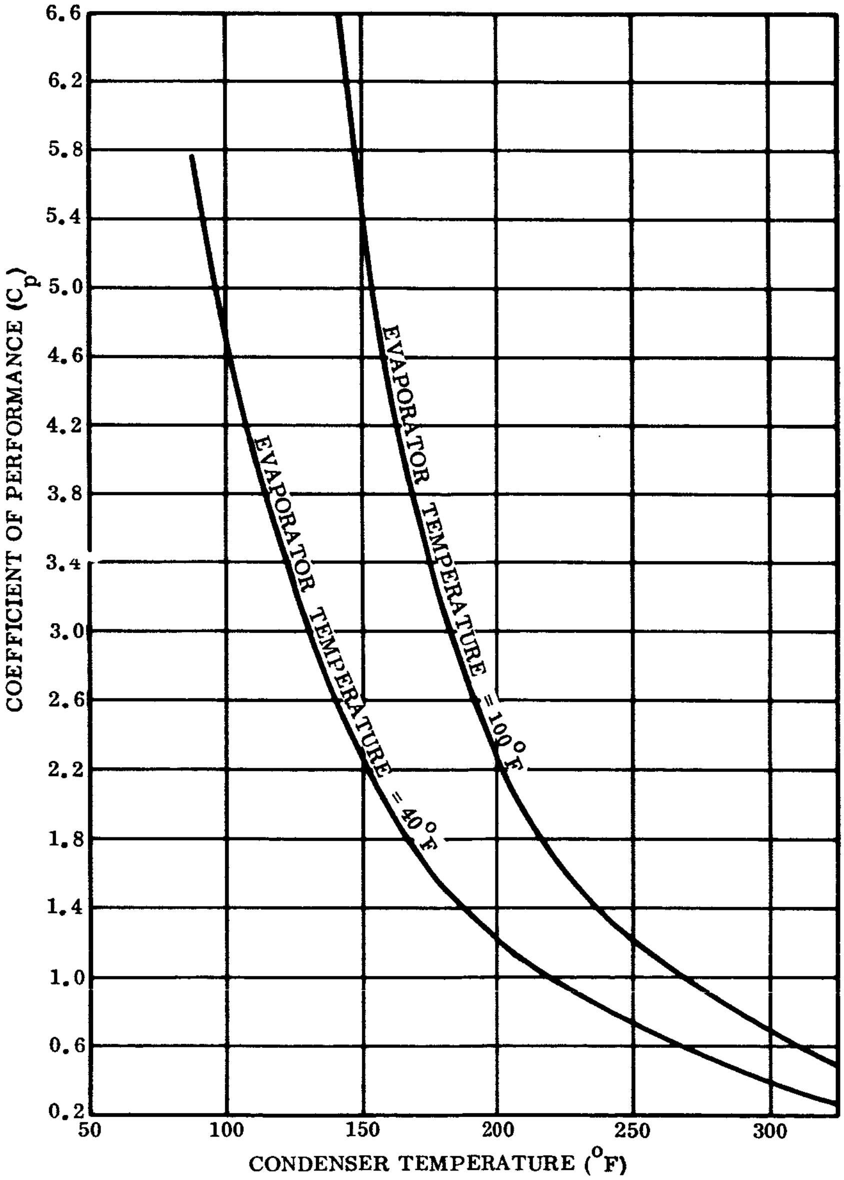


Figure 11-17. Vapor Compression Coefficient of Performance Versus Condenser Temperature

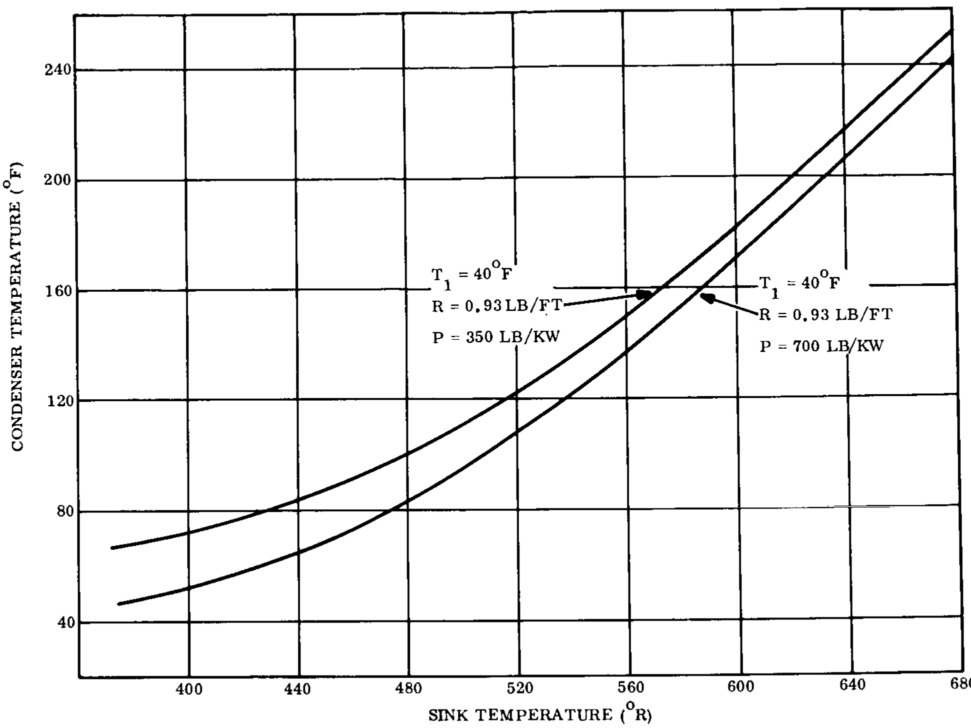


Figure 11-18. Heat Pump Optimum Condenser Temperature Versus Effective Sink Temperature

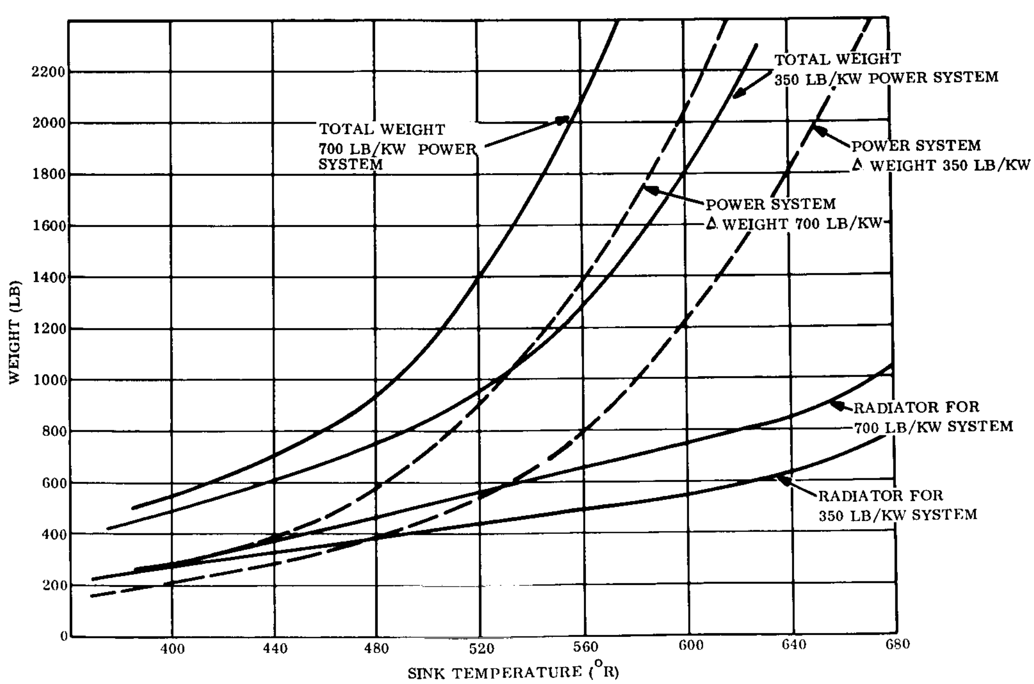


Figure 11-19. Heat Pump System Weight, 18000 Btu/Hr Capacity

11.6 ABSORPTION REFRIGERATION CONSIDERATIONS

The analysis of absorption refrigeration systems, design of equipment, and integration with the MORL vehicle have all been fully explored for the orbiting space station in Section 7. Rejection of low temperature heat is again a possible problem area for the lunar shelter. Use of absorption refrigeration is a possible asset, waste power cycle heat being used to operate an absorption refrigeration cycle which raises low temperature waste heat to a higher temperature. The orbiting design discussed in Section 7 will be used in the lunar shelter, the principal modification is the radiator characteristics.

11.6.1 HEAT ENERGY TO BE REJECTED

For the shelter life support analysis, a complete cabin heat balance is made in Section 11.7. Table 11-6 summarizes process heat, heat used, and heat rejected. The net cabin energy rejected, including electronic equipment (exclusive of power system components) is 17,470 Btu/hr for the Mercury Rankine cycle and 18,568 Btu/hr for the Brayton cycle. For the purposes of this study the air and liquid loops are combined into one heat load. The inlet temperature to the cabin cooling loops is 40°F for both. The absorption refrigeration equipment, for convenience, is sized for 18,000 Btu/hr load as in the orbiting space station case.

11.6.2 CYCLE CONSIDERATIONS

The orbiting space station (MORL) application in Section 7 considered a cycle with 200°F generator temperature, 40°F evaporator temperature, and 100°F absorber and condenser temperatures. The evaporator temperature remains the same with the same life support cooling loop temperatures. The other temperatures follow as optimum for the lithium bromide-water working fluid and the 40°F evaporator temperature. As in the orbiting case, there is not sufficient waste Brayton Cycle heat available to operate more than one refrigeration stage. Higher rejection temperatures would be possible only for a reduced refrigeration load.

The Mercury Rankine cycle has 35.3-kw waste heat at approximately 600°F in the integrated system. There is then, additional energy for powering the refrigeration system beyond what the Brayton cycle could provide. Realizing the advantages of higher rejection temperatures for the LSS radiators, a dual cycle is considered for use with the Mercury Rankine power system. Table 7-3 presents an idealized dual cycle which can be applied to the lunar shelter.

11.6.3 EQUIPMENT DESIGN

The single-stage design can be the same as described in Section 7 and Appendix B. Table 7-6 summarizes the weight and pump power requirements (weight is 43 pounds and pump power is 69 watts). Further research is required to determine the result of using lunar gravity for fluid/gas separation and the effect of the gravity on heat transfer coefficients vis-a-vis spinning the whole system as in the orbiting case. Details of the lunar radiators are presented in Table 11-1. The heat rejection capabilities were taken from Figure 11-11 representing the most severe thermal environment, lunar noon.

The dual-stage design is similar to using two single-stage systems and then adding some additional inter-unit integration (such as low temperature loop condenser and absorber plumbing to the high temperature loop evaporator). Figure 7-4 is a schematic of the equipment requirements. The weight and power for the dual system can be approximated by calculation on a heat input basis. COP of single system is about 0.75 and the dual system is about 0.32. A ratio for approximating weight and power is $\frac{0.75}{0.32} = 2.33$. The dual system weight is then 100 pounds and the power required for pumping 161 watts. Details of the lunar radiators are presented in Table 11-2. The heat rejection capabilities were again taken from Figure 11-11.

TABLE 11-1. ABSORPTION REFRIGERATION
BRAYTON POWER SYSTEM RADIATOR PARAMETERS

		CONDENSER	ABSORBER	LOW TEMPERATURE RADIATOR
T_{in}	°F	95	90	50
T_{out}	°F	85	70	40
T_{eff}	°F	90	80	43
η_{fin}		0.90	0.90	0.90
ϵ		0.84	0.84	0.84
Q/A	Btu/hr-ft^2	135	114.2	63
w	lb/ft^2	0.93	0.93	0.93
Q	Btu	19,200	22,800	18,000
A	ft^2	142	200	286
W	lb	132	186	266

TABLE 11-2. DUAL LiBr ABSORPTION REFRIGERATION
AND
MERCURY RANKINE POWER SYSTEM RADIATOR PARAMETERS

		CONDENSER	ABSORBER	LOW TEMPERATURE RADIATOR SYSTEM
T_{in}	°F	200°	180°	50
T_{out}	°F	185°	170°	40
T_{eff}	°F	190°	175°	43
η_{fin}		0.85	0.85	0.9
ϵ		0.84	0.84	0.84
Q/A	Btu/hr-ft^2	340	302	63
w	lb/ft^2	0.93	0.93	0.93
Q	Btu	22,500	51,800	18,000
A	ft^2	66.2	172 ft^2	286
W	lb	61.5	160 lb	266

11.7 TWO-MAN LUNAR SHELTER LIFE SUPPORT SYSTEM

11.7.1 PROCESS POWER REQUIREMENTS

The Life Support requirments for a two-man lunar shelter are obtained by scaling down, where applicable, the requirements calculated for the six-man Orbiting Space Station. The subsystems considered are again:

a. Oxygen Recovery
b. Urine water recovery
c. Waste water recovery
d. Solid Waste management
e. Food preparation
f. Cabin environmental control

The design parameters remain basically the same except for the cabin environmental control due to the changes in the external and internal configuration of the lunar shelter as compared with the orbiting space station. In addition, since the power requirements for the cabin environmental control depends largely on the heat load contributed by the first five life support subsystems, the cabin environmental control will be analyzed separately.

11.7.1.1 Oxygen Recovery

A description of the balanced process of recovery of oxygen from carbon dioxide is given in Appendix A. The recovery is accomplished in four distinct processes:

a. Concentration of CO_2
b. Breakdown of CO_2 into CH_4 and H_2O by the Sabatier method
c. Pyrolysis of CH_4 into carbon and hydrogen
d. Electrolytic disassociation of H_2O into hydrogen and oxygen

These processes are basically unaffected by the changed environment for a two-man lunar shelter. It is assumed that the equipment is sheltered and insulated from the extremes of temperatures of the lunar long day-night cycle. The mechanical design is then relatively simplified due to the absence of some of the zero-g environment problems such as the gas-liquid phase separation required for the vapor condensers and the electrolysis unit.

The endothermic power requirements can be scaled down in all processes except the carbon dioxide concentration unit. In this unit a considerable amount of power is expended to heat both the desiccant and CO_2 adsorbing canisters during the desorbing process. The reduced requirements for the two man shelter are then:

Air bypass for CO_2 concentration

$$20 \text{ cfm} \times 2/6 = 6.67 \text{ cfm or } 0.267 \text{ lb/min}$$

Amount of moisture to be removed in the desiccant canister

$$6.67 \text{ ft}^3/\text{min} \times \frac{1 \text{ lb } H_2O}{1703.2 \text{ cu ft}} \times 60 \text{ min/hr} = 0.235 \text{ lb/hr}$$

Heat of reaction (cooling requirement)

$$0.235 \text{ lb/hr} \times 1800 \text{ Btu/lb} = 423 \text{ Btu/hr}$$

Additional cooling of dry air from 50°F to 40°F

$$0.267 \text{ lb/min} \times 10°\text{F} \times 0.232 \text{ Btu/lb} \times 60 \text{ min/hr} = 37 \text{ Btu/hr}$$

Heat of reaction of CO_2 adsorbing sieve

$$0.6 \text{ lb/hr } CO_2 \times 2/6 \times 300 \text{ Btu/lb} = 60 \text{ Btu/hr}$$

Blower = 1/3 of 40 watts for 6 men = 14 watts

Heat required to bring the desiccant canister from approximately 60°F to 250°F minimum for desorbing

$$Q_{air} = 0.267 \text{ lb/min} \times 60 \text{ min/hr} \times 0.23 \text{ Btu/lb °F} \times 180 \text{ °F} = 660 \text{ Btu/hr}$$

$$Q_{moisture} = 0.235 \text{ lb water/hr} \times 1800 \text{ Btu/lb water} = 423 \text{ Btu/hr}$$

$$Q_{canister} = 10 \text{ lb} \times 0.25 \text{ Btu/lb °F} \times 180 \text{ °F} = \underline{450 \text{ Btu/hr}}$$

$$1,533 \text{ Btu/hr}$$

Heat required to desorb CO_2 from canister:

$$Q_{canister} = 20 \text{ lb} \times 0.25 \text{ Btu/lb °F} \times 200\text{°F} = 1000 \text{ Btu/hr}$$

$$Q_{desorbtion} = 0.2 \text{ lb } CO_2\text{/hr} \times 300 \text{ Btu/lb } CO_2 = \underline{60 \text{ Btu/hr}}$$

$$1060 \text{ Btu/hr}$$

Process heat from Sabatier reactor:

$$Q = 1610 \text{ Btu/lb } CO_2 \times 0.2 \text{ lb } CO_2\text{/hr} = 320 \text{ Btu/hr}$$

Energy requirement for water electrolysis:

$$Q = 3,365 \text{ Btu/hr} \times 2/6 = 1,122 \text{ Btu/hr}$$

Energy requirement for pyrolyzation of methane:

$$Q = 440 \text{ Btu/hr} \times 2/6 = 146 \text{ Btu/hr}$$

11.7.1.2 Urine Water Recovery

The power requirements for both the evaporation and condensation process are:

$$Q = 830 \text{ Btu/hr} \times 2/6 = 276 \text{ Btu/hr}$$

For the pyrolysis unit, $Q = 60 \text{ watts} \times 2/6 = 20 \text{ watts}$

11.7.1.3 Waste Water

The waste water subsystem utilizes the urine water pyrolysis unit. The heat of vaporization and condensation are:

$Q = 1500$ Btu/hr x2/6 = 500 Btu/hr

11.7.1.4 Solid Waste

The power requirement for the solid waste management is:

For the evaporator and condenser Q = 1000 Btu/hr x 2/6 =	333 Btu/hr
For sludge dehydration Q = 50 Btu/hr x 2/6 =	17 Btu/hr
For waste disinfection Q = 40 Btu/hr x 2/6 =	13 Btu/hr

11.7.1.5 Food Preparation

The total power requirement for the six-man station was 228 Btu/hr. Reduced to two men, the requirement is 228 x 2/6 = 76 Btu/hr

11.7.1.6 Cabin Environmental Control

The power requirements for the cabin environmental control depend largely on the heat load contributed by the five life support subsystems thus far analyzed and their support equipment. For this reason, the cabin environmental control subsystem will be treated separately after the overall system design and power requirements have been established.

11.7.2 NON-INTEGRATED LIFE SUPPORT SYSTEM DESIGN

The non-integrated life support system derives all of its energy from the lunar shelter electric power generating equipment. It is assumed that the process equipment is located within the shelter, mostly because of necessity or safety reasons. Storage tanks and other supplies could be located outside the shelter. On this basis then, the support equipment power can be scaled down directly from the six-man orbiting station. Similarly, the two cooling systems, one for the oxygen recovery and the other for the remaining life support

TABLE 11-3. NON-INTEGRATED LIFE SUPPORT AND ENVIRONMENTAL CONTROL SYSTEM PARAMETERS

		Weight (lbs)	Process Power (watts)	Support Power (watts)	Life Support System		Heat Out	
					Exo-Thermic	Endo-Thermic	Liquid (Btu/hr)	Air (Btu/hr)
Oxygen Recovery	CO_2 Concentrator	60	760	14			2593	48
	Electrolysis	25	330			915	215	
	Pyrolyzation	10	43			147	348	21
	CO_2 Pump	5		33	320	64		
	Accumulator, Reactor, Condenser	15						
	System Cooling	40		23			78	
	Heat Leakage			30				102
	Miscellaneous	12		7				24
Food and Waste Management	Urine Water Recovery	30	101	7		68	276	24
	Solid Waste Treatment	20	107	7			365	24
	Waste Water Recovery	25	147	7			500	24
	Food Management	19	23					78
	Heat Leakage			60				205
	Cooling System	16		20			68	
	Miscellaneous	5						
Cabin Environmental Control	External Heat							68
	Fan	4		46				157
	Metabolic Heat							927
	Ducting	10						
	Heat Exchanger	6						
	Filters and Housings	2						
	Glycol Pump/ Fluid	4		25			(85)	(to air loop)
	Lamp	2		25				85
	Catalytic Burner	2		50				171
	Radiator	40						
Totals		352	1511	354	320	1194	4443	2043
Operations Heat Load	Liquid Radiator	100					8555	
	Air Load Radiator	39						2140
		491						

subsystems, can also be scaled down since their function is strictly that of transporting the life support waste heat to a suitable space radiator or heat exchanger. Weights have been estimated according to the reduced volume of the equipment. In Table 11-3 the process power requirements calculated in Paragraphs 11.7.1.1 through 11.7.1.5 have been listed separately from the support equipment power for ease of correlation. The energy balance for the non-integrated systems is shown in Table 11-4. A system schematic is shown in Figure 11-20.

11.7.3 INTEGRATED LIFE SUPPORT SYSTEM

The integrated power systems weights and power requirements are shown in Table 11-5. The modifications for integration with the power systems are to include the additional heat requirements, losses, and pumping power given below. Figure 11-21 is a system schematic.

11.7.3.1 High Temperature Loop

Waste heat from the power generating unit (600°F) is used to heat the CO_2 desorber and the desiccant desorbing canisters.

High temperature loop heat leakages:

Canister heat leakage = 30 watts x 3.41 Btu/hr-watt = 103 Btu/hr

$$\text{Tubing losses} = 1060\ \text{Btu/hr} \times \frac{7\ \text{ft for 2 men}}{40\ \text{ft for 6 men}} = 186\ \text{Btu/hr}$$

289 Btu/hr

TABLE 11-4. NON-INTEGRATED LIFE SUPPORT SYSTEM ENERGY BALANCE

HEAT INPUT		HEAT REMOVED	
Metabolic	927 Btu/hr	Air Conditioning	4183
Cabin Leakage	68	Liquid Cooling	12,998
Exothermic	320	Process Endothermic	1194
Electrical	17,065		
	18,380		18,375

11.7.3.2 Low-Temperature Loop

A 155°F source (low-temperature loop) is used to heat all of the remaining processes with the exception of sterilization and disinfection, which derive the small quantity of energy needed directly from the Sabatier exothermic reaction.

Low-temperature loop heat leakages:

Canister heat leakage = 60 watts x 3.41 = 205 Btu/hr

Tubing = 225 Btu/hr x $\frac{15 \text{ ft} - 2 \text{ men}}{60 \text{ ft} - 6 \text{ men}}$ = 57

262 Btu/hr

11.7.3.3 High-Temperature Loop Therminol FR-1 Flow

Mass flow rate = 578 lb/hr

Equivalent length of tubing = 41 ft

Total pressure drop = 5.3 psi

Pumping power = 5 watts

11.7.3.4 Integration with Brayton Cycle

The Brayton Cycle cannot provide a 600°F waste heat source as efficiently as it can provide a 400°F waste heat source. This section compares the effect on system weight and power requirements due to the lowering of the heat source from 600°F to 400°F.

TABLE 11-5. LIFE SUPPORT AND ENVIRONMENTAL CONTROL INTEGRATED SYSTEM PARAMETERS

	Integrated System	Weight (lbs) Rankine	Weight (lbs) Brayton	Process Power (watts)	Support Power (watts)	Rankine Heat In	Rankine Heat Out (Btu/hr) Liquid	Rankine Heat Out (Btu/hr) Air	Brayton Heat In	Brayton Heat Out (Btu/hr) Liquid	Brayton Heat Out (Btu/hr) Air	Exo-Thermic Btu/hr	Endo-Thermic Btu/hr
Oxygen Recovery	CO_2 Concentrator	60	103		14	2882	2593	337	3922	3633	337		
	Electrolysis	25	25	330			215			215			
	Pyrolization	10	10	43									915
	CO_2 Pump, Accumulator,	5	5		33		348	21		348	21	320	147
	Condenser, Reactor	15	15										64
	System Cooling	40	47		23		78						
	Heat Leakage				30			102			102		
	Heat Exchanger and Miscellaneous	26	26		7			24			24		
Food and Waste Management	Urine Water Recovery	30	30	20	7	276	276	24	276	276	24		68
	Solid Waste Treatment	20	20		7	365	365	24	365	365	24		
	Waste Water Recovery	25	25		7	500	500	24	500	500	24		
	Food Management	19	19		6	58		78	58		78		
	Heat Leakage				60			205			205		
	Cooling System	16	16		20		68			68			
	Miscellaneous	5	5										
	Heat Exchanger	10	10										
Cabin Environmental Control	External Heat					(68)		68	(68)		68		
	Fan	4	4		46			157			157		
	Metabolic Heat					(927)		927	(927)		927		
	Ducting	10	10										
	Heat Exchanger	6	6										
	Filters and Housing	2	2										
	Glycol Pump and Fluid	4	4		25		(85)	To air Loop 85		(85)	To air Loop 85		
	Lamp	2	2		25								
	Radiator	40	41										
	Catalytic Burner	2	2		0/17	171		171	171		229		
	TOTALS	376	427	398	310/327	4252	4443	2332	5292	5405	2390	320	1194
Operations Heat Loads	Liquid Radiator	100	100				8555			8555			
	Air Radiator	39	39					2140			2140		
		515	566										

a. Desiccant Desorbing Canister

Weight

	600°F System	400°F System
Sieve	2.25	5.25
Container	4.0	7.0
Equipment	3.75	3.75
	10 lb	16 lb

Desorption Heat Input

Atmosphere	660	660
Sieve	450	720
Desorption	423	423
	1533 Btu/hr	1803 Btu/hr

b. CO_2 Desorbing

Weight

	600°F System	400°F System
Sieve	4.6	10.7
Container	7.0	16.3
Equipment	8.4	8.4
	20.0	35.4

Process Heat Input

	1060 Btu/hr	1830 Btu/hr

c. Total Weight for 400°F System

Desiccant Desorbing	= 16 x 2 =	32 lb
CO_2 Desorption	= 35.4 x 2 =	70.8 lb
		102.8 lb

d. Total Power Requirements for a 400°F System

Desiccant Desorption	= 1803 Btu/hr
CO_2 Desorption	= 1830 Btu/hr
	3633 Btu/hr

Total weight increase = 102.8 - 60 = 42.8 lb

Total power increase = 3633 - 2593 = 1040 Btu/hr

The energy balances for both integrated power systems are listed in Table 11-6.

TABLE 11-6. INTEGRATED SYSTEMS ENERGY BALANCE

MERCURY RANKINE SYSTEM			
HEAT INPUT		HEAT REMOVED	
Metabolic	927 Btu/hr		
Cabin Leakage	68	Endothermic	1194 Btu/hr
Exothermic	320	Liquid Cooling	12998
Electric Power (LSS)	2400	Air Conditioning	4472
Electric Power (Station)	10695		18664 Btu/hr
Thermal Power	4252		
	18662 Btu/hr		
BRAYTON SYSTEM			
HEAT INPUT		HEAT REMOVED	
Metabolic	927 Btu/hr		
Cabin Leakage	68	Endothermic	1194 Btu/hr
Exothermic	320	Liquid Cooling	14038
Electric Power (LSS)	2458	Air Conditioning	4530
Electric Power (Station)	10695		19,762 Btu/hr
Thermal Power	5292		
	19,757 Btu/hr		

11.7.4 CABIN COOLING

Shelter design parameters are:

a. Free Volume = 200 ft^3

b. Cabin Pressure = 7 psia

c. Air Composition = 50% O_2 - 50% N_2

d. Relative Humidity = 50% nominal

e. Cabin Temperature = 75°F ± 5°F

f. Air Flow = 330 cfm

g. Humidity Control = 6.67 cfm bypass to CO_2 removal

11.7.4.1 Cabin Heat Load

Leakage due to cabin walls:

Maximum in (during lunar day) = 68 Btu/hr
Maximum out (during lunar night) = 385 Btu/hr

[Worst case in lunar day 68 Btu/hr input (from Reference 11-2)]

To provide heat make-up during the lunar night, the coolant flow between the ECS heat exchanger and the radiator would be regulated.

a. Cabin Air Heat Load (minus ECS experiments) = 1847 Btu/hr

b. Experiments - Electronic Load

(1) Power source output is given as 5 kw

Non-integrated = 1.719
Air Conditioning = 0.146
1.865 kw

Power available for experiments = 5 - 1.87 = 3.13 kw

(2) Heat to be removed by cabin air = 3.13 x 3414 x 0.2 = 2144 Btu/hr

(3) Heat dumped by air conditioning equipment = 0.149 x 3414 = 509 Btu/hr

Total heat load = 4568 Btu/hr including the lunar day heat leakage through the cabin walls.

11.7.4.2 Cabin Heat Removal

A counter-flow heat exchanger is used to remove the cabin heat from the air and transport it to the space radiator. The liquid transport fluid used is a glycol solution (30% glycol, 70% water). The heat exchanger coil is made of 3/8-inch OD finned tubing with a sectional flow area of 0.000525 square foot and an outside surface area of 0.775 square foot/foot.

Mass of air $= 200\ ft^3 \times 0.07\ lb/ft^3 \times 7/14.7 = 6.7\ lb/volume$

Cabin air turnover $= \dfrac{330\ ft^3/min}{200\ ft^3/volume} \times 60\ min/hr = 99\ volume/hr$

$$\text{Air } \Delta T = \frac{Q}{WC_p} = \frac{4568\ Btu/hr}{6.7\ lb/vol \times 99\ vol/hr \times 0.232\ Btu/lb\ ^\circ F} = 29.7^\circ F$$

The temperature at the exit of the heat exchanger is taken as 55°F which is the dew point for 50% relative humidity.

The entrance temperature is then $55^\circ F + 29.7^\circ F = 84.7^\circ F$

Glycol $\Delta T = 10^\circ F$ (lowest temperature assumed = 40°F (Radiator)
(highest temperature assumed = 50°F)

Glycol flow $= W = Q/C_p\ \Delta T = 532\ lb/hr$

Glycol film coefficient of heat transfer

$$h = \frac{0.023\ C_p G}{(N_{pr})\ 273\ (N_{re}).2} = 480\ Btu/hr\text{-}ft^2 - {}^\circ F$$

Air film coefficient of heat transfer

$h = 11.9\ Btu/hr\text{-}ft^2$ (Calculated from data given in Reference 11-11)

Log mean temperature difference $\Delta T_m = \dfrac{(84.7 - 50) - (55 - 40)}{\ln \dfrac{84.7 - 50}{55 - 40}} = 23.5^\circ F$

Average overall coefficient of heat transfer

$Q = UA \times \Delta T_m$, $UA = 4568/23.5 = 195\ Btu/hr^\circ F$

Heat transfer surface

$1/UA = 1/(Ah)\ air = 1/(Ah)\ glycol$

$A\ (air) = 1987\ ft^2$

Feet of finned tubing $= 19.7/0.775 = 25.5\ ft$

Δ P Glycol Line = f x $1/d\, n^2/2$ = 0.25 psi/ft of tubing

Δ P in Heat Exchanger = 0.25 x 25.5 =	6.4 psi
Δ P in radiator	15.1
Δ P in interconnecting lines and fittings, total equivalent length = 20 ft	
Δ P = 0.25 x 20 =	5.0
	26.5 psi

$$\text{Glycol Pump Power HP} = \frac{Q\,\Delta P \times 144}{33{,}000 \times 0.5} = 0.032 \text{ HP} = 25 \text{ watts}$$

11.7.4.3 Cabin Ventilation

The air velocity in the ducts has been selected at 1000 feet/minute. Air flow is set at 330 CFM in order to achieve an acceptable temperature rise in the cabin with the large (5kw) power source for the small (200 cubic feet) volume of the cabin.

Duct size A = 330/2 branches cubic foot/minute x 1/1000 cubic foot/minute = 0.165 square foot which corresponds to a circular duct of 5.5 inches diameter. The ΔP/100 feet of length for this duct is 0.33 inch of water (Reference 11-12).

Assuming 15 feet of ducting:

Δ P = 30 x 0.33 in/100 ft = 0.1 in. water

Δ P in the diffusers = 0.2 in. water

Δ P in 90°elbows, L = 17D = 7 ft

Δ P = 7 ft x 2 x 0.33 in./100 ft = 0.05 in. water

Total Δ P = 0.05 + 0.2 + 0.1 + 0.05 = 0.4 in. water

Correcting for 7 psia atmosphere, 0.4 x 7/14.7 = 0.2 in. water

Total system pressure drop:

Ducting System	= 0.2 in. water
Heat exchanger	= 0.2
Filter and Charcoal	= 0.2
	0.6 in. water

Fan power required for 0.6 in. water pressure drop and 330 CFM

$$HP = AHP/0.5 = CFM \times \Delta P/6536 \times 0.5 = 0.061\ HP = 46\ watts$$

11.7.4.4 Air Purification

a. Non-Integrated Design - Air purification involves the removal of contaminants produced by the human body, compartment components and supplies.

The oxygen recovery system discussed in the first phase of the thermal integration study, Reference 11-10, provides for an effective removal of carbon dioxide. Odors and some trace contaminants are removed by the charcoal bed. However, a catalytic burner must be used to ensure the removal, by oxidation, of CO, H_2, CH_4 and 0_3. The catalyst considered for this study is palladium, electronically heated to 700°F. It is estimated that 50 watts of power is required to operate the unit.

Bacteria floating or suspended in the cabin air are killed by the ultraviolet energy of a germicidal lamp located in the air duct. One standard G25T8 bulb is assumed, with a power consumption of 25 watts. Component weight estimates are shown in Table 11-3.

b. Integrated Design - The only component in the cooling and ventilation system which lends itself to thermal integration with the electrical generating power system is the catalytic burner. The small amount of air that is by-passed to the catalytic bed can be heated to approximately 600°F in the Rankine cycle system and 400°F in the Brayton cycle. These temperatures are sufficiently high to cause the reduction of all commonly known trace contaminant gases except methane which is not noticeably affected at temperatures below 425°F regardless of the catalysts presently in use. Published data on catalyst characteristics, operating temperatures, and reaction rates is fragmentary and incomplete. The best available information shows that for a cabin environment similar to that of the MORL vehicle, as much as 80% of the CH_4 gas can be oxidized at 600°F over a palladium catalyst. On this basis it is concluded that no additional electrical power input is required for air purification in the Rankine cycle integrated system while in the Brayton cycle integrated system additional power will be required to bring the catalyst from 400°F to the 500-600°F range.

A conservative estimate for the additional power required is 17 watts. This based on the assumption that 0_3, $C0$, H_2 and carbon compounds are reduced with a catalyst operating at low temperatures up to 400°F while only the methane gas is reduced over the high temperature catalyst. Component weight estimates are shown in Table 11-5.

11.7.4.5 Cooling Power Requirements

a. Non-Integrated Design

Fan Power	46 watts
Glycol Pump	25
Germicidal Lamp	25
Catalytic Burner	50
	146 watts total

b. Integrated Design

1. Rankine Cycle

Fan Power	46 watts
Glycol Pump	25
Germicidal Lamp	25
Catalytic Burner	0
	96 watts total

2. Brayton Cycle

Fan Power	46 watts
Glycol Pump	25
Germicidal Lamp	25
Catalytic Burner	17
	113 watts total

11.7.5 SUMMARY, POWER AND WEIGHT REQUIREMENTS OF LIFE SUPPORT SYSTEM

a. Non-Integrated System

Power requirement = 1,719 + 146 = 1,865 watts

Total weight = 491 pounds

b. Integrated Systems

1. Rankine cycle integrated system

   Total electric power requirement = 703 watts

   Total weight = 515 pounds

2. Brayton Cycle integrated system

   Total electric power requirement = 720 watts

   Total weight = 566 pounds

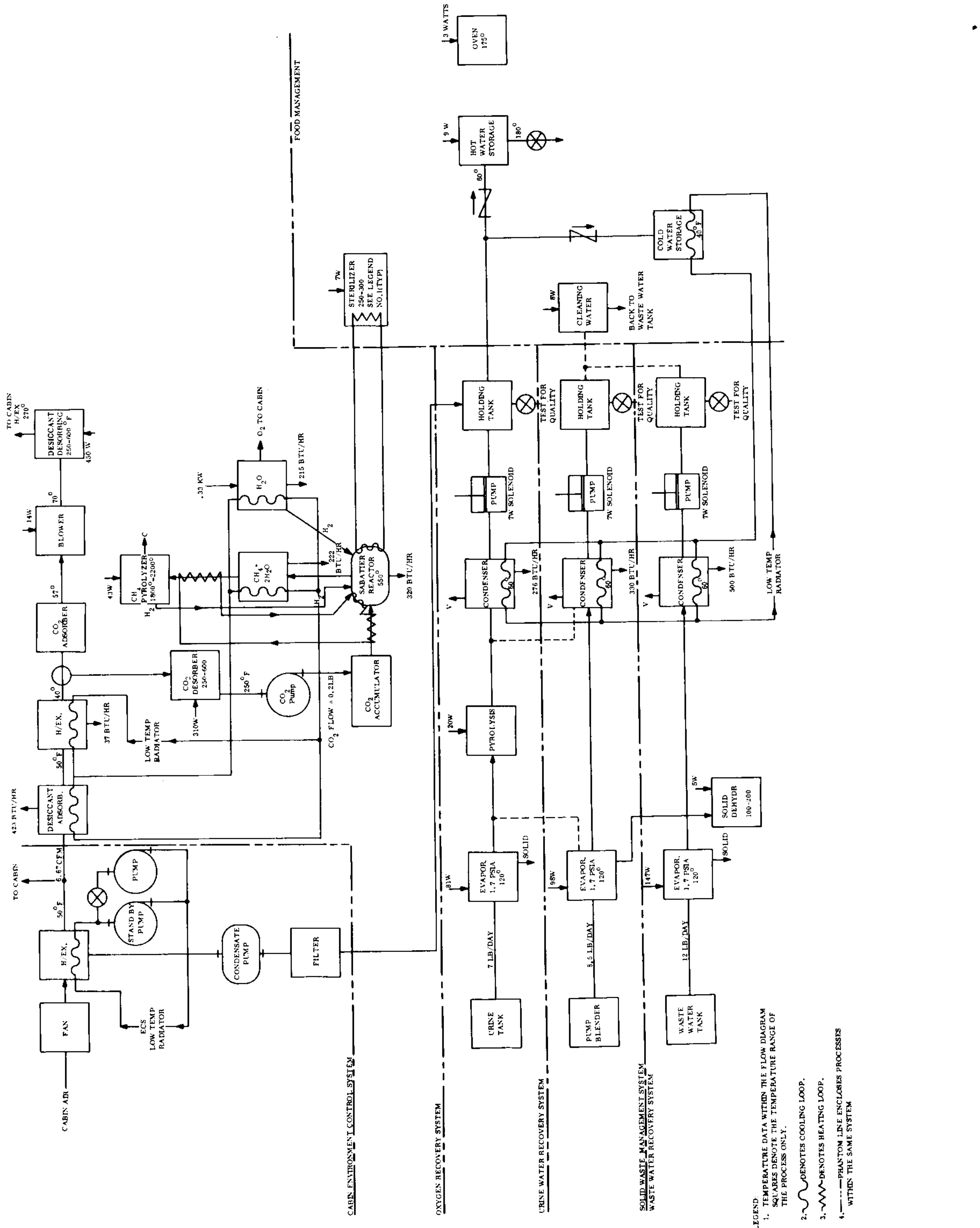


Figure 11-20. Non-Integrated Life Support System, Flow Diagram

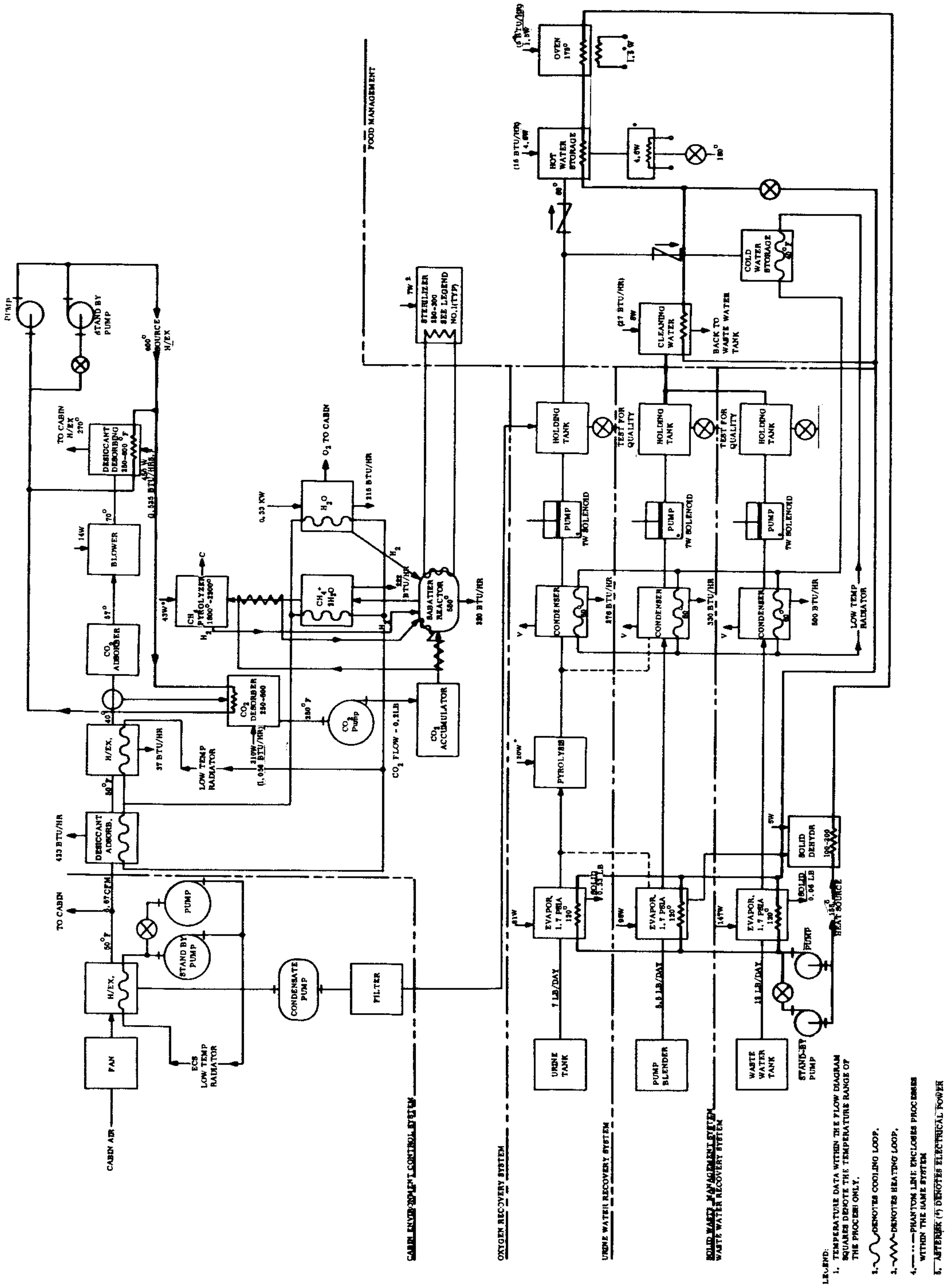


Figure 11-21. Integrated Life Support System, Flow Diagram

11.8 ISOTOPE BRAYTON FIVE KW_e POWER SYSTEMS

11.8.1 INTRODUCTION

The solar energy profile for a fixed point on the lunar surface precludes use of a solar power system since there are two weeks of darkness. Possible power sources that can function include radioisotope dynamic systems. This section describes Isotope Brayton Systems, the following Section (11.9) describes Isotope Mercury Rankine Systems for use on the moon.

The Isotope Brayton Power Systems convert thermal energy absorbed from decaying Pu_{238} to electrical energy with a turbo-generator, using argon as a working fluid. These systems are quite similar to those described in Section 8 for the MORL orbiting space station; the lunar application is scaled down from $8KW_e$ to $5KW_e$. A system schematic is shown in Figure 11-22. A temperature/entropy diagram on the Brayton Cycle is shown in Figure 11-23. The cycle analysis and assumptions are shown in Table 11-7.

Major components are:

a. Pu_{238} Isotope Source and Heat Exchangers
b. Isotope Shield
c. Source/Shield Thermal Control
d. Turbo-Generators
e. Turbo-Compressors
f. Recuperator
g. Radiator

Two power systems are considered, a non-integrated system, and a system integrated with life support and absorption refrigeration cooling.

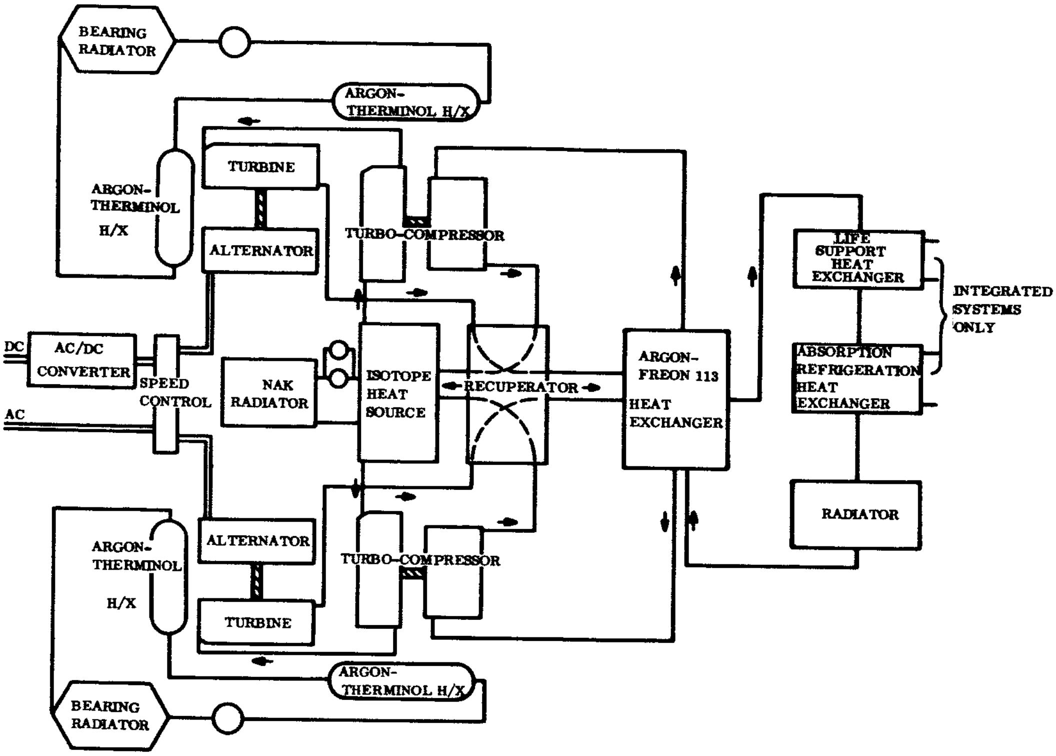


Figure 11-22. Isotope Brayton Power System, Schematic Diagram

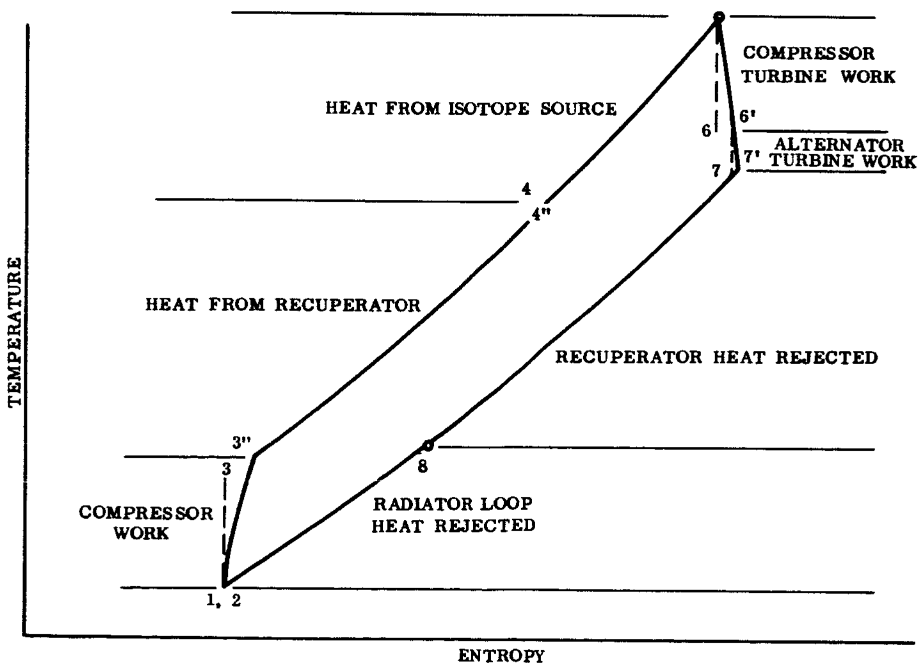


Figure 11-23. Brayton Cycle Diagram

TABLE 11-7. FIVE KW_e LUNAR BRAYTON SYSTEM CYCLE ANALYSIS

Compressor Efficiency = 0.79

Compressor Turbine Efficiency = 0.82

Generator Turbine Efficiency = 0.83

Fluid = Argon, properties from Reference 11-12

Assume bearing losses of:

1.8 Btu lb Compressor (non-integrated)
2.1 Btu lb Compressor (integrated)
0.6 Btu lb Generator (non-integrated)
0.8 Btu lb Generator (integrated

State	T	h	P_r	P	
1, 2	536	66.5	12.23	6.0	
3	750	93.1	28.15	13.8	
3'	808.3	100.2			
3"		98.4			
4	1488	184.9			
4"		183.1			
5	1950	242.5	304.7	12.97	1.635
6	1603.4	199.2	186.5	7.93	
6'	1670	207.0	206.8		1.245
7	1529	190.0	166	6.37	
7'	1561	193.7			
8	875	109.0			

$h_8 - h_1 = h_{rejected}$

<u>Heat Balance</u>		<u>Non-integrated</u>	<u>Integrated</u>
Energy Added	=	59.4 Btu/lb	59.4 Btu/lb
Compressor Losses	=	1.8 Btu/lb	2.1 Btu/lb
Generator Losses	=	0.6 Btu/lb	0.8 Btu/lb
Heat Rejected	=	42.5 Btu/lb	42.5 Btu/lb
Available Energy	=	15.1 Btu/lb	14.8 Btu/lb

(Generator Bearing Losses Taken from Available Energy)

11.8.2 NON-INTEGRATED SYSTEM POWER

The shelter power level is defined as $5KW_e$. To simplify analysis, the profile is defined as steady-state, with no circuit transient provisions (other than speed control) required. The load is divided into one-half 400 cycle ac and one-half dc. The ac is supplied directly by the alternator, while 220 watts is required for ac/dc conversion to supply 2.5 KW_e dc output power. Five percent of the net load is allocated for speed control for the turbo-generator unit (250 watts). Provision is made for an additional 100 watts for pump power for the bearing coolant loops (see Table 8-4). From the analysis of an Isotope Stirling System in Reference 11-1, pump power requirements for a NaK loop source/shield thermal control is estimated as 200 watts. The total non-integrated turbo-alternator output is 5.77 kw.

11.8.3 INTEGRATED SYSTEM POWER

As described in Section 11.7, the non-integrated power to the life support system is 1.865 kw. The integrated system electrical power requirement is 673 watts. The resultant saving is 1192 watts, such that the integrated power output is 3.81 kw. The ac power and dc power are each 1.905 kw. The ac/dc conversion loss is 170 watts. Speed control of the turbo-alternator requires 190 watts. The pump power for the bearing coolant loops is 80 watts. The NaK pump work for the shield/source thermal control loop is estimated at 150 watts. Total integrated power system alternator output is 4.46 kw.

11.8.4 HEAT BALANCE

The energy calculations for both the non-integrated and the integrated Isotope Brayton Systems are summarized in Table 11-8.

TABLE 11-8. LUNAR BRAYTON SYSTEMS ENERGY BALANCE

	Non-Integrated System	Integrated System
Output, kw	5.00	3.81
AC/DC Conversion	0.22	0.17
Cycle Coolant	0.10	0.08
Shield Coolant Pump Power	0.20	0.15
	5.52	4.21
Speed Control, kw	0.25	0.19
	5.77	4.40
Generator Efficiency	86%	85.2%
Generator Power, kw	6.71	5.18
Turbo-generator Bearing Losses, kw	0.30	0.30
Turbine Work, kw	7.01	5.48
Btu/sec	6.65	5.19
Net Energy Available, Btu/lb (from Table 11-7)	15.3	14.6
Required Argon Flow, lb/sec	0.435	0.354
Heat Added, Btu/lb	59.4	59.4
Compressor Bearing Losses, Btu/lb	1.75	2.1
Radiator Rejection, Btu/lb	42.5	42.5
Compressor Bearing Losses, kw	0.80	0.80
Heat Rejected, Main Radiator, kw	19.5	16.0
Isotope Source Size, kw	29	24
Heat Losses (Thermal Control, etc.), kw	1.7	1.7

11.8.4.1 Generator Losses

The alternator efficiencies were determined from Figure 6-23 of Reference 11-10. The non-integrated losses are 0.94 kw, while the integrated system losses are 0.78 kw.

The bearing losses were estimated at 0.30 kw for both systems, based on Table 6-9 of Reference 11-10.

11.8.4.2 Compressor Losses

The turbocompressor bearing losses were estimated at 0.8 kw for both systems, based on Table 6-9 of Reference 11-10.

11.8.4.3 Turbine Work

The compressor efficiency is taken as 79% and the compressor turbine efficiency as 82%. These are the same as the integrated compressor performance for the Reference 11-10 Study and are shown in Table 6-6 of that reference. The compressor turbine work must also supply the bearing losses energy of 0.8 kw. The compressor work is determined by the pressure ratio and flow rate of the working fluid (see Table 11-7 and 11-8). The non-integrated compressor turbine work is 16.3 kw, while the integrated turbine work is 13.4 kw.

The turbo-alternator turbine efficiency is 83%, from Table 6-6 of Reference 11-10. The required turbine work is the sum of generator work and bearing losses, 7.01 kw for the non-integrated case and 5.48 kw for the integrated case (from Table 11-8). The system flow is then based on this power turbine work.

11.8.4.4 Heat Recovered

The recuperator recovers or transfers heat from the lower pressure side of the cycle to the higher pressure side of the cycle. The amount transferred is 84.7 Btu/lb or 38.8 kw for the non-integrated case and 32.1 kw for the integrated system.

11.8.4.5 Heat Rejected

Following the recuperator, additional heat is rejected from the argon fluid such that the gas temperature at compressor inlet is 536°R (see Figures 11-22 and 11-23). The amount rejected is then the difference in enthalpy between state point 8 and state point 1 times the flow rate; 19.5 kw for the non-integrated system and 16.1 kw for the integrated system. The heat lost to the compressor and turbine bearings and the generator heat losses are rejected with separate radiator-pump systems. Their characteristics are summarized in Table 11-9. The non-integrated heat load is 2.04 kw and the integrated system rejects 1.88 kw.

A portion of the isotope source heat is rejected in the source-shield thermal control loop. Some cooling is always necessary to maintain the shield below its melting point. This loop is, in addition, sized to reject all of the source heat for periods (such as storage) when the power system is not operating.

11.8.4.6 Isotope Source Size

The heat added is 59.4 Btu/lb (from cycle analysis) times the flow rate. This is 27.3 kw for the non-integrated case and 22.4 kw for the integrated system. The source is assumed to consist of parallel fuel element-heat exchanger tubes with 1 kw output each (from Reference 11-10). The necessary number of elements is 28 and 23 for the non-integrated and integrated systems to satisfy the power systems requirements. It is assumed that 1 kw additional energy is required in each case to make up the heat removed in shield cooling. Total non-integrated source energy is 29 kw, integrated source is 24 kw.

TABLE 11-9. BRAYTON SYSTEM RADIATORS

	Non-Integrated	Integrated
Power System Radiator		
Waste Heat	19.5 kw	16.1 kw
Life Support Heat		1.58 kw
Absorption Refrigeration Heat		7.03 kw
Net Heat Load	19.5 kw	7.49 kw
Btu/Hr	66,600	25,550
Radiator Inlet-Outlet Temperature	860-500	670-500
Radiator Effective Temperature	615	566
ηfin	0.85	0.85
Specific Weight, Lb/Ft^2	0.93	0.93
Area, Ft^2	261	161
Total Weight, Lb.	243	150
Q/A, Btu/Hr-Ft^2	255	157.26
Heat Exchanger Weight	124	150
Generator/Bearing Coolant Radiators		
Heat Rejected, kw	2.04	1.88
Btu/Hr	6970	6410
Radiator Inlet-Outlet Temperature	660-640	660-640
Effective Temperature	645	645
η fin	0.8	0.8
Specific Weight Lb/Ft^2	0.93	0.93
Area, Ft^2	22.6	20.8
Total Weight, lb	21	19.5
Source/Shield Thermal Control and Radiator		
Pump Weight (2 units) lb	32	28
Shield Insulating and Heat Removal Loop, lb	7	6
PuO_2 Heat Exchanger Piping and NaK Inventory, lb	12	10
Radiator Weight, lb	14	12
Total Weight, lb	65	56

11.8.5 POWER SYSTEM COMPONENTS

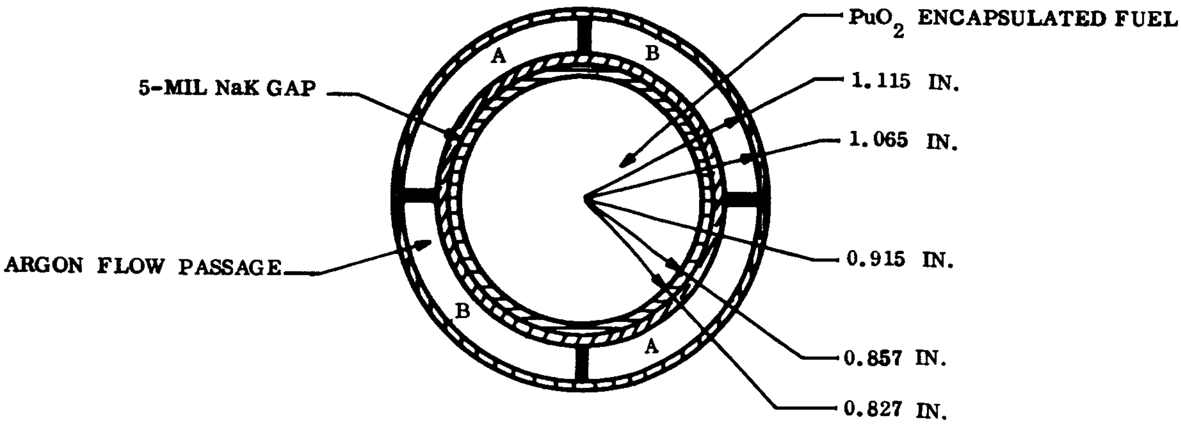
The two power system weights are summarized in Table 11-10. The non-integrated power system weighs 3444 pounds, the integrated power system weighs 2939 pounds.

TABLE 11-10. LUNAR BRAYTON SYSTEMS, WEIGHT SUMMARY

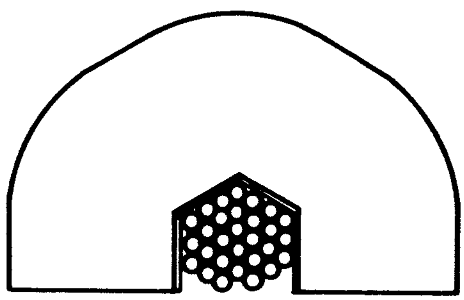
Item	Weight (lb) Non-integrated	Weight (lb) Integrated
Isotope Source	185	153
Heat Exchangers	530	438
Shielding	1390	1270
Recuperator	230	185
Turbo compressors	68	68
Turbo alternators	98	78
Support Structure and Piping	395	350
Controls	95	66
Main Radiators	243	150
Radiator Loop Heat Exchanger	124	105
Bearing Radiator System	21	20
Shield Thermal Control and Radiator	65	56
Total	3444	2939

Isotope Fuel and Heat Exchanger

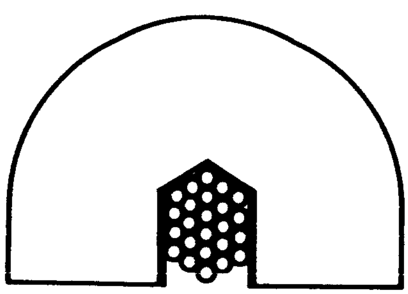
The fuel-heat exchanger design is identical to that for the Isotope Brayton System in Reference 11-10, also the same as that in Section 8 of this report. A cross section of a fuel-heat exchanger tube is shown in Figure 11-24. Total fuel and heat exchanger weights are plotted in Figure 11-25 as a function of source heat. The fuel weight (30% density PuO_2) is 185 pounds non-integrated and 153 pounds for the integrated system. The heat exchangers' weights are 530 pounds non-integrated and 438 pounds integrated.



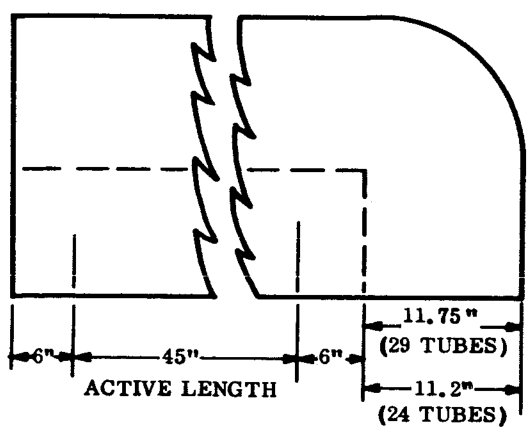
FUEL - HEAT EXCHANGER ELEMENT (FROM REFERENCE 11-10)



NON-INTEGRATED ISOTOPE SOURCE
29 TUBES



INTEGRATED ISOTOPE SOURCE
24 TUBES



SIDE VIEW (BOTH SYSTEMS)

Figure 11-24. Isotope Source and Shield Characteristics

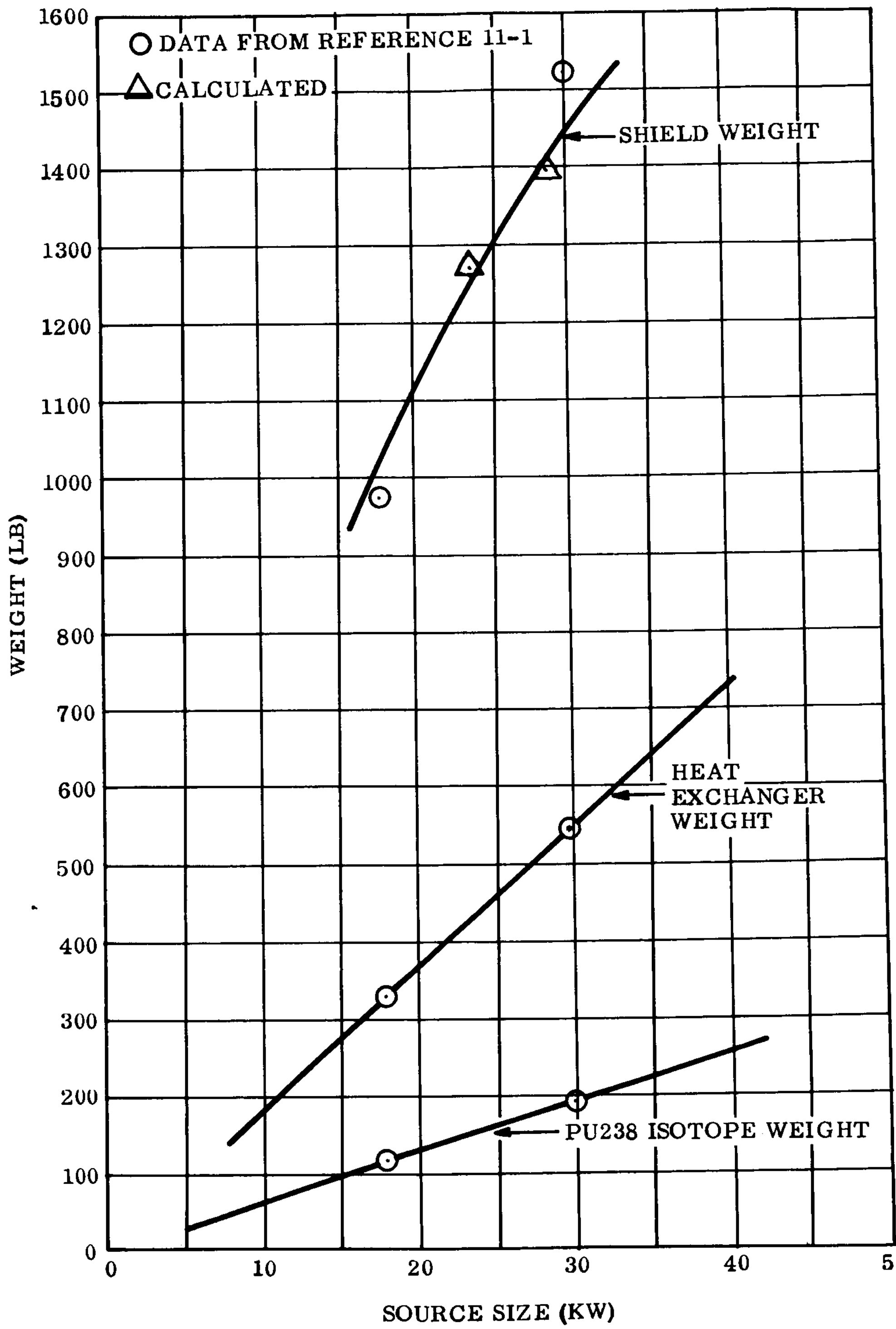


Figure 11-25. Isotope Source Components

The shield design followed the guidelines of the study in Reference 11-10. The fuel-heat exchanger cluster is shielded on four of six sides. The configuration is shown in Figure 11-24 for both power systems. The shield thickness criteria in Reference 11-10 is to limit the radiation dose to 30 rem per year (page 5-28, Section 5.4.2.1, "Shielding"), composed of 9.6 rem gamma radiation and 20.4 rem neutron dose. The time-averaged distance to crew stations in the orbiting space station is in the order of 13 feet. For the lunar shelter crew station, time-averaged distances from the isotope source are expected to be in the order of 7 feet. Thus, for a given unit area, the flux might be 3 to 4 times as great as the orbiting case, but time-of-stay is planned for a shorter duration. For real applications, a study would be made of possible mission radiation profiles from man earth launch to earth return to arrive at optimized shelter shielding requirements. The assumption here is to use the same shield thickness requirements as for the orbiting station. Shield thicknesses are 11.75 inches of reinforced lithium hydridge for the non-integrated case and 11.2 inches for the integrated system (see Figure 11-24). The weights are 1390 pounds non-integrated and 1270 pounds integrated. Weight data is presented as a function of source size in Figure 11-25.

Source Thermal Control

The argon temperature out of the isotope source is 1490°F by definition. It is found that LiH melts at 1272°F. Therefore, a thermal barrier and cooling loop is required between the shield and the heat exchanger tubes. The isotope fuel elements require heat removal to avoid damage. When the power system is operating, the argon does this job, but for standby conditions a separate provision must be made to reject the isotope thermal energy. To satisfy both of the above conditions a NaK loop is required, with pumps and a radiator. Table 11-9 lists the cooling loop estima ted characteristics. The radiator is sized to handle all of the source energy output. Two pumps are provided for redundancy.

Turbo Machinery

The 67,500-rpm turbo compressors are identical to those in Section 8. At the low output level of these systems, the weight is practically independent of output. Each system contains two units weighing 34 pounds each.

Two 12,000-rpm homopolar 4-pole turbo alternators are used for each power system. These units are identical in design to those of Section 6, Reference 11-10. The non-integrated weight is 49 pounds for each turbo alternator and the integrated weight is 39 pounds each. The weights are plotted as a function of generator output in Figure 11-26.

Radiator Loop

The power system radiator characteristics are shown in Table 11-9. The loop schematic can be found as a part of Figure 11-22. The Freon 113 loop used is based on the same principles presented in Section 8. The heat exchanger was scaled from Table 8-6. The radiator configuration, based on the study of Section 11.4, is a vertical east-west oriented fin, with a local surface modifier having characteristics $\alpha s/\epsilon = 0.06/0.06$. The specific weight could not be determined directly, but was estimated at 0.93 pounds/square foot of fin (see Section 11.4 for further discussion).

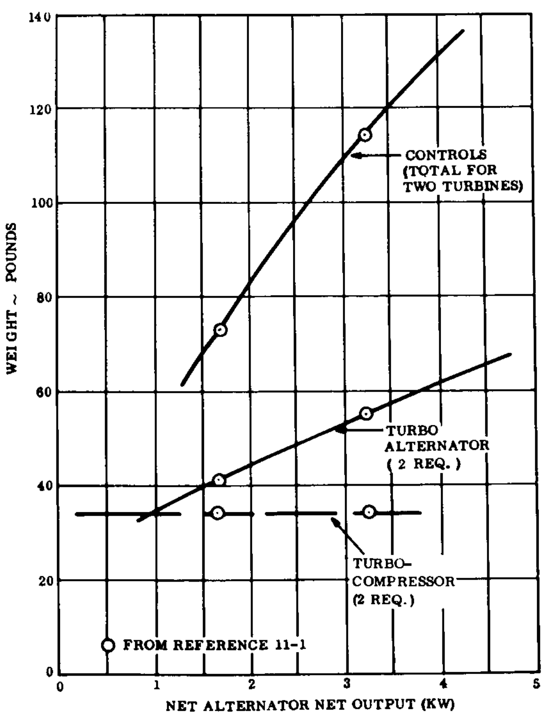


Figure 11-26. Turbo Machinery and Controls

Other Components

The power system controls, recuperator and support structure are identical in principle to those described in Section 6 of Reference 11-10. Figures 11-26 and 11-27 show their weights as a function of system size.

11.8.6 DISCUSSION

NaK Loop

The NaK Pumps require their rated power during standby, but some smaller amount when the loop function is shield thermal control only. The rated power was included as an addition to station power, although conservative, since the shield cooling loop was not designed in depth. If the NaK was always circulated at the same flow rate, NaK radiator area would have to be reduced while the station was operating. Another, and perhaps

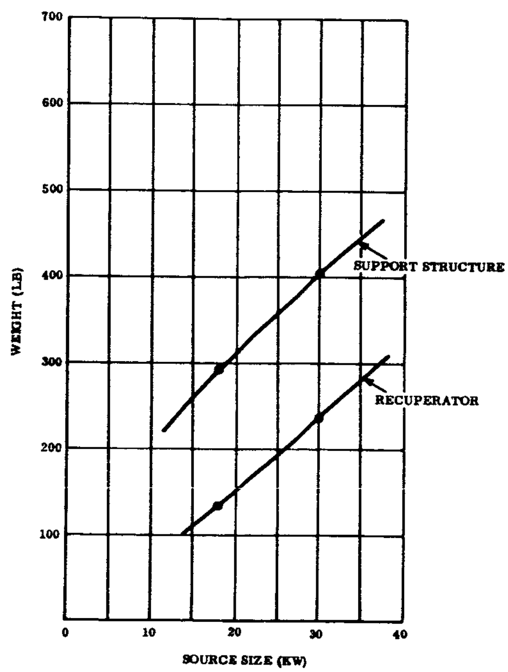


Figure 11-27. Support Structure and Recuperator

more desirable solution, would be to have a liquid loop source with a NaK-Argon Heat Exchanger. NaK pump power would then be always nearly the same, and a considerable weight saving in the source and shield would be realized (in the order of 1100 pounds).

For a standby condition, a separate power source must be provided to operate the NaK pumps. A possible solution is to incorporate a thermoelectric generator into the NaK radiator.

Cycle Calculations

In the Brayton Cycle calculations (Table 11-7) bearing losses were assumed as some Btu/lb energy loss. The actual losses were fixed at 0.3 and 0.8 kw, from the study in Reference 11-10. Since the cycle calculation had to be made on a per pound basis, the resulting bearing losses are not exact, although the differences are negligible in terms of the total power systems, so further cycle reiterations were not deemed necessary.

Radiator

The integrated radiator actually rejects only 7.49 kw out of 16.1 kw of the total heat to be rejected. Thus, for a comparison with the non-integrated power system only, 50 pounds of additional radiator would have to be included in the integrated system to account for the rejection of 16.1 kw. Note that the specific heat rejection capability of the integrated radiator is much lower than the non-integrated radiator, since thermal integration uses the higher temperature waste heat.

Power System Weight Reduction

Considering the power system only, the integrated weight is 2989 pounds, while the non-integrated weight was 3444 pounds. This is a 13.2% reduction in weight.

11.9 FIVE KW_e ISOTOPE MERCURY RANKINE POWER SYSTEM

11.9.1 INTRODUCTION

The Isotope Mercury Rankine Power System is an Isotope thermal energy conversion. system using Mercury vapor to drive a permament magnet alternator, producing a-c power at 2000cps. There are two liquid loops, a NaK 78%-22% Eutectic in the isotope-heat exchanger loop, and mercury in the working loop. The isotope loop is derived from the isotope Stirling work performed on Study NAS 3-2799; the results of which are presented in Reference 11-10. The working loop is derived from the information presented in this report in Section 9. Figure 11-28 shows a schematic of the Dynamic Power System.

The Major Components are:

a. Pu238 Isotope Thermal Source and Heat Exchanger.
b. Isotope Shield
c. Isotope Loop Thermal Controller
d. NaK-Hg Heat Exchanger (Hg Boiler)
e. Two single-shaft, sealed, turbine-alternator-pump units of the TRW Sunflower Type (References 11-14 and 11-15).
f. Mercury Radiator-Condenser

Four power systems cases are considered; non-integrated and integrated; redundant turbo-alternators; and single turbo-alternators with stand-by capability. These cases are the same as were examined for the Orbiting Station in Section 9.

11.9.2 NON-INTEGRATED SYSTEMS POWER

The shelter power level is defined as 5 kw. To simplify analysis, the profile is defined as steady-state, with no circuit transient provisions (other than speed control) required.

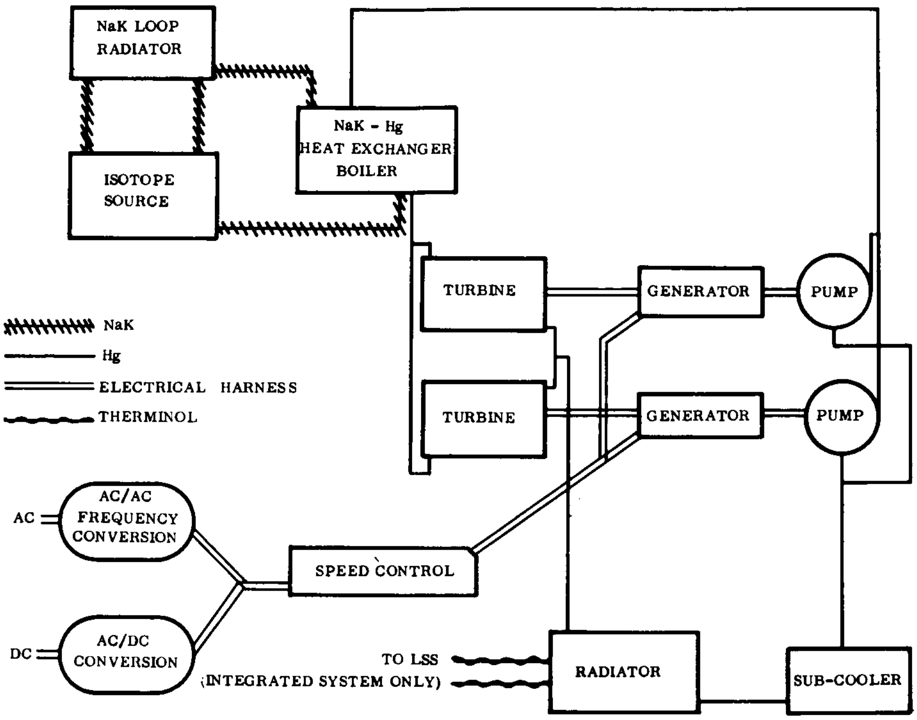


Figure 11-28. Isotope Mercury Rankine Power System, Schematic Diagram

The power is split half and half to 400-cps ac and dc distribution networks. The conversion of 2.5 KW_e from 2000-cps ac to 400-cps ac at an efficiency of 0.91 requires 250 watts of additional power. The conversion of 2.5 KW_e 2000-cps ac to dc at an efficiency of 0.92 requires 220 watts of additional power. Turbo alternator speed control uses 5% of the total power or 250 watts. The isotope source requires 350 watts to pump the NaK through the radiator and heat exchanger, maintaining a constant isotope source temperature. The total electrical load required from the generators is 6.07 kw.

11.9.3 INTEGRATED SYSTEMS POWER

As described in Section 11.7, the non-integrated power to the Life Support System is 1.865 kw. The integrated Life Support Electrical Requirement is 656 watts. The resultant saving is 1209 watts, such that the integrated Power System level is 3.79 kw. With the equal division of power into ac and dc, the conversion requirements are 200 watts for ac/ac and 170 watts for ac/dc. The speed control is 5%, or 190 watts. The NaK pump work is 270 watts. Total electrical power is then 4.62 kw for the integrated systems.

11.9.4 HEAT BALANCE

The energy calculations for the four system configurations considered are summarized in Table 11-11.

The alternator efficiencies were determined from Figure 7-1 of Reference 11-10 (Mercury Rankine PM Alternator Efficiency vs Alternator Output). The two turbine systems efficiency were found for half-size alternators, or outputs of 3 kw for the non-integrated case and 2 kw for the integrated case. The "1+1" systems were sized for 6.1 and 4.6 kw outputs. Two turbine systems alternator losess are 1.16 kw and 1.01 kw, while the "1+1" losses are 0.85 kw and 0.69 kw.

The criteria used for the Orbiting Station was to have a mercury circulating pump for each turbo-alternator. The power per pump was assumed to be independent of flow. The pump power used in the Sunflower System (Reference 11-14 and 11-15) was 0.18 kw. The two-turbine systems then have 0.36 kw pump power, while the 1 + 1 systems have 0.18 kw Mercury pump power required.

The NaK loop characteristics are extrapolated from the NaK loop used in the Stirling cycle analysis for contract NAS 3-2799, Reference 11-10. The NaK pump power is 0.35 kw for the non-integrated systems and 0.27 kw for the integrated systems.

The turbine(s) supplies shaft power to the generator and the pump (see Figure 11-13). The efficiency of the turbines is found in Figure 9-1, considering one turbine for the 1+1 system and two turbines, each rated for half the load for the two-turbine system. The inlet and outlet temperatures and pressures are in every case the same as those conditions achieved in the Sunflower Program (References 11-14 and 11-15); Pinlet = 240 psia, T_{inlet} = 1260°F, P_{outlet} = 7 psia, T_{outlet} = 605°F. The isentropic enthalpy change is 43.2 Btu/lb. The actual turbine work is (43.2)(turbine efficiency). The Mercury flow rate is then shaft work divided by work per pound. The turbine outlet Mercury quality is over 93% in every case, which is above the condensation point (some supersaturation occurs, more so than with water).

TABLE 11-11. FIVE KW_e MERCURY RANKINE POWER SYSTEM

	Non-Integrated System		Integrated System	
	1 + 1 Turbine	2 Turbines	1 + 1 Turbine	2 Turbines
Electric Power	5.00	5.00	3.791	3.791
AC/DC Conversion	0.22	0.22	0.17	0.17
AC/AC Conversion	0.25	0.25	0.20	0.20
NaK Pump Work	0.35	0.35	0.27	0.27
Speed Control	0.25	0.25	0.19	0.19
	6.07	6.07	4.62	4.62
Generator Efficiency	0.89%	84%	87%	82%
Generator Losses	0.85	1.16	0.69	1.01
Pump Work	0.18	0.36	0.18	0.36
Turbine Power Required	7.10	7.59	5.49	5.99
Generator Power for Turbine Efficiency (Total Power minus speed control and pump losses)	6.67	7.00	5.12	5.44
Turbine Efficiency	60%	51%	56.7	48.1%
Turbine Δh/sec	6.73	7.19	5.21	5.68
Theory Δh/sec	11.21	14.1	9.18	11.8
Flow, lb/sec	0.26	0.327	0.213	0.274
Heat Rejected Btu/lb	117.2	121.1	119.7	122.3
Heat Rejected Btu/sec	30.5	39.6	25.5	33.5
(KW Rejected)	32.1	41.8	26.9	35.3
Boiler KW_t Required	39.2	49.4	32.4	41.3
Actual Boiler Heat, KW_t	40.0	50.3	33.3	42.7
NaK Radiator Dissipation, KW_T	0.8	0.9	0.9	1.4
Shield Thickness, inches	12.7	13.3	12.2	12.8
No. of Heat Exchanger Tubes	4.0	5.0	3.0	4.0

Energy is removed from the working fluid until it is a liquid by condensing radiator. The amount of energy is the difference between turbine outlet enthalpy and the saturated liquid enthalpy. There is also a slight additional rejection of heat energy in a sub-cooler, the final enthalpy is 41.1 Btu/lb.

The required energy in the Mercury Rankine Boiler is the sum of the turbine work and the heat rejected by the radiators (it is assumed that the energy gained by the working fluid in cooling the generator balances system thermal losses).

11.9.5 POWER SYSTEM COMPONENTS

11.9.5.1 Isotope Thermal Energy Source

The heat energy requirement is supplied by P_u238. The fuel element and fuel-NaK heat exchanger design is taken from the Stirling Cycle Analysis of Reference 11-10, Section 5. Minor changes have been made to the active fuel length for the three-tube and four-tube configurations to accomodate the specific heat requirements of the power systems. A five-tube cluster was scaled to provide sufficient heat for the non-integrated two-turbine system. Table 11-12 lists the isotope source parameters. The weights are plotted in Figure 11-29.

11.9.5.2 Isotope Shield

The shield configuration is based on the element grouping and protection requirements. The protection requirements used in Reference 11-10 consisted of a maximum dose of 30 rem per year (page 5-28, Section 5.4.2.1, "Shielding"). The breakdown was 9.6 rem gamma radiation and 20.4 rem neutron dose. It was only necessary to shield two sides, or approximately half the total surface area. For the Lunar Shelter System, it is expected that full coverage might be required. Also, the average distance from crew accomodations to the isotope package would be less than the orbiting station. However, the mission life, or crew occupancy, would be much shorter than one year, so that using an identical shield

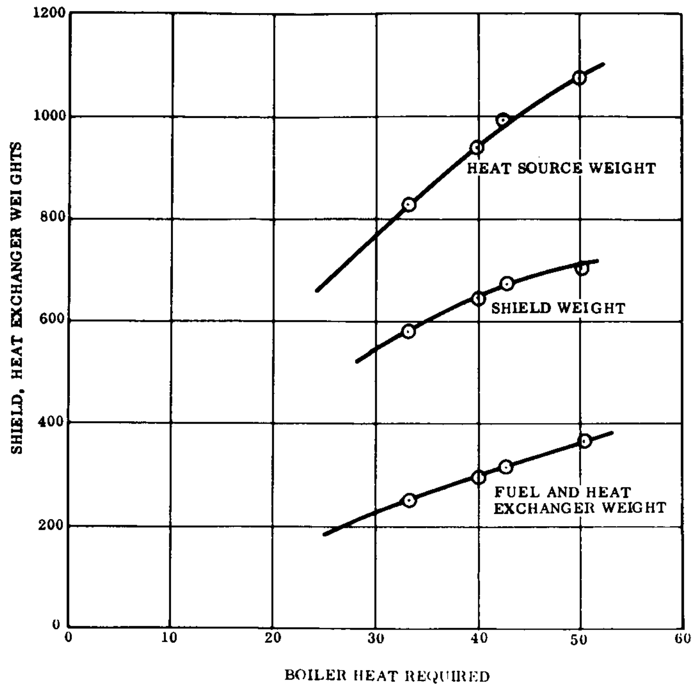


Figure 11-29. Isotope Source Weight for Mercury Rankine System

design to the orbiting station would yield reasonable weights. The shield weights and thickness are listed in Table 11-12. Figure 11-29 shows the weight as a function of heat required.

11.9.5.3 NaK Heat Exchanger, Radiator, and Pumps

The isotope source energy is removed by a NaK 78%-22% eutectic fluid loop. The heat is transmitted to Mercury in a NaK-Mercury heat exchanger, excess heat energy being dissipated in a NaK radiator. A pump is required to move the NaK in the loop. Figure 11-28 includes a schematic of this loop as a part of the power system.

Figure 11-30 shows the estimated NaK-Hg heat exchanger weight and NaK inventory as a function of boiler heat. The NaK radiator is sized from Section 5 of Reference 11-10. Figure 11-31 depicts radiator area and weight as a function of heat dissipated. Note that the radiator must be sized to reject all of the source heat to prevent source damage when

TABLE 11-12. ISOTOPE SOURCE PARAMETERS

Basic Isotope Sources			Weight	Power
3-tube	23.6 in.	active length		32 kw
		Shield	545	
		Heat Elements	237	
4-tube	23.6 in.	active length		42.7 kw
		Shield	676	
		Heat Elements	313	
5-tube	23.6 in.	active length		53.4 kw
		Shield	749	
		Heat Elements	391	

Selected Source Characteristics	Non-Integrated		Integrated	
	1+1	2	1+1	2
Number of Tubes	4	5	3	4
Shield Thickness	12.7 in.	13.3 in.	12.2 in.	12.8 in.
Active Fuel Length	22.1 in.	22.1 in.	24.6 in.	23.6 in.
Shield Weight	645 lb	706	580	676
Heal Element Weight	293	366	247	313
Flow	20,400 lb/hr	25,200 lb/hr	16,700 lb/hr	21,500 lb/hr
Δ h	6.8 Btu/lb	6.8 Btu/lb	6.8 Btu/lb	6.8 Btu/lb
Δ h	32.4^{o}F	32.4^{o}F	32.4^{o}F	32.4^{o}F
Nak Inventory	10 lb	10 lb	10 lb	10 lb
Nak Heat Exchanger Weight	4 lb	4 lb	4 lb	4 lb
Nak Radiator Weight	35 lb	43.3 lb	29 lb	37 lb
Area	19 ft^2	24 ft^2	15.6 ft^2	20.5 ft^2

the power system does not demand heat. The pumps are also scaled from Reference 11-10. To provide maximum safety three pumps are used, two on-line and one on standy. The pump design is canned rotor, with an assumed efficiency of 20%. The pumps weight is shown in Figure 11-30.

11.9.5.4 Turbo-Alternator

The turbine-alternator-pump units are identical in design to those used previously in this study (see Section 9). The weight as a function of generator output is shown in Figure 11-32.

11.9.5.5 Other Mercury Loop Equipment

The other equipment requirements are:

a. Speed control
b. Starter
c. Mercury inventory
d. Support structure

These items are similar in design to the equipment discussed in Section 9. The origin of the information is the Sunflower System described in References 11-14 and 11-5. Figure 11-32 presents weights as a function of generator output.

11.9.5.6 Condensing Radiator

The radiator design is taken from Reference 11-9. The surface coating, $\alpha_S/\epsilon = 0.25/0.9$ is used as a reasonable coating for the high operating temperature. This results in a slight reduction in heat rejection capability on a unit area basis. The weight and area as a function of heat load are presented in Figure 11-31.

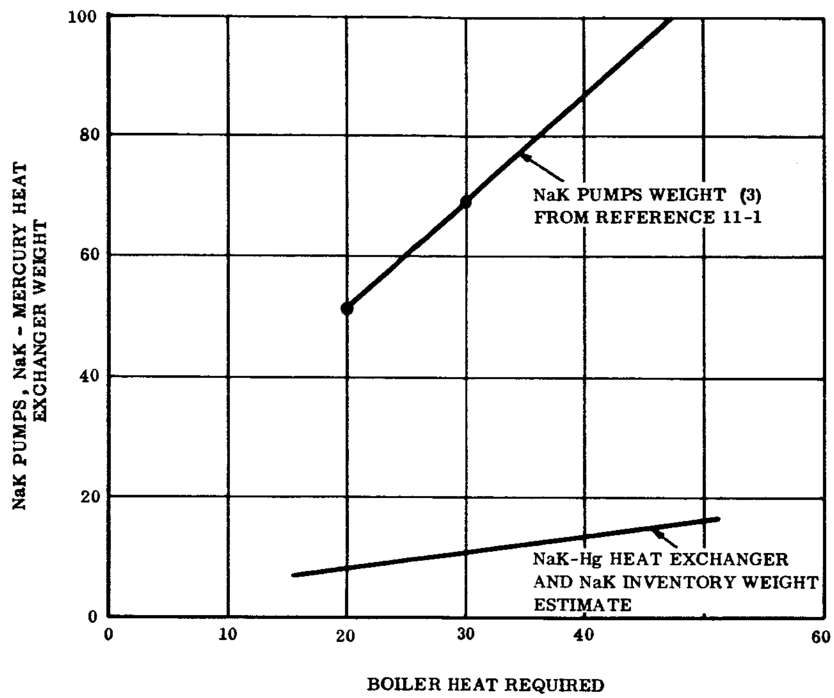


Figure 11-30. Isotope Mercury Rankine NaK Loop Heat Exchanger and Pump Weights

11.9.6 DISCUSSION

11.9.6.1 Total Weights

Table 11-13 presents a weight summary of the four power systems studied. The reduction in power required is 24% for the integrated case. The weight reductions are 9% for the two-turbine systems and 13% for the one-plus-one turbine systems. The reason for the weight reductions being so small is that the turbo-alternator efficiencies decrease rapidly with decreasing system size.

11.9.6.2 NaK Pump Power

The NaK pumps require continuous power to remove heat from the isotope source. This means that either the power system must always be operating or that a separate power source is required. A possible solution is to incorporate a thermoelectric generator into the NaK radiator.

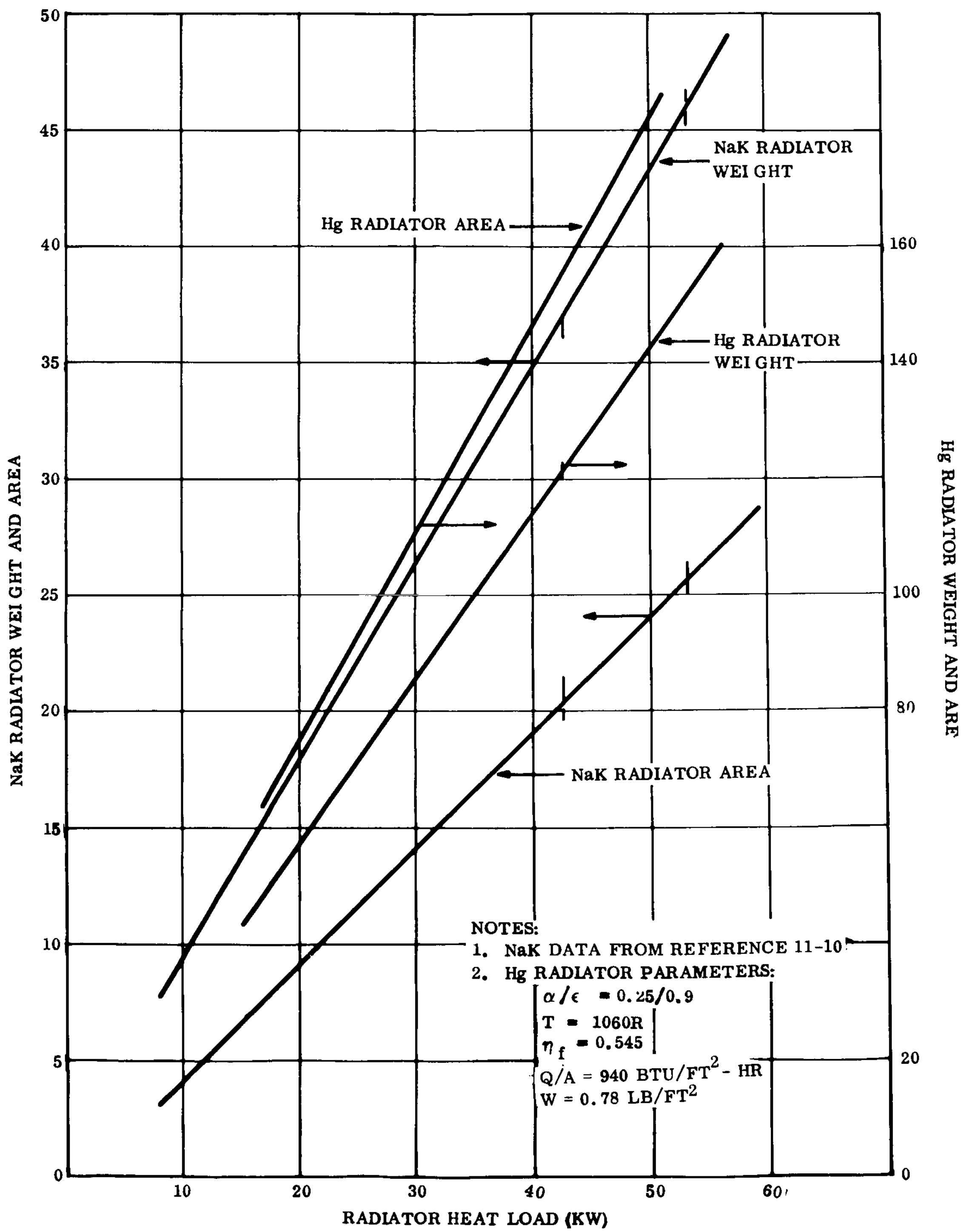


Figure 11-31. Isotope Mercury Rankine System NaK and Hg Radiator Properties

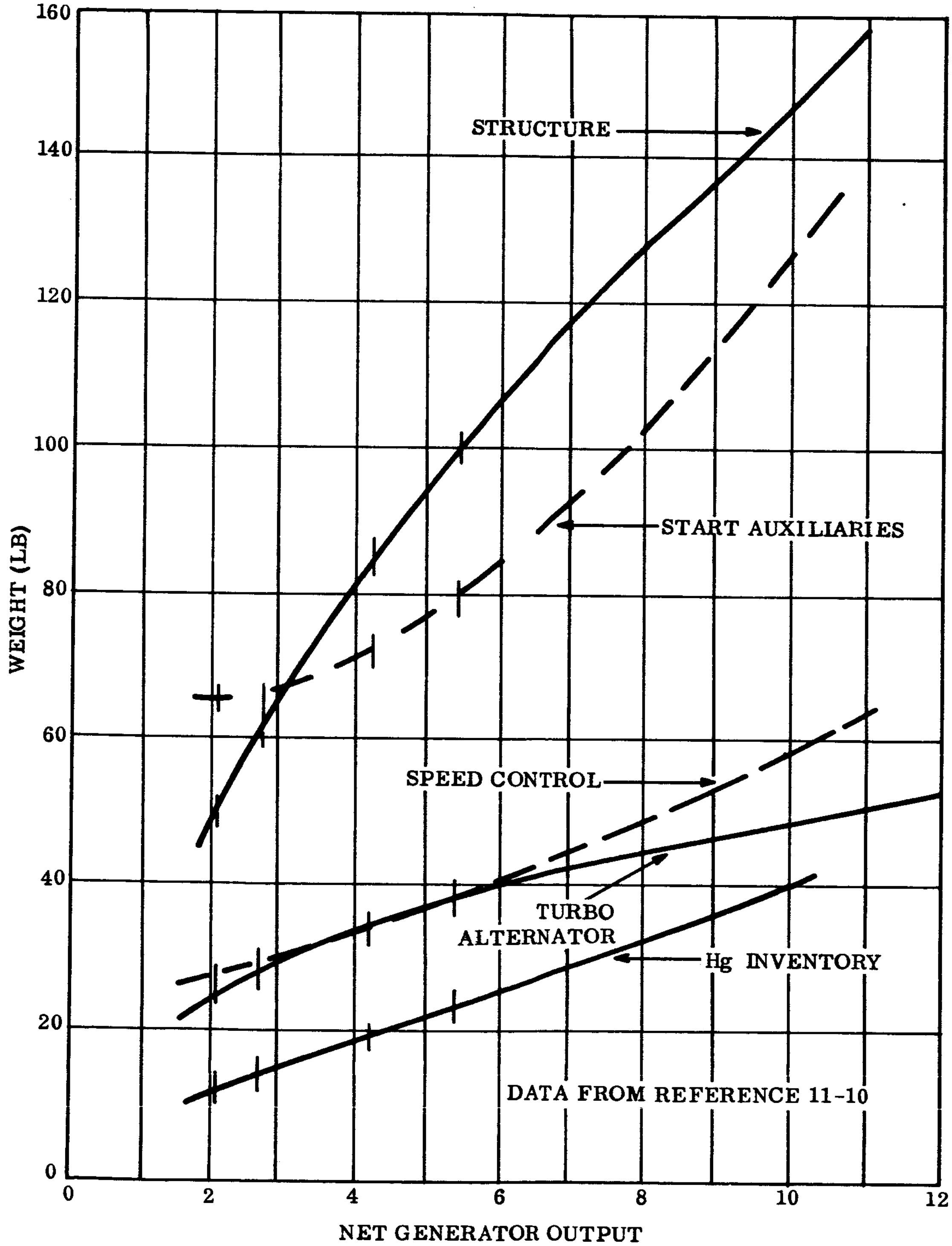


Figure 11-32. Mercury Rankine System Component Weights

TABLE 11-13. ISOTOPE MERCURY RANKINE SYSTEMS WEIGHT SUMMARY

ITEM	Non-Integrated		Integrated	
	1 + 1 Turbine System	2 Turbine System	1 + 1 Turbine System	2 Turbine System
Isotope Shield	645 lb	706 lb	580 lb	676 lb
Fuel and Heat Exchanger	293	366	247	313
NaK Pumps	87	105	75	92
NaK-Hg Heat Exchanger	13	16	12	14
Turbo-Alternator	38	55	34	50
Speed Control	38	59	34	56
Start Auxiliaries	80	133	72	130
Hg Inventory	24	27	19	23
NaK Radiator	35	43	29	37
Hg Radiator	114	142	94	121
Structure	100	123	85	100
Second Turbo-Alternator	38		34	
Second Mercury Inventory	24		19	
Additional Structure	50		40	
	1579	1775	1374	1612
Reduction in Weight Due to Life Support Heat Use			-3.6	-3.6
Reduction in Weight Due to Absorption Refrigeration Heat Use				
Single Cycle			-19.9	-19.9
Cascade (Dual) Cycle			-46.8	-46.8
Totals	1579	1775	Single 1350	1588
			Dual 1324	1562

11.10 THERMALLY INTEGRATED SYSTEMS

This section describes the integration of the Life Support, Environmental Control and Electronic Equipment cooling equipment with the power systems in the two-man Lunar Shelter. The effect of integration with a heat pump and absorption refrigeration cycle cooling system is also discussed. The shelter Life Support and Environmental Control systems are discussed in Section 11.7. The Electronic Equipment cooling is considered only for radiator sizing from Section 5. The Brayton and Mercury Rankine Power systems are described in Sections 11.8 and 11.9 for the Lunar Shelter.

11.10.1 BRAYTON POWER SYSTEM THERMAL INTEGRATION

Table 11-14 summarizes the integration with the Brayton Power system and compares the three methods of cabin heat rejection. As can be seen, the lightest weight system is the one integrated with the low temperature radiator. In each case, the three Cabin-Life Support loop radiators used in Section 11.7 were replaced with a single heat rejection system, Low Temperature Radiator (LTR), Absorption Refrigeration (AR), or Heat Pump (HP). The mechanical connection is identical to all three. This is an inlet pipe carrying fluid to be cooled, and an outlet pipe carrying cooled fluid. The three heat loads are combined, such that the necessary return temperature of $40^{o}F$ is identical for all three. The radiator sizes for the ultimate heat rejection to space are sized solely on their effective temperature. The LTR radiator weighs 266 pounds, the AR radiator 318 pounds, and the HP radiator 259 pounds. The LTR system requires no additional equipment while the AR system requires 43 pounds (from Section 11.6) and the HP system requires an estimated 25 pounds for compressor, motor, heat exchangers, and refrigerant inventory. The Life Support equipment weighs 156 pounds in all cases, minus radiators. The Integrated Power system weighs 2989 pounds. For AR comparison, a 50-pound credit for radiator reduction is subtracted and a 48-pound penalty for additional pump power is added. For the HP case 302 pounds is added as a weight penalty for the additional electric power to drive the vapor compressor. Note that the increases in electric power for the AR and HP systems is replaced in the net output. The lightest weight Life Support-Brayton Power System is then 3611 pounds, which is 370 pounds lighter than the non-integrated system.

TABLE 11-14. LUNAR SHELTER POWER SYSTEMS COMPARISONS

	Radiators Effective Temp. (^{o}F)	Radiator Area (Ft^2)	Radiator Weight (lb.)	Heat Rejection Equipment Weight (lb.)	Life Support Equipment Weight (lb.)	Power System Weight (lb.)	Total System Weight (lb.)	Net Electrical Power (kw)
<u>BRAYTON SYSTEM</u>								
A. Integrated with low temperature radiator	43	286	266	-	356	2989	3611	3.81
B. Integration with Absorption Refrigeration	19,200 Btu at 90^{o}F 22,800 Btu at 80^{o}F	142 200	318	43	356	2987	3704	3.88
C. Integrated with Heat Pump	47	278	259	25	356	3273	3913	4.22
D. Non Integrated System	43	286	266	-	271	3444	3981	5.00
<u>MERCURY RANKINE SYSTEM</u>								
A. Integrated with low temperature radiator	43	286	266	-	295	1609	2170	3.79
B. Integrated with Absorption Refrigeration	22,500 Btu at 190^{o}F 51,800 Btu at 175^{o}F	66.5 172	221.5	100	295	1618	2234	3.95
C. Integrated with Heat Pump	67	204	190	35	295	1824	2344	4.40
D. Non-integrated System	43	286	266	-	271	1775	2312	5.00

11.10.2 MERCURY RANKINE POWER SYSTEM THERMAL INTEGRATION

The characteristics of the Integrated Mercury Rankine Power System and the three methods of cabin heat rejection are also presented in Table 11-14. The LTR heat rejection system is lightest of the three alternatives when considering the Mercury Rankine system as a heat/electrical power source. The radiator weights are 266 pounds for the LTR system, 221 pounds for the AR system (dual cycle) and 190 pounds for the HP system. Additional equipment requirements consist of 100 pounds for Absorption Refrigeration equipment (as described in Section 11.6.3) or an estimated 35 pounds of vapor compression equipment. The Life Support equipment weight minus radiators is 295 pounds (from Table 11-14). The Integrated Power system weight is 1609 pounds. A weight saving of 47 pounds is experienced with the radiator in integrating with the dual AR system, however, the 161 watts additional power requires 56 pounds more power system weight. For the HP application the electric power weight penalty is 215 pounds. The lightest Life Support-Mercury Rankine power system is 2170 pounds, 142 pounds lighter than the non-integrated system.

11.10.3 DISCUSSION

11.10.3.1 Heat Rejection

The comparisons just made show that Absorption Refrigeration is not of benefit for the application considered. The heat pump system requires a higher weight than that of the Absorption Refrigeration cycle to supply the electrical driving power. Also note that the optimization point of radiator-power system weight for the HP does not allow much reduction in radiator weight/area.

The low temperature, as considered for this application (no lunar surface view factor) provides better heat rejection than the radiator used for the orbiting space station. This is because the provided thermal coating on the radiator ($\alpha_s/\epsilon = 0.05/0.84$) results in an effective sink temperature of $\sim 370^{\circ}$R, while the stipulated orbital sink temperature was 400°R.

The precaution to be used in using a low temperature radiator is to select a shelter site where the radiator space oriented surface will not "see" any of the Lunar surface. The lunar surface approaches $710^{o}R$ during the day and will (a) reject heat to a LTR, and (b) block potential space radiation. The vertical fin radiator used as a heat rejector for higher effective temperatures only partially overcomes this problem by shielding some of the lunar surface and rejecting heat from both sides. Where the lunar shelter site can not be chosen or where it varies such as with a mobile vehicle, it is possible that a method of raising effective radiator temperatures (such as the AR or HP systems) would be used to assure adequate heat rejection.

11.10.3.2 Heat Pump Optimization

The heat pump optimization of the radiator weight and station power carried out in Section 11.5 dropped two terms for calculation simplification. For the special case where the condenser and evaporator temperatures are very close together, such as considered in the shelter integration, this simplification predicts the optimum temperature slightly low. However, choice of the condenser temperature as $T_{effective}$ tends to negate this bias.

11.10.3.3 Life Support System Radiators Combination

The liquid loop Life Support radiators can operate at a higher temperature ($T_{eff} = 68^{o}F$). In the interests of clarity, these two were combined with the air cooling loop with a T_{eff} of $43^{o}F$ to allow a single calculation for each heat rejection method. This results in a slightly conservative total radiator weight, but it does not affect the comparison of methods.

11.10.3.4 Power System Weight Penalties

The specific weight of the power systems was taken as 350 lb/kw for the Mercury Rankine System and 700 lb/kw for the Brayton System. These were also used in the HP plot in Section 11.5. The resulting specific weights for gross generator outputs, 349 lb/kw and 670 lb/kw are sufficiently close such that the assumed numbers provide a reliable comparison.

11.10.3.5 Systems Comparison

Under the guidelines used, the Integrated Mercury Rankine power system at 1609 pounds is 1380 pounds lighter than the Integrated Brayton Power system. However, this difference is not very great when the approximately 1100 pound weight savings for a liquid cooled Brayton Isotope source is considered. It is suggested that future consideration of shelter power systems include a liquid loop for cooling Brayton Cycle thermal energy sources. Future work should also include:

a. Multiple stage refrigeration systems for use with adverse terrain.

b. Radiator configuration studies, integral structure with cabin walls (especially power system radiators).

c. Heat rejection system weight optimizations to determine best lunar fin efficiencies.

11.11 REFERENCES FOR SECTION 11

11-1. Apollo Logistics Support System Payloads, The Boeing Company Document D2-36072-3, April, 1965, Volume 1.

11-2. Apollo Logistics Support System Payloads, The Boeing Company Document D2-36072-3, April, 1965, Volume 2.

11-3. Pettit, E. and Nicholson, S.G., "Lunar Radiation and Temperature," Ap. J 71, 1930.

11-4. Sytinskaya, as reported in "Natural Environment of the Moon," N.T. Bobrovnikoff, J'59, AD 242177 Tech. Note 847-3.

11-5. Johnson, R.W. "Lunar Surface Characteristics and Problems," Presented to the 13th Annual Soil Conference, Univ. of Minnesota, Minneapolis, Minn. April 15, 1965.

11-6. Baldwin, R.B., "The Faces of the Moon," U. of Chicago Press, 1949.

11-7. "ALSS PAYLOADS," Bendix Corporation, Final Presentation Summary for NASA Contract NAS 8-11287, 12 April, 1965.

11-8. "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 SS 3243-3 (Volume III Subsystem Studies).

11-9. Costello, F.E., "A Note on the Optimunization of Vapor Compressor Heat Pump Systems," G.E. Document No. PIR-ATE-117-079, 9/1/61.

11-10. Thermal Integration of Electrical Power and Life Support Systems for Manned Space Stations, Final Report for Contract NAS 3-2799, Issued in preliminary form as G.E. Report 65SD226.

11-11. Aero Space Applied Thermodynamics Manual, Society of Automotive Engineers, Technical Division, New York, N.Y., Revised Edition January 1962.

11-12. Air Conditioning and Refrigeration Data Book, The American Society of Refrigerating Engineers, New York, N.Y., 1955 Edition.

11-13. Gas Tables, Joseph H. Keenan and Joseph Kaye, Massachusetts Institute of Technology, John Wiley and Sons, Inc., Sixth Printing October 1957.

11-14. Status Review of Sunflower, PIC-SOL209/4, Section B-1, TRW, Inc., 24 September 1963.

11-15. Sunflower Status and Application Considerations, Bulletin 311-MRD-3, TRW, Inc., November 1963.

APPENDIX A
LIFE SUPPORT SYSTEM ANALYSIS

A.1 INTRODUCTION

Long duration manned space missions, such as required for the establishment of lunar bases or for interplanetary travel, would imposed prohibitive size and weight requirements on the space vehicle if the vital life support functions were maintained solely by the continuous depletion of on-board stored materials. For example, approximately eight tons of drinking water and two tons of oxygen are consumed by six men during a one-year period.

Recovery and reuse of water and oxygen in a semiclosed loop life support system provides a significant reduction in vehicle launch weight and physical size. However, the recovery of water and oxygen as well as the management of other life support functions require the expenditure of thermal and/or electrical energy.

The advent of large solar and nuclear electrical power systems for the future space vehicles greatly enhances the prospects for long duration manned space flights. These systems collect large quantities of thermal energy utilizing a portion to generate electricity, and rejecting the unused portion as waste heat. The waste heat rejected in large quantities from the space radiator becomes a convenient direct source of energy when integrated with life support systems.

This section describes the basic life support functions during a prolonged space flight, analyzes the process endothermic power requirements for each function and discusses and evaluates the possibility and benefits of thermal integration of the processes with the waste heat source.

A. 2 STATION CONFIGURATION

The space station basic configuration is of the MORL (Manned Orbiting Research Laboratory) type, 260 inches in diameter with a docking hanger and airlocks. The crew cabin is spherical in shape and is divided into three sections, a living quarters, a laboratory for experiments, and a centrifuge which periodically reconditions the crew to a normal gravity environment. The station is designed for a six-man crew for a one-year mission with 180 days resupply.

The life support design parameters are:

a. Cabin Atmosphere

Total pressure	-	7 psia
Oxygen partial pressure	-	160 mm Hg
Carbon dioxide partial pressure	-	3.8 mm Hg
Nitrogen partial pressure	-	Dilueuent
Relative humidity	-	50%
Nominal dry bulb temperature	-	$72^{\circ}F$
Ventilation rate	-	35 cfm/man
Cabin atmosphere volume	-	$3470\ ft^3$

b. Metabolic Data

Oxygen consumption	-	1.87 lb/man-day
Water allowance	-	7.72 lb/man-day
Food (dry)	-	1.38 lb/man-day
Carbon dioxide output	-	2.32 lb/man-day
Urine water output	-	3.30 lb/man-day
Fecal water output	-	0.25 lb/man-day
Urine and fecal solids output	-	0.22 lb/man-day

Evaporative water loss (respiration and perspiration) Metabolic water	-	0.72 lb/man-day
Latent and sensible heat output	-	11,112 Btu/man-day
Wash water	-	6.0 lb/man-day

Since this section deals primarily with the utilization of waste heat, only the endothermic power requirements of the five major life support systems are considered in detail. The systems discussed are:

a. Oxygen recovery from carbon dioxide by the Sabatier method.
b. Water recovery from urine by distillation and vapor pyrolysis.
c. Wash water recovery by distillation.
d. Food management utilizing dehydrated foods.
e. Solid waste management by an automatic process.

Each discussion concludes with a summary of the endothermic power requirements at the various temperature ranges. The endothermic power requirements are than tabulated and the advantages and disadvantages of utilizing waste heat for these functions are enumerated.

A.3 OXYGEN RECOVERY

Oxygen recovery from carbon dioxide is a complicated, but logistically necessary process for extended mission space vehicles with large crews. There are several workable methods for oxygen recovery; however, one of the more promising processes utilizes the Sabatier reaction. The process is accomplished in four distinct phases:

a. Collection of Carbon Dioxide from the Cabin Atmosphere
b. Sabatier Reaction

$$\left.\begin{array}{l} CO_2 + H_2 \rightarrow CO + H_2O \\ \\ CO + 3H_2 \rightarrow H_2O + CH_4 \end{array}\right\} CO_2 + 4H_2 \rightarrow CH_4 + 2H_2O$$

c. Water Electrolysis

$2H_2O \rightarrow 2H_2 + O_2$

d. Catalytic Pyrolyzation of Methane

$CH_4 \rightarrow 2H_2 + C$

A.3.1 CARBON DIOXIDE COLLECTION

The carbon dioxide generated by the crew must be separated from the cabin atmosphere and concentrated prior to insertion into the Sabatier Reactor. Several collection methods have been or are being developed. Prominent among these is a regenerable adsorption process which is presently designed to jettison the carbon dioxide to space. Modifications to this state-of-the-art system will permit collection of the carbon dioxide for use in the Sabatier Reaction. A schematic flow diagram of the CO_2 concentration process is presented in Figure A-1.

The system consists of two sets of canisters alternately adsorbing and desorbing. During the adscrption cycle, a small by-pass of air from the cabin environmental control system is passed through an adsorption canister where all water vapor is removed, thus drying the air to a dew point of-75°F. The amount of water removed by the desiccant is 0.71 lb/hr. The desiccant used for this purpose is a molecular sieve or silica gel.

The dry air is then passed through the second molecular sieve where the carbon dioxide is removed from the air stream by adsorption of the sieve material. This is an entirely exothermic cycle requiring removal of the heat of reaction, due to the condensation of water vapors and CO_2 adsorption. It is calculated that in order to remove the 0.562 lb/hr of CO_2 generated by the six-man crew, and to maintain the partial pressure of the CO_2 within the design limits, a by-pass flow of 20 cfm is sufficient.

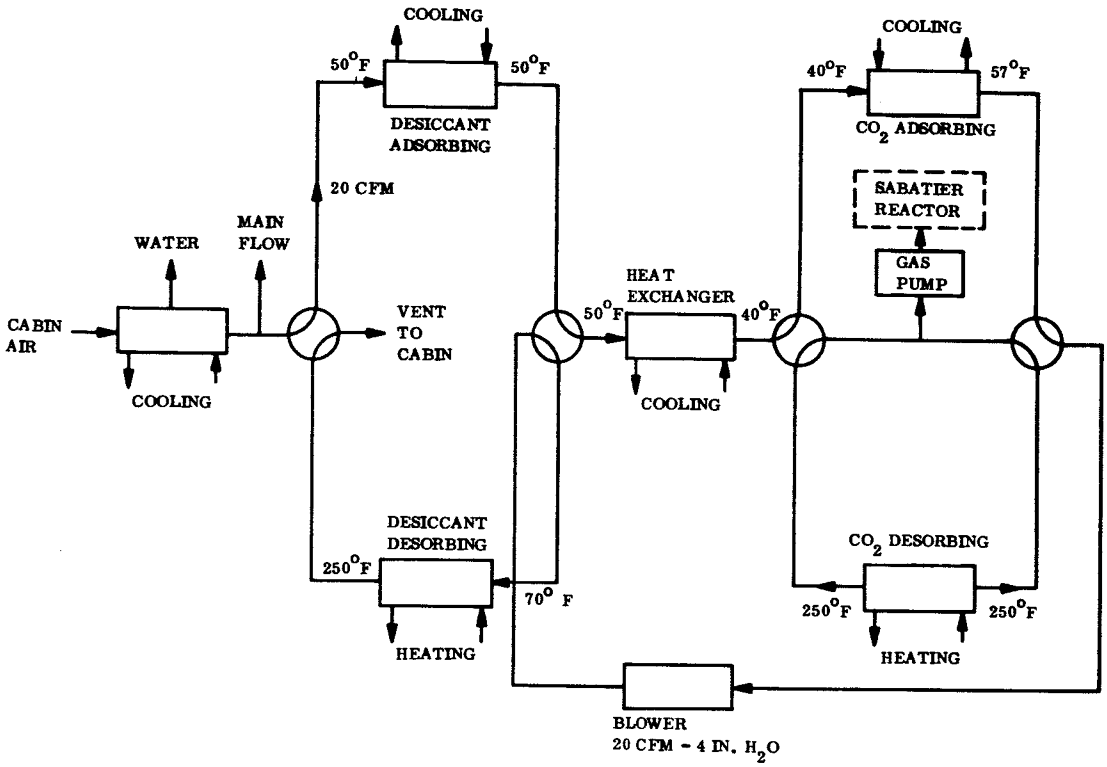


Figure A-1. Carbon Dioxide Concentration

On the desorption cycle, the dried and purified air is circulated back through the first molecular sieve (or silica gel) canister where by heating the canister, the water vapor is returned to the air stream. Independently and concurrently CO_2 is being driven out from the second molecular sieve by the addition of heat. Both desorption processes are then endothermic. The heat required to desorb the desiccant canister to a low residual water content is 4168 Btu/hr. Of this amount, 1988 Btu/hr are required to heat the 20 cfm of air to 250°F, 900 Btu/hr are required to raise the mass of the canister to the same temperature and 1280 Btu/hr is the heat of desorption for the 0.71 pound of water in the canister.

The desorption of the CO_2 canister requires 2180 Btu/hr of which 2000 Btu/hr are required to heat up the canister and 180 Btu/hr is the heat of reaction. The amount of energy required to heat the residual air in the canister is negligible. The process temperature for the CO_2 canister desorbing is 250°F. This is a minimum temperature. Higher temperatures up to 600°F would improve the system performance and increase the desorption rate for both the water vapor and CO_2 canisters.

The total endothermic power requirements for this process are:

Item	Operating Temperature Range (^{o}F)	Endothermic Power Requirement (Watts)
Carbon Dioxide Collection		
Desiccant Desorbing	250-600	1225 (@250^{o}F)
Sieve Desorbing	250-600	642 (@250^{o}F)
	TOTAL	1867

A.3.2 SABATIER REACTION

The collected carbon dioxide is compressed into a storage and surge tank to provide a continuous metered flow to the Sabatier Reactor. The Sabatier reaction mechanism takes place in two steps. First the carbon dioxide is hydrogenated to form carbon monoxide and water:

$$CO_2 + H_2 \rightarrow CO + CH_4$$

Secondly the carbon monoxide is hydrogenated to form methane and water:

$$CO + 3H_2 \rightarrow H_2O + CH_4$$

Combining the steps given the total reaction:

$$CO_2 + 4H_2 \rightarrow CH_4 + 2H_2O$$

A thermodynamic analysis of the reaction shows the reaction to be exothermic so that for each gram mole of carbon dioxide consumed in the reactor, 156.4 Btu/hr are evolved or 1610 Btu/lb of CO_2. Thus, for a continuously operating reactor processing 0.6 pound of CO_2 per hour, approximately 1000 Btu/hr must be removed from the reactor.

The react ion will take place at approximately 15 psia in the presence of a catalyst. The catalyst material and reaction chamber temperature are critical.

A nickel catalyst at a 550°F chamber temperature has proven to have good operational characteristics for the reaction. The weight of nickel catalyst required for the reaction is a function of operating temperature and carbon dioxide flow rate. For the previously described system, approximately 1,72 pounds of catalyst are required. Also, the catalyst is consumed at a rate of approximately 0.027 pound per day, consequently a method of replacing the catalyst or a large catalyst bed must be provided for extended missions. The resulting heated gases (methane and water) which exit from the reaction can be in heat exchange contact with the cooler incoming carbon dioxide. Consequently, heat is exchanged and the reaction is self-supporting. Initial start-up of the system requires a heater to boost the incoming gas temperature to begin the reaction. The methane and water gases are separated by condensation of the water. Each compound is then ready for further processing, electrolysis of water and pyrolyzation of methane.

A. 3.3 WATER ELECTROLYSIS

Water electrolysis may be accomplished by a reverse fuel-cell process.

$$2H_2O \rightarrow 2H_2 + O_2$$

The heat of dissociation of liquid water into gaseous hydrogen and oxygen is endothermic and amounts to 271.5 Btu/gram mole (liquid). The 6.2 gram moles of carbon dioxide which enter the Sabatier reactor per hour produce 12.4 gram moles or approximately 0.5 pounds of water per hour. The electrolysis cell will thus, theoretically consume 271.5 Btu/gram mole x 12.4 gram moles/hr or 3365 Btu/hr or nearly one kilowatt of electrical power.

A functional schematic of a single electrolysis cell is shown in Figure A-2. The unit is an ion-exchange type electrolysis cell developed by General Electric for fuel cell applications, operating at 1.75 volts per cell. A multi-cell stack will be required to provide the desired flow characteristics. The water supplied from the condenser is stored in an accumulator with a liquid electrolyte (sulfuric acid). The resulting solution is maintained in the chamber between two membrane electrodes. When electrical power is applied to the ion-exchange membrane, the water in solution is dissociated such that oxygen ions collect at the positive membrane/electrode. The pure gases are thus formed on the far side of the membrane where the oxygen is vented to the cabin atmosphere and the hydrogen is returned to the Sabatier reactor.

The electrolyte does not permeate the membrane or enter into a dissociation; thus the concentration is not affected.

The most important characteristics of the electrolysis unit are its high efficiency and the inherent capability to separate the gases drom the liquid even in zero gravity. Of the 0.5 pound of water dissociated, approximately 0.44 pound of oxygen and 0.06 pound of hydrogen are produced per hour.

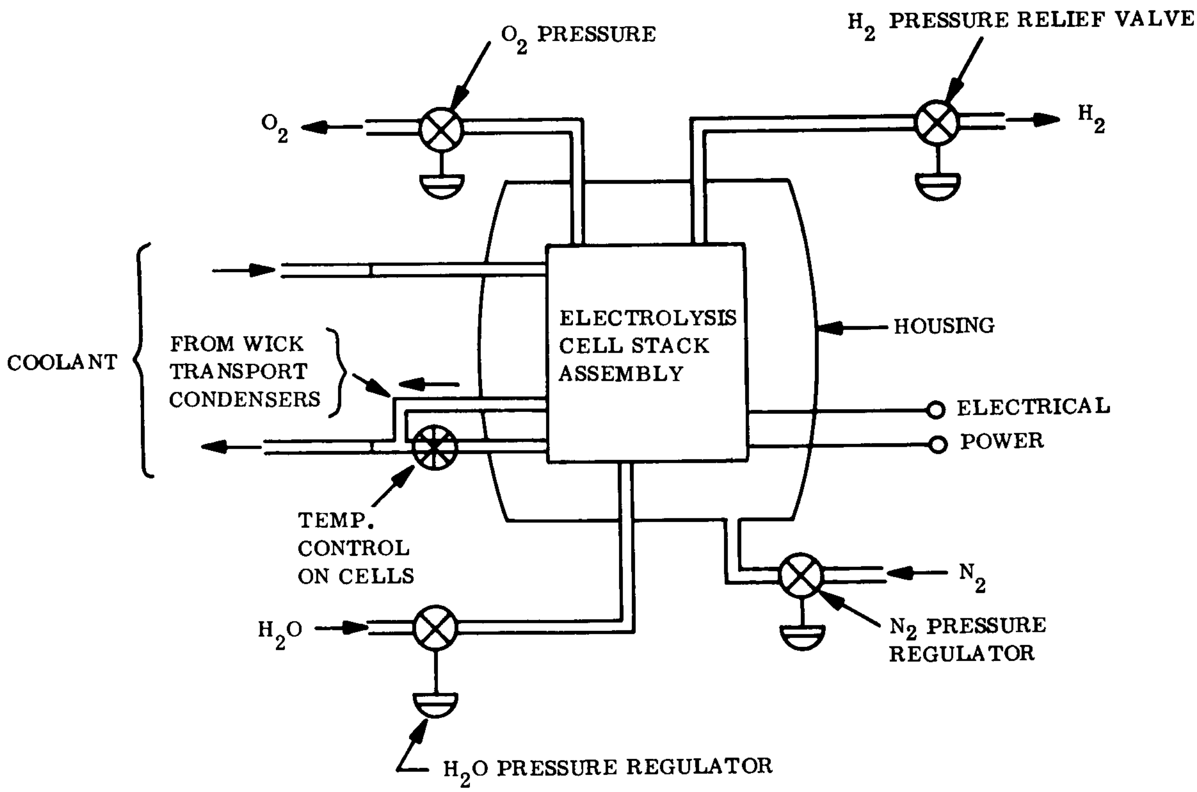


Figure A-2. Water Electrolysis Subsystem, Functional Schematic

A.3.4 CATALYTIC PYROLYZATION OF METHANE

The methane that is separated from the water vapor after the Sabatier reaction must be decomposed to recover sufficient hydrogen to close the cycle. The carbon is stored or jettisoned to space. The rather simple expression:

$$CH_4 \rightarrow 2H_2 + C$$

represents the most difficult process in the oxygen recovery system. Consequently, the penalties of recovering the hydrogen versus merely jettisoning of the methane must be traded off for various mission lengths. Jettisoning of the methane would require storage of 0.06 pound of hydrogen (same as recovered from water) for each hour of operation of the Sabatier Reactor. This amounts to 1.44 pounds per day or approximately 526 pounds per year.

Cracking of the methane is accomplished by pyrolyzation at 1800–2200°F without a catalyst and 1500–1800°F with a catalyst. The main problems are the high temperatures of operation and the removal of the carbon from the reactor and from the catalyst, if used. The higher reactor temperature presents a problem of construction materials. A scraping method, or equal, is required to remove the hard carbon scales from the walls. This amounts to 0.1635 pound of carbon produced per hour or 1432 pounds per year. Similar problems are encountered when a catalyst is utilized since the carbon inactivates the catalyst by reducing surface area or possibly even combining with the catalytic material. Consequently, the catalyst must periodically be replaced. The heat of dissociation of methane is endothermic (71 Btu/mole or 2010 Btu/lb). This amounts to 440 Btu/hr or 130 watts for the process flow rate of 6.2 moles per hour.

The four integrated phases of the total system for oxygen recovery from carbon dioxide are illustrated in Figure A-3 along with part of the cabin environmental control system. The process power requirements (endothermic only) are:

Item	Operating Temperature Range (°F)	Endothermic Power Requirement (Watts)
Carbon Dioxide Collection		
Desiccant Desorbing	250-600	1225 @250°F
Sieve Desorbing	250-600	642 @250°F
Pyrolyzation of Methane	1800-2200	130
	TOTAL	1997

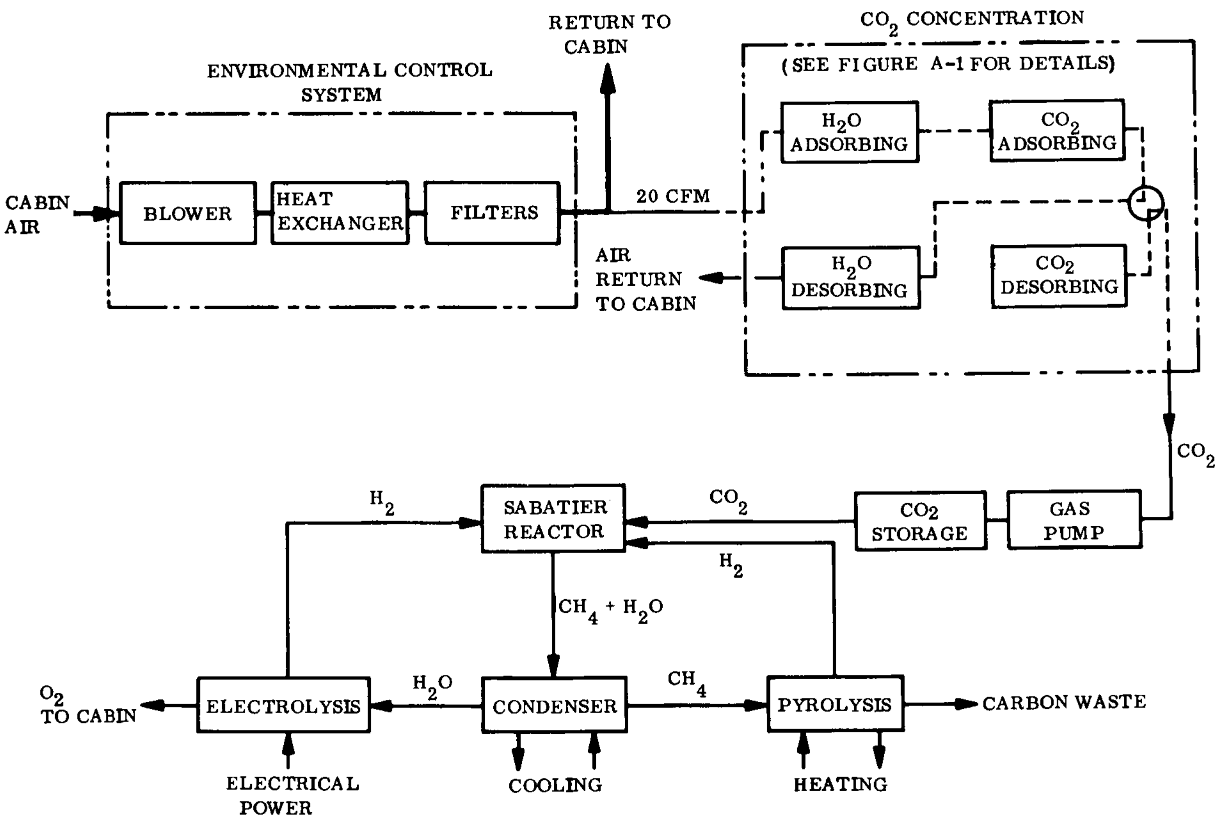


Figure A-3. Oxygen Recovery from Carbon Dioxide

A.4 WATER RECOVERY FROM URINE

The recovery of potable water from urine represents one of the more significant methods of reducing storage material. A possible daily water balance for each crew member is given in Figure A-4. Considering that all the respiration and perspiration water collected by the vehicle environmental control system is recovered, an additional source of 2.33 pounds of water per man day must be supplied for drinking and food preparation. This would amount to 5150 pounds of water for a six-man, one-year mission. Obviously, the launch weight and storage resupply logistics even for this type of a partially open system cannot be tolerated. The water deficiency can be more easily and efficiently overcome by the recovery of water contained in the urine.

The 3.3 pounds-man/day of water in the urine represents a source of potable water which along with the recovered respiration and perspiration water will more than fulfill the needs for drinking and food preparation. In fact an excess of water is shown (Figure A-4) since the water stored in the food adds to the system balance and the metabolic water produced by the men more than equals the water losses in waste solids, cabin leakage and system vacuum vents.

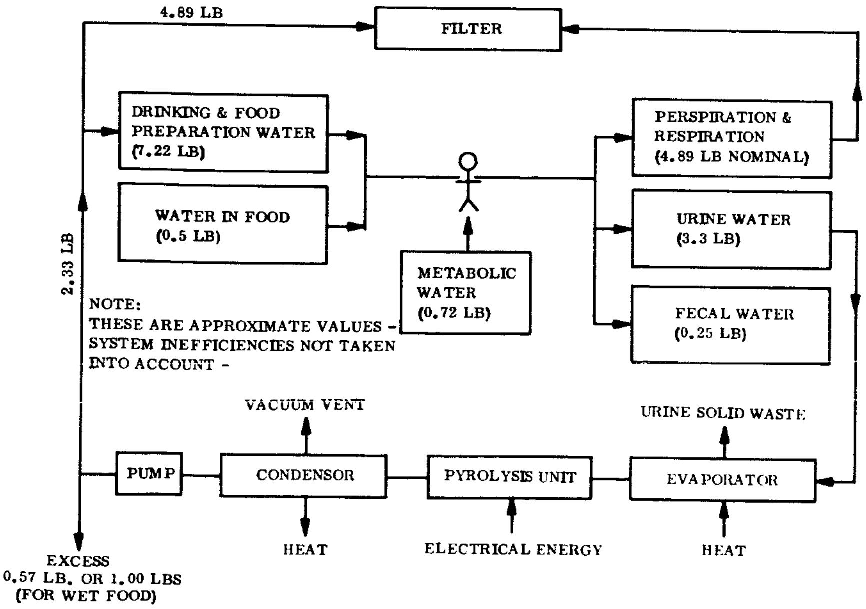


Figure A-4. Water Recovery from Urine

A.4.1 SYSTEM DESCRIPTION

The quality of the water utilized for drinking and food preparation is of prime importance since the same water may be ingested several hundred times during the mission.

Such high quality water can be obtained efficiently and reliably from urine by the method of distillation and vapor pyrolysis. A typical analysis of reclaimed water by the vapor pyrolysis process is shown in Table A-1. This water was recovered from a month-old fecal-urine slurry and was analyzed by an independent laboratory. The points analyzed were adjudged to be well within the chemical standards established by the U.S. Public Health Service. Also, bacteriological tests for members of the Coliform Group and Streptococcus were negative.

TABLE A-1. WATER ANALYSIS OF PYROLYZED WATER FROM METABOLIC WASTES*

Ammonia as N, ppm	2.0
Phenolphthalein Alkalinity as Ca Co3, ppm	0.0
Methyl Orange Alkalinity as Ca CO3, ppm	6.0
pH	6.7
Specific Conductance Micromhos 18°C	17.0
Specific Conductance Micromhos 18°C (corrected)	2.5
Nitrite as N, ppm	0.19
Nitrate as N, ppm	0.0
Odor	None
Phenol, ppb	0.0

*Part of the Hydro John Waste Management System as developed by General Electric under contract NAS 9-1301. Analysis by Betz Laboratories, Inc., April 20, 1964, based on sample standards established by the United States Public Health Service.

The distillation and pyrolysis system requires no chemical additives or filtering processes as illustrated in Figure A-5. Therefore, for a given crew size, the system weight penalty for water reclamation does not increase appreciably as the mission length increases. Urine is added to an evaporator either directly from the source or from an intermediate storage container. The 21 pounds of urine per day will contain approximately one pound of solid waste.

After an initial batch of waste liquid enters the evaporator, the internal pressure of the evaporator is reduced to approximately 1.7 psia and sufficient thermal energy is transferred to the liquid from a hot fluid heat transport medium (waste heat) source to cause boiling at approximately 120°F. An average boiling rate of 0.83 pound of water per hour will require approximately 830 Btu/hr waste heat transferred to the evaporator. The water vapor is then passed through a vapor pyrolysis unit which heats the vapor to 1800°F and oxidizes the water vapor impurities in the presence of a catalyst. The oxygen which combines with the impurities is bled into the evaporator at a rate of approximately six atmospheric cubic centimeters per minute.

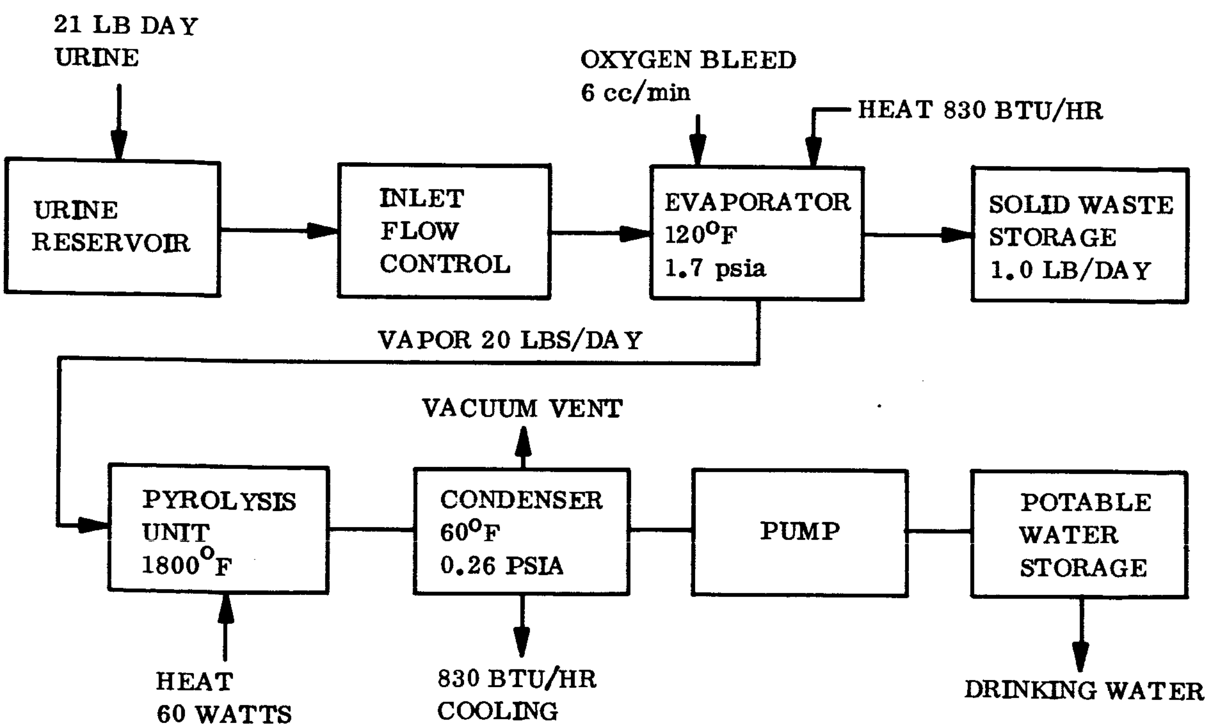


Figure A-5. Urine Water Recovery System

The vapor pyrolysis unit contains a counter flow heat exchanger and is jacketed with superinsulation to minimize heat losses.

The overall energy requirements for the vapor pyrolysis process is 72 watt-hours per pound of water recovered or 60 watts continuous. The pyrolyzed vapor is then liquidified in a condenser while the gaseous noncondensables (impurities) are vented to space vacuum. A cold fluid liquid heat transport medium provides the sink to remove approximately 830 Btu/hr for condensation. A pump periodically removes the potable condensate to storage.

The only expendables of the system are the gases lost through the vacuum vent pressure control. This amounts to less than 20 pounds per year.

The endothermic power requirement for this process in then:

Item	Operating Temperature Range ($^{\circ}$F)	Endothermic Power Requirement (Watt)
Evaporator	100-120	243
Pyrolysis	1800	60
	TOTAL	303

Waste water is defined as the water which has been used to cleanse the bodies of the crew members (showers, washing, etc.), washing of clothing, eating utensils, and food preparation surface areas. The resulting liquid will contain many of the impurities commonly found in urine, such as urea, chlorine, calcium, etc., although at a decreased concentration. It is considered that benzalkonium chloride, a washing compound, will be mixed with the water prior to use.

An assumed break-down of the wash water requirements is given below:

Hand and face washing	1 lb/man day
Showering	3 lb/man day accumulated over 3 days
Eating utensils cleansing	1 lb/man day
Washing of clothing	1 lb/man day accumulated over 6 days
Total	6 lb/man day

Based on the above table the men will shower once every three days and will wash their clothing once every 6 days. These water requirements may be reduced by the use of wash-dry cleansing pads and expendable clothing. However, since a system must be provided for the recovery of shower water, no appreciable advantage is seen by the use of the expendable cleansing pads and clothing. Also, the weight of expendable clothing is quite high over a year long mission exceeding 110 pounds for the six men.

A.5.1 SYSTEM DESCRIPTION

The system utilized for the recovery of waste water is very similar to the system for the recovery of urine water except that pyrolyzation of the water vapor may not be required. However, the impurities may accumulate over a long period to an intolerable level. Thus, pre-or post-treatment of the liquid may be required or a portion or all the distilled waste water vapor may be pyrolyzed. Commensurate with the philosophy of maintaining expendable materials to a minimum and obtaining optimum utilization of equipment already existing in the system, the pyrolysis unit of the water recovery system is utilized to maintain the wash water at an acceptable purity level by purifying only a small portion of the vapor flow.

* This is due to the fact that the amount of dissolved impurities in the water after distillation is much lower than that found in the urine distillate so that continuous full flow pyrolyzation is not required.

A block diagram of the system would look much like the one shown in Figure A-5.

In a manner similar to the urine recovery system, the waste water is added to the evaporator either directly from the source or from an intermediate storage container. The approximately 36 pounds of wash water which enter the system per day contain from 0.25 to 1 percent solids. An average boiling rate of 1.5 pounds of water per hour requires approximately 1500 Btu/hr waste heat transferred to the evaporator. The water vapor flow from the evaporator is divided so that a small amount is diverted to the pyrolysis unit of the urine water recovery system. The recombined water vapor flow is liquidified in a condenser. A cold fluid heat transport medium provides the heat sink to remove approximately 1500 Btu/hr for condensation. A pump periodically removes the condensate to the wash water storage vessel where it is mixed with the small diverted liquid flow from the potable water recovery system. The purity of the stored wash water is thus determined by the amount of water by-passed to, and purified by, the pyrolysis unit, and then returned to wash water storage. Experimental data will establish the desired wash water purity, and thus, the required by-pass flow rate. However, this will not significantly affect the total endothermic power requirement. The process endothermic power requirement for the water recovery is then:

Item	Operating Temperature Range (^{o}F)	Endothermic Power Requirements (Watts)
Evaporator	100-200	442

A.6 FOOD MANAGEMENT

Adequate nutrition must possess the quantity and quality of ingredients to maintain the body mind of the human biological machine in its best operating conditions. The essential nutrients in food, such as vitamins, aminoacids, and minerals, can be supplied in the form of tablets, liquids, purees, etc., to satisfy the basic physical requirements. Special low residue diets have been studied to reduce some of the normal body functions, thus simplifying, at least apparently, the problem of supporting man's life during prolonged space flight.

However, studies conducted at the General Electric Company and other major aerospace industry firms indicate that in addition to the nutritive qualities, food must be palatable; and equally important, it must look like real food. The importance of this requirement becomes more obvious when we consider the wide variety of unusual stress imposed on men living in a space station, and the role of food as the only reward in an otherwise drab and routine environment.

Fresh or frozen food, although highly desirable, would pose enormous problems because of additional storage requirements, need for for refrigeration equipment to prevent spoilage, and decreased payload capability. The problem is greatly minimized by the use of dehydrated food, in which the original water content has been reduced to less than 5 percent. Payload savings realized are at least 2.5 lb/man-day. The food itself when reconstituted with 170°F water recovers the texture, flavor, color and palatability of fresh food.

A.6.1 DESCRIPTION

The average solid food requirements for optimum reconstitution per man day has been set at approximately four pounds. Assuming that all food is of the dehydrated type, the four pounds would be made up from 1.0 pound of dehydrated solid and 3.0 pounds of hot water at 180°F. After absorption of the water, helped by kneading, the food must be held at 175°F for approximately five minutes for optimum reconstitution. All food would be precooked and stored in expandable plastic containers capable of receiving the additional volume of reconstituting water.

An additional 2.3 lb/man-day of water and 0.35 lb/ man-day of dehydrated solid are required for beverages. See Figure A-6.

Tests, such as the 30-day, four-man simulated space flight conducted at General Electric MSD during October, 1963 showed that the dehydrated food was highly acceptable and rewarding.

- SOLID FOOD 1.35 LB/DAY
  - 1.0 LB AS FOOD
    - 0.25 LB AS IS
    - .75 LB REHYDRATED + 3.0 LB OF WATER @ 180°F
  - 0.35 LB AS BEVERAGE + 2.3 LB OF WATER
    - 1.0 LB @ ROOM TEMPERATURE
    - 1.3 LB @ 40°F

Figure A-6. Food Requirements

In estimating peak power requirements, it must be considered that man does not like to eat alone and that most likely two of the six men will be together on a "work shift". Assuming that the largest meal may be half of the daily ration and that the heating is done in ten minutes, the peak power requirement would be:

$$P = \frac{2 \text{ lb food}}{\text{man}} \times \frac{2 \text{ men}}{\text{meal}} \times \frac{0.9 \text{ Btu}}{\text{lb } ^{\circ}\text{F}} \times \frac{27^{\circ}\text{F}}{10 \text{ min}} \times \frac{60 \text{ min}}{\text{hr}} = \frac{584 \text{ Btu}}{\text{hr-meal}}$$

or 171 watts/meal.

Requirements for providing water at 40°F for cold beverages are not discussed due to the confinement of this discussion to endothermic power applications.

A. 6.2 STERILIZATION OF UTENSILS

To provide the true feeling of a good meal, the man in a space cabin can use earth-type "silverware" slightly magnetized to stay on a tray when not in use. Special "sticky" sauces* can hold the food to the tray or dish so that meals would be as natural as possible.

* Such as those developed by Libby, McNeil and Libby Food Scientists and successfully used during the GE 30-day, four-man simulated space flight.

This, of course, complicates the cleaning and sterilizing of eating utensils. Some of the dishes, especially when sticky sauces are used could be made from edible material similar to the "bread sheets" used by the early Romans and some modern nomadic people, and consumed at the end of a meal. This would ensure complete consumption of the prepared calories and reduce waste storage.

One pound per man-day of non-potable, clean, hot water is sufficient for cleaning personal eating utensils. The water with some cleaning additives would be contained in plastic bags together with the utensils to be cleaned. The hot water would be coming from the same reservoir which supplied hot water for personal hygiene. It is estimated that an additional five lb/man-day is needed for personal hygiene and clothes washing. The total hot water requirements would then be six lb/man-day. To utilize the same heating system as for the potable water, only half of the amount would be heated and then mixed. The average heat input to bring three pounds of water to $170^{o}F$ is 83 Btu/hr.

Sterilization of eating utensils can be done with $240^{o}F$ superheated steam. A continuous operating sterilizer would require approximately 25 Btu/hr.

Another possible means of sterilizing is by using germicidal ultraviolet lamps. Two lamps of five watts each used intermittently a few times a day would provide good sterilization at a lower power input than required by the hot-steam concept. In addition, the same lamps could be used to sterilize the cabin air if connected to the cabin environmental control system. The disadvantage of this system is that it requires electrical power rather than the available waste heat.

The endothermic power requirement can be summarized as follows:

Item	Operating Temperature (°F)	Average Endothermic Power Requirement (Watts)
Food Preparation	180	27
Food Baking	175	9
Cleaning	170	25
Sterilization	250-300	8
	Total	69

A. 7 SOLID WASTE MANAGEMENT

One of the most difficult problems of extended mission space flights is the management of waste solids. Included in this category are human excreta, worn-out clothing, hair clippings, skin tissue, toe and fingernail clippings, waste paper, hand tissues, expended air filters, etc. In obeisance to the philosophy of noncontamination of space, the vehicle must then become a variable garbage truck in the collection and storage of the solid waste product. The state-of-the-art has not progressed sufficiently to utilize solid waste products to close the ecological cycle of the man in space.

A. 7. 1 SYSTEM DESCRIPTION

Human excreta is of prime concern because of the high concentration of bacteria found in feces. Consequently, the fecal solid waste management must be completely sanitary in the manner of collection, transportation, and storage. Also the system must be psychologically acceptable to the crew over the entire mission duration. A system design which would meet these requirements is shown in Figure A-7. The feces enters the hopper and is carried by air flow (for zero gravity operation) through a transport tube to the pump-blender section. On command, the rectal area is cleansed with warm flush water and dried with warm air.

The flush water mixes with the feces in the pump-blender. The resulting slurry is pumped into a still-pot. The water in the still-pot is boiled at reduced pressure and temperature. A space vacuum port vents non-condensables to provide pressure control. The resulting water vapor flow is divided so that the majority is condensed for re-use as flush water. The small portion of the vapor is passed through a pyrolysis unit where the water impurities are oxidized and the resulting water is returned to storage to dilute the flush water contaminant concentration.

Mixing of the stool with water provides self-cleaning features; also, since the water is initially utilized to cleanse the rectal area after defecation, the use of toilet tissue is eliminated. The water utilized for flushing and cleansing is separated from the solid waste by distillation and is condensed for re-use as flush water. The solid sludge is periodically removed from the evaporator, thus minimizing the production of ammonia and organic gases. The solid wastes are stored in large porous bags and are vacuum dried to inhibit bacterial growth. Also, a small amount of ammonia is generated in the fecal slurry and is dissolved in the condensed water. This ammonia is sufficient to sanitize the unit during flushing, but is of sufficiently low concentration so as not to irritate the human skin.

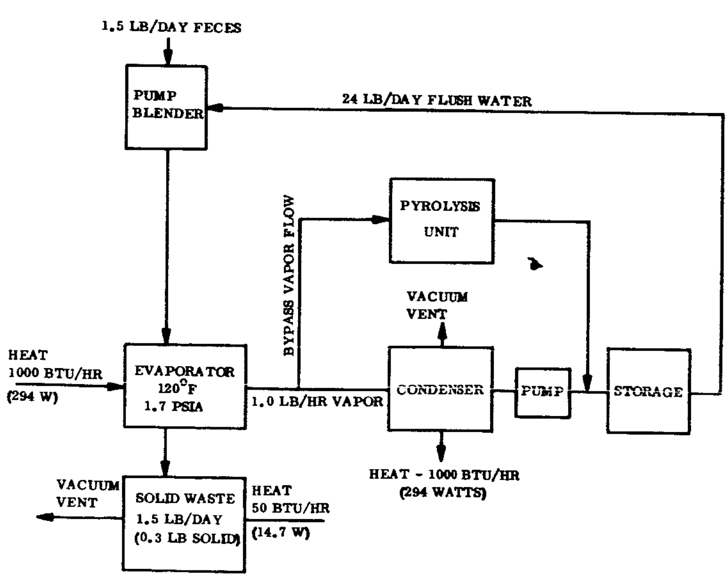


Figure A-7. Fecal Solid Waste Management System

Approximately 0.25 pound of feces is excreted per man day and approximately four pounds of water per flush is required to cleanse the rectal area and to sanitize the unit.

The vaporization and pyrolyzation processes are again similar to those of the urine water recovery system. Approximately 1000 Btu/hr or 294 watts of waste heat is required for the vaporization of approximately one pound of water per hour. The by-pass is purified of ammonia and organics and utilized to dilute the flush water to an acceptable purity level in the storage reservoir. In this manner, pre-or post-treatment of the flush water is eliminated and expendable filters are not required. The condenser heat sink is a liquid heat transport medium which removes approximately 1000 Btu/hr or 294 watts to condense the water vapor. A small amount of ammonia and organics are also carried over to the condensate. The ammonia acts as a disinfectant so that the flush water also sanitizes the system.

The evaporator and heat exchanger surface is periodically cleaned when the fecal sludge is expelled. This assures a high heat transfer coefficient throughout the mission duration. The expelled fecal sludge is forced into a porous bag and the bag is subjected to low pressure and heating from a radiative waste heat source to cause drying. The 3.0 pounds of fecal sludge collected per day contains approximately 1.5 pounds of water. As shown in Figure A-4, there is an excess of water in the system. This excess is produced by the men at a rate of 4.3 pounds per day and is contained in the food at a rate of 3.0 pounds per day. Thus, even if 1.5 pounds of fecal water is jettisoned to space, 5.8 pounds of excess water, not counting recovery process inefficiencies, is produced per day. This excess will more than make up for system inefficiencies and losses due to cabin leakage.

The entrained water is removed from the fecal sludge by heating at a rate of 50 Btu/hr or 15.0 watts. The sludge is thus dried so that bacterial growth is inhibited. The 0.3 pound of of dried feces will still have a volume approximately the same as the fecal sludge. This amounts to nearly a nine-cubic-foot volume per year. The dried feces may be compressed to the density of water, minimizing the storage volume to approximately 1.75 cubic feet per year.

A.7.1 DISPOSAL OF OTHER SOLID WASTE MATERIALS

The urine solids and waste water solids are handled in a similar manner as fecal solids. Thus, they are vacuum dried and stored. It is considered that such items as skin tissue, hair clippings, toe and fingernail clippings are all part of waste water solids.

Such items as expended food containers, worn out clothing waste paper, hand tissues, air filters, etc., will be compressed and bailed in a pneumatically operated press, disinfected, and sealed in plastic bags. Disinfecting the materials is accomplished by heating them in excess of 250°F for several hours. This will also provide a minimum size bundle since most of the material will be thermoplastic, thus they will exude into most of the voids in the bail. A crude estimate of 11 watts of waste heat required is made by estimating that one pound of waste is processed per hour and that the mean specific heat is 0.2 Btu/lb°F.

The endothermic power requirements for the solid waste management can be summarized:

Item	Temperature (°F)	Endothermic Power (Watts)
Evaporator	120	294
Sludge Dehydration	100-200	15
Waste Disinfection	250	11

A.3 SUMMARY OF POWER REQUIREMENTS

A summary of the endothermic power requirements identified in the discussion of the five life support subsystems is shown in Table A-2.

The various energy requirements have been tabulated according to the process temperature. It must be noted that these values do not include system inefficiencies and support power needs such as pumps and control equipment. In addition, the one kilowatt of electrical energy required for the electrolysis of water (see oxygen recovery discussion) has not been included. This is because the purpose of the table is to present a review of the thermodynamic processed of the life support system that theoretically could be integrated with the space station power generating system.

The various power cycles known to have been considered for the six man MORL type station are:

- Solar Mercury Rankine cycle
- Solar and Isotope Brayton cycles

In all cases waste heat in a gaseous or liquid transport medium is available. This fluid could be piped directly through the life support system, were it safe to do so, or utilize by means of proper intermediate heat transfer equipment. The only life support processes that cannot be integrated are the methane and the water vapor pyrolysis which require temperatures much higher than available with the waste heat fluid.

The piping of hot fluid imposes, of course, additional requirements such as circulation pumps, increased take-off weight, increased volume, and increased heat leakage losses. To evaluate quantitatively the advantages of integration, the total requirement of relative weight and power must be calculated for a nonintegrated system where all process energy is obtained directly from the station power supply, and an integrated system where waste energy is utilized instead of electrical energy, whereever possible. The effect of volume increase is negligible. Heat leakage can be considerable especially in the high temperature range and has been taken into account.

A.8.1 NON-INTEGRATED SYSTEM TOTAL REQUIREMENTS

The non-integrated life support system power and weight requirements are based on a preliminary, nonoptimized system design. Two cooling loops with glycol as the heat transport medium are utilized to remove the process heat to a space radiator. One loop

TABLE A-2. ENDOTHERMIC POWER REQUIREMENT

Subsystem	Operation	Range (^oF)	Average Continuous Endothermic power	1800-2200 ^oF	250-600 ^oF	180-170 ^oF	100-120 ^oF
Oxygen Recovery	Desiccant Desorbing	250-600	1225		1225		
	Sieve Desorbing	250-600	642		642		
	Pyrolyzation of Methane	1800-2200	130	130			
Water Recovery	Evaporator	120	243				243
	Pyrolysis	1800	60	60			
Wash Water Recovery	Evaporator	120	442				442
Food Preparation	Food Preparation	130	27			27	
	Baking	175	9			9	
	Cleaning	170	25			25	
	Sterilization	250-300	8		8		
Solid Waste Management	Evaporator	120	294				294
	Sludge Dehydration	100-200	15			15	
	Waste Disinfection	250	11		11		
			3131 Watts	190	1886	76	979

serves the higher temperature oxygen recovery equipment; the other ties into all remaining low temperature processes such as the vapor condenser for the water recovery unit.

A breakdown of the power requirements and weight estimates for this mode of operation is shown in Table A-3. The total non-integrated power requirement is 4566 watts. The total take-off equipment weight is 622 lbs.

A.8.2 INTEGRATED SYSTEM TOTAL REQUIREMENTS

The total power and weight requirement for an integrated life support system are shown side-by-side with the non-integrated system (Table A-3) for easy comparison. It can be seen that the relative additional weight due to the waste heating equipment is negligible when compared to the saving in process power reduction.

The total integrated power requirement is 1601 watts. The total weight requirement is 678 pounds. The equipment weight is based on a 600°F waste heat fluid temperature and will vary according to the temperature of the fluid. It is estimated that an additional 102 pounds are required where the temperature is reduced to the 400°F temperature range such as in the case of the Brayton cycle.

A.9 CONCLUSION

A comparison of the weight and power requirements arrived at in Table A-3 shows that by integrating the life support system with the space station power generating equipment, a total of 2965 watts can be saved at an estimated weight increase of from 56 to 158 pounds over a non-integrated life support system. The small increase in weight in the life area is more than offset by the relatively large reduction in electrical power requirement so that an overall vehicle system weight saving is achieved. This can be shown by comparing the weight increase in the integrated life support equipment to the weight equivalent of the electrical power saved using an average power penalty of 0.7 lb/watt. The weight equivalent of 2965 watts is 2075 pounds so that in terms of take-off weight the saving that can be realized is 2075-158=1915 pounds.

TABLE A-3. POWER REQUIREMENTS AND WEIGHTS ESTIMATE FOR NON-INTEGRATED AND INTEGRATED LIFE SUPPORT SYSTEMS

SYSTEM	NON-INTEGRATED Weight lbs.	NON-INTEGRATED Process Power Watts	NON-INTEGRATED Support Power Watts	INTEGRATED Weight lbs.	INTEGRATED Process Electric Power Watts	INTEGRATED Process Thermal Power Watts
Oxygen Recovery						
CO_2 Recovery Canisters	120	1867	40	120	40	1867
CO_2 Cannister System Heat Leakage	-	-	-	-	-	351
Electrolysis Cells	100	1000	-	100	1000	-
Pyrolyzation Unit	20	130	-	20	130	
CO_2 Accumulator Pump	7	-	100	7	100	-
Accumulator, Reactor Condenser	26	-	-	26	-	-
System Cooling	100	-	70	100	70	-
Heat Leakage (Loss)	-	-	42	-	-	42
Misc. Hardware	12	-	7	12	7	-
Fluid Heating System	-	-	-	40	93	-
TOTALS	385	2997	259	425	1440	2260
Urine Water Recovery	43	303	7	43	67	243
Wash Water Recovery	35	442	7	35	7	442
Food-Management	19	69	-	19	-	69
Solid Waste Treatment	29	320	7	29	7	320
Heat Leakage (Loss)	-	-	95	-	-	95
Cooling System	106	-	60	106	40	-
Misc. Hardware	5	-	-	5	-	-
Fluid Heating System	-	-	-	16	40	-
TOTALS	237	1134	176	253	161	1169

NONINTEGRATED LIFE SUPPORT SYSTEMS

WEIGHT = 622 LB

POWER REQUIREMENT = 4566 WATTS

INTEGRATED SYSTEMS	RANKINE SYSTEM	BRAYTON SYSTEM
WEIGHT	678 LB	780 LB
ELECTRIC POWER REQUIREMENTS	1601 WATTS	1601 WATTS
THERMAL POWER REQUIREMENTS	3429 WATTS	3872 WATTS

The ultimate meaning of an integrated power system is that it makes available for experiments over 60 percent of the energy which would otherwise be used for life support.

In terms of power for a space station such as the one considered in this report, the emphasis of the mission is thus shifted from the mere support of a crew in space to the efficient manned exploration of space.

APPENDIX B

MORL ATTITUDE CONTROL REQUIREMENTS CALCULATIONS

B.1 MORL MOMENTS OF INERTIA

From Reference B-1, $\Delta V = 7$ fps

where $F = 800$ lb, $t = 10.66$ sec

To find vehicle mass,

$$\int F \, dt = \int m \, dv$$

For constant force and acceleration,

$$F \Delta t = m \Delta V$$

$$800 (10.66) = 7 m$$

$$\underline{m = 1220 \text{ slugs}, \ W = 1220g, \ 39,300 \text{ lb}}$$

MORL Diameter = 260 inches

MORL Length = 531 inches

Assume radii of gyration of 8 ft, 15 ft, 15 ft for roll, yaw, pitch.

Then $I_{Roll} = m \bar{r}^2 = 1220 (64) = \underline{78,000 \text{ slug-ft}^2}$

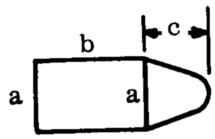
I_{Pitch} and $I_{Yaw} = m \bar{r}^2 = 1220 (225) = \underline{275,000 \text{ slug-ft}^2}$

B.2 DRAG CONSIDERATIONS

Minimum vehicle area = $(21.7)^2 \frac{\pi}{4} = 370 \text{ ft}^2$

Side Area $\approx$

$$\overset{b}{\left(\frac{321}{12}\right)} \quad \overset{a}{21.7} + \overset{a/2}{\frac{21.7}{2}} \quad \overset{c}{\left(\frac{210}{12}\right)}$$



$$\overset{b}{26.75} \ \overset{a}{(21.7)} + \overset{a/2}{\frac{21.7}{2}} \quad \overset{c}{(17.5)}$$

$$580 + 317 = 897 \text{ ft}^2$$

Sun oriented average area = $370 + \left(\frac{897-370}{2}\right) = 634 \text{ ft}^2$

Drag = 0.73 lb/year/sq ft @ 250 nm for $I_{sp} = 300$
(See Figure 6-2.)
Drag weight, earth oriented = 270 lb/year
sun oriented = 463 lb/year

For solar dish, area dish = $(30)^2 \frac{\pi}{4} = 710 \text{ ft}^2$

Average = $710 + \left(\frac{897-710}{2}\right) = 803.5 \text{ ft}^2$

Solar paddles, area = 2180 ft^2

Average = $897 + \left(\frac{2180+370-897}{2}\right) = 1723 \text{ ft}^2$

Drag Weight, concentrator = (803.5) 0.73 = 586 lb/year

Solar paddles = (1723) (0.73) = 1258 lb/year

B.3 LIMIT CYCLE CONTROL

Scale from MOL type properties

	I_{sp}	I_R	$I_{Y,P}$	Roll Arm (ft)	Pitch, Yaw Arm (ft)
MOL Type	140	9000	126,000	10	25
MORL	250	78,000	275,000	22	40
Factors	0.56	8.65	2.18	0.455	0.625

Engine sizes are 25 lb pitch, yaw for MOL,

60 lb pitch, yaw for MORL such that cycle rates will be similar.

25 lb roll for MOL

100 lb roll for MORL (Relax roll constraints slightly).

The additional impulse for MORL can be found by multiplying the specific impulse, inertia and arm factors and comparing with the MOL data presented in Reference B-2.

Roll control sizing = (8.65) (0.455) = 3.9

Pitch, Yaw control sizing = (2.18) (0.625) = 1.36

If 1/3 of the impulse is allotted to each axis, then scale factor

$$= \underset{I_{sp}}{\underset{\uparrow}{(0.56)}} \left[1/3\ (1.36) + 1/3\ (1.36) + 1/3\ (3.9)\right]$$

$$= (0.56) \left[0.453 + 0.453 + 1.30\right]$$

$$= 0.56\ (2.206) = 1.18\ \frac{\text{MORL requirement}}{\text{MOL requirement}}$$

For 2-degree limit cycle control, 25 lb was used for 30 days. For 365 days, the MORL amount would be $25 \left(\frac{365}{30}\right) 1.18$ equals 359 pounds.

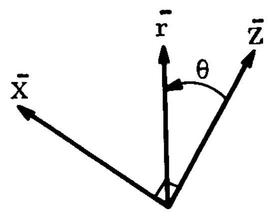
Note that the solar system will have I_{roll} about 60% greater and resultant requirements will be larger by 20% for that case.

B.4 GRAVITY GRADIENT IMPULSE REQUIREMENTS

	I_{Roll} slug-ft^2	$I_{Pitch, Yaw}$ slug-ft^2	ΔI slug-ft^2
Case 1 (Isotope)	78,000	275,000	198×10^3
Case 2 (Concentrator)	83,000	280,000	198×10^3
Case 3 (Solar Paddles)	135,500	332,000	198×10^3

Although the inertias go up the difference is the same, so gravity gradient requirements are the same (neglecting other effects such as products of inertia).

The gravity gradient torque $= \frac{3K}{R^3} r_x r_z (I_{xx} - I_{zz})$



where r is the unit radius vector

x, z are inertial vehicle coordinates

$$\frac{K}{R^3} = \omega_o^2$$

$$r_x r_z = \cos\theta \sin\theta$$

$$= 1/2 \sin 2\theta$$

$$\text{Torque} = \frac{3}{2}\,\omega_o^{\,2}\,(\Delta I)\sin 2\theta$$

$$\text{Average torque} = \int_0^{\pi/2} \frac{T\,d\theta}{\pi/2}$$

For orbit period of 5660 seconds, $\text{torque} = \frac{3}{\pi}\,\omega_o^{\,2}\,(\Delta I) = \frac{3}{\pi}\left(\frac{2\pi}{5660}\right)^2 (198,000) = 0.233$ ft-lb

Impulse/Orbit = 0.233 (5660 seconds) = 1310 ft-lb-sec

Moment Arm = 40 ft

$\frac{1310}{40} = 32.7$ lb-sec/rev

365 x 15 x 32.7 = lb-sec/year = 179,000

$I_{sp} = 250$ sec, weight $= \frac{179,000}{250} = 715$ lbs/year/engine

Two engines $W_{tot} = 1430$ lb/year

B.5 CENTRIFUGE CONSIDERATIONS

Data from Reference B-3

Speed of flywheel = (20.5) (52) (4.32) = 4600 rpm

Speed of flywheel =	(20.5)	(52)	(4.32)	= 4600 rpm
	Rpm Centrifuge	Drive Reduction	Gear Reduction	

Moment of Inertia $\quad I = \frac{m\,r^2}{2} = \left(\frac{133}{32.2}\right)\left(\frac{1.25^2}{2}\right) = 3.22 \text{ slug-ft}^2$

Momentum $\quad I_\omega = 3.22\ (4600) \left(\frac{2\pi}{60}\right) = 1550$ ft-lb-sec

Centrifuge

Moment of Inertia $\quad I = m\,\bar{r}^2 = \frac{I_\omega}{} = \frac{1550}{20.6\ \frac{2\pi}{60}} = 718$ slug ft^2

$$= \frac{170 + 340}{32.2}\ \bar{r}^2 = \frac{510}{32.2}\ \bar{r}^2 \qquad \bar{r} = 6.73 \text{ ft}$$

Majority of inertia is mass and distance of astronauts. For ten-pound mass change, ΔI is 15 slug-ft^2 (2%). For 6-inch radial position change of astronauts, ΔI is about 10%. Assume average I deviation to be 5% from theoretical, or flywheel compensated. This is then 80 ft-lb-sec per start or stop, 160-ft-lb-sec required per day.

Total = 365 (160) = 58,500 ft-lb-sec/year

Station moment arm = 21.7 ft

$$\frac{58,500}{21.7} = 2700 \text{ lb-sec}$$

Two engines, = 5400 lb-sec

= 21.6 lb/year

For re-entry simulation (uncompensated)

$I_{\omega_{cont.}} = 718\ (3)\ 20.5 \left(\frac{2\pi}{60}\right)$ (for 9 gravity re-entry)

= 4660 ft-lb-sec

Assume six astronauts, five rides each, = 30 rides/year

$$\Delta \text{ Imp.} = \left[4660\ (2) - 1550\ (2)\right] = 3110\ (2)$$

$$3110\ (2)\ (30) = 185{,}700 \text{ ft-lb-sec/year}$$

Couple = 21.7 ft

$$\frac{185{,}700}{21.7} = 8610 \text{ lb-sec}$$

Two engines = 17220 lb-sec

$$\frac{17{,}200}{250} = 69 \text{ lbs/year}$$

Centrifuge 9g Time ≈ 1 Hr/Year

Gyro Effort
(Earth Orientation)

$$\Delta I\, \omega = 3110$$

$$C = I\, \omega\, \Omega = \frac{3110\ (2\pi)}{5660} = 3.45 \text{ ft-lb.}$$

Moment arm = 40 ft

Engine force $= \frac{3.45}{40} = 0.0863$ lb

Two engines = $I_{sp} = 250$

Weight Required $= \frac{(0.0863)\ (2)\ (3600)}{250} = 2.5$ lb

For 1g operation, 5% deviation

$$C = 80 \left(\frac{2\pi}{5660} \right)$$

$$= 0.0887 \text{ ft-lb}$$

$$\text{Impulse} = (0.0887)(3600)(365) = 1.165 \times 10^5 \text{ ft-lb-sec}$$

Moment arm = 40 ft

$$\frac{116500}{40} = 2900 \text{ lb. seconds}$$

$$\text{with two engines, fuel} = \frac{5800}{250} = 23.2 \text{ lb}$$

Sun oriented total	Earth oriented total
21.6	21.6
69	69
290 lb	2.5
	23.2
	116.3 or 117 lb

B.6 REFERENCES FOR APPENDIX B

B-1. Report on a System Comparison and Selection Study of a Manned Orbital Research Laboratory, Douglas Report SM-44612, Appendix 16, Propulsion and Reaction Control, September 1963.

B-2. MOL Stabilization Disturbance Effects and Boundaries, General Electric MSD, Internal Memo 4-35, February 2, 1965.

B-3. Report on a System Comparison and Selection Study of a Manned Orbital Research Laboratory, Douglas Report SM-44612, Appendix B, Artificial Gravity Provisions, September 1963.

APPENDIX C

ABSORPTION REFRIGERATION EQUIPMENT DESIGN

C.1 EVAPORATOR DESIGN

Heat Load = 18,000 Btu/hr

Evaporator Vapor Temperature = 40°F

Life Support/Equipment Cooling Fluid = Ethylene Glycol 30%-Water 70% (By Weight).

Evaporator Inlet Fluid Temperature = 55°F

Assume Evaporator Outlet Fluid = 45°F

Then log mean $\Delta t \approx \frac{15 - 5}{1.1} \approx 9.1\ ^\circ F$

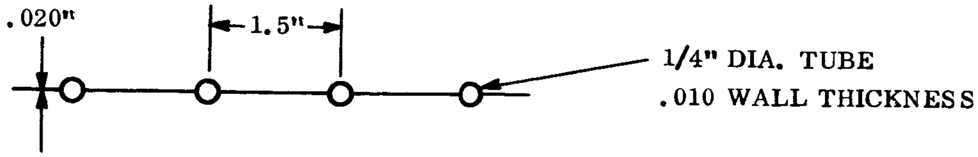
$$UA = \frac{Q}{\Delta t} = 19\ 80$$

$$W = \frac{18,000}{(0.875)10} = 2060 \text{ lb/hr}$$

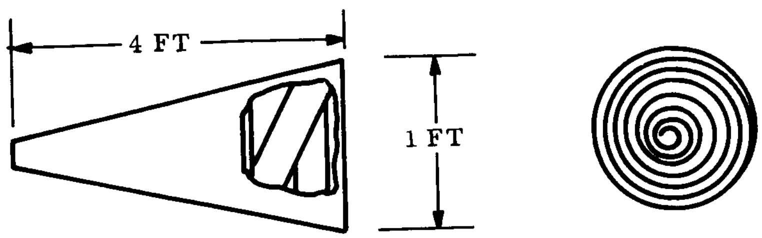
Assume an overall heat transfer coefficient of 100.

Then $A = \frac{1980}{100} = 19.8 \text{ ft}^2$

A typical element cross section might be:



The required length is 20 feet. The element is coiled in the evaporator container.



C.1.1 EVAPORATOR FLOW CALCULATIONS

$$\text{Reynolds Number} = \frac{DV\rho}{\mu}$$

$D = 1/4$ in.

$$V = \frac{Q}{A} = \frac{W}{\gamma A} = \frac{2060}{(65)\ (3600)\ (4)\ (0.00034)}$$

μ = See Figure C-1

γ = See Figure C-2

$$A = \frac{\pi}{4}\left(\frac{1}{4}\right)^2\left(\frac{1}{144}\right)$$

$V = 6.47$ ft/sec

$$N_{RE} = \frac{\left(\frac{1}{48}\right)\left(6.47\right)\left(2.02\right)}{\frac{6.3}{10^5}}$$

$$= 4320$$

From Figure 6-4, Reference 7-5, f (friction factor) = 0.038

$$\text{The pressure drop, } \Delta P = \frac{fV^2 L\gamma}{2gD}$$

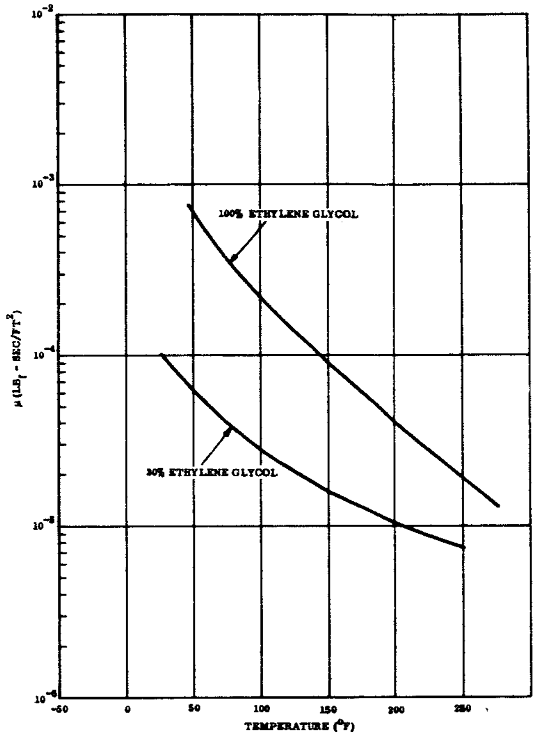


Figure C-1. Ethylene Glycol Viscosity

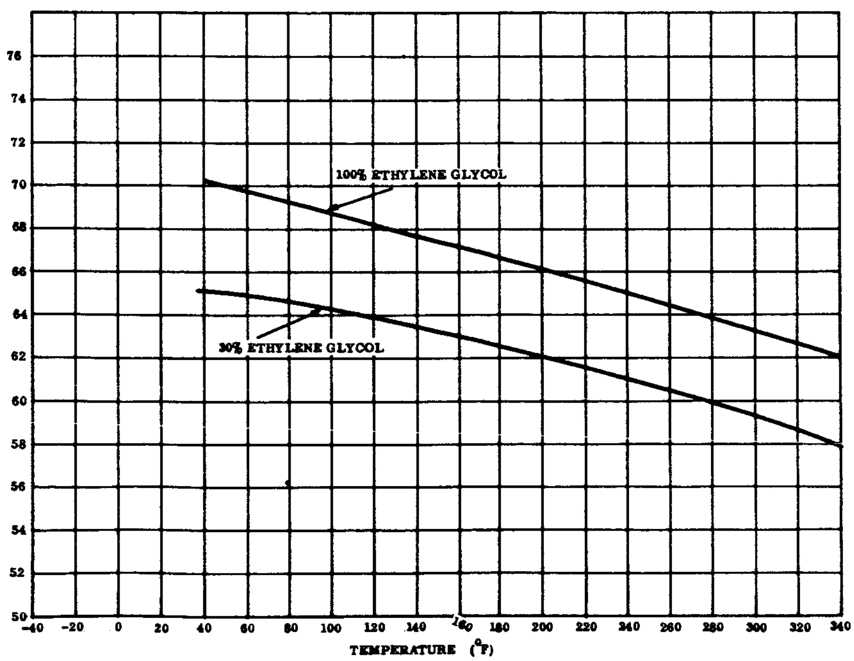


Figure C-2. Ethylene Glycol Density

$$\Delta P = \frac{(0.038)\,(6.47)^2\,(20)\,65}{2\,(32.2)\,\frac{1}{48}}$$

$$\Delta P = 1540 = 10.7 \text{ psi}$$

$$\text{Work} = Q\,\Delta P = \frac{1540\,(2060)}{(65)(3600)} = \frac{13.5 \text{ ft-lb}}{\text{sec}}$$

$$\text{Pump Motor Power} = \frac{\text{Work}}{\eta} = \frac{13.5\,(746)}{550\,(5)} = 36.7 \text{ watts}$$

C.1.2 WEIGHT CALCULATIONS

$$t = 0.020 \text{ in.}$$

Equivalent Width = 7.2 in.

$$\text{Volume} = 20 \times 12 \times 7.2 \times 0.020 = 34.5 \text{ in.}^3$$

$$\gamma = 0.1 \text{ lb/in.}^3 \qquad w = 3.45 \text{ lb}$$

$$\text{Glycol Volume} = (1/4)^2\,\frac{\pi}{4}\,(4)\,(20)\,(12) = 47 \text{ in.}^3$$

$$W = \frac{47}{1728}\,(65) = 1.77 \text{ lb}$$

Evaporator = 0.020 Wall Thickness

$$\text{Area} = \pi r S = \pi\,(6)\,60.3 - \pi\,(1.2)\,(12) \qquad S = \sqrt{36 + 3600}$$

$$= 1130 - 45 \qquad S = 60.3$$

$$= 1085$$

$$V = (0.02)\,1085 = 21.7 \text{ in.}^3$$

$$W = 2.2 \text{ lb}$$

Consider 0.2-inch layer of insulation, density 5 lb/ft^3 for outside surface, $W_{insulation} \sim$ 0.3 lb

Internal supports for container and coolant loop - estimate 0.5 lb

Evaporator Summary Weight	3.45
	1.77
	2.2
	0.3
	0.5
	8.22 lb
Coolant Pump and Motor	1.0
	9.2 lb

C.2 CONDENSER DESIGN

Temperature = 100°F

$$\text{30\% Ethylene Glycol Viscosity} = \frac{2.8}{10^5} \text{ lb sec/ft}^2$$

The overall heat transfer coefficient, U is largely dependent on the lowest component coefficient, which is taken, in this case as the fin-to-vapor film coefficient. If the condenser is built on the same lines as the evaporator, then the vapor flow will be turbulent, and the U is conservatively estimated to be 200 Btu/hr-ft^2-$^\circ$F.

C.2.1 CONDENSER WEIGHT

With the higher temperature and pressure, and consequent higher heat transfer in the condenser, a reasonable weight estimate is one-half the evaporator weight.

$$W_c = \frac{8.22}{2} = 4.1 \text{ lb}$$

C.2.2 PUMPING POWER

The heat to be removed is 19,200 Btu/hr

$$W = \frac{19(200)}{(0.875)(10)} = 2200$$

If the same tubing configuration is maintained, the tube diameter and flow velocity can be changed due to the lower viscosity and still maintain turbulent flow.

$$N_{Re} = \frac{DV\rho}{\mu}$$

$$4320 = \frac{D(3.2)(2)}{\frac{2.8}{105}} \qquad D = \frac{(2.8)\,(4320)}{105\,(3.2)(2)}$$

$$D = 0.0189 \text{ ft}$$

$$= 0.227 \text{ in.}$$

$$A = \frac{Q}{V} = \frac{2200}{64.4(3600)(3.2)} = 0.00297 \text{ ft}^2$$

$$A = 0.427 \text{ in.}^2$$

$$A_{\text{one tube}} = (0.227)^2 \frac{\pi}{4} = 0.0405 \text{ in.}^2$$

$$\text{Number of Tubes} = \frac{0.427}{0.0405} = 10.5, \text{ say 11 tubes}$$

$$\Delta P = \frac{f(V^2)L\gamma}{2_g D} = \frac{0.038(3.2)^2(10)64.4}{2(32.2)\,\frac{(0.227)}{12}}$$

$$\Delta P = 206 \text{ psf} = 1.71 \text{ psi}$$

Assume Radiator $\Delta P = 3$ psi

Then Power $= Q\,\Delta P = \dfrac{22000\ (4.71)(144)}{(64.4)(3600)} = 6.44\ \dfrac{\text{ft-lb}}{\text{sec}}$

Pump Power $= \dfrac{6.44(746)}{550(0.5)} = 17.5$ watts

Coolant Pump Weight = 0.7 lbs

Total Weight = 4.8 lbs

C.3 GENERATOR DESIGN

Heat Required = 24,000 Btu/hr.

Assume $U = 300$

Assume $\Delta t = 20^{\circ}F$ $\quad S = \dfrac{24,000}{300(20)} = 4\ ft^2$

Let $N_{Re} = 12,000$

$$12,000 = \frac{DV\rho}{\mu}$$

Average Temperature = $220^{\circ}F$

$$\gamma = 61.5\ lb/ft^3$$

$$\mu = \frac{9.2}{10^6}$$

Let V = 3 ft/sec

$$D = \frac{12,000\ \frac{(9.2)}{10^6}}{3\ (1.91)} = 0.0193 = 0.211\ in.$$

$$V = \frac{Q}{A} \qquad A = \frac{Q}{V} = \frac{W}{\gamma V}$$

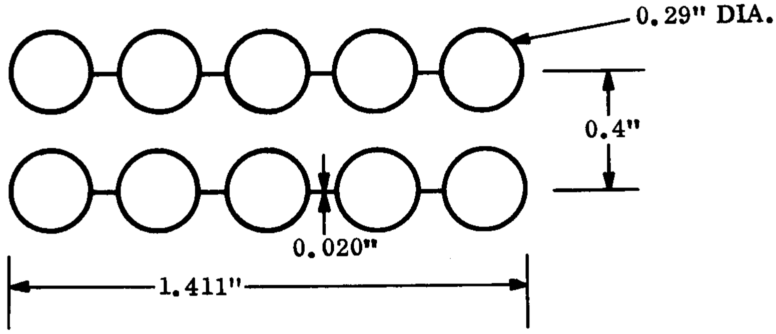
$$W = \frac{24,000}{0.875(20)} = 1370 \text{ lb/hr}$$

$$A = \frac{1370}{61.5(3)(3600)} = 0.00207 \text{ ft}^2 = 0.298 \text{ in.}^2$$

$$A_{\text{one tube}} = 0.066$$

Number of tubes = 4.5, say 5 tubes

C.3.1 TUBE GROUPING



Tube Circumference = πd	=	0.653 in.,
for 5 tubes, total	=	3.265 in.
Spacers = 0.080 x 8	=	0.71
		3.975 in.

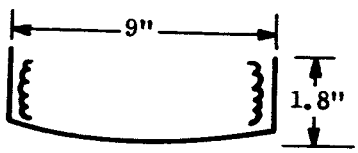
$$\text{The Length Required} = \frac{4 \text{ ft}^2}{\frac{3.975 \text{ in.}}{12 \text{ in/ft}}} = 12.1 \text{ ft}$$

$$\text{Diameter} = \frac{\text{Perimeter}}{\pi}$$

$$\text{Diameters Required } \frac{12.1 \times 12}{\pi} = 46.2 \text{ inches}$$

Let tube rings be 0.4-inch apart on Center Line, then sum of rings from 1.2-inch diameter to 8.4-inch diameter is sufficient to supply total surface.

C.3.2 GENERATOR WEIGHT



Let Diameter = 9 in.
Let Equivalent Depth = 1.8 in.

$$\text{Generator Surface} = \frac{\pi}{4}(9)^2 + 9\pi(1.8)$$

$$= 64.4 + 50.8$$

$$= 115.2 \text{ in.}^2$$

Let t = 0.030 in.

$V = 3.46 \text{ in.}^3$, For Aluminum w = 0.35 lb

1/2" insulation @10#/ft^3 = 0.33 lb

$$\text{Generator Volume} = \frac{\pi}{4} d^2h = \frac{\pi}{4}(9)^2\, 1.8 = 114.5 \text{ in.}^3$$

Heat Exchanger Volume (Aluminum)

$$= \left[(0.211)(\pi)(0.020)(5) + (0.089)(0.020)(4)\right](12.1)(12)$$

$$= \left[0.0662 + 0.0071\right] 145 = 10.62 \text{ in.}^3 = 1.1 \text{ lb}$$

Heat Exchanger Volume (Ethylene Glycol and Water)

$$= \frac{\pi}{4}(0.211)^2(5)(145) = 25.4 \text{ in.}^3$$

$$\gamma = 61.6, \quad W = \frac{25.4}{1728}(61.6) = 0.905 \text{ lb}$$

Volume for Refrigerant = 114.5-25.4-10.62

$= 78.5 \text{ in.}^3$

From Figure 7-2, density is about 1.8 (62.4) then refrigerant inventory =

$$\frac{(1.8)(62.4)(78.5)}{1728} = 5.1 \text{ lb}$$

For connection with the condenser, an 18-inch flue length is estimated

Let $t_{connection} = 0.020$ in.

Volume $= \pi$ (9) (18) (0.020) = 10.17 in.^{3}

W = 1 lb

Total Generator Weight

0.35
0.33
1.1
0.9
5.1
1.0
8.8 lb

C.3.3 PUMP POWER

$$P = \frac{f V^2 L \gamma}{2_g D}$$

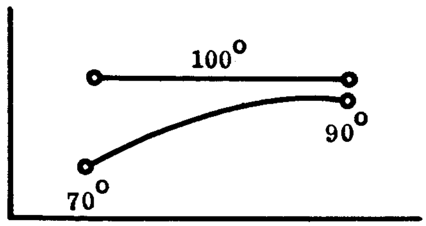
$$= \frac{0.03\ (9)\ (12.1)\ 61.5}{2\ (32.2)\ \frac{0.211}{12}} = 177.5 \text{ lb/ft}^2$$

$$\text{Pump Work} = Q \Delta P = \frac{1370}{61.5} \ \frac{(177.5)}{3600} = 1.1 \text{ ft-lb/sec} = \frac{1.1\,(746)}{550(0.5)} = 3 \text{ Watts}$$

C.4 ABSORBER DESIGN

Heat Rejected - 22,800 Btu/hr

$$W = \frac{22,800}{(0.875)18.2} = 1430 \text{ lb/hr}$$



$\mu = 3.7 \times 10^{-5}$

$\gamma = 64.6$

$\Delta t = \frac{20}{1.1} = 18.2^{\circ}$

Let U = 200

$S = \frac{22,800}{200(18.2)} = 6.26 \text{ ft}^2$

Assume $N_{Re} = 5,000$ $\quad$ V = 4

$N_{Re} = \frac{DV\rho}{\mu}$

$5000 = \frac{D(2)(4)}{3.7 \times 10^{-5}}$

$D = \frac{5 \times 10^3 \times 3.7 \times 10^{-5}}{8} = 2.33 \times 10^{-2}$

$= 0.28$ in.

$Q = \frac{1430}{64.6(3600)} = 0.00615$ $\quad A = 0.00154$
$= 0.222$

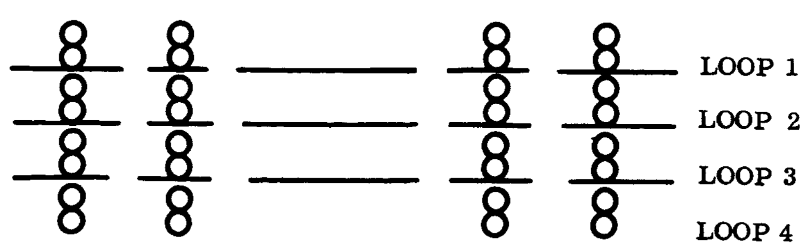
A = 0.0615/tube

3.6 or 4 tubes required

$V_{actual} = 3.6$ ft-sec

$N_{Re\ actual} = 4500$

Consider tubing configuration with four paralleled loops, each loop passing twice around per diametral ring.



Number of tubes per ring is eight. Total length = 8 π D per ring.

Tube circumference = π D = 0.88 in.

$$\frac{\times\ 8}{6.84}\ \text{in. per ring}$$

Required surface = 6.26 ft^2

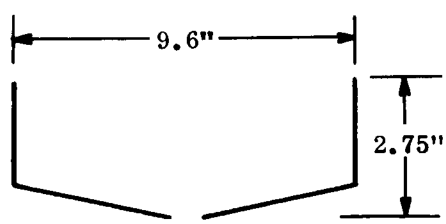
One-foot circumferential length = 0.57 square feet of area

Total circumferential length $= \frac{6.26}{0.57} = 11$ ft or 132 inches

Sum of the Diameters $= \frac{132}{\pi} = 42$ in.

Let tube space be 1/2 in. apart, ℄ to ℄

$$\left.\begin{array}{l}1\\2\\3\\4\\5\\6\\7\\8\\9\end{array}\right\}\ 45\ \text{in.}$$



9.6-inch absorber diameter

Absorber volume = 199 $in.^3$

Area $= 9.6\,(\pi)(2.75) + (9.6)^2\,\frac{\pi}{4} = 82.9 + 72.3 = 155.2\ in.^2$

Absorber Can Volume = 155.2 x 0.02 = 3.1 $in.^3$

W = 0.3 lb

Tubes, t = 0.010 in, πD = 0.88 in., Area = 0.0088 $in.^2$, V = 9.3 $in.^3$, W = 0.93 lb

Ethylene Glycol-Water Volume $= \left[(0.28\,\frac{\pi}{4}\,)(1056)\right] = 65\ in.^3$

$$W = \frac{64.6(65)}{1728} = 2.43 \text{ lb}$$

$$\begin{array}{rl} \text{Refrigerant Volume} = & 199 \\ & \underline{-9.3} \\ & 189.7 \\ & \underline{-65} \\ & 124.7 \text{ in.}^3 \end{array}$$

$$W = \frac{62.4(1.7)\ 124.7}{1728} = 7.65 \text{ lb}$$

Assume 24 in. x 0.020 Flue from Evaporator

$$V = 10\ (\pi)\ (24)\ (0.020) = 15.1 \text{ in.}^3$$

$$W = 1.51 \text{ lb}$$

Total Absorber Weight

$$\begin{array}{r} 0.3 \\ 0.93 \\ 2.43 \\ 7.65 \\ \underline{1.51} \\ 12.8 \text{ lb} \end{array}$$

Absorber Pump Pressure

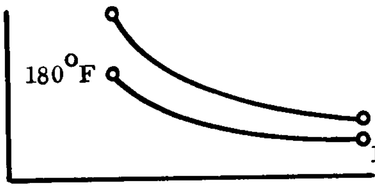
$$P = \frac{fV^2 L\gamma}{2g\ D} = \frac{0.038\ (3.6)^2 11(64.6)}{2(32.2)\left(\frac{0.28}{12}\right)} = \begin{array}{rl} & 232 \text{ lb/ft}^2 \\ + & \underline{432} \text{ (radiator)} \\ & 654 \text{ lb/ft}^2 \end{array}$$

$$\text{Work} = Q\ \Delta P = 0.00615\ (654) = 4.02 \text{ ft-lb/sec}$$

$$= \frac{4.02(746)}{550(0.5)} = 10.9 \text{ watts}$$

C.5 HEAT EXCHANGER (RECUPERATOR)

Net Q = 135 Btu/min. = 8100 Btu/hr



$$t = \frac{10}{0.695} = 14.4^{\circ}F$$

Let U = 200

$$\text{Required Area} = \frac{8100}{200\ (14.4)} = 2.81\ ft^2$$

Let $N_{Re} = 1000, \quad V = 1/2$

$$\rho = 3.5$$

μ, Assume $\frac{1}{10^5}$ $\qquad 1000 = \frac{D(1/2)(3.5)}{\frac{1}{10^5}}$

$$\frac{1000}{1.75x10^5} = D$$

$$D = 0.00572\ ft = 0.0685\ in.$$

$$\pi D = P = 215\ in.$$

Flow = 232 lb/hr by definition from Table 7-2.

$$\text{For } V = 1,\ Q = \frac{232}{(1.8)624\ (3600)} = 0.000568\ ft^3/sec$$

$$A = \frac{Q}{V} = \frac{0.000568}{1/2} = 0.001137$$
$$= 0.01365\ in.^2$$

3.74 tubes required.

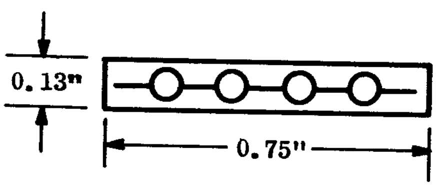
Use 4 tubes

$$S = 0.215 \text{ in.} \times 12 = 2.58 \text{ in.}^2/\text{ft per tube}$$
$$\text{or } 10.32 \text{ in.}^2/\text{ft for 4 tubes}$$

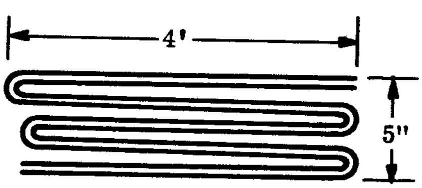
Assume Fin Surface = Tube Surface

Then $S = 20.6 \text{ in.}^2/\text{ft}$

$$L = \frac{144}{20.6}\ (2.81) = 19.6 \text{ ft}$$



H/X CROSS SECTION



C.5.1 RECUPERATOR WEIGHT

Fin-Tube Weight

$$15.46 \times 0.020 \times 20 = 6.18 \text{ in.}^3$$

$$W = 0.62 \text{ lb}$$

Refrigerant Volume = $0.13\ (0.75) \times 240 = 23.4 \text{ in.}^3$

$$\underline{-6.2}$$
$$= 17.2 \text{ in.}^3$$

$$W = \frac{(1.8)(62.4)17.2}{1728} = 1.2 \text{ lb}$$

External Surface = 2 (0.75+0.13) x 0.020 x 240

$= 8.54 \text{ in.}^3$

$= 0.85$ lb

Insulation = 170 in.^3 at 0.005 lb/cu in.

W = 0.85 lb

Recuperator Weight = 0.62
1.2
0.85
0.85
3.5 lb

C.5.2 RECUPERATOR PUMP WORK

$$P = \frac{fV^2 L\gamma}{2_g D}$$

$$P = \frac{(0.064)(0.25)(20)(1.8)(62.4)(12)}{2(32.2)\ (0.068)}$$

$$= 98.5 \text{ lb/ft}^2$$

$$\text{Work} = Q\,\Delta P = 0.000568\ (98.5)$$

$$= 0.056 \text{ ft-lb/sec}$$

$$= 0.15 \text{ watt}$$

C.6 SYSTEM PRESSURIZING AND RECIRCULATING PUMP

System Pressurizing Requirements = Q Δ P

$$= \frac{(3.86)(60)}{3600(1.8)(62.4)} (144)\ (1.2) = 0.000568\ (144)\ (1.2)$$

$$= 0.0984 \text{ ft-lb/sec}$$

Assume recirculation is twice pressurizing flow and same Δ P

Work = 0.2952 ft-lb/sec

$$\text{Pump Work} = \frac{(0.295)(746)}{550\ (0.5)} = 0.8 \text{ watt}$$