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PRELIMINARY PARAMETRIC EVALUATION
OF A KILOWATT THERMOELECTRIC
PU-238 POWER SYSTEM FOR
LUNAR SURFACE OPERATION.

MND 3483

March 15, 1965

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I. INTRODUCTION

This report has been prepared by the Nuclear Division of the Martin Marietta Corporation, Baltimore, Maryland under Contract AT (30-1)-3483.

The objective of this study was to provide preliminary parametric technical information for use in evaluation of kilowatt electric radioisotope power supplies for lunar surface missions. Provisions of Contract AT(30-1)-3483 stipulated that the Po-210 1.5 kw(e) thermoelectric isotopic power system designed by Martin for the Apollo manned spacecraft under USAEC Contract AT(30-1)-3296 be used as a reference design. This design was to be modified for Pu-238 fuel and as required by lunar surface environmental conditions to produce 1.5 kilowatts of electrical power for periods up to one year.

The referenced Po-210 power system, as fully described in Martin MND 3296, Vol. 3, has accordingly been modified as the basis for conducting a parametric design analysis. Four 1.5 kw(e) Pu-238 fueled preliminary designs have been examined having radiator areas respectively of 30, 50, 75, and 100 square feet per 500 watt (e) module. Performance characteristics of the four Pu-238 fueled power systems have been estimated for varying lunar environmental conditions and for different radiator orientations relative to the lunar surface.

Shielding required for astronaut vehicle occupants has been incorporated to comply with a 3 REM dose limit. It was however beyond the contract scope to investigate further the use of lunar obstacles or separation of the heat source from the spacecraft as a means to reduce shielding weight.

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II. APPLICATION AND CRITERIA

A. SYSTEM APPLICATION

There are two thermoelectric generator system approaches that can be considered for lunar surface missions. The first type has a static, self-contained heat rejection subsystem similar to those used in previous SNAP radioisotope power supplies. The second type utilizes a dynamic heat rejection subsystem consisting of a liquid cooling system with remote heat rejection surfaces.

The static and dynamic system designs are intended for use with different types of spacecraft and therefore consideration must be given to the basic payload configurations to which each is applicable.

1. Dynamic System

A dynamic system, with its large, low temperature, radiator is intended for use on those missions where the payload, last boost stage, or a transition section combined with the payload makes large areas of vehicle skin available for power system waste heat rejection. Since the metallic skin is present whether or not a radioisotope power supply is on board, a large radiator area can be made available to the radioisotope power system for a comparatively small weight increase. The radiator weight charged against the radioisotope power supply consists of the pumps, manifolds, and lines necessary to conduct the cooling fluid out to the proper skin location, and the heat transfer tubes on the skin, including any meteoroid protection armor that may be required.

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On the other hand, if the vehicle has no available skin, the weight of a large radiator would make the dynamic system much heavier. It has been estimated that the weight of skin and tubes amounts to approximately 75 lbs/100 ft² of radiator area. Most of this weight is attributable to the skin. Since additional structure would probably be needed to support the skin and tubes during launch, it can be seen that a static system will provide a lighter system for vehicles without available skin area.

2. Static System

A static system is entirely self contained, using the surface of the generator itself for high temperature waste heat rejection. It is somewhat heavier than the dynamic system because of the surface requirements for heat rejection. The static system should be considered for those vehicles which have no skin available for a large low temperature radiator.

The space vehicle designer should also remember that the static system has to be exposed to space on as many sides as possible so that it may reject its waste heat with a minimum weight radiator configuration. Generally, this means that the static system can be used on any vehicle but must be installed on the outside of the spacecraft or better, deployed on a boom away from the spacecraft. Boom deployment serves to reduce or eliminate both the thermal input to the vehicle and the amount of nuclear radiation shielding required.

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3. Thermoelectric Materials

Three families of thermoelectric materials have been developed to the point where they may be considered operational: lead telluride (Pb Te), bismuth telluride ($\text{Bi}_2 \text{Te}_3$), and silicon germanium (Si Ge) alloys. The PbTe has been used for energy conversion in a number of isotope power supplies, including SNAP 3, 7 series, 9A, 11 and 19. The $\text{Bi}_2 \text{Te}_3$ has been used in various low temperature commercial applications. SiGe has been used in SNAP 10A and 17A.

Each of these materials has an optimum operating temperature range where its figure of merit is highest. These optimum ranges are: $\text{Bi}_2 \text{Te}_3$ - - 70° to 400°F ; PbTe - - 400°F to 1000°F and SiGe - - 1000°F to 1800°F . It is possible to increase generator efficiency by fabricating the thermoelectric elements out of two or more materials so each material operates over its optimum temperature range. Thus, maximum efficiency is obtained by using segmented elements. The PbTe - $\text{Bi}_2 \text{Te}_3$ segmented couples are currently being developed under the SNAP 21 program.

PbTe generally offers higher efficiency than SiGe but, to prevent oxidation and sublimation in hard vacuum, it must be operated in a sealed environment. SiGe, on the other hand, can operate in both air and hard vacuum without deterioration. The higher SiGe conversion system radiator temperature also reduces required radiator area and hence overall generator weight and size. Therefore, a definite cross-over point is indicated in radiator area, above which a PbTe system is more efficient, and below which a SiGe system is better.

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B. PARAMETRIC ANALYSIS CRITERIA

There are several design requirements, criteria and assumptions that have been established for this preliminary parametric analysis. These requirements and criteria fall into two general categories: those established by the contracting agency and those assumed in order to develop the system design parameters. These requirements and criteria are discussed.

1. Design

a. Power Supply

As a reference design point, the dynamic system conceptual design for the thermoelectric, Po-210, 14 day mission, radioisotope power supply defined in "Feasibility of Isotopic Power for Manned Lunar Missions" (Reference 1) will be used.

b. Fuel

The fuel will be Pu-238 in the oxide form. Fuel characteristics are per "Plutonium Dioxide Data Sheet" (Reference 2). See Appendix A.

c. Thermoelectric Materials

The thermoelectric elements will be of homogeneous lead telluride material with a maximum thermoelectric hot junction temperature of 1000°F.

2. Performance

a. Power Output

The power system will deliver 1500 watts to the load at the end of a one year mission. Any excess power available at the beginning of the mission will be consumed by regulatory circuits in the power system.

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b. Voltage

The power system output voltage will be 28 volts $\pm$ 10%.

c. Load

It has been assumed that the load appears to the power system to be resistive only, and time invariant.

3. Environmental Conditions and Mission Parameters

a. Launch vehicle and trajectory

The Apollo spacecraft (service module) configuration, flight profile, and flight schedule parameters are to be used as guidelines in deriving power supply data.

b. Lunar vehicle

The lunar vehicle will be considered to have surfaces both parallel and perpendicular to the lunar surface which embody power supply heat rejection radiators.

c. Mission life

The mission life will be a minimum of one year from the time of launch with a maximum launch hold time of 30 days.

d. Effective sink temperatures

The effective sink temperatures will be based on a solar constant on the lunar surface of $442.8 \text{ Btu/hr-ft}^2\text{-}^\circ\text{R}^4$, an albedo of the lunar surface of 0.095 and a 210°R lunar night sink temperature (References 3 and 4).

e. Ambient vehicle temperatures

It is assumed that a location in the payload electronics bay that has a regulated ambient temperature between $+30^\circ$ and $+130^\circ\text{F}$ for the duration of the mission will be available for the power supply voltage converter and regulator.

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f. Shock and vibration

The structural design dynamic load criteria has been assumed to be equivalent to a static load design criteria of 100 g.

4. Nuclear Radiation

Shielding will be employed as required by the location of radioisotope heat sources within the Apollo service module and as necessary to limit the total heat source radiation dose to any crew member to 3 REM.

5. Safety

The basic safety philosophy governing design of the Radioisotopic Thermoelectric Generator (RTG) requires that the general public not be exposed to undue hazards resulting from normal use of the generator or from any credible accident attendant to its use.

The safety design criteria governing the life of the generator system are as follows:

a. Shipment

Shipment of the radioisotopic fuel shall be in accordance with applicable Federal, State and local regulations.

b. Containment

The design will provide absolute containment or confinement of the fuel under all credible missile abort induced conditions.

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c. Impact

The fuel capsule shall maintain structural integrity when subjected to terminal velocity impact upon a rigid medium.

d. Pressure

The fuel capsule shall have sufficient structural integrity to contain the internal helium pressure buildup due to radioactive decay or external pressure due to ocean burial.

e. Aerospace earth re-entry

The power supply fuel source will be designed for random, intact re-entry at superorbital velocity as a consequence of mission aborts.

f. Corrosion resistance

The fuel capsules shall maintain integrity (corrosion resistance) to permit adequate time for sea recovery operations in the event the capsule is impacted within recoverable sea depths.

The capsule shall be resistant to the corrosive action of missile propellants and their by-products and be chemically compatible with the isotopic fuel form.

The fuel assembly shall also exhibit high corrosion and sublimation resistance at high operating temperatures.

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III. SYSTEM DESCRIPTION

A. GENERAL

The overall electrical power system will consist of three completely independent thermoelectric generators, each producing, continuously, 530 watts (e) or more through the end of mission life. This results in a gross output of 1590 watts (e) of which 90 watts (e) are required for pumps and other auxiliary equipment that support the power generation system. A generator system schematic is shown in Figure III-1. Three such generators are required for a 1.5 kw system.

Each generator consists of a heat source radiating energy to thermoelectric packages, which in turn convert part of the thermal energy to electrical power and reject the remaining thermal energy at the cold junction to a fluid coolant. This waste energy is then transported, via the coolant, through tubes to the radiator panel and radiated to space. The general configuration of each 500-watt generator, shown in Figure III-2, consists of the isotope heat source holding 14 capsules of $\text{PuO}_2\text{-238}$, two 250-watt fluid-cooled thermoelectric modules on either side of the fuel block, a door and insulation and all required support structures. The fuel block is beryllium in order to provide ablative protection for the encapsulated radioisotope to meet the intact re-entry requirement. System fueling at the launch site (and defueling should the launch be scrubbed) is permitted via a rotating door. This door also provides for cooling the fuel block in an emergency such as complete coolant system failure. To do this, the block can be freed of insulation on one side, providing for a highly efficient heat dump.

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THERMOELECTRIC GENERATOR SCHEMATIC

Figure III-1. Thermoelectric Generator Schematic
(three required for power system)

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GENERATOR ASSEMBLY
(TYPICAL)
FIGURE II-2

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The primary heat rejection system is a radiator composed of the aluminum skin of the space vehicle to which aluminum tubes are brazed. Coolant carrying the waste heat from the thermoelectric modules is circulated through the tubes, transferring its heat to the vehicle skin. The heat is then dissipated by radiation to space. The radiator tubes are protected from meteoroid damage by the vehicle skin and aluminum shields sandwiched between the skin and tubes.

Redundant pumps, the redundancy depending on reliability requirements, propel the fluid through the coolant loop. Check valves in the system automatically prevent fluid recirculation through dormant pump(s). In the event of pump failure, which will be detected by suitable pressure or temperature-sensing elements, a control circuit will shut down the defective pump and initiate operation of an alternate pump or releases the spring-loaded door to expose the fuel block in case of a complete cooling system failure. Power is supplied to the pumps and control circuit from the generator electrical output.

The thermoelectric generator system will produce the required power at 28 volts dc, thereby eliminating need for extensive power conditioning equipment. A voltage regulator will be used to maintain each generator output at 28 volts $\pm 1\%$ so that several such units could be placed in electrical parallel for increased reliability and higher power outputs.

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B. MAJOR COMPONENTS

The radioisotope fuel selected for this lunar surface application is Plutonium-238 in the form of plutonium dioxide (PuO_2). The fuel characteristics are presented in Appendix A. The basic safety philosophy required for this parametric study is intact re-entry and total containment of the radioisotope under all conditions.

The fuel therefore will be assembled into capsules and 14 capsules placed in a beryllium ablator block to form the fuel block assembly and heat source for each of the 500-watt (e) generators (Figure III-2).

1. Fuel Capsules

The fundamental constraint in the design of a radioisotope heat source is that the general populace must not be subjected to undue radiological hazards resulting, directly or indirectly, from its use. This constraint, for an alpha-emitting isotope, is virtually tantamount to requiring containment* of the fuel within its encapsulation under planned pre-mission, operational, and post-mission conditions and all credible accidental environments excepting specific cases in which it is demonstrated the fundamental constraint remains inviolate. The latter exceptions are few in number and include, for example, nonterrestrial environments and, perhaps, deep earth burial and ocean deposition.

There are many conditions or environments in which the fuel container will or may exist and which require safety analysis. The objective here, however, is to evolve a preliminary capsule design appropriate for the Pu-238 source material and to meet

* Containment is defined either for the period of time required for recovery and disposal, or until the fuel material ceases to imply a radiological hazard.

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the overall system requirements of the heat source. Accordingly, only those requirements which control capsule design are considered. Verification that the selected design satisfies the fundamental safety constraint is an elaborate process and may be accomplished only during a development program itself.

The feasibility of the proposed design has been investigated with respect to available mission requirements and nuclear safety considerations based upon potential radiological hazards that may evolve from possible accidents during any phase of the mission prelaunch, launch, launch abort, orbital and inter-lunar.

The preliminary nuclear safety criteria established as the basis of this study include:

- (1) The fuel containment structure (FCS) must provide for fuel containment under all prelaunch environments of transportation, storage and fueling.
- (2) The FCS must retain its integrity under the most severe mechanical, thermal, and chemical forces of a launch pad abort environment.
- (3) The FCS must contain the fuel after earth impact at terminal velocity.
- (4) The FCS must have sufficient structural integrity to contain the internal helium pressure buildup due to radioactive decay or external hydrostatic pressure due to ocean burial, including the effects of sea water corrosion, which could present radiological hazards, if release occurred.
- (5) The FCS materials shall be chemically compatible with the radioisotope fuel.

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- (6) The FCS shall be incorporated into sufficient ablative material to provide for intact earth re-entry under both orbital and superorbital entry condition.

The following section lists and discusses those requirements which control capsule material and configuration selection.

a. Material selection

The foregoing discussion points out that an appropriate fuel capsule material must be compatible: (1) with the fuel form; (2) retain structural integrity during earth impact at terminal velocity; (3) possess good resistance to oxidation and corrosion; and (4) maintain structural integrity when exposed to maximum internal or external pressures.

No single material possesses all of these properties within the stated operating temperature range. For this reason, laminated capsules were considered for this study. They consist of an inner capsule wall of TZM (0.5 Ti, 0.08 Zn, Mo) and an outer wall of Haynes-25. Within the capsule, the fuel is enclosed in a tantalum liner.

TZM was selected primarily for its capability of containing the helium pressure buildup at temperatures in the vicinity of 1800°F.

Haynes-25 was selected for the fuel capsule impact material based on its properties at 1800°F and prior tests conducted for the SNAP 17A and other programs. Impact strength was a predominate consideration, but saltwater corrosion, oxidation and sublimation characteristics were also evaluated. Hastelloy C appears to be an acceptable alternate material. The Haynes-25 alloy is slightly better for impact, while the Hastelloy is somewhat better for saltwater corrosion. Hastelloy C saltwater

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corrosion rates were evaluated under the SNAP 7 program, and Haynes-25 corrosion has been evaluated under the SNAP 17A program. Final selection should be based on capsule impact tests.

TZM is now being produced on a commercial basis to meet guaranteed properties. Recent studies at Climax Molybdenum Company revealed that TZM achieves its strength at elevated temperatures by the precipitation of complex carbides in addition to normal strain hardening.

Creep properties obtained from actual tests reveal the superiority of TZM to the older Mo-0.5% Ti alloy. Weldability of TZM is good, although any slight change in the weld parameters (speed, amperage, atmosphere, etc.) would have an effect upon the quality of the weld. Mound Laboratories are now working on a weldability study using TZM. Martin has successfully welded and brazed TZM fuel capsules and subjected the capsules to leak tests without any apparent failures.

TZM, however, has poor oxidation resistance at elevated temperatures. For this reason, it is necessary to encapsulate the molybdenum alloy capsule in an oxidation resistant material such as Haynes-25. Haynes-25 by itself is an excellent high temperature material and possesses the necessary oxidation resistance for the present application. In addition, it is a very ductile material and exhibits excellent impact strength.

Compatibility studies have been conducted between the fuel form and TZM. Test results showed that diffusion occurred between the fuel form and TZM although the rate of diffusion could not be measured.

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Since compatibility problems exist between the fuel form and TZM, an inner liner encapsulating the fuel form is required. It appears that the compatibility of Ta with the proposed fuel form is adequate for the temperature range and the mission lives contemplated. (Mound Laboratories is currently using Ta and Ta-10% W liners in other programs.)

Based upon the material requirements discussed above, a capsule concept (Figure III-3) has evolved for the specified application. The fuel capsule is composed of a tantalum inner liner, surrounded by a TZM pressure container with an outer cladding of Haynes-25. In the following sections the wall thickness of the different fuel capsule materials will be determined with the exception of the tantalum inner liner. A ten mil tantalum liner was selected since any thickness less than this would pose a welding problem.

b. Helium pressure buildup

Associated with the alpha-emitting fuels is the inherent problem of helium buildup which occurs as a result of the radioactive decay of the fuel, in which alpha particles are ejected from the nucleus and capture electrons to form a helium atom.

In the design of a fuel capsule, therefore, consideration must be given to the pressure buildup. Helium pressure can be contained by two methods: (1) by adjusting the wall thickness to contain the high pressure; and, (2) by adjusting the void volume to reduce the internal pressure to an acceptable value. The capsule wall thickness is directly related to the pressure a capsule can withstand; therefore, an increase in pressure would require a thicker capsule wall and an increase in weight.

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Figure III-3. Fuel Capsule

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Using volume control, an increase in pressure would necessitate an increase in void volume and a corresponding increase in weight. The pressure which a thickness, L , of the capsule wall may sustain varies linearly with L , providing L is small compared with the capsule radius. Under the same conditions, the capsule weight varies linearly with L also. On the other hand, the pressure contained by the capsule varies as V^{-1} , while the weight is directly proportional to V if the capsule length is varied. From these remarks, it is apparent that capsule weight does not depend sensitively on which technique is employed to contain the calculated pressure buildup.

An internal pressure analysis was made for the proposed fuel capsule design, see Appendix B. The internal pressure buildup was calculated using the perfect gas law. The maximum helium pressure was found to be 17,500 psi and occurred 135 years after fuel encapsulation. By applying the thick wall cylindrical stress formula and the allowable working stress for the TZM material, it was found that 130 mils would provide sufficient structural strength to contain the helium pressure.

c. Impact requirements

Of paramount importance in all fuel capsule designs is the maintenance of structural integrity at impact. The terminal velocity of the body is related to the ballistic coefficient of the fuel capsule. To maintain structural integrity while limiting structural weight, it is obviously desirable to minimize the impact velocity. The impact velocity may be evaluated from

$$V_I = \sqrt{\frac{2 W_I}{\rho C_D A}}$$

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where

ρ = atmospheric density at selected altitude

W = weight

C_D = drag coefficient

A = cross sectional area

Numerical data obtained using this equation indicate that for acceptable impact speeds based on previous capsule test experience (< 450 fps) the ballistic coefficient should not exceed about 200 slugs/ft².

To optimize the fuel capsule design from an impact standpoint, the capsule wall material selected must exhibit a high energy absorption capacity. A satisfactory material for this purpose, as demonstrated in the SNAP 17A and other programs, is the Haynes-25 alloy. The determination of required wall thickness for a capsule of given L/D, weight and volume may not be done theoretically at the present time. In practice, a test program is required, first to determine a design thickness and then to verify the reliability of the design. The preliminary capsule design developed herein is, therefore, based primarily on experience.

From previous impact test programs at Aberdeen Proving Grounds and at Martin Facilities, the Haynes-25 outer containment vessel was selected with an 80 mil wall thickness to augment the 130 mils of TZM, used as the helium pressure containment vessel, in order to achieve fuel containment.

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d. Launch pad incidents

Launch vehicle malfunction on the launch pad could subject the fuel containment structure to shock, heating and the aggressive corrosive action of the propellants.

An explosion could produce shock overpressure of up to approximately 3000 psi, if multiple stages exploded simultaneously. However, the individual capsule, considering only the TZM structure, can withstand an external collapsing pressure of approximately 13,500 psi at 1800°F. Therefore, any explosion generated by missile propellants would not rupture the fuel capsule.

Since the fuel containment structure is designed for intact re-entry (see Section 2 - Fuel Block Assembly) in which higher heat fluxes are experienced and for longer periods of time than are created on the launch pad, it is reasonable to assume that a launch pad fire would not cause melt down of the containment structure.

e. Hydrostatic pressure

Impact in water is a greater probability than on land if an abort occurred during the ascent portion of the trajectory. The generator and the fuel capsules would consequently be subject to external pressure depending upon the depth of water. The maximum possible hydrostatic pressure corresponding to the greatest ocean depth is approximately 14,000 psi. Considering only the TZM component of the containment structure a hydrostatic pressure of approximately 20,000 psi at ambient conditions could be resisted.

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In summary, based upon the selection of materials for the capsule structure which exhibits high-temperature strength, high impact energy absorption capacity, good corrosion resistance and the design of the structure considering the requirements of internal helium pressure, earth impact, launch pad accidents and external hydrostatic pressure, confidence is expressed in the design that it will satisfy the containment structure safety criteria.

2. Fuel Block Assembly

The plutonium oxide fuel is pressed and sintered into annular shapes and slipped into the tantalum liners. The fueled liners are seal welded and decontaminated, then placed into the TZM cylinders which in turn are placed into the Haynes-25 outer shells. Each layer is an individual encapsulation and is individually welded. Minimum tolerance fits and assembly under one atmosphere of helium minimizes the temperature difference from the surface of the capsules to the fuel. For each 500 watt(e) generator, 14 fuel capsules are required for the heat source (see Figure III-2).

Beryllium is used as the heat distributor and the ablative heat shield for the fuel source to conform to the nuclear safety design criterion of intact re-entry. Beryllium was selected because of its unique combination of properties.

The shape of the beryllium block is a rectilinear parallelogram. This shape of block was found to furnish sufficient area on the large edge so that during an emergency heat dump situation the fuel block temperature would not exceed the melting temperatures of the block and capsule materials.

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The operations of installing encapsulated fuel in the beryllium block are performed in a hot cell by remote control. The 14 capsules are inserted into the prebored cavities of the block. Helium gas is introduced to the interior of the block to enhance thermal conductance. Plugs, machined from hot pressed beryllium, are inserted and screwed into place. The plugs are sealed by TIG welding to the block. The cladding end cap is pressed into place over the edge, and welded to the rest of the cladding to form a gas-tight envelope.

Each fuel block assembly will be inserted into its generator through the spring-loaded (emergency heat dump) door during the launch countdown. It will lock into the bellows support assembly which is designed to provide adequate clearance for the fuel block insertion. During normal operation, the fuel block assembly radiates heat from each large surface to the adjacent 250-watt(e) thermoelectric module.

a. Ablative materials requirements

The method which has been selected to provide protection for the fuel block during re-entry heating is to use an ablative material around the fuel capsules. While many ablative materials are available, unique requirements of the system under consideration limit the number of acceptable materials.

Several organic and inorganic materials are available which can absorb large amounts of heat per unit weight, transmit little or no heat to internal components and are lightweight. Organic materials, however, tend to degrade at about 800°F and char at 1400°F. Since normal operating temperatures of the fuel source are in excess of 1400°F, the organic materials cannot be used.

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Pyrolytic graphite has a high melting temperature, low conductivity and good heat radiation properties; but is difficult to join to other materials and has poor structural strength.

An examination of the ablator materials available has lead to the adoption of beryllium for many re-entry situations. Beryllium is lightweight (1136 lb/ft³) has a high heat of sublimation (470 Btu/lb), a high emissivity (0.61) and a reasonably low melting temperature (2345°F). Although the beryllium transmits re-entry heat readily (44 Btu/hr-ft-°R) and is also toxic when inhaled or ingested, its advantages counterbalance these few penalties involved in its utilization.

During entry, the heat source will experience temperatures greater than those of normal operation. At 2000°F, the tensile strength of beryllium is down to about 200 psi. For a large heat source, the maximum temperature during entry occurs near the time of maximum stagnation pressure. The combination of forces would exceed the strength of the beryllium, and tear the heat source apart. As a solution, a structural matrix is considered as an integral part of the heat source. This could be in the form of a wire basket of refractory material, or of additives, such as vacuum-grown crystals of refractories, which would be included in the beryllium powder during fabrication prior to hot pressing and sintering. Another form of re-enforcing might be to incorporate interlocking rods of a superalloy in the beryllium block. Rods could serve the dual function of holding the heat source intact during entry, and of securing the cladding material to the heat source.

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The fuel block cladding serves various purposes. It provides a high-emissivity coating for thermal radiation, and it prevents beryllium sublimation during high temperature operation in outer space. The cladding material for the heat source is selected to obtain good high temperature mechanical properties and corrosion resistance. It is presently planned to use helium gas between the cladding and the heat source to improve the heat interface characteristics. A better method to assure a good interface is to employ welding or brazing of the cladding to the heat source. This technique would be employed, if available. However, development is required to perfect the process. Successful brazes have been made using a silver-lithium alloy as the braze medium, but these were on a laboratory scale. Problems might be encountered in adapting the method to a heat source configuration with adequate quality control. These problems are not seen to be extensive, and they appear likely to lend themselves to normal development approaches.

In the determination of the ablator thickness for intact re-entry a nominal spacecraft flight trajectory from lift-off to interlunar injection of the vehicle from an earth orbit or direct ingestion into a heliocentric interlunar orbit were evaluated to determine the most stringent aerothermodynamic forces that could be seen by the fuel block, for either normal or abort conditions. The most stringent condition was determined to be re-entry into the earth atmosphere at superorbital velocity.

The following specific superorbit re-entry condition at the outer edge of the earth atmosphere (400,000 feet) was used to determine the total heat available for burnup:

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Inertial Velocity, $V_0 = 36,000$ fps

Inertial flight path angle, $\nu_0 = -6$ degrees

Based upon the total heat available, the required thickness of ablator material was determined by a heat balance on the material to be ablated. The equation used defining the heat balance was:

$$\begin{array}{rclcl} \text{Available Heat} & & \text{Heat Absorbed} & & \text{Heat of fusion} \\ \text{During Re-entry} & = & \text{By Ablator} & + & \text{for Ablated Material} \\ & & & & + \\ & & \text{Heat Rejected} & & \\ & & \text{By Radiation} & & \end{array}$$

$$Q_{\text{avail}} = m C_p \Delta T + m h_f + \sigma \epsilon A T^4 t$$

The minimum beryllium ablator thickness was determined when the total heat available was equal to the heat required to expose the fuel capsules, as established by the above equation.

The results of the study indicate that the fuel capsules of the 500 watt (e) radioisotopic generator can be protected from aerodynamic burnup by placing approximately 2 inches of beryllium ablator around the capsules.

Since re-entry heating under superorbital conditions and ablation mechanism are not precisely known, the results of this study should not be considered as absolute, but rather as an indication of the feasibility of intact re-entry for this type of generator.

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FIGURE III-4
THERMO-ELECTRIC MODULE SECTION
DYNAMIC SYSTEM

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3. Thermoelectric Modules

The energy conversion portion of each generator consists of two 250-watt (e) thermoelectric modules (Figure III-2) placed on two sides of the fuel block assembly. The two modules are electrically connected in series and each contains 350 or more lead telluride thermocouples. The couples are connected in a series-parallel arrangement such that the failure of a single couple will only effect that couple. Since each couple is in parallel with one other couple and that pair is in series with the next pair, the loss of one couple (open circuit) would cause an almost negligible power drop (i. e. $< 1/700$ of the total power).

The individual couples (Figure III-4) consist of a hot shoe, a P-type element, an N-type element, and FerroVacE high purity iron diffusion barriers at either end of the elements. The hot shoe is a flexible strap made of 0.005 inch copper. Copper is used because of its low electrical resistance and flexibility. The iron diffusion barrier is ultrasonically spotwelded or brazed to both ends of the hot shoe. This barrier is used to prevent diffusion of the copper into the thermoelement. It has been found that high purity iron is the best barrier material since it does not poison the thermoelements and reasonable bonds may be obtained. The thermoelements are bonded directly to the iron. Copper connector straps are soldered to the iron discs on the cold end of the thermoelements to complete the series-parallel couple connections for the module.

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Thermal insulation is required around the elements. This is accomplished by using Min-K sheet insulation which has been preformed to receive the couples. Assembly is accomplished by simply inserting the element legs through holes in the insulation. The Min-K sheet may be large enough to satisfy the whole module, or it may be divided upon into sections to facilitate assembly and repair. Min-K has the lowest thermal conductivity of any of the available insulations.

Due to the marked tendency of the lead telluride thermoelements to sublime in a vacuum environment and their susceptibility to oxidize, each thermoelectric module must be operated in an inert gas atmosphere for sustained service. Therefore, a rectangular stainless steel bellows surrounds the thermoelectric module. The bellows provide a high thermal resistance path to reduce heat loss. It is brazed to both the bellows support and the cold junction plate to form a sealed chamber. The chamber is evacuated and backfilled at assembly with an inert gas such as argon at six to eight inches of Hg absolute pressure when cold. The gas pressure increases to approximately 7 psia at operating temperature. The electrical leads from the module are brought through the cold junction plate by use of hermetically sealed connectors.

To keep the thermoelectric module from shorting out, it must be electrically insulated. This is accomplished on the hot side by coating the face of the bellows with aluminum oxide and inserting a sheet of mica between the oxide and the hot shoes for a second barrier. The mica will also permit the hot shoes to slide during thermal expansion and contraction with a minimum bending and shearing stress in the thermoelement. The cold end is insulated with a sheet of mica inserted between the connecting

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straps and the stainless steel diaphragm. Electrical insulation of the cold junction is not as difficult due to the lower temperature, and the mica is sufficient. Aluminum oxide and mica have been used for electrical insulation of thermoelements in other generators, and were used together in both the SNAP 7 and the SNAP 9A generators.

The heat sink at the cold junction of the thermoelements is formed by a liquid-cooled diaphragm which is brazed to a cold plate for structural strength and to complete the liquid passage. The liquid pressure acting on the diaphragm provides the necessary force (approximately 35 psi) on the thermoelements to maintain good thermal conduction through the elements and to prevent the elements from breaking when subjected to dynamic loads. A variation of the module pressure loading system has been used successfully on a chemically fueled generator currently under development at Martin. There the module pressure was held at half the external air pressure, providing a differential pressure of approximately 7 psi. The diaphragm is made of stainless steel to satisfy the requirements for a thin flexible material which is not appreciably effected by the coolant fluid or radiation. The cold junction plates are made of beryllium because of its high strength-to-weight ratio. The plates are of a ribbed design to provide the required stiffness at a minimum weight. All of the structural loads of the generator are carried through these plates. The loads from the fuel block normal to the cold plate are carried through a ring of insulation surrounding the module. The loads in the other two axis are carried through the beryllium bellows support and thermal insulation on the sides.

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4. Thermal Insulation

To achieve a reasonable conversion efficiency, it is necessary to thoroughly insulate the fuel block. This forces approximately 90% of the thermal energy to flow through the elements. The module insulation has been covered in the thermoelectric module discussion. The area around the modules and on each of the four remaining sides is also insulated with Johns Manville Min-K, chosen for its excellent insulation properties under the conditions experienced in the dynamic generator. Min-K can be molded or machined to a desired section, and can carry fairly substantial loads (as high as 200 psi), provided they are distributed over a large area and the operating temperature is under 1000°F. The insulation on the front face of the generator is attached directly to the rotating door and protected by a thin sheet of metal. Sheet metal panels form the remainder of the enclosure on the back, top and bottom. These beryllium sheet metal panels form a tension tie between the cold junction plates.

5. Cooling System

The cold junction temperature of the generator is maintained by pumping coolant through the passage formed between the stainless steel diaphragm and the cold junction plates. A pump is required to maintain the required flow of approximately 750 pounds per hour at a maximum pressure rise of 2 psi across the pump. The design life for the system is one year which is long for any single pump to operate on a continuous basis with the reliability desired. Therefore, it is necessary to place a number of pumps in parallel to assure that the reliability requirements of the system will be attained. Only one pump will operate at any given time. When that pump fails to deliver the required flow, the failure is sensed by the corresponding temperature increase, and the next pump is automatically (or manually) started.

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The pressure supplied by the pumps is only that pressure required to maintain the coolant flow. An additional 35 psia will be superimposed on the coolant system by a small accumulator. It is this pressure that supplies the mechanical loading on the thermoelements. The accumulator uses a spring to supply the required force on a small piston-bellows arrangement to obtain the required pressure on the coolant. The accumulator also compensates for small volumetric changes in the coolant system caused by thermal expansion or contraction.

The radiator (see Figure III-5) will utilize the skin of the vehicle, and the skin thickness has been estimated at 0.050 inch for purposes of thermal and meteoroid armor calculations. To facilitate assembly and heat transfer, the aluminum radiator tube will be swaged into a D shape with the flat side brazed to the vehicle skin. Coolant is circulated through the tubes, giving up its heat to the vehicle skin which dissipates it by radiation to space. The coolant will be either de-ionized water or an organic fluid, such as Sanowax. The coolant selected will depend upon the radiator area available which limits the operating temperature of the coolant and, therefore, its vapor pressure.

Should a loss of the cooling system occur which would result in excessive fuel block temperatures, the control system (or manual override) will activate the spring-loaded emergency heat dump door. This will expose an area of 1.2 square feet on the front side of the fuel block to space. This exposed area will dissipate over 15,000 watts(t) at a fuel block temperature of 2000°F which is greater than the beginning-of-life heat available.

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RADIATOR TUBE DETAIL

Figure III-5. Radiator Tube Detail

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IV. PARAMETRIC EVALUATION AND PERFORMANCE

The primary purpose of this study is to develop preliminary parametric data relating the effect of the lunar surface environment on the performance of a 1.5 kw(e) radioisotope power supply. Specifically the effects of lunar day and night environment on the power supply specific power, system weight, heat rejection radiator parameters (such as effective radiator area and coolant temperature and flow) and system efficiency must be studied.

A. PARAMETRIC EVALUATIONS

For a parametric study of this system where the radioisotopic generator power out is fixed and the sink temperature can vary due to lunar day and night conditions, then the heat to be rejected becomes the controlling factor. In an examination of the heat radiation relationship, the logical approach when the sink temperature varies is to select several radiator areas and determine the resulting radiator temperature and hence the system temperatures and efficiency. The selection of radiator areas of 30, 50, 75 and 100 sq. ft. for each 500 watt generator will essentially define four generators whose power output will vary as the sink temperature varies. For this study, the generators were designed to produce 500 watts during the lunar nights with power drop-off during the lunar day. The hot junction temperature of the thermoelectric system was selected so that it would not exceed 1000°F during the lunar day operation.

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1. Thermoelectric Parametric Study

The thermoelectric analysis for the modules of the generator was performed by the Vince-Tom thermoelectric optimization code for the IBM 709 (Reference 5). The program input includes conversion material properties, element lengths, electrical contact resistivities, and hot and cold junction temperatures. The program output includes thermoelectric efficiency, number and size of elements for a desired electrical output power. The optimization and input data is detailed in Appendix C.

The parametric study was performed for hot junction temperatures of 850 to 1000°F, cold junction temperatures of 100 to 700°F, lead telluride elements length of 0.375 inch and an electrical contact resistance of $150 \mu \Omega \text{cm}^2$. The thermoelectric efficiencies of these thermoelements over the temperature ranges are presented in Figure IV-1.

2. Effective Sink Temperature

To evaluate the effect of the lunar environment it is necessary to determine what the effective sink temperatures of the radiator are. Since there is no lunar atmosphere and it is assumed that no heat is conducted away, then all heat rejection is accomplished by radiation. Therefore, the effective sink temperatures are a function of:

- (1) the radiation view factor to both the sun and the lunar surface
- (2) the solar constant
- (3) the lunar albedo
- (4) the thermal emissivity of the radiator
- (5) the solar absorptivity of the radiator
- (6) the earth shine

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FIGURE IV-1

THERMOELECTRIC EFFICIENCY
LEAD TELLURIDE-2N & 2P

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The sun was considered to be a point source of radiation at an infinite distance, and the lunar surface as taken to be an infinite flat plane. Lambert's cosine law (view factor = $\cos \theta$) was assumed to be valid for this study even through the distribution of planetary heat across the disk of the full moon does not follow Lambert's law, but rather it follows the $2/3$ power of the cosine of the phase angle (Reference 6). However, to avoid unnecessary complications in the analysis, Lambert's law was assumed to apply to a small area. The assumption was also made that the lunar surface was in radiation equilibrium with the incident solar flux since there is an apparent fast temperature response of the lunar surface during an eclipse (Reference 3) and the maximum lunar surface temperature occurs at the subsolar point without a phase lag (Reference 4). The solar constant used was $442.8 \text{ Btu/hr ft}^2 \text{ } ^\circ\text{F}^4$ and the lunar albedo was taken as 0.095.

The earth shine was neglected in the analysis in Appendix E and since, the effective sink temperature cannot go to zero, the minimum sink temperature was taken as the lunar surface night temperature of 210°R . In the analysis, there is no allowance for residual lunar surface heat, so the temperature transition from lunar day to lunar night would not be as sharp as results shown in Figure IV-2. Neither of these factors, however, will influence the maximum sink temperatures, therefore they were not considered to be critical for this study.

It is desired that the radiator surface act as a solar reflector. This normally is accomplished by either conversion coatings, nonmetallic paints, transparent nonpigmented paints, vitreous enamel, transparent conversion or inorganic bonded coatings. Of these,

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FIGURE II-2
RADIATOR
EFFECTIVE SINK TEMPERATURE
ON LUNAR SURFACE

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RADIATOR EMISSIVITY, $\epsilon \sim 0.9$

LUNAR ALBEDO, $\alpha \sim 0.095$

RADIATOR POSITIONS WITH
RESPECT TO LUNAR
SURFACE

e.g. UP
 NORTH
OR
SOUTH

RADIATOR
SOLAR
ABSORPTIVITY, α

0.25 ——— OR ———
0.30 ———

SOLAR AZIMUTH ANGLE, $\phi \sim$ DEGREES

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the lack of information and/or the problems of use (Reference 7) on a large surface rule out all but nonmetallic paints for immediate use. These paints have been studied by Martin, Lockheed, Philco and others, and the results have shown that the solar absorptivity degrades (increases) with exposure to ultraviolet radiation. To date, there has been a negligible effect on the thermal emissivity of the paints. The results of tests by Philco have shown that ZnO methyl silicone (TC-46-20) has an emissivity of 0.89 while the solar absorptivity increases from 0.169 to 0.329 after 700 hours exposure to a 10 sun ultraviolet intensity while the average surface temperature is 495°F. Similar tests have TiO₂ in methyl silicone (Dow Corning #Q9-0090) to have an emissivity of 0.91 while the solar absorptivity increases from 0.24 to 0.28 after 140 hours exposure to 10 sun ultraviolet intensity (Reference 8). Since the possible exposure time to ultraviolet radiation can be 2200 hours in a year, a conservative value of $\epsilon = 0.9$ and $\alpha = 0.25$ were used with a possible degradation to $\alpha = 0.30$. The effects of the degradation are shown in Figure IV-2 for the radiator facing away from the lunar surface. There is a negligible effect on the temperature of a radiator surface facing the lunar surface since here only the input of the lunar albedo (which is small) is affected by degradation. The interaction of the radiator and lunar emissivities is assumed to be negligible since the lunar surface is microscopically assumed to be a black body, therefore the emissivity of the radiator is that of the coating.

The effect of dust on the RIPS is difficult to evaluate since it is presently unknown how thick the layer of dust is on the lunar surface, and hence, how thick a dust layer could be expected on the vehicle. Comparisons of the polarization of moonlight and the polarization of light from various substances in the laboratory have indicated that the

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whole surface of the moon is covered by a volcanic ash which resembles a fine powder in texture. Temperature measurements of the lunar surface during lunar eclipses along with laboratory measurements and theoretical computations have also indicated that the major portion of the lunar surface is covered by a few millimeters of pumice dust which has a thermal conductivity of .000726 Btu/hr-ft-⁰F. (Reference 3). The particle size has been estimated to be 4 mils in diameter, so if a dust layer of 4 mils covered the whole radiator, the temperature increase within the system would be 190⁰F. This is sufficient to discourage the use of a horizontal radiator (facing up) in a severe dust environment.

3. Radiator System

The thermal inventory will vary with the radiator area and sink temperatures since these determine the radiator temperature, the cold junction temperature, and hence the thermoelectric efficiency. The smaller radiator areas will require the larger thermal inventories and hence the larger system weights.

Analyses were performed to determine the trade off between thermal inventory and system weight as the number of radiator tubes varies for a given radiator area. These analyses are detailed in Appendix F.

A previous evaluation determined that the pressure drop across the thermoelectric module coolant passages did not become excessive for coolant passage heights as small as 0.060 inch with a 2000 pounds per hour flow. The temperature differential across the coolant boundary layer and the pressure drop in the radiator tubes became the controlling criteria for optimizing thermal inventory and system weight. Pressure drop

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and manufacturing considerations both indicate that a 0.25 inch diameter D-shape tube is the minimum tube size. A flow rate of 700 to 900 pounds per hour gives reasonable temperature differentials and pressure drops for the module and tubes.

For a given radiator size, the radiator efficiency can be varied by changing the number or spacing of radiator tubes. This changes the radiator temperature and the pumping power required for the coolant. As the number of radiator tubes and the radiator efficiency increase, the pumping power decreases. Eventually however, the resulting increase in system weight will correspond with a negligible gain in system efficiency and hence a minor decrease in the thermal inventory.

Conversely, as the number of radiator tubes decreases, the system becomes less efficient. A decrease down to a few tubes would result in such poor radiator efficiencies that the thermal inventory weight increase would more than offset any weight savings produced by the reduction in the number of tubes.

Therefore, for a given radiator area there exists an optimum number of radiator tubes which will result in a minimum weight system. Due to the gradual change in slope of the weight - thermal inventory curve around the minimum point it is feasible to accept a radiator configuration which is slightly heavier than the minimum weight to take advantage of a further decrease in fuel inventory.

An investigation of the effects of sink temperature on the thermoelectric system (Appendix F) shows that besides the greater weight and thermal inventory resulting from a higher sink temperature, there is also an increase in the number of radiator tubes required for the optimum design. For a given system design, the number of

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radiator tubes must be optimized for the higher sink temperature which will result in the highest radiator efficiencies.

Due to the limitations of the contract scope of work, the optimum configuration for each radiator area was not determined. However, from the analyses in Appendix F, a nine tube radiator has been selected as optimum at lunar night for a 100 square feet radiator. This number of tubes will result in conservatively high values of generator weight and required thermal inventory for all other radiator areas.

B. PERFORMANCE

This work is an extension of the "Feasibility of Isotopic Power for Manned Lunar Missions" (Reference 1), whose purpose was to define a 1.5 kw(e) isotopic power supply for a 14 day lunar orbiting mission. This power supply concept has been examined for possible future use in a manned vehicle on the lunar surface with a design life of one year. The study has been a preliminary parametric evaluation of the effects of the lunar environment on several radiator areas and the resulting power supply performance.

1. Generator Performance

To define the effects of the lunar environment, sink temperature and radiator temperatures on the radioisotope thermoelectric power supply, four representative radiator areas were selected (30, 50, 75 and 100 square feet) for which four thermoelectric generators were examined. The radiators were assumed to be in various positions both parallel and perpendicular to the surface of the moon at the lunar equator. The radiator surfaces were assumed to have an emissivity of 0.90 and a solar absorptivity of 0.25. The degradation to be expected for the radiator coating

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in an ultraviolet environment was found to have little effect on the sink temperatures. The extremes of daytime sink temperatures were estimated to be 510°R maximum for a radiator parallel to the lunar surface and facing up, and 715°R maximum for a radiator parallel to the lunar surface but facing down. The sink temperature for lunar night operation was assumed to be 210°R .

The generators examined produce a net electrical power of 500 watts at $28 \pm 1\%$ volts during lunar night operations while requiring the thermoelectric hot junction temperatures to be 1000°F or less during daylight operation. The generators use sintered high density plutonium oxide (Pu-238) fuel and unsegmented lead telluride thermocouples. The fuel sources were designed as rectangular parallelograms with two-inch beryllium ablators to meet the emergency heat dump and intact re-entry requirements. The two-inch diameter fuel capsules, which were designed to meet the impact and pressure containment requirements, have variable lengths due to the difference in fuel inventory between the generators.

The performance details for each generator are presented in Table IV-1. The radiator area, thermal inventory and weight for a 1500 watt (e) power supply would consist of three such generators. The variations of system efficiency with sink temperature are shown in Figure IV-3. It is seen, if care is taken in positioning the radiator so that the effective sink temperature is less than 600°R , that the loss in power for a 75 square feet radiator between night and day operation is less than 1.5%. This variation is also indicative of the effects of lunar dust on the radiator surface.

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TABLE IV-1

RIPS CHARACTERISTICS
(for each 500 watt T/E generator)

Radiator Surface Area, ft ²	30	50	75	100
At minimum sink temp = 210°R				
EOL net power, watts	500	500	500	500
EOL Gross power, watts	560	560	560	560
EOL load voltage, volts	28	28	28	28
EOL Efficiency, %				
Thermoelectric	4.00	6.29	6.72	6.90
System	3.48	5.28	5.59	5.72
EOL Thermoelectric temperatures, °F				
Hot junction	950	910	870	850
Cold junction	595	385	305	262
At maximum sink temp = 715°R				
EOL net power, watts (e)	444	451	474	480
EOL Efficiency, %				
Thermoelectric	3.60	5.73	6.40	6.68
System	3.09	4.77	5.29	5.50
EOL Thermoelectric temperatures, °F				
Hot junction	~1000	~1000	~1000	~1000
Cold junction	648	488	433	408
Fuel block surface temp. °F	1400	1365	1357	1355
Fuel temp. °F	1517	1482	1474	1472
EOL Thermal inventory, KW (th)	14.30	9.47	8.95	8.74
BOL fuel inventory, KW (th)	14.60	9.61	9.10	8.86
BOL fuel inventory, K gms.	36.87	24.25	22.93	22.39
BOL fuel inventory, K curies	455	300	283	270
Thermoelectric Characteristics				
N&P elements material	Pb Te (unsegmented)			
Length of elements, in.	0.375	0.375	0.375	0.375
Number of thermocouples	984	726	716	708
N-element diameter, in.	0.616	0.466	0.431	0.415
P-element diameter, in.	0.640	0.479	0.441	0.422

EOL = End-of-life

BOL = Beginning-of-life

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FIGURE IV-3

RIPS 500 WATT GENERATOR

SYSTEM EFFICIENCIES

DYNAMIC COOLING SYSTEM

HOT JUNCTION TEMP = 1000 °F AT

 $T_h = 715 °R$

POWER OUT = 500 WATTS AT

 $T_c = 210 °R$

RADIATOR AREA

50 FT

100

75

50

30

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EFFICIENCY ~ %

0

2

4

6

8

100

200

300

400

500

600

700

SINK TEMPERATURE, T_c ~ °R~~CONFIDENTIAL~~

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2. Weight

The estimated weights of the generator components are presented in Table IV-2 as a function of radiator area. Each structural member of the generator has been conservatively sized to assure it has sufficient strength to meet the 100-g loading criteria.

It should be emphasized that the weights presented are not optimized. However, the weights are believed to be conservative for the generators examined. Variations of specific power with sink temperature are presented in Figure IV-4.

3. Conclusions

The generators as discussed in this report are not intended to be the optimum configurations for use on the lunar surface, since complete design, analysis and optimization studies have not been made. However, the more significant lunar environmental effects on the generator and its dynamic cooling system have been determined by this preliminary parametric study.

a. Generator

The worst daylight condition to be expected is when the total generator's radiator is parallel to and facing the lunar surface. If the radiator surface has only a small view of space, generator performance will increase. This performance will continue to increase until the radiator has a complete view of space and is facing away from the lunar surface. However, with this radiator orientation the probability of collecting lunar dust on the surface of the radiator is highest. It has been estimated that a 4 mil covering of dust would increase the effective sink temperature by 190⁰F.

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TABLE IV-2

GENERATOR WEIGHTS
(for each 500 watt (e) generator)

Item	Radiator Area (sq. ft)			
	30	50	75	100
	Weight (lbs)			
Fuel	81.3	53.6	50.7	49.3
Fuel capsules	98.4	64.4	60.7	59.0
Ablator and cladding	175.5	136.0	132.0	128.7
Bellows Support	9.6	7.6	7.2	7.2
T/E Elements and shoes	13.9	4.9	4.2	3.8
Bellows and diaphragm	3.6	2.7	2.6	2.5
Electrical Straps	4.0	2.9	2.8	2.7
ZrO ₂ , Mica, Al ₂ O ₃ sheets	2.8	2.0	1.9	1.8
Min-K Insulation (T/E module & sides)	18.9	15.8	15.6	15.4
Cold Plates	22.0	15.7	14.6	14.1
Door and Back Plates	2.3	2.2	2.1	2.0
Side Plates	2.1	1.5	1.4	1.4
Radiator Sys (tubes, headers, armor and water)*	15.7	14.6	14.4	14.3
Pumps, motors, inverters and controls	7.0	7.0	7.0	7.0
	443.0	331.9	317.2	309.2

*These weights do not include the weight of the
vehicle skin (radiator fins).

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FIGURE II-4

RIPS 500 WATT GENERATOR
SPECIFIC POWER

DYNAMIC COOLING SYSTEM

POWER = 500 W(e) AT $T_s = 715^\circ R$

HOT JUNCTION TEMPERATURE = $1000^\circ F$ AT $T_s = 715^\circ R$

UNSHIELDED WEIGHTS

SPECIFIC POWER ~ WATTS (e) / POUND

2.0

1.5

1.0

0

100

200

300

400

500

600

700

SINK TEMPERATURE, $T_s \sim ^\circ RANKINE$

RADIATOR
AREA
~ SQ. FT.

100

75

50

30

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The possibility exists that this power supply would be utilized in a roving vehicle. If so, it would be desirable to keep the radiator size as small as possible. An investigation of Figures IV-3 and -4 reveals that for a 1.5 kw(e) power supply, radiator sizes below 150 square feet produce heavy, inefficient power sources. Therefore, it is recommended that dynamic radiator areas for lead telluride generators which are less than 150 square feet for a 1.5 kw(e) power supply be used with caution.

A definite crossover point is indicated in radiator area, above which a lead telluride system is more efficient and below which a germanium silicon thermoelectric system is better. (See Chapter II). The extreme thermal environment of the lunar surface might make it profitable to study a dynamic generator design which uses germanium silicon as the conversion material instead of lead telluride. Germanium silicon offers possible weight savings since it can operate in a vacuum and at higher hot and cold junction temperatures. Thus, it may be possible to operate with a radiator area of 100 square feet or less for a 1.5 kw(e) dynamic system and accept the higher radiator temperature. If the radiator surface area available on the vehicle is limited, a static system should also be considered.

b. Fuel source

The use of an ablator to achieve intact re-entry of the fuel capsules was made. However, with a plutonium oxide fuel form of plasma torch micro-spheres (Reference 2), it might be possible to allow the capsules to burnup and disperse the fuel micro-spheres during an abort re-entry. If the amount of fuel would not allow capsule burnup and fuel dispersion, it might be possible to design the system with the

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fuel source in the Command Module during the translunar phase of the mission and eliminate the need for the ablator. With the fuel source in the Command Module, there is still an extremely remote possibility of fuel dispersion if the Command Module would fail to re-enter intact.

The safety requirement of a pressure containment (TZM) for helium buildup adds a considerable amount of weight to the fuel source. This pressure containment might not be required or feasible. If the capsule can be designed to leak the helium as the pressure increases during life, this would eliminate the need for this pressure shell. Also, it might not be possible to obtain in a hot cell the required weld integrity that would prevent helium diffusion or leakage from a capsule operating at 1800°F with high internal pressure. Since the helium pressure does not become a problem for a number of years after fuel encapsulation and the possibility of returning the fuel source is quite high, the real need for this pressure containment is questionable. The capsule material under an abort condition could possibly corrode away before the internal pressure presented a serious problem.

The feasibility of a multiwalled capsule meeting the pressure and impact containment requirements has never been demonstrated. Since there exists a high probability that a detrimental reaction will occur between a TZM pressure containment structure and a Haynes-25 impact structure under conditions of high-temperature impact, any preliminary design using this concept cannot be considered adequate unless the concept is evaluated. The detrimental reactions that could cause concern are:

- (1) High impact energy being transferred to the TZM pressure shell and weakening its structural integrity.

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- (2) Defects in one material propagating through the other material (a crack in the Haynes-25 propagating through the TZM).
- (3) Brittle fracture of the TZM and puncture of the Haynes-25.
- (4) High stress areas, due to welding, shrink fitting, or thermal expansions, causing impact containment failure.

Since no test data exist for a laminated fuel capsule design of this type, an impact test program would provide these data; and a comparison could be made with known results from other impact test programs utilizing Haynes-25 as the fuel capsule material.

Although the problem of criticality (Reference 9) and its effects on the heat source has not been investigated due to contract limitations, it is believed that the heat source can be designed to avoid this problem. However, this is an area in which future design analyses should be preformed. Another consideration for these heat sources is the availability of plutonium. A comparison of the quantities of fuel required and the proposed plutonium production levels (Reference 10) indicates that it will require 2 to 3 years production to fuel a large power system.

c. General

The cooling fluid used in Reference 1 and also for this preliminary parametric study has been water. In actual practice it will not be feasible to use water as a coolant on the moon since the water temperature can be expected to exceed 330°F during lunar day operation. At this temperature, the water vapor pressure exceeds 100 psia which will unnecessarily penalize the system weight.

The total changes in the system required by the mission and lunar environment from the system of Reference 1 is shown in Table IV-3.

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TABLE IV-3

SYSTEM CHANGES REQUIRED FROM THE 14 DAY APOLLO DESIGN*

<u>System</u>	<u>Change</u>	<u>Reason for Change</u>
Heat Source	From Po-210 (Gd Po) to Pu-238 (PuO ₂) fuel	Longer mission requirements
	Larger fuel capsules	Lower power density of Pu-238.
Thermoelectric	Segmented to non-segmented	High cold junction temperatures in lunar day.
	Increased number and size	Lower system efficiencies and temperature differential.
Cooling	From water to other coolant	High coolant temperatures.

*Reference 1

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V. RADIATION SHIELDING

An analysis was made to obtain shielding thicknesses and weights required for the thermoelectric power supplies with different beginning-of-life thermal inventories. The shielding required was that which would limit radiation exposure of crewman to 3 REM for mission lifetimes of 90, 180 and 360 days. Spacecraft configurations and shielding thicknesses for the various systems and mission times were obtained from data presented in Reference 11. To use this data, it was necessary to assume the same spacecraft configuration and power supply location as in the referenced report. Three individual thermoelectric generators are required to supply the total 1.5 kw(e) power for the spacecraft. These were assumed to be located next to each other to reduce shielding weight. The power supply location and the various dose points used to establish shield thicknesses are shown in Figure V-1.

Due to the lack of specific information concerning crewman activities in the space station, several points were chosen as being representative of the activity-time requirements. These points are designated as D_1 , D_2 and D_3 in Figure V-1. Point D_1 is approximately at the center of an assumed living quarters, D_2 the operational area and D_3 the sleeping quarters. Crewman occupancy times assumed at each of these points and each of the missions are given in Table V-1. The sum of the product of these times and the dose rates at the respective points give the total dose to the crewman for the mission.

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FIG V-1 LOCATION OF DOSE POINTS AND GENERATORS
IN RELATION TO SPACECRAFT

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V-3

TABLE V-1

Occupancy Time at Different Stations

Dose Point	<u>Mission Time (days)</u>		
	90	180	360
	Time at Dose Point (hrs)		
D ₁	682	1364	2775
D ₂	756	1562	3065
D ₃	722	1444	2920

According to the data and configurations in Reference 11, the neutron radiation is predominant and requires shielding for all the power supplies and missions examined. Lithium hydride was chosen as the shield material because of its light weight and its ability to attenuate fast neutron fluxes. The lithium hydride maximum thicknesses required to limit the total mission dose to 3 REM are shown in Table V-2. The X, Y and Z directions are shown in Figure V-1.

TABLE V-2

Lithium Hydride Maximum Shield Thickness to give 3 REM for Mission

Watts (t) per generator	Total watt (t) per power supply	<u>Mission Time (days)</u>					
		90		180		360	
		X	Y, Z	X	Y, Z	X	Y, Z
		Shield Thickness (inches)					
8,860	26,580	2.75	6.5	4.25	9.0	5.75	11.75
9,095	27,285	3.0	6.75	4.5	9.25	6.0	12.0
9,610	28,830	3.25	7.0	4.75	9.5	6.25	12.25
14,600	43,800	4.0	8.25	5.0	10.75	6.5	13.75

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V-4

Maximum shield weights to limit the total dose to 3 REM for various mission times are given in Table V-3. The shield weights and thicknesses, assume the occupancy time of Table V-1 and neglect the effect on attenuation of spacecraft structure, equipment, supply tanks, etc.

TABLE V-3

Maximum Shield Weights to give 3 REM for Mission

Watts (t) per generator	Total watt (t) per power supply	Mission Time (days)		
		90	180	360
		Shield Weight (pounds)		
8,860	26,580	330	525	780
9,095	27,285	350	550	815
9,610	28,830	370	580	855
14,600	43,800	470	670	975

The weights given in Table V-3 are upper limits of estimated shield weights.

More exact estimates cannot be made until:

1. The spacecraft configuration is known.
2. The activities of the crew members, locations and the times they will spend performing these activities are known more precisely.
3. The location of the generator or generators in the spacecraft are established.
4. The location of equipment, tanks structure, etc. are known in relation to activity centers and generator location.

With this more detailed information, considerable weight savings could be obtained in shaping the shield by reducing the thickness where it is not required.

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V-5

It should also be emphasized that distance between the radioisotope heat source and operating personnel should be maximized since the dose is directly proportional to the inverse of the distance squared.

When lunar surface personnel mission activities can be detailed, the distinct possibility arises that significant savings in system shield weight can be realized. For longer crew stay times, the possibility of making use of natural terrain obstructions or materials for shielding the fuel source can be factored into isotopic power system/lunar vehicle integration and design.

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APPENDIX A

PLUTONIUM DIOXIDE DATA SHEET

(All values are for Pu-238 O₂ except where indicated otherwise)

A. Physical Properties

1. Melting Point:

Pu-239 O₂ - 2280 ± 50°C (Reference A-1); 2300°C
(Mound Laboratory data)

2. Boiling Point:

4140°K (from vapor pressure data)

3. Vapor Pressure (Reference A-1):

$$\log p(\text{atm}) = \frac{(-29240 \pm 530)}{T} + (8.072 \pm 0.239) \quad (1)$$

4. Density:

Theoretical - 11.46 g/cc
Sintered Particles

Sintering Temp., 1 hour (°C)	Apparent Density (g/cc)
800	5.4
900	5.8
1000	6.4
1100	7.0
1200	7.5
1300	8.2
1400	9.0
1500	9.8
~ 2000	~11.4

Plasma Torch Spheres (best obtained to date)

Pu-239 O₂ - 11.0 g/cc

Pu-238 O₂ - 9.7 g/cc

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B. Thermal Properties

1. Heat of Vaporization:

Pu-239 O₂ (ΔH_{298}) - 147.1 $\pm$ 4.3 kcal/mole at 1 atm He
(Reference A-1); 153.8 kcal/mole at
1 atm O₂ (Reference A-2).

2. Thermal Expansion Coefficient for Pu-239 O₂ (Reference A-1):

<u>Temperature (°C)</u>	<u>$\Delta L/L \times 10^8$</u>
25	110
100	480
200	1380
300	2345
400	3355
500	4415
600	5530
700	6680
800	7950
900	8210
1000	10630

3. Heat Capacity:

C_p at 325°K = 17.4 cal/°K per gfw

4. Thermal Conductivity:

0.0065 cal/cm²/cm/sec./°C at 700°C (Reference A-5)

C. Mechanical Properties

1. Hardness:

Pu-239 O₂ - 1163 kg/mm² (Mound Laboratory data), 105 kg/mm²
(Reference A-2)

No variation from center to edge of particles was observed
on Mound Laboratory measurements.

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2. Crush Strength:

Pu-238 O₂ (74-149 μ diameter) > 1 kg

D. X-Ray Data for Pu-239 O₂

1. Reference A-3:

Symmetry - FCC, C_a F₂ type

$$a = 5.3960 \pm 0.0003 \text{ \AA}$$

$$Z = 4$$

$$D_X = 11.46 \text{ g/cc}$$

Space Group - Fm 3m

Interatomic Distances:

$$\text{Pu} - \text{O} = 2.336$$

$$\text{O} - 4 \text{ Pu} = 2.336$$

Isotypes - Ce O₂, Pr O₂, Tb O₂, Th O₂, Pa O_{2.2},

UO₂, Np O₂, Am O₂, Cm O₂

2. Mound Laboratory data for microspheres:

Composition Pu O_{1.98}

Symmetry - FCC

$$a = 5.405 \pm 0.006 \text{ \AA}$$

$$D_X = 11.34 \text{ g/cc}$$

Space Group - Fm 3m

E. Solubility

1×10^{-9} g/ml after 53 days in Maine sea water; 1.80×10^{-9} g/ml after 81 days in Maine sea water (Mound Laboratory data); 2.76×10^{-8} g/ml in sea water (Reference A-4).

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F. Particle Properties

1. Packing Fraction:

37 - 250 μ diameter Pu-239 O₂ spheres - 75% packing fraction

2. Particle Size:

32 - 250 μ diameter spheres have been produced

3. Particle Distribution:

The following plot gives particle distribution for some 74 - 150 μ diameter particles.

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A-5

G. Radiation Characteristics for Pu-238 O₂

1. Specific Activity - 17.4 curies/g Pu-238.
2. Neutron flux ~ 4×10^4 n/sec/g.

H. Plutonium-238 Fuel Form Characteristics

<u>Form</u>	<u>Density (g/cc)</u>	<u>Melting Point (°C)</u>	<u>Thermal Density (watts/cc)</u>	<u>Specific Power (watts/g)</u>
Metal	15-16	630 ± 10	7.13	0.460
Pu O ₂	11.4	~ 2300	4.51	0.396*
Pu N	14.23	~ 2600	6.17	0.434
Pu C	13.6	~ 1850	5.95	0.438

*Value based on 80% isotopic purity.

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APPENDIX B

FUEL BLOCK ASSEMBLY DESIGN DETAILS

A. Fuel Capsules

The design of the fuel capsule within the fuel block assembly is predicated upon the TZM thickness necessary to withstand the helium pressure buildup during the life of the fuel, and the Haynes 25 thickness required to survive earth impact at terminal velocity.

1. Pressure Containment - The pressure buildup was computed using the perfect gas law which states:

$$PV = \eta RT \quad (1)$$

where:

P - pressure (atmospheres)
V - void volume (liters)
 η - no. moles of helium gas
R - universal gas constant $(8.207 \times 10^{-2} \frac{\text{ATM Liters}}{\text{Mole} \cdot ^\circ\text{K}})$
T - temperature ($^\circ\text{K}$)

The number of helium gas moles, η , is determined from

$$\eta = \frac{A}{A_T W_T} [1 - \text{EXP}(-\lambda t)] \quad (2)$$

A = Initial mass of radioisotope (gm)
 $A_T W_T$ = Atomic weight of helium (gm/mole)
 λ = Decay constant (yr^{-1})
t = Time (yr)

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and if the conductive or convective heat removal is assumed, the temperature, T , with the capsule as a function of time is given as:*

$$T = [T_0 - T_a] \text{EXP}(-\lambda t) + T_a \quad (3)$$

T_0 = Initial fuel temperature, °K

T_a = Ambient temperature, 320°K

The helium pressure and maximum fuel temperature as a function of time are shown in Figure B-1.

The wall thickness, then, is determined from the standard stress relationship:

$$\sigma = P \frac{R^2 + r^2}{R^2 - r^2}$$

P = maximum pressure (psi) = 17,770

σ = tensile strength (psi) = 120,000

R = Outer radius (in.)

r = Inner radius (in.) = 0.790

$$\frac{P}{\sigma} = \frac{17,770}{120,000} = .148$$

$$R = r \sqrt{\frac{1 + .148}{1 - .148}}$$

$$R = r \sqrt{1.34}$$

$$R = (0.790) (1.16) = 0.917$$

The wall thickness is therefore 0.127 inch. To be conservative a 0.130 inch was used as the required thickness for the TZM containment structure.

* During operation, the fuel block is designed such that the temperature is nearly constant; however, equation (3) is correct at times well beyond the mission duration.

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Figure B-1
Helium Pressure Buildup

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2. Impact containment - The velocity at impact is the impact body's terminal velocity - that velocity which a body has attained when the gravity and drag forces on it are equal. This velocity may be evaluated from:

$$V_I = \sqrt{\frac{2W}{\rho C_D A}} \quad (4)$$

where

V_I = impact velocity

ρ = atmospheric density at altitude under consideration

W = weight

C_D = drag coefficient

A = cross sectional area

Using this relationship, the terminal velocity for the present fuel capsule design was calculated and found to be 296 ft/sec.

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APPENDIX C

THERMOELECTRIC CALCULATIONS

1. Thermoelectric Efficiency

The thermoelectric efficiencies used for the lead telluride materials of this study have been calculated from the Vince-Tom IBM program developed by Martin Nuclear. The analysis is set up as follows:

The thermoelectric efficiency is defined as the electrical power produced, P_e , divided by the thermal power, P_t , required or:

$$\eta_{T/E} = P_e / P_t \quad (1)$$

But

$$P_e = I^2 R_e \quad (2)$$

the thermal energy supplied to the thermocouple is:

$$P_t = q_I + q - 1/2 I^2 R_i - I^2 R_c \quad (3)$$

where

I = current

R_e = external load resistance

R_i = internal resistance of thermoelectric elements

R_c = thermoelement contact resistance

q_I = Peltier cooling

q = heat conduction

Now Peltier heat is calculated by:

$$q_I = \alpha I T_H \quad (4)$$

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where

T_H = Hot junction temperature of thermoelectric element
(absolute temperature units)

$\bar{\alpha}$ = Seebeck coefficient = $|\alpha_N| + |\alpha_P|$

N refers to negative element

P refers to positive element

The heat due to conduction is:

$$q = \left[\frac{k_N A_N}{l_N} + \frac{k_P A_P}{l_P} \right] (T_H - T_C) \quad (5)$$

where:

k = element thermal conductivity

A = cross-sectional area of element

l = length of element

The open circuit voltage, E_{oc} , is given by:

$$E_{oc} = \bar{\alpha} (T_H - T_C) = \bar{\alpha} \Delta T \quad (6)$$

$$\text{Define; } S_N = \frac{A_N}{l_N} \text{ and } S_P = \frac{A_P}{l_P} \quad (7)$$

Substituting equations (2), (3), (4), (5), (6) and (7) into equation (1) we

get:

$$\eta_{T/E} = \frac{\bar{\alpha}^2 \Delta T^2 R_e}{(R_1 + R_e + R_c)^2} \quad (8)$$

$$\frac{\bar{\alpha}^2 T_H \Delta T}{R_1 + R_c + R_e} + (k_N S_N + k_P S_P) \Delta T - 1/2 \frac{\bar{\alpha}^2 \Delta T^2 R_1}{(R_1 + R_e + R_c)^2} - \frac{\bar{\alpha}^2 \Delta T^2 R_c}{(R_1 + R_e + R_c)^2}$$

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C-3

Define:

$$X = \frac{R_e}{R_i + R_c} \quad (9)$$

and

$$K = k_N S_N + k_P S_P \quad (10)$$

Rearranging equation (8) using equations (9) and (10) we get:

$$\eta_{T/E} = \frac{\Delta T}{T_H} \frac{X}{X+1 + \frac{K(R_c + R_i)(X+1)^2}{\alpha^2 T_H} - \frac{\Delta T(R_i/2 + R_c)}{T_H(R_i + R_c)}} \quad (11)$$

We desire to maximize the efficiency for all possible variations of the independent variable. This is accomplished as follows: First we maximize with respect to X with all other variables held constant. Thus:

$$\frac{d \eta_{T/E}}{d X} = \left[\frac{\Delta T}{T_H} \right] \frac{X + 1 + \frac{K(X+1)^2}{\alpha^2 T_H} (R_c + R_i) - \frac{\Delta T(R_i/2 + R_c)}{T_H(R_i + R_c)}}{\text{Denominator}} - X \left[1 + \frac{2(X+1) K (R_i + R_c)}{\alpha^2 T_H} \right] \quad (12)$$

Setting the numerator equal to 0 for maximum efficiency

$$\frac{1 + K(X+1)^2 (R_c + R_i)}{\alpha^2 T_H} - \frac{\Delta T(R_i/2 + R_c)}{T_H(R_i + R_c)} - (2X^2 + 2X) \frac{K(R_i + R_c)}{\alpha^2 T_H} = 0 \quad (13)$$

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Rearranging and solving for X

$$X_{\max} = \left[1 + \frac{\alpha^2 T_H}{K (R_c + R_i)} - \frac{\alpha^2 \Delta T (R_i/2 + R_c)}{K (R_c + R_i)^2} \right]^{1/2} \quad (14)$$

$$X_{\max} = \frac{R_e}{R_i + R_c}$$

Now we define a variable B by:

$$B = \frac{S_N}{S_P} \quad (15)$$

For maximum efficiency, the product of the conductance and resistance, $K (R_c + R_i)$ must also be minimized with respect to the element shape factors.

Thus

$$K = (k_P + k_N B) S_P \quad (16)$$

$$R_i = \left(\rho_P + \frac{\rho_N}{B} \right) \frac{1}{S_P} \quad (17)$$

$$R_c = \left(\frac{C_P}{l_P} + \frac{C_N}{l_N B} \right) \frac{1}{S_P} \quad (18)$$

where

 ρ = thermoelement electrical resistivity

C = contact resistivity

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Therefore

$$K(R_i + R_c) = k_P \rho_P + \frac{k_P \rho_N}{B} + \frac{k_P C_P}{l_P} + \frac{k_P C_N}{l_N B} + k_N \rho_P B + k_N \rho_N$$

$$+ \frac{k_N C_P B}{P} + \frac{k_N C_N}{l_N}$$

$$\frac{d[K(R_i + R_c)]}{dB} = 0 = -\frac{k_P \rho_N}{B^2} - \frac{k_P C_N}{l_N B^2} + k_N \rho_P + \frac{k_N C_P}{l_P}$$

Solving for B,

$$B = \left[\frac{k_P \rho_N + \frac{k_P C_N}{l_N}}{k_N \rho_P + \frac{k_N C_P}{l_P}} \right]^{1/2} = \frac{S_N}{S_P} \quad (19)$$

These equations fully define the system, and the efficiencies are plotted in Figure C-1 and based on the characteristics of lead telluride in Figures C-2 thru -7.

2. System Characteristics

After selecting the radiator area -- 75 ft² per generator has been selected as an example -- the radiator and hence the cold junction temperature of the thermoelectrics can be determined. To obtain these values, it must be known how much heat is to be radiated from the system.

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$$q_T = \begin{array}{l} \text{Heat flow} \\ \text{through T/E} \end{array} + \begin{array}{l} \text{Parasitic} \\ \text{heat loss} \end{array} \quad (20)$$

$$q_T = \frac{1}{\eta_{T/E}} (P_G) + k \frac{A}{L} \Delta T$$

q_T - Total radioisotope heat, watts

$\eta_{T/E}$ - Thermoelectric efficiency

P_G - Power out of thermoelements, 560 watts per generator

$k \frac{A}{L}$ - Heat loss through insulation, bellows, etc. 1.068 watts/ $^{\circ}\text{F}$

ΔT - Hot junction minus cold junction temperature, $^{\circ}\text{F}$

The extra 60 watts per generator (560-500) has been allotted for the following items:

Regulator losses	5
Control system	20
Coolant pump	5
Strap losses	30

The parasitic heat losses are based on the analysis in Appendix D. Even though it is recognized that this value isn't a constant for all four generators, a ten percent variation in this value will result in less than one percent variation in the total heat. Therefore the $k \frac{A}{L}$ value of 1.068 watts/ $^{\circ}\text{F}$ is taken to be the same for all generators.

The cold junction temperatures are a given temperature differential above the radiator temperature. These values have been determined as shown in Appendix E.

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At the given sink temperature (210°R) it is necessary to assume a hot and cold junction temperature.

$$T_H = 870^\circ\text{F} \quad T_C = 305^\circ\text{F} \quad T = 870 - 305 = 565^\circ\text{F}$$

$$q_T = \frac{1}{.0671} (560) + 600 = 8950 \text{ watts}$$

The heat to be radiated is

$$\begin{aligned} q_D &= q_T - (500 + 5 + 20 + 5) \\ &= 8950 - 530 = 8420 \text{ watts} = 28,700 \text{ Btu/hr} \end{aligned} \quad (21)$$

The radiator relationship is

$$q_D = \eta_E \sigma \epsilon A_R (T_R^4 - T_S^4) \quad (22)$$

q_D = heat to be dissipated, Btu/hr

σ = Stefan-Boltzman constant, $.1714 \times 10^{-8}$ Btu/hr ft² °R⁴

ϵ = Emissivity, 0.9

η_E = Radiator effectiveness

A_R = Area of radiator, ft²

T_R = Temperature of radiator, °R

T_S = Sink temperature, °R

For the 75 sq. ft. radiator, the radiator temperature is 272°F and the cold junction temperature is 305°F. The system efficiency is the power out (500 watts) divided by the total fuel inventory or $\eta_S = 500/8950 = 5.59\%$.

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C-8

When the sink temperature is increased, the use of equation (22) will give the new radiator temperature. For a sink temperature of 715°R $T_R = 400^\circ\text{F}$ and $T_C = 433^\circ\text{F}$. The hot junction temperature is arrived at by taking a heat balance on the generator.

$$q = \begin{array}{c} \text{parasitic} \\ \text{heat} \\ \text{loss} \end{array} + \begin{array}{c} \text{element} \\ \text{heat} \\ \text{conduction} \end{array} + \begin{array}{c} \text{peltier} \\ \text{cooling} \end{array} \quad (23)$$

$$= k \frac{A}{L} \Delta T + \left[\left(k \frac{A}{L} \right)_P + \left(k \frac{A}{L} \right)_N \right] N \Delta T + N I \alpha T_H$$

T_H - hot junction temperature, °R

N - Number of couples (estimated)

I - Electrical current through couples, amps

α - Average Seebeck coefficient, volts/°F

Subscripts P and N are for the positive and negative elements.

An evaluation of this equation at the sink temperature of 210°F, $I = 9.125$ amp, $\left(k \frac{A}{L} \right)_P = .00672$ watts/°R, $\left(k \frac{A}{L} \right)_N = .00647$ watts/°R, $N = 290$ and $\alpha = 265.7$ volts/°R gives a value of q of 3705 watts. With this value, and a re-evaluation of the thermoelectric properties of k and α , a new hot junction temperature of 997°R is found for a sink temperature of 715°R.

The power out of the system is found by determining the total heat flow through the thermoelectrics; multiplying by the T/E efficiency and subtracting out the power for controls, etc.

$$P = (q_T - k \frac{A}{L} \Delta T) (\eta_{T/E}) - 60 \quad (24)$$

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For this configuration, the power is 474 watts, and the system efficiency is 474/8950 or 5.29%.

3. Thermoelectric Sizing

The number of thermocouples required depends upon the voltage output requirements. Since this generator design is based upon series-parallel circuits so that one half of the generator thermocouples produce the required voltages, the voltage required is 28 volts, and allowing 2 volts for strap losses, the required voltage is 30 volts. From these relationships,

$$V_L = \frac{X}{X + 1} E_{oc} \quad (25)$$

V_L - Required voltage

$$X = \frac{R_e}{R_i + R_c}$$

E_{oc} - Open circuit voltage

$$E_{oc} = N \alpha \Delta T \quad (26)$$

the number of thermocouples required were found to be 358 for each series circuit.

The size of the thermoelements were determined from

$$D_P = \sqrt{\frac{4 S_P L}{\pi}} \quad \text{and} \quad D_N = \sqrt{B} \quad D_P \quad (27)$$

D - diameter

B - defined by equation 19 = .958

L - thermoelement length = .953 cm

$$S_P = \frac{A_P}{l_P}$$

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$$S_P = \frac{[(R_i + R_c) / S_P] N X}{R_L} \quad (28)$$

R_i / S_P - Defined by eq. 17 = 5808 micro ohms

R_c / S_P - Defined by eq. 18 = 1598 micro ohms

R_L - Load resistance, ohm

$$R_L = \frac{V_L^2}{P} \quad (29)$$

V_L = Load voltage, 30 volts

P - Power produced by one series circuit, 280 watts.

The resulting element diameters are $D_P = 0.441$ inch and $D_N = 0.431$ inch.

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FIGURE C-1

THEMEOELECTRIC EFFICIENCY
LEAD TELLURIDE-2N-82P

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C-12

LEAD TELLURIDE
TEGS-2N COLD PRESSED SINTERED

FIGURE C-2
SEEBECK COEFFICIENT

FORM 4-60200

K-E
ATTENTION

15744

84

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LEAD TELLURIDE
TEGS-12M COLD PRESSED & SINTERED
FIGURE C-3
ELECTRICAL RESISTIVITY

NOVA 50000 K.M. 1000000 1000

85

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LEAD TELLURIDE
TAGS-2N, GOLD PRESSED & SINTERED

FIGURE C-4

THERMAL CONDUCTIVITY

8000 & 80000

K-E

86

86

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C-15

LEAD TELLURIDE
TEGS-AP, COLD PRESSED & SINTERED

FIGURE C-5

ELECTRICAL RESISTIVITY

1000 & 0.0005
K&E
JANUARY 1964
1000

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LEAD TELLURIDE
TAGS-2P COLD PRESSED & SINTERED

FIGURE C-6

SEEBECK COEFFICIENT

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FORM 8-64280 K&E

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APPENDIX D

PARASITIC HEAT LOSSES

To determine the required end of life fuel loading for the thermoelectric generator, it is necessary to know not only the thermoelectric efficiency but also the so-called parasitic heat losses from the system. These parasitic losses consist of all heat produced by the radioisotope which escapes the system without having been previously conducted through the thermoelements. In the particular design presented, the parasitic heat losses may be classified into three major categories.

1. Module Insulation

Thermal energy is lost from the system when it passes through the Min-K insulation surrounding the thermoelements. This insulation is also in thermal parallel to the thermoelements and the loss is strictly conduction.

2. Module Bellows

The module bellows is also in thermal parallel to the thermoelements and provides another parasitic conduction loss. The bellows is given an extended length to reduce the conduction loss from this component.

3. Fuel Block

The fuel block is faced by the thermoelectric modules on the two large faces only. Therefore, any heat escaping from the four lateral surfaces is not conducted through the thermoelements and constitutes another parasitic loss. This loss was considered in this study, in order to be conservative, as a conduction loss.

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D-2

4. Summary

The heat losses were therefore determined by the following relationship:

$$q = k \frac{A}{L} \Delta T$$

q = heat flow, Btu/hr

k = heat conduction, Btu/hr-ft-°F

A = heat flow path area, ft²

L = heat flow path length, ft

ΔT = temperature differential, hot to cold sides, °F

Using the particular design studied and a hot junction temperature of 925°F with a cold junction temperature of 240°F, the total parasitic heat losses were conservatively calculated to be 730 watts (t) per generator.

Since the dimensions between generators will not vary greatly as a function of radiator size and the parasitic heat loss is less than 10% of the thermal inventory requirements (end-of-life), the variation in parasitic heat loss from generator to generator was assumed to be proportional to the temperature differential across the thermocouples.

$$\begin{aligned} q_x &= \frac{q_1}{\Delta T_1} \Delta T_x \\ &= \left[\frac{730 \text{ watts}}{925^\circ\text{F} - 240^\circ\text{F}} \right] \Delta T_x \\ q_x &= 1.068 \Delta T_x \end{aligned}$$

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APPENDIX E

EFFECTIVE SINK TEMPERATURE

The external heat input to the radiator is that from the sun, the lunar Albedo and the thermal radiation from the lunar surface. The effective sink temperature can be determined by:

$$\dot{Q}_S + \dot{Q}_A + \dot{Q}_L = \sigma \epsilon T_S^4 \quad (1)$$

$\dot{Q}_S$ - Solar heat flux per unit area, Btu/hr ft²

$\dot{Q}_A$ - Lunar Albedo flux per unit area, Btu/hr ft²

$\dot{Q}_L$ - Lunar thermal flux per unit area, Btu/hr ft²

σ - Stefan-Boltzmann constant, 0.1714×10^{-8} Btu/hr ft² °R⁴

ϵ - Radiator thermal emissivity

T_S - Effective sink temperature, °R

Since the heat input from earth shine is not accounted for in this relationship, it would be possible for night time temperatures to go to zero °R; however, since this does not actually happen, the night sink temperature is arbitrarily limited to the 210°R measured temperature of the lunar night surface (Reference 3).

The heat inputs are affected by the view factor of the radiator to the sun and the lunar surface. For this study, the lunar surface is assumed to be flat, and therefore its primary direction can be represented by the X-Y-Z orthoganol axis of Figure E-1, and the lunar surface itself is represented by the X-Y plane. The radiator can have any attitude, which is represented by the unit vector normal to the radiator surface $\bar{N}$.

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E-2

Since the most severe heating conditions are those at the lunar equator, the radiator is placed there so that the sun will always lie in the Y-Z plane, and the attitude of the sun with respect to the lunar surface is shown by the unit vector $\bar{S}$. Therefore, the view factor of the sun on the radiator can best be expressed by the vector dot product:

$$F_S = \bar{S} \cdot \bar{N}$$

$$F_S = S_x N_x + S_y N_y + S_z N_z$$

$$F_S = \cos \gamma \cos \phi - \sin \gamma \sin \phi \cos \beta$$

F_S - View factor of sun to radiator surface

$\bar{S}$ - Unit vector in direction of sun

$\bar{N}$ - Radiator surface normal unit vector

S_x, S_y, S_z - Scalar value of $\bar{S}$ on X, Y or Z axis

N_x, N_y, N_z - Scalar value of $\bar{N}$ on X, Y or Z axis

γ, ϕ, β - Defined as in Figure E-1

The view factor of the radiator to the lunar surface is determined by applying Lambert's cosine law -

$$F_L = 1/2 (1 - \cos \gamma)$$

F_L - View factor of lunar surface to radiator surface.

Therefore, for a radiator whose surface normal vector lies in the vertical plane, at an angle of 45° from the vertical toward the rising sun, the view factors to the sun and lunar surface are as follows when the solar azimuth is -30° .

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E-3

$$F_S = (\cos 45) \cos (-30) - (\sin 45) \sin (-30) (\cos 90) \\ = 0.966$$

$$F_L = (1 - \cos 45)/2 = 0.3535$$

The heat inputs to the radiator are then defined.

$$\dot{Q}_S = S \alpha F_S$$

$$\dot{Q}_A = S \nu \alpha F_L \cos \phi$$

$$\dot{Q}_L = S (1 - \nu) F_L \cos \phi$$

S - Solar constant on lunar surface, 442.8 Btu/hr ft² °R⁴

α - Solar absorptivity of radiator surface

ν - Lunar Albedo, 0.095

The quantity of heat input for the previously discussed radiator which has a solar absorptivity of 0.25 and an emissivity of 0.9 is -

$$\dot{Q}_S = 107 \text{ Btu/hr}$$

$$\dot{Q}_A = 3 \text{ Btu/hr}$$

$$\dot{Q}_L = 122 \text{ Btu/hr}$$

The sum of these heat inputs will result in a sink temperature of 624°R

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E-4

Figure E-1

- $\vec{N}$ = Radiator surface normal vector
- $\vec{S}$ = Unit vector in direction of sun
- β = Angle between solar - vertical plane and normal-vertical plane
- γ = angle between vertical and surface normal
- ϕ = solar azimuth angle

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APPENDIX F COOLING SYSTEM DESIGN

The cooling system shown schematically in Figure F-1 has to be evaluated for both minimum system weight and minimum thermal inventory considerations.

The solution of the problem became an iterative process, the first step of which was to determine the energy which had to be dissipated by the cooling system. This was accomplished by the relationship

$$q_c = \frac{1}{\eta_{T/E}} \left[\frac{P_o}{\eta_R} + \frac{P_c}{\eta_I} + \frac{P_P}{\eta_I} + P_s \right] + K \frac{A}{L} (T_H - T_C) - \left[\frac{P_o}{\eta_R} + \frac{P_c}{\eta_I} + \frac{P_P}{\eta_I} \right]$$

q_c - Heat to be dissipated, watts

P_o - Power to load, watts

P_c - Controls power, 20 watts

P_P - Pump power, watts

P_s - Strap losses

$\eta_{T/E}$ - Thermoelectric efficiency

η_R - Regulator efficiency, 99%

η_I - Inverter efficiency, 90%

$K \frac{A}{L}$ - Thermal loss through insulation, 1.068 watts/ $^{\circ}$ F

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F-2

T_H - Hot junction temperature, 925°F

T_C - Cold junction temperature, °F

The pump power is a function of the coolant flow rate and pressure drop.

$$P_P = 3.764 \times 10^{-4} \frac{\dot{W} \Delta P}{\eta_P}$$

$\dot{W}$ - coolant flow, lbs/hr

ΔP - Pressure drop through cooling system, ft of water

η_P - Pump efficiency, 15%

The coolant flow rate is equated to the dissipated heat by -

$$\dot{W} = 3.412 q_c / C_p \Delta T_m$$

C_p - Specific heat of coolant, 1.0 Btu/lb - °F

ΔT_m - Average coolant temperature rise across module, °F

The pressure drop through the system is a function of the system geometry. All entrance, exit, bend and expansion pressure losses, K , are assumed to have no pressure recovery. The pressure drop is the sum of the pressure drop through the module, connecting lines and radiator.

$$\Delta P = \left[\left(K + f \frac{L}{D} \right) \frac{(\dot{W}/3600)^2}{2g \rho^2 A^2} \right]_m + \left[\left(K + f \frac{L}{D} \right) \frac{(\dot{W}/3600)^2}{2g \rho^2 A^2} \right]_L + \left[\left(K + f \frac{L}{D} \right) \frac{\dot{W}/3600^2}{2g \rho^2 A^2} \right]_R$$

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F-3

f = coefficient of friction

K_m = 1.5

L_m = 1.083 ft

D_m = $1.333 \frac{H}{H + 16}$ ft

$\dot{\omega}_m$ = $1/2 \dot{\omega}$

g = 32.17 ft/sec²

ρ = 60.1 lb/cu ft

A_m = .1111 H ft²

H = coolant passage height, in.

D_L = $0.118 r_i \sqrt{N}$ ft

$\dot{\omega}_L$ = $\dot{\omega}$

A_L = $.0109 N r_i^2$ ft

N = number of radiator tubes

K_R = 1.5

L_R = 12 ft

D_R = $.02545 r_i$ ft

W_R = W/N

K_L = 4.3

L_L = 2 ft

Δ_R = $.0109 r_i^2$ ft²

r_i = radiator tube internal radius, in.

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The coefficient of friction, f , is the Darcy factor which is based on the function $f = f(Re)$.

$$Re = \frac{(\dot{W}/3600)}{\mu} \cdot \frac{D}{A}$$

μ - viscosity, 0.205×10^{-3} lb/ft-sec

The combination of these equations will give the heat to be dissipated, q_c , for given values of cold junction temperature, T_C , module coolant passage height, H , coolant temperature increases across the module, ΔT_m , radiator tube internal diameter, r_i , and the number of radiator tubes, N .

To determine the temperature differential between the T/E cold junction and the radiator fin root, it is first assumed based on previous studies that the temperature differential between the cold junction and the module-fluid interface is 20°F .

The heat coming off the thermoelectrics and the parasitic heat losses are considered in the temperature difference across the module-fluid interface, ΔT_2 . This is slightly conservative since a portion of the parasitic heat does not flow across this interface.

$$\Delta T_2 = \frac{q_c (3.412)}{h_b A_D}$$

A_D = Surface area of diaphragm, ft^2

$$h_b = 0.664 \frac{k}{L_D} Re^{1/2} Pr^{1/3} \quad \text{P. 268 of Reference F-1.}$$

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$$Re = \frac{\dot{W}/3600}{\mu} \frac{L_D}{A_m}$$

L_D - Length of module in direction of coolant flow, feet

Pr - Prandtl number of water, 1.88

k - Thermal conductivity of water, 0.394 Btu/hr-ft-°F

A_m - Area of module coolant flow passage, .1111 H sq-ft

H - Coolant passage height, inches

The temperature differential across the fluid-radiator interface is -

$$\Delta T_3 = \frac{q_c (3.412)}{h_c A_c}$$

A_c = Internal area of cooling tubes, sq. ft.

$A_c = 5.14 r_i N$

h_c = Heat transfer coefficient, coolant to tubes, Btu/hr ft² °F

$h_c = 0.023 (C_p \frac{\dot{W}/N}{A_T}) Re^2 Pr^{-2/3}$ p. 347 of Reference F-1

C_p - Specific heat of water, 1.00 Btu/hr-ft-°F

A_T - Radiator tube area, .0109 r_i^2 sq. ft.

Pr - Prandtl number 1.88

$$Re = \frac{\dot{W}/3600}{\mu} \frac{D_e}{A_T N}$$

D_e - Radiator tube hydraulic diameters, .1017 r_i ft

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The temperature drop from the fluid-radiator interface to the fin root is

$$\Delta T_4 = q_c \frac{\Delta X_m}{k_m A_R} \quad (3.412)$$

ΔX_m - Distance from interface to skin surface, .0228 feet

k_m - Thermal conductivity of metal, 100 Btu/ft-hr-°F

A_R - Area of heat flow path, sq. feet

$$A_R = 2 r_i L_T N = 24 r_i N$$

The equations for ΔT_2 , ΔT_3 and ΔT_4 plus temperature differential between the cold junction and diaphragm and the assumed T/E cold junction temperature define the fin root temperature of the radiator. Here it is necessary to assume a radiator area, A_R , from which the fin length, L_F , (Figure F-1) can be computed.

$$A_R = 2 \left[\frac{L_F + (r_i + .02)}{12} \right] N L_T$$

or

$$L_F = \frac{A_R}{2N} - (r_i + .02) \quad \text{where } L_T = 12 \text{ feet}$$

The fin efficiency can be determined from the standard fin relationship

$$\eta_F = \frac{\tanh J}{J}$$

where

$$J = \sqrt{\frac{2 h_R}{k_m t_s}} \quad (L_F)$$

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F-7

- k_m - Thermal conductivity of metal, 100 Btu/hr-ft-°F
- t_s - Thickness of vehicle skin, 0.050 in.
- h_R - Equivalent heat transfer coefficient, radiator to space,
Btu/hr-ft² - °F

$$h_R = \frac{.1704 \left[\epsilon \left(\frac{T_R}{100} \right)^4 - \left(\frac{T_S}{100} \right)^4 \right]}{T_R - T_S}$$

- ϵ - skin emissivity, 0.90
- T_R - Temperature of radiator, °R
- T_S - Sink temperature, °R

The surface effectiveness of the radiator can be determined by -

$$\eta_E = \frac{\eta_F L_F + r_i + .02}{L_F + r_i + .02}$$

From this information, the quantity of heat flow from the radiator can be determined by -

$$q_R = \eta_E A_R h_R (T_R - T_S)$$

The heat flow from the radiator can then be compared with the heat to be dissipated.

Discrepancies between the two quantities can be adjusted by one of three methods.

1. Adjust the number of radiator tubes for minor adjustments
2. Vary the cold junction temperatures
3. Change the radiator area

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Once a consistent set of numbers have been arrived at, the fuel inventory q_{ET} , at end of life can be determined.

$$q_{ET} = q_c + \left[\frac{P_o}{\eta_R} + \frac{P_c}{\eta_I} + \frac{P_P}{\eta_I} \right]$$

For a one year mission, the fuel inventory at beginning-of-life is -

$$q_{ET} = 1.015 \ q_{ET}$$

The weight of the following parameters was included.

1. fuel, Pu-238
2. capsule with 100% void volume, Tantalum, TZM and Haynes-25
3. radiator tubing, aluminum
4. radiator armor, aluminum
5. coolant, water

The radiator armor thickness was determined by the relationship of Appendix G.

The weight of the system was defined as

$$M = (\text{Fuel} + \text{Capsule}) + (\text{Tubing}) + (\text{Armor}) + (\text{Coolant})$$

From these relationships, the thermal inventory and weights were computed at a 300°R sink temperature for radiator areas of 50, 75 and 100 sq. ft., and at a 600°R sink temperature for a radiator area of 75 sq feet (Figure F-2). The optimum number of radiator tubes is found to increase as the radiator area decreases or the sink temperature increases. From this, a nine tube radiator has been selected as optimum at lunar night for a 100 sq. ft. radiator. This number of tubes, then, will result in

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conservatively high values of generator weight and required thermal inventory for all other radiator areas.

From these results, the radiator efficiency and the temperature differential, radiator to T/E cold junction, required in Appendix C are tabulated below.

Radiator Area ft ²	Radiator Efficiency %	Temperature Efferential Radiator to Module °F
30	95	40
50	92	36
75	87	33
100	80	32

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FIGURE F-1

FIGURE F-2

COOLING SYSTEM OPTIMIZATION

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APPENDIX G

DYNAMIC SYSTEM RADIATOR ARMOR REQUIREMENTS

The radiator tubing is an aluminum "D" shaped tube, 20 mils thick. The radiator tubes are attached to the vehicle skin which is 50 mils thick. Between these two items is a strip of aluminum which will provide additional armor requirements.

As given in Reference G-1, the expression for the probability of no penetration of meteoroids through a given thickness of metal in space is:

$$P_s = \left[\frac{1}{e} \right]^{T \propto A (\psi)^3 \beta (t)^{-3} F}$$

where:

P_s = probability of no penetrations or probability of success

T = length of time exposed

A = area exposed to meteorite bombardment

t = thickness of metal (or depth of penetration)

$\propto$ = meteoroid influx rate

ψ = a constant dependent on properties of the meteoroids, the target metal and the directional distribution of the meteoroids

β = a constant related to the meteoroid influx distribution

F = a correction factor for oblique incidence

In References G-2 and G-3, ψ is given as:

$$\psi = \frac{a \gamma}{2.54} \left(\frac{6}{\pi} \right)^{1/3} (\rho_p)^{-1/3} \left(\frac{62.4 \rho_p}{\rho_t} \right)^{\phi} \left(\frac{V}{C} \right)^{\theta}$$

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G-2

and from the expression for P_s , the thickness, t , required for a given probability of success is:

$$t = \frac{a \bar{r}}{2.54} \left(\frac{6}{\pi} \right)^{1/3} (\rho_p)^{-1/3} \left(\frac{62.4 \rho_p}{\rho_t} \right)^\phi \left(\frac{\bar{V}}{C} \right)^\theta \left[\frac{\alpha AT}{-\ln P_s} \right]^{1/3\beta} \left[\frac{2}{3n\theta\beta + 2} \right]^{1/3\beta}$$

where:

t = required single material armor thickness (in.)

a = thin plate and spall adjustment

$\bar{r}$ = 2, an experimental constant

ρ_p = meteoroid particle density (0.44 gm/cm³)

ρ_t = target material density (172 lb/ft³)

ϕ = 1/2, an experimental constant

$\bar{V}$ = average meteoroid velocity (98,400 fps)

c = speed of sound in target material

$$c = 12 \sqrt{\frac{E_t g}{t}}$$

E_t = Young's modulus at target temperature (10⁷ lb/in.²)

g = 32.2 fps²

θ = 2/3

α = 2.66 x 10⁻¹¹

constants for meteoroid flux distribution
suggested by NASA as of December 1963.

β = 1.34

A = exposed target surface (ft²)

T = mission time (days)

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G-3

P_s = probability of no penetration of metal of thickness t

$n = 1$, for isotropic flux distribution where oblique penetrations are dependent on the normal component of velocity.

The final term, $F = \frac{2}{3n\theta + 2}$, is a correction factor to change oblique incidence for isotropic flux distributions to effective normal incidence.

By appropriate rearrangement and insertion of the suggested values, the expression for thickness can be written as a function of vehicle and mission quantities alone:

$$t = (\text{constant}) [E_t^2 \rho_t]^{-1/6} \left[\frac{AT}{-\ln P_s} \right]^{1/4}$$

where the constant is

$$C = \alpha^{1/4} g^{-1/3} (12)^{-2/3} \left[\frac{a \gamma}{2.54} \left(\frac{6}{\pi} \right)^{1/3} (\rho_p)^{-1/3} (62.4 \rho_p)^{1/2} (V)^{2/3} \right] \cdot \left[\frac{2}{3n\theta + 2} \right]^{1/4}$$

Therefore

$$C = 2.6 \text{ and } t = 2.6 [E_t^2 \rho_t]^{-1/6} \left[\frac{AT}{-\ln P_s} \right]^{1/4}$$

The area, A , was computed for 9 radiator tubes each of which has an internal diameter of 0.25 inches and is 12 feet long. The probability of success was taken as 0.9999 for a one year mission. This resulted in an armor thickness of 0.204 inches.

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