MND-P-2373

PHYSICS AND MATHEMATICS

13-WATT CURIUM FUELED THERMOELECTRIC GENERATOR FOR A SIX-MONTH SPACE MISSION

Final Report - Subtask 5.8

By
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July 1960

Martin Company
Baltimore, Maryland

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FINAL REPORT
13-WATT CURIUM FUELED THERMOELECTRIC
GENERATOR FOR A
SIX-MONTH SPACE MISSION

July 1960

Subtask 5.8

Nuclear Division
Martin Company
Baltimore, Maryland

Project Engineer

FOREWORD

This is the final report on Subtask 5.8 of Atomic Energy Commission Contract AT(30-3)-217 with The Martin Company. The work was completed in July 1960.

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SUMMARY

This report outlines the development by The Martin Company of a design concept for a thermoelectric generator suitable for use in extra-terrestrial space missions for periods up to six months. The generator derives its power from the radioactive decay of the isotope Curium-242. It has been optimized on the basis of minimum weight consistent with environmental safety and performance requirements. It weighs about 16.6 pounds and produces 13 watts of d-c power (at 3 volts output) continuously over its six-month operational life.

I. INTRODUCTION

This program was initiated by The Martin Company in October 1959 as part of a general developmental effort in the field of isotopic power application, sponsored by the Atomic Energy Commission's Missile Projects Branch in the Division of Reactor Development.

The general objective of the program was to design a reliable, light-weight and efficient power supply suitable for operating electronic instruments aboard extraterrestrial space probes. Thermoelectric semiconductors were to be used for conversion of thermal energy to electrical energy so that the generator would contain no rapidly moving parts and would operate independently of its environment. Curium-242 was specified as the isotopic fuel because:

- (1) Its half life was of the same order of magnitude as most of the missions envisaged.
- (2) Projections of production costs indicated that it would be relatively inexpensive.
- (3) Problems of external radiation from the heat source would be minimized.

Work on the conceptual design of a generator for a six-month space mission proceeded for about four months; it was then suspended in favor of a design of a generator specifically for a hard lunar impact mission. Development of the latter concept is described in a parallel report, MND-P-2374 (Ref. 1). The original study was resumed at the request of the AEC in April 1960 and was carried to conclusion.

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II. DESIGN APPROACH

A. SPECIFICATIONS

Development of the generator was coordinated with the Jet Propulsion Laboratory of the California Institute of Technology, so that the unit would meet the general requirements of space probe packages, under concurrent development at CIT for the National Aeronautics and Space Administration. The criteria established for the conceptual design proved to be of the utmost significance in the work which ensued. Those criteria were:

- (1) Electrical power--The generator was to deliver 13 watts at 3 volts d-c of continuous, unregulated electrical power for a period of at least six months following launching into space.
- (2) Weight--The integrated power supply, consisting of generator, d-c to d-c converter and voltage regulator, was not to exceed 18 pounds in weight, of which 15 pounds were allocated to the generator itself. The scope of this subtask did not include design of the voltage converter or regulator.
- (3) Environment--The unit was to operate in normal fashion in an undefined space trajectory (for example, a circumlunar probe), following subjection to missile-induced conditions typical of the Vega launch vehicle:
 - (a) Acceleration--rotation about any axis at 700 rpm.
 - (b) Vibration--15 g of white Gaussian noise from 20 to 2000 cps for 10 minutes, in each of three mutually perpendicular planes.
 - (c) Shock--4 axial shocks of 25 g, each lasting 20 milliseconds, with a rise time of 1 millisecond.

Development of the Vega vehicle was cancelled during the course of the program; these conditions were then held to be illustrative only.

- (4) External radiation--Because of the presence of radiation-sensitive equipment in the JPL payloads, external radiation emanating from the generator was specified not to exceed 7 gamma photons/cm²/sec above 100 kv, at a point 10 cm from the surface. In the case of an anisotropic radiation field, the generator could be

oriented with respect to the rest of the payload so that the minimum radiation level would be seen by the payload. No restriction was placed on neutron fluxes external to the generator during its operational life.

For safe ground handling, external radiation levels were to be reduced to a maximum of 60 mrem/hr at a distance of 1 meter. Design of a suitable shipping cask to limit external radiation to this figure was included in the scope of the work.

- (5) Fuel form--Curium-242 was specified as the radioisotope to be evaluated in the study. An extensive program aimed at developing this isotope as a heat source was being undertaken concurrently as Task 6 under Contract AT(30-3)-217. The chemical form or geometry to be employed was not specified and was to be determined by the state of curium technology and the demands of the generator design.

Its half life of 163 days and specific thermal power of 122 watts/gram make Curium-242 a suitable heat source for minimum size generators having moderate operational lives of 3 to 6 months. Biological-shielding problems are minimized with Cm-242 because of the relatively low level of ionizing radiation emanating from sealed sources of the isotope.

B. ALTERNATIVE DESIGNS CONSIDERED

In the approach to this study, several alternative designs were evaluated with respect to the specifications established. They included:

- (1) Several units of the SNAP III design, fueled with curium instead of polonium, connected electrically. In SNAP III, heat is conducted directly from the source to the thermoelectric elements.
- (2) An enlarged version of SNAP III.
- (3) A generator in which heat is transferred from the fuel container to a surrounding spherical collector shell by radiative processes. Thermoelectric elements are mounted normal to the shell and are in turn enclosed within a spherical radiator shell (Fig. 1).

Fig. 1. Spherical 13-Watt Generator Concept

- (4) A modification of Item (3) in which the spherical shells are replaced by 14-sided polyhedrons so that the thermoelectric elements can be joined to flat surfaces rather than to spherical surfaces (Fig. 2).
- (5) A radiative-type generator consisting of coaxial right circular cylindrical surfaces, with the cylindrical heat source mounted on the axis (Fig. 3).

Detailed preliminary analyses of the various types of generator design may be found in other Martin Nuclear Division internal and external reports (Refs. 2, 3, 4 and 5).

Heat-transfer analyses performed during evaluation of the various design concepts indicated that a spherical configuration utilizing radiative heat transfer would have optimum weight and performance characteristics. Manufacturing and assembly considerations, however, suggested that modifications of the spherical configuration would be necessary. In particular, the making and retaining of good physical contact between the thermoelectric elements and their connectors and heat collector or radiator surfaces appeared particularly troublesome. In addition, access to the central void region for purposes of assembly, disassembly or repair was obviously restricted; and addition of equipment for mounting the generator or for providing a thermal bypass was made more complex.

As a compromise between the conflicting requirements of manufacturing ease, minimum weight and maximum performance, the cylindrical radiative design was chosen for further study.

Fig. 2. Polyhedral 13-Watt Generator Concept

Fig. 3. Cylindrical 13-Watt Generator Concept

III. DEVELOPMENT OF FINAL DESIGN

A. PRELIMINARY SELECTION AND SIZING OF COMPONENTS

As the first step in arriving at the final design, a general layout of the cylindrical configuration was made, and the overall weight and performance of its component parts were assessed.

The five basic components evaluated were:

(1) Heat source

The design of the long, cylindrical heat source featured a capsule of curium oxide at one end, with the remainder of the rod as a gamma shield. Gold was chosen as the shielding material because of its excellent gamma-attenuation properties and high thermal conductivity. The entire rod acted as a thermal radiator.

(2) Collector (or hot junction)

The collector was a thin cylindrical shell of aluminum alloy, coaxial with the heat source. (Aluminum was used only to simplify analysis. It was replaced by a more suitable material at a later stage of development.)

(3) Converter

The thermal-electrical conversion system consisted of pairs of cylindrical P- and N-type thermoelectric semiconductors, with their axes normal to the surface of the collector. Doped lead telluride was selected as the thermoelectric material because of the availability of detailed knowledge of its thermal, mechanical and electrical properties.

(4) Insulation

A layer of electrical and thermal insulation was distributed around the thermoelectric elements. Johns-Manville Min-K insulation was chosen because of its fabricability and low thermal conductivity.

(5) Radiator

The outer enclosure, a thin aluminum cylinder, served as the surface from which unconverted heat was dissipated to the environment. Electrical connectors and other structural members could be attached to the aluminum shell. Figure 3, as noted before, shows the resultant geometry at a later stage in the design.

The following assumptions were made in the required calculations:

- (1) No heat is lost except that which passes through the insulation.
- (2) The entire heat source exhibits a uniform surface temperature.
- (3) The collector surface can be maintained at a uniform temperature of 1000° F (the maximum permissible for lead telluride elements).
- (4) The collector and radiator are assumed to be aluminum for weight and thermal analysis, although other materials may be found to be more suitable.
- (5) The radiator has a fixed area and a constant, uniform temperature.
- (6) Maximum overall thermal-electrical conversion efficiency is attained.
- (7) A full vacuum exists in the region between the heat source and the collector.
- (8) No special consideration is given at this point to the weight of the device to be required for bypassing surplus heat at the beginning of the operational life of the generator.
- (9) For the preliminary study, no losses due to electrical-contact resistance at the ends of the thermoelectric elements are considered.

The procedure followed in the calculations was to size the heat source on the basis of an assumed overall efficiency. The radiator temperature for a given size of generator was determined with Singer's equation for radiation to infinite space. This figure, together with the preselected hot junction temperature, fixed the thermoelectric and thermal efficiencies. If the resulting calculated overall efficiency did not agree with the assumed value, another overall value was selected and the process repeated until the two values coincided. The results obtained are given in Table I.

TABLE I
Characteristics of Cylindrical Radiative-Type Generator

<u>Results</u>	<u>All Units 8 Inches in Height</u>		
Unit diameter (inches)	5.5	6	6.5
Efficiency (%)			
Thermoelectric	6.72	6.83	6.97
Thermal	91.3	91.4	92.6
Overall	6.1	6.2	6.4
Temperature (°F)			
Hot junction	1000	1000	1000
Cold junction	330	310	300
Heat loss (watts)	21	18	15
Weight (pounds)			
Heat source and container	2.41	2.41	2.41
Outer shell	0.68	0.83	0.93
Inner shell	0.34	0.32	0.30
Thermoelectrics	1.43	1.34	2.32
Insulation	1.53	2.01	2.51
Structures (25%)	<u>1.60</u>	<u>1.73</u>	<u>2.12</u>
Total	7.99	8.64	10.59
Thermoelectrics			
Number of pairs	29	28	28
Length (inches)	0.75	0.75	1.00
Area			
P element (in. ²)	0.1218	0.1194	0.1485
N element (in. ²)	0.1019	0.0984	0.125

B. DETAILED DESIGN AND EVALUATION OF COMPONENTS

1. Fuel Capsule and Heat Source

a. Mechanical Design

Initially, using oxide pellets as the curium source was considered, but this approach was discarded because of concern over radiation damage to the compound and increased external radiation caused by alpha-neutron reactions on oxygen. Subsequently, a fuel form consisting of an alloy of gold and curium (5 Au:1 Cm) was selected because of its potential ease of fabrication and good heat-transfer characteristics. The gold also serves to lower the specific power of curium to a more appropriate level.

The shape of the fuel--a right circular hollow cylinder--was a compromise between heat-transfer and mechanical or fabrication concerns. It is desirable to place the fuel at the end of the heat source, close to the area where surplus heat is to be dissipated, to avoid a large temperature gradient over the length of the heat source. At the same time, a solid cylinder, while preferable for fabrication, gives rise to excessive axial temperatures. A hollow cylinder is more difficult to manufacture, but it provides an internal void space to be used as a reservoir for storage of the helium gas which accrues from the alpha decay of curium. It also eliminates the centerline-temperature problem.

Because Curium-242 decays to Plutonium-238, it is necessary to enclose the curium in a thin tantalum outer liner, to prevent corrosive attack of the primary encapsulation material by the plutonium. Tantalum is essentially inert to plutonium at the temperatures to be experienced by the heat source.

An inner liner of tantalum is provided for the curium-gold mixture to support it in the event of unpredictable deterioration of the alloy. Very small holes drilled in the liner permit escape of helium to the central void region.

Based on a final overall conversion efficiency of 4.8%, 6.3 grams of Curium-242 are required to give an initial thermal output (at the time of encapsulation) of 755 watts. It is expected that the curium will not be separated in pure form, but as an Americium-241 alloy containing 45% curium by weight. Therefore, the weight of the Am-Cm mixture required is 14 grams, and another 70 grams of gold diluent bring the weight of the fuel to 84 grams in a volume of 0.284 cubic inch.

According to calculations made by the Safety Analysis Group, the pressure buildup of helium in the void space can be expected to rise to 7870 psi at the end of the useful life of the generator (240 days after encapsulation). The resultant unsupported hoop stress induced in the tantalum container (1/32-inch wall thickness) would approach 110,000 psi, about 10 times the ultimate stress for this metal at operating temperatures. The tantalum vessel cannot be made thicker because of the problems of assured burnup upon re-entry to the earth's atmosphere or upon final-stage failure at near-orbital velocities. The tantalum is therefore backed up by an outer container of Hastelloy C, a material with excellent high temperature strength and resistance to corrosive attack by sea water, but readily burnable under re-entry conditions. The outer container must be installed within a few days after encapsulation of the curium in the tantalum, or distortion and ultimate failure could occur in the 60-day period between encapsulation and launch.

If the dimensions of the Hastelloy C vessel are set at 15/16 inch ID and 1-1/4 inches OD, the maximum hoop stress induced in it by evolution of helium after six months of operation is 27,900 psi. The ultimate strength of this material is 43,000 psi, and the yield strength is 36,000 psi at 1600° F. The mean operating temperature of the Hastelloy C is expected to be about 1450° F. While this gives an additional small factor of safety, any unexpected temperature rise much above 1600° F would cause rupture of the container, since the strength of the material decreases rapidly with increased temperature in this region.

At 60 days after encapsulation (the anticipated time of launch), the maximum hoop stress would be 9770 psi. Maximum permissible capsule temperature at this time would be about 2000° F, and launch pad aborts resulting in combustion of launch-vehicle fuel could conceivably cause rupture of the container, assuming that no bulging of the container occurs prior to rupture. A more detailed analysis of potential failure during operational sequences may be found in a companion report, MND-P-2366 (Ref. 6). A detailed drawing of the heat source assembly is shown in Fig. 4.

As stated in Section II, the fuel capsule was to be attached to one end of a long metallic cylinder for optimum heat transfer and reduction of temperature gradients. During the course of the final design, gold was replaced by tungsten in the heat source since tungsten offered superior gamma-shielding properties, greater high temperature strength (no encapsulation required) and good thermal conductivity.

b. Thermal analysis

Figure 5 represents the simplified configuration of the heat source employed for thermal analysis. On the assumptions that the outer surface of the fuel capsule is at a uniform temperature and that there is a uni-

Fig. 4. Fuel Capsule Space Mission Generator

Fig. 5. Fuel Capsule and Radiator

form heat flux over the inner surface of the hollow cylinder, the fuel capsule surface temperature was computed from a modified Stefan-Boltzmann equation.

The heat flow was selected as 140 watts, based on a total thermal power of 271 watts (overall generator efficiency = 4.8%). The remaining 131 watts are radiated to the collector by the tungsten shield. These proportions seem reasonable, since the temperature gradient along the heat source is found to be less than 50° F by a method described in Appendix A.

Because of the laminar nature of the fuel capsule, two composite thermal conductivities were derived--one in the axial direction, one in the radial direction--to overcome the difficulty of calculating heat flow paths for three different metals. From these values, the temperature of the inner surface of the tantalum container was calculated. Derivation of the equations employed is given in Appendix A, and the various temperatures obtained are shown in Fig. 5.

The inner temperature of the hollow curium cylinder was also computed, with the assumption that Curium-242 comprised 45% of an americium-curium mixture. A figure of 1620° F was obtained. Appendix A shows the equation employed.

2. Thermal Bypass Mechanism (heat dump)

In the earlier conceptual study leading to the final design, a novel means of rejecting surplus heat to the environment at the beginning of life of the generator was considered. Such a device was required to ensure constant electrical output during the operational life of six months by maintaining the thermoelectric hot-junction temperature at a constant value. It would also prevent high temperature damage to the heat source, thermoelectric elements and other structural components of the generator.

Figure 3 shows in outline form the first method conceived for heat rejection. Curved bimetallic springs were to sense the interior temperature of the generator by expansion or contraction with changes in temperature. They were to move a sliding thermal conductor and attached radiator, so that the surface available for conduction of heat would change and the interior temperature remain essentially constant. The design was rejected upon further analysis because it appeared to be too heavy; also, losses of heat were excessive at the end of life, causing a significant increase in the amount of curium fuel required. Further, the transfer of heat through a conductor and film gap of varying area could not be predicted with sufficient accuracy without experimental verification.

The design selected has been considered for other SNAP generators, in particular the 100-watt curium fueled generator designed previously by The Martin Company (Ref. 7). It employs a fixed radiator attached to the heat source at the end containing the fuel capsule. Covering the radiator is an insulated shutter which is actuated by the thermal expansion of molten metal contained in a loop situated in the hot junction region. Because of the size of the loop and the type of linkage connecting it to the shutter, the shutter is fully open (exposing maximum radiator surface to space) at the beginning of life, and is fully closed at the end of life (reducing parasitic heat losses to a minimum). The molten metal is a sodium-potassium mixture (NaK).

An analysis of the actuator and its operation may be found in Appendix B. Figure 6 shows the actuator design in detail.

The factor of interest in this mechanism is whether the excess heat available at the beginning of life can be dissipated at a rate which will stabilize the thermoelement hot-junction temperature at the correct value. If the temperature of the radiator surface can be established, and if the heat sink temperature is known, the thermal power which can be dissipated may be determined.

The thermal power of the generator at the beginning of its operational life is 583 watts, based on an overall efficiency of 4.8%. Of this amount, 312 watts--more than 50% of the total power--must be bypassed through the radiator. A conservative approach to determining dissipable thermal power began with the calculation of the difference between the average temperature of the fuel capsule surface and the temperature of the center of the radiator, with the assumption that all of the heat must be conducted from the midpoint of the fuel capsule through copper at the bottom of the capsule. Substitution in the familiar conduction equation gave a temperature-drop value of 52° F. The efficiency of the radiator was determined to be 92%, and its heat-radiating capacity was found to be 10% higher than the required 312 watts.

3. Thermoelectric Converter

Figure 6 shows the details of the thermoelectric converter.

The lead telluride thermoelectric elements are designed to be inserted into the generator from the outside, so that individual elements may be replaced if necessary. Silicone rubber O-rings on the element-follower caps permit charging the converter region with an inert gas to retard sublimation of the lead telluride. The method of bonding the hot ends of the elements to the electrical connections is not stipulated in this conceptual design.

Fig. 6. 13-Watt Generator for 6-Month Space Mission

SECT. E-E

Fig. 6. (continued)

All electrical connections at the colder ends of the elements are made externally, for ease of assembly and repair.

C. ASSEMBLY PROCEDURES

The discussion which follows is made with reference to Fig. 4. The fuel form is placed in the tantalum capsule and followed by the tantalum snap ring or cylinder. It may be helpful to include tabs on the ends of the snap ring, turned toward the center, to make insertion easier. The cap is welded in by fusion, so that its insertion into the Hastelloy capsule will not be impeded. After the tantalum capsule is inserted into the Hastelloy C capsule, the cap is screwed in and seal-welded; then, the capsule is welded (or screwed, as an alternate design) to the tungsten shield block. This subassembly is now ready to be screwed into the previously calibrated radiator-actuator-door subassembly.

The radiator-actuator-door subassembly is designed to be vacuum-filled at high temperature with NaK, calibrated in a test jig and stored. Allowing it to cool to room temperature while filled will not damage the mechanism, even if the NaK solidifies completely.

The copper radiator collar must be heated to allow the fuel-shield subassembly to be screwed into it. It may be necessary to heat the copper to a temperature slightly higher than the normal operating temperature to allow shrinkage to a low resistance fit.

The whole fuel and heat-dump assembly (consisting of radiator-actuator-door and fuel capsule-shielding subassemblies) is then slipped into the converter section and fastened with the three retaining screws. The door should be very nearly wide open at this time, but if it is not, it can be pulled open against the spring to allow clearance for screw fastening.

D. GENERATOR CHARACTERISTICS

The foregoing discussion leads to the summary of generator characteristics listed in Table II.

TABLE II

Generator Characteristics
Final Conceptual Design (Fig. 6)

Operational life (months)	6
Size	
Diameter (inches)	7.5
• Height (inches)	8.38 (shutter closed)
Efficiency (%)	
Thermoelectric	6.51
Thermal	73.5
Overall	4.8
Temperatures (°F)	
Hot junction	1000
Cold junction	370 (in space)
Heat loss (watts)	36
Insulation thickness (inches)	1.25
Thermoelectric elements	
Output voltage (volts)	3
Number of pairs	30
Cross-sectional area (square inches)	
P-type	0.1300
N-type	0.1089
Doping	
P-type	1.0% Na
N-type	0.03% PbI
Length (inches)	0.75
Fuel	
Isotope	Curium-242
Purity (%)	45
Dilution (Au to Cm by weight)	5 to 1
Weight of isotope (grams)	6.3
Thermal power (watts)	
At encapsulation	752
At launch	582
At end of life	270
Estimated weight (pounds)	
Shell insulation	1.178
Collector	1.286
Thermoelectric elements and bosses	4.80
Radiator	1.766

TABLE II (continued)

Internal gamma shield	3.88
Actuator reservoir	0.491
NaK	0.077
Actuator and hinge	0.033
Shutter cover and liner	0.500
Shutter insulation	0.170
Fuel capsule	0.529
Radiator	1.857
Reservoir supports	0.043
Fuel	<u>0.031</u>
TOTAL	16.641

IV. PHOTON AND NEUTRON SHIELDING

In the early stages of this study, it was believed that the JPL shielding specifications for external photon fluxes could be met completely by use of an internal shield. The internal shield might have served for primary gamma rays produced during the decay of curium and its daughters, and during the spontaneous fission of curium. However, more comprehensive calculations indicated that significant photon dosages resulted from inelastic scattering of the neutrons produced by curium fission, and from (n, γ) reactions. This gamma radiation was produced both within the internal shield and in the surrounding generator structure; it was therefore impractical to provide enough shielding over an entire end of the generator to attenuate the radiation without adding excessive weight. To avoid compromising generator performance or design, shielding of only about 60% of the required thickness was placed with the generator. In an actual mission, the equivalent of about 1.5 inches of tungsten must be attached externally to reduce the photon dose to the prescribed 7 photons/cm²/sec at a distance of 10 cm from the surface. The weight of the latter increment is not computed, since it will be a function of the radiation-sensitive area to be protected. The tungsten shielding provided within the generator is 2.5 inches thick.

Based on the amount of fuel remaining after the 2-month period between encapsulation and launch, the shielding required for purposes of safe ground handling of the generator is 1.6 inches of water. This shielding reduces the radiation level to the prescribed 60 mrem/hr at 1 meter. The relative amounts of various types of radiation and the effect of the water shield upon them is given in Table III.

TABLE III

Ground Handling Radiation Dose Rates

Type of Radiation	Dose Rate, (mrem/hr at 1 meter)	
	<u>No shield</u>	<u>1.6-in. water shield</u>
Neutron	110	48.6
Decay gamma	12.6	8.8
Fission gamma	2.7	2.2
Capture gamma	0.03	0.026
Inelastic gamma	<u>0.015</u>	<u>0.013</u>
TOTAL	125.3	59.64

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V. GROUND HANDLING AND SHIPPING CASK

Although only a nominal thickness of water is required for radiation protection in handling or shipping the generator, a shipping cask which will dissipate the thermal energy liberated by the generator at a rate fast enough to avoid excessive temperature rises is more cumbersome. The design which has evolved is based on the considerations that the water shield-coolant contained in the cask will not boil under all anticipated external ambient conditions and that no forced circulation of water by mechanical means will be required. Figure 7 shows the cask design.

The weight of the empty cask is estimated to be 128 pounds, and it will contain 238 pounds of water. Water is permitted to enter the interior of the generator, to cool the fuel-radiator assembly directly. Heat is transferred by convection and conduction to the outer wall of the cask. Enough room is provided for the generator, to permit the radiator shutter to open and close as temperatures dictate. Approximately 2000 Btu/hr must be dissipated from the exterior of the cask.

Fig. 7. Ground Cask for 13-Watt Thermoelectric Generator

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VI. PARALLEL STUDIES

Concurrent with the study discussed in this report, several other projects in the same area of interest have been carried out under this subtask. The design effort on a generator for a hard lunar impact mission (Ref. 1) and the preliminary safety analysis of the impact generator and the unit described herein (Ref. 6) have already been mentioned. Considerable effort has been expended also in developing an IBM computer code for general use in thermoelectric generator design. This code optimizes the size and number of thermoelectric elements on the basis of their point-variable thermal, thermoelectric and electrical properties and the desired electrical output of the generator under consideration (Ref. 8).

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APPENDIX A

DERIVATION OF ANALYTICAL EXPRESSIONS

A. TEMPERATURE OF A BODY RADIATING TO SPACE

Singer's equation is:

$$T_r = \left[\frac{(1 - \alpha_v) (1400) \left(\mu + \frac{\alpha_E}{\pi} \cos \rho \right) + \epsilon_{ir} (188.5) + X}{4 \epsilon_{ir} \sigma} \right]^{1/4}$$

where

X = heat generation rate per unit cross-sectional area

α_v = reflectivity

μ = eclipse factor

α_E = earth's albedo

ρ = angle between orbit and sun

ϵ_{ir} = infrared emissivity

σ = Stefan-Boltzmann constant.

The values assumed in this equation were:

$$1 - \alpha_v = 0.1$$

$$\mu + \frac{\alpha_E}{\pi} \cos \rho = 0.7$$

$$\epsilon_{ir} = 0.94.$$

B. RADIOISOTOPE DECAY

When the design power has been determined, the initial power necessary to allow for isotope decay can be calculated from:

$$N_0 = N_d e^{\lambda t}$$

where

$$N_0 = \text{initial power}$$

$$N_d = \text{design power}$$

$$\lambda = 0.693/\text{half life}$$

$$t = \text{time.}$$

The initial isotope loading can then be determined from the known power density.

C. HEAT LOSS THROUGH GENERATOR

The heat loss for the space generator is calculated from Fourier's third law:

$$q = -k A(r) \frac{dT(r)}{dr}$$

where

$$q = \text{heat flow}$$

$$k = \text{thermal conductivity}$$

$$r = \text{radius or thickness}$$

$$A = \text{area as a function of position}$$

T = temperature as a function of position.

Integration between the inner and outer surfaces gives:

$$\int_{r_7}^{r_8} \frac{dr}{A(r)} = \frac{k}{q} [T_1 - T_2]$$

where

r_7 = distance from center to inner shell

r_8 = distance from center to outer shell

T_1 = inner-shell temperature

T_2 = outer-shell temperature.

This equation can be used to compute the heat loss when the area term has been expressed so as to include heat flow through the sides and ends.

D. TEMPERATURE DISTRIBUTION OF HEAT SOURCE

The temperature gradient in the fuel capsule can be determined approximately by considering the sketch below and assuming the temperature varies only in the axial direction. Then the differential equation describing the cylindrical fuel capsule can be shown to be:

$$\frac{d^2 T(x)}{dx^2} = \frac{k_1 P}{k A} [T(x) - T_\infty]$$

The solution to this equation is:

$$T_{(x)} = T_{\infty} + C_1 e^{mx} + C_2 e^{-mx}$$

where

T_{∞} = ambient temperature

$T_{(x)}$ = temperature

C = constant of integration

$$m = \sqrt{\frac{h P}{k A_{cs}}}$$

h_1 = combined coefficient of radiation and convection

P = perimeter

k = thermal conductivity

A_{cs} = cross-sectional area.

The constants of integration can be evaluated by choosing the temperature at $x = 0$ to be T_0 and by assuming the heat removed at $x = L$ to be removed to the film-coefficient concept. Thus:

$$C_1 = \frac{T_0 - T_{\infty}}{1 - ae^{-2mL}}$$

$$C_2 = \frac{T_0 - T_{\infty}}{1 + ae^{-2mL}}$$

$$a = \frac{mk - h}{mk + h}.$$

E. CAPSULE SURFACE TEMPERATURE

The surface temperature of the fuel capsule was determined using the following equation:

$$q = \sigma F_{\epsilon} A_c (T_3^4 - T_4^4)$$

where

σ = Stefan-Boltzmann constant

F_{ϵ} = emissivity (including view factor) of the capsule surface

A_c = area of capsule surface

T_3 = capsule surface temperature

T_4 = hot junction temperature.

F. COMPOSITE THERMAL CONDUCTIVITY OF LAMINAR FUEL CAPSULE

The composite thermal conductivities for the fuel capsule were obtained from the following equations.

For the axial direction

$$q = \frac{k A \Delta T}{L} = (k_1 A_1 + k_2 A_2 + k_3 A_3) \frac{\Delta T}{L}$$

where

subscript 1 refers to the tantalum liner

subscript 2 refers to the Hastelloy C container

subscript 3 refers to the copper jacket

k = thermal conductivity

A = the heat flow area

L = the length of the flow path

$$k_{\text{composite}} = \frac{(k_1 A_1 + k_2 A_2 + k_3 A_3)}{A_1 + A_2 + A_3}.$$

For the radial direction

$$\frac{2\pi L \Delta T}{q} = \frac{\ln \frac{r_2}{r_1}}{k_1} + \frac{\ln \frac{r_3}{r_2}}{k_2} + \frac{\ln \frac{r_4}{r_3}}{k_3}$$

where the subscripts refer to the conductivity and inner status of the same materials as above.

$$\frac{\ln \frac{r_2}{r_1}}{k_1} + \frac{\ln \frac{r_3}{r_2}}{k_2} + \frac{\ln \frac{r_4}{r_3}}{k_3} = \frac{\ln \frac{r_4}{r_1}}{k_{\text{composite}}}$$

$$k_{\text{composite}} = \frac{\ln \frac{r_4}{r_1}}{\frac{\ln \frac{r_2}{r_1}}{k_1} + \frac{\ln \frac{r_3}{r_2}}{k_2} + \frac{\ln \frac{r_4}{r_3}}{k_3}}$$

In the following, $k_{\text{composite}}$ in the axial direction will be designated k_z , while the radial conductivity will be k_r . The differential equation

$$k \nabla^2 T(r, z) = 0$$

or

$$k_r \frac{\partial^2 T(r, z)}{\partial r^2} + \frac{k_r}{r} \frac{\partial T(r, z)}{\partial r} + k_z \frac{\partial^2 T(r, z)}{\partial z^2} = 0$$

gives the solution

$$T(r, z) = \left[D I_0(\beta r) + E K_0(\beta r) \right] \sin \sqrt{\frac{k_r}{k_z}} \beta z$$

where

D, E and β are constants

I and K are Bessel functions

The boundary conditions are:

$$(1) \quad T(r, 0) = 0$$

$$(2) \quad T(r, L) = 0$$

$$(3) \quad T(r_4, z) = 0$$

$$(4) \quad \left(\frac{\partial T}{\partial r} \right) \bigg|_{r=r_1} = - \frac{q}{k_r}.$$

With these boundary conditions, the solution becomes

$$T(r, z) = \frac{4q' L}{\sqrt{k_z k_r} \pi^2} \sum_{m=0}^{\infty} \frac{\left\{ -K_0 \left[\sqrt{\frac{k_z}{k_r}} (2m+1) \frac{\pi r_4}{L} \right] I_0 \left[\sqrt{\frac{k_z}{k_r}} (2m+1) \frac{\pi r}{L} \right] \right.}{(2m+1)^2 \left\{ K_0 \left[\sqrt{\frac{k_z}{k_r}} (2m+1) \frac{\pi r_4}{L} \right] I_1 \left[\sqrt{\frac{k_z}{k_r}} (2m+1) \frac{\pi r_1}{L} \right] \right.}$$

$$+ \left. I_0 \left[\sqrt{\frac{k_z}{k_r}} (2m+1) \frac{\pi r_4}{L} \right] K_0 \left[\sqrt{\frac{k_z}{k_r}} (2m+1) \frac{\pi r}{L} \right] \right\} \sin (2m+1) \frac{\pi z}{L}$$

$$+ \left. K_1 \left[\sqrt{\frac{k_z}{k_r}} (2m+1) \frac{\pi r_1}{L} \right] I_0 \left[\sqrt{\frac{k_z}{k_r}} (2m+1) \frac{\pi r_4}{L} \right] \right\}$$

G. CONDUCTION OF HEAT IN A CYLINDRICAL SOURCE

The equation for the presence of a heat source in a cylinder conducting in the radial direction only is

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{s}{k} = 0$$

where

- s = volumetric heat production rate (Btu/hr-ft³)
 k = thermal conductivity (Btu/hr-ft-°F)
 T = temperature (°F)
 r = radius (ft)

H. RADIATION OF HEAT TO SPACE FROM HEAT-DUMP RADIATOR

Radiation from the excess heat-dump radiator is given by the following calculation:

$$e_f = \frac{\sqrt{2/\zeta}}{\left(1 + \frac{r_6}{r_5}\right)} \left[\frac{I_1 R_a \zeta K_1 R_b \zeta - I_1 R_b \zeta K_1 R_a \zeta}{I_1 R_a \zeta K_0 R_b \zeta + I_0 R_b \zeta K_1 R_a \zeta} \right] *$$

where

- e_f = efficiency of fin
 ζ = $(r_6 - r_5)^{3/2} (h_2/k_f A_4)^{1/2}$
 R_a = $\sqrt{2} / \left(1 - \frac{r_5}{r_6}\right)$
 R_b = $\frac{r_5}{r_6} R_a$
 r_5 = inner radius
 r_6 = outer radius
 h_2 = convective heat transfer coefficient computed as $\frac{4\sigma F_\epsilon T_5^3}{\epsilon}$

where T_5 is the temperature at the inner radius of the copper radiator and the other terms are as previously defined

- k_f = the thermal conductivity of the fin

*See page 84, Heat Conduction Transfer, by Schneider.

$$A_4 = 2 (r_6 - r_5) \times \text{fin thickness.}$$

Computation gives a fin efficiency of about 92%. For calculating the heat dump

$$q = \sigma F_e A_r e_f T_5^4$$

where the effective space temperature is negligible and where

$$A_r = \text{the radiating area of the dump.}$$

Thus

$$q = 980 \text{ Btu/hr for the circular fin portion.}$$

For the center part of the fin of 1.75-inch diameter at uniform temperature of 1752° R

$$q = 190 \text{ Btu/hr.}$$

Total $q = 1170 \text{ Btu/hr}$, which exceeds the required heat rejection capacity of 312 watts, or 1065 Btu/hr. The equation for two-dimensional steady-state heat flow in a finite hollow cylinder is

$$\frac{\partial^2 T(r, z)}{\partial r^2} + \frac{1}{r} \frac{\partial T(r, z)}{\partial r} + \frac{\partial^2 T(r, z)}{\partial z^2} = 0$$

The boundary conditions are:

$$T(r_2, z) = T_c$$

$$T(r, \frac{L}{2}) = 0$$

$$T(r, 0) = T_c$$

$$T(r_1, z) = [T_h - T_c] \frac{2z}{L} + T_c$$

The solution to the above equation with the specified boundary conditions is

$$q = -kR_1 \cdot 16 [T_h - T_c] \sum_{n=1}^{\infty} \frac{1}{(2n-1)} \frac{K_0 \left[\frac{(2n-1) \pi r_2}{L} \right] I_1 \left[\frac{(2n-1) \pi r_1}{L} \right]}{I_0 \left[\frac{(2n-1) \pi r_1}{L} \right] K_0 \left[\frac{(2n-1) \pi r_2}{L} \right]} \dots$$

$$--- \frac{+I_0 \left[\frac{(2n-1) \pi r_2}{L} \right] K_1 \left[\frac{(2n-1) \pi r_1}{L} \right]}{-I_0 \left[\frac{(2n-1) \pi r_2}{L} \right] K_0 \left[\frac{(2n-1) \pi r_1}{L} \right]}$$

where

- k = thermal conductivity
 r_1 = internal radius
 r_2 = external radius
 T_h = hot junction temperature
 T_c = cold junction temperature
 q = heat flow

I. THERMOELECTRIC EFFICIENCY

Thermoelectric efficiency, including contact resistance, is computed from Eqs (A) and (B), which were derived in report MNEA-AJS-011.

$$(A) \quad \eta_{\max} = \frac{\Delta T}{T_h} \frac{\frac{x_m}{x_m + 1}}{(x_m + 1) + \frac{K (x_m + 1)^2}{\alpha^2 T_h} (R_c + R_i) - \frac{\Delta T}{T_h (R_i + R_c)} \left(\frac{R_i}{2} + R_c \right)}$$

$$(B) \quad x_m = \frac{R_e}{R_i + R_e} = \sqrt{1 + \frac{\alpha^2 T_h}{K (R_c + R_i)} - \frac{2 \Delta T \left(\frac{R_i}{2} + R_c \right)}{K (R_c + R_i)^2}}$$

where

- R_e = load resistance
 R_i = element internal resistance
 R_c = electrical contact resistance

$$K = (k_p + k_n B) S_p, \text{ thermal conductance}$$

$$B = S_n / S_p$$

$$S_n = A_n / L_n$$

$$S_p = A_p / L_p$$

$$k_p = \text{thermal conductivity, P elements}$$

$$k_n = \text{thermal conductivity, N elements}$$

$$\Delta T = T_h - T_c$$

$$T_h = \text{hot junction temperature}$$

$$T_c = \text{cold junction temperature}$$

$$\alpha = \text{thermoelectric power}$$

NOTE:

All temperatures absolute

APPENDIX B

ACTUATOR FOR RADIATOR SHUTTER

A spring-opposed liquid-metal actuator design was attempted because of the possibility of making a sealed, precalibrated system. Also, other vapor and organic-liquid schemes have been designed. A study of about nine liquid metals from the Liquid Metal Handbook resulted in the choice of NaK, 22:78% alloy, primarily because of its volume coefficient of expansion. However, the melting point, scattering cross section, thermal conductivity and vapor pressure were considered also.

The volume coefficient of expansion had to be derived from temperature-density data as follows:

$$D = \frac{1}{v}.$$

By differentiation,

$$\frac{dD}{D} = - \frac{dv}{v}.$$

Dividing by temperature, assuming constant unit mass, gives

$$\frac{dv}{V^{\circ}F} = - \frac{dD}{D^{\circ}F} = \beta$$

where

β = volume coefficient of expansion

D = density

v = specific volume

V = volume.

The chosen NaK composition has a volume coefficient of expansion of $1.81 \times 10^{-4}/^{\circ}F$ between data points 550° and 700° C (1022° and 1292° F). The melting point of this alloy (probably the eutectic) is 12° F. Corrosive properties tabulated in the Liquid Metal Handbook indicate that only Al or Zn and Sn alloys of Cu show poor corrosion resistance to Na, K or NaK up to 800° F. There is therefore a wide choice of construction materials.

The required system volume may be calculated with the assumptions that a 1/2-inch stroke is needed for good control of the door through a 90° angle over a temperature range of 1000° to 1050° F, and that a bellows of 1/4-inch nominal diameter is used.

$$\begin{aligned}\Delta V &= \beta V_{1000} \Delta T = 1/2 (\pi/4) (1/4)^2 \\ &= 1.7 \times 10^{-4} / ^\circ\text{F} (V_{1000}) 50^\circ \text{ F}\end{aligned}$$

$$V_{1000} = 2.89 \text{ in.}^3$$

Bellows compressed volume at 1000 ° F is

$$V_B = 7/4 \frac{\pi}{4} (1/4)^2 = 0.0855 \text{ in.}^3$$

Therefore, the reservoir volume including tubing is

$$V_R = V_{1000} - V_B = 2.89 - 0.0855 = 2.8 \text{ in.}^3$$

Fig. 1.

This reservoir may be formed from a length of standard 14 to 22 ga rectangular 1/2 x 1 inch stainless steel tubing filled with Woods Metal and rolled or bent to the approximate inside shape of the generator. The ends of the tube are capped. A schematic of the system is shown in Fig. 1. The required length of tubing will be approximately

$$V = (1/2 \times 1 \times 0.9) L = 2.8 \text{ in.}^3 ; L = 6.22 \text{ inch. ,}$$

where 0.9 is an arbitrary void fraction used because 1/2 inch and 1 inch are outside dimensions, and because corners are rounded.

The vapor pressure of 22:78% NaK at 1022° F is 100 mm Hg (1.94 psi) or the minimum pressure to prevent formation of a vapor pocket in the system. This pressure is to be supplied by a spring. The necessary spring force will be

$$F = 1.94 A = 1.94 (\pi/64) = 0.0951 \text{ lb.}$$

A heavier spring should be used, since the modulus of elasticity may drop to about 1/3 of its room temperature value at 1000° F.

The expansion of the containing system was not taken into account. However, the volume coefficient of expansion for stainless steels is about 10% of that for NaK. The small error can be compensated for in a more detailed design.

If the unit were filled at 1000° F and allowed to cool to room temperature, the vapor pressure of NaK would drop below 1 mm Hg; there would therefore be a crushing force due to full atmospheric pressure on nearly all of the actuator.

Fig. 2.

If there is a 15 psi crushing force on a 1 inch length of 1/2 x 1-inch rectangular tube (Fig. 2a.), the free-body diagram for treatment as a restrained beam is as shown in Fig. 2b. It is assumed that forces P and M can be neglected since they have nearly opposite effects, and that the modulus of elasticity is $E = 20 \times 10^6$ psi at 1000° F.

$$M_{\max} = \frac{WL^2}{8} = \frac{15(1)}{8} = 1.88 \text{ in.-lb}$$

$$I_0 = \frac{bh^3}{12} = \frac{1(0.03)^3}{12} = 2.25 \times 10^{-6} \text{ in.}^4$$

$$c = 0.015 \text{ in.}$$

$$S = \frac{Mc}{I_0} = \frac{1.88(0.015)}{2.25 \times 10^{-6}} = 12.6 \times 10^3 \text{ psi}$$

where

S = stress

M = bending moment

I_0 = moment of inertia around neutral axis

c = distance of stress from neutral axis.

The yield stress for many stainless steels is 2 to 6 times the calculated value at operating temperature. The radius of curvature will be

$$\rho = \frac{EI}{M} = 23.9 \text{ inches.}$$

It appears that the deflection of the system will not be very large. The bellows will stand the pressure well, since it is designed to handle over 400 psi internal pressure, and its convoluted construction indicates that it will resist almost as large an external pressure.

If the holes and pins are ± 0.002 inch, the maximum slack for 6 pivots will be

$$6 \times 0.004 = 0.024, \tan \theta = \frac{0.024}{1/4} = 0.096$$

$$\theta \approx 6 \text{ degrees.}$$

Since at least four of the pivots will be under spring tension, the slack in the door system should be 2 degrees or less.

Fig. 3.

Spring tension to hold the door (Fig. 3) should be at least

$$(5/2 \text{ inch}) = T (1/4 \text{ inch}); \quad T = 5 \text{ lb.}$$

This is much greater than the previous requirements for vapor pressure; therefore, the spring should be designed for door tension.

Under acceleration on launch the door will normally be fully open, but it should be analyzed for vibrations in any subsequent hardware design.

The push-rod loading is small if checked by intermediate-length column theory. The radius of gyration for a rod 1/8 inch in diameter and 1-3/8 inches long will be

$$r = \sqrt{I/A} = \frac{D}{4}$$

where

r = radius of gyration

I = moment of inertia

A = cross-sectional area

D = diameter of cross section.

The ratio of length to radius of gyration is

$$\frac{L}{r} = \frac{1 1/8}{1/4 \times 1/8} = 44$$

For round-ended intermediate-length steel columns of $\frac{L}{r} = 44$,

$P/A \approx 33,000$ psi.

For the push rod,

$$\frac{P}{A} = \frac{5}{\pi/4 (1/8)^2} = 408 \text{ psi, a very safe stress level.}$$