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DESIGN OPTIMIZATION OF RTG FOR SOLAR-POLAR MISSION

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ABSTRACT

The paper presents the conceptual design of an RTG, employing stacked radioisotope heat source modules and silicon-germanium thermoelectric couples, for use on the joint NASA/ESA International Solar-Polar Mission. It describes a computer program which, for a given set of assumptions and ground rules, can rapidly examine thousands of design variations and determine the combination of design parameters which will minimize the RTG weight for a given power output. Graphical results are presented to illustrate design trends and clarify the optimization procedure. The generated results lead to interesting and somewhat unexpected conclusions about optimum design parameters, the relative constancy of RTG specific power over a wide power range, comparisons between single and split RTGs, sensitivity to variations in heat source module design, weight comparisons with the MMW RTG, and comparative RTG weights for different housing and radiator fin materials.

AT THE REQUEST OF THE Department of Energy's Advanced Nuclear Systems and Products Division, Fairchild Space and Electronics Company* has been conducting conceptual design studies for a light-weight Radioisotope Thermoelectric Generator (RTG), to be used on the joint NASA/ESA International Solar-Polar Mission (ISPM). That mission calls for two spacecraft, developed under the respective direction of NASA/JPL and ESA, for scientific explorations outside the earth's ecliptic plane. The two spacecraft are to be launched simultaneously aboard the shuttle, and boosted by the IUS upper stage into long elliptical orbits towards Jupiter, whose gravitational pull helps to lift them out of the ecliptic plane. As illustrated in Figure 1, the two spacecraft will arrive above the sun's North and South poles about four years after launch.

Fig. 1 - SOLAR POLAR MISSION PLAN

*under Contract EN 77-C-02-4281

Each spacecraft needs a sun-independent nuclear power system, capable of delivering about 250 watts at the end of mission. Since both spacecraft appear to be severely weight-limited, minimization of the RTG's weight (without sacrifice of reliability) is of crucial importance.

The Solar-Polar RTG design is to be based on the use of discrete heat source modules which are currently under development at Los Alamos (LASL). The design concept under development is called the General Purpose Heat Source (GPHS), since the modules can satisfy a wide range of power conversion systems, power levels, and space missions. Each module contains within it the necessary safety provisions to ensure fuel retention under all credible abort modes.

Since some of the required safety verification tests are still incomplete, the exact heat source module weight is not yet defined at the time this is written. However, preliminary results indicate that the use of a modular heat source will permit a significantly lower RTG weight than the monolithic type of heat source used in previous missions. For example, with the heat source of the Multi-Hundred-Watt (MMW) RTG used previously on the Voyager mission, it would not be possible to meet JPL's present weight goal for the ISPM power system. By contrast, current designs of the modular heat source make it possible to meet that goal, with some margin left over to cope with anticipated contingencies. This explains the motivation for using a new heat source, still under development.

In addition to the new heat source, DOE had also planned to use a new thermoelectric conversion material in the ISPM flight generators. The intended material consisted of copper selenide for the P-legs and gadolinium selenide for the N-legs. Based on single-element tests under idealized laboratory conditions, these selenide materials offered the possibility of significantly higher conversion efficiencies than could be obtained with the silicon-germanium thermoelements that were used on previous missions.

By the end of 1978, the promised performance level of the selenide elements had not yet fully materialized. Nor had the developer demonstrated the levels of reproducibility and performance stability, in a practical engineering configuration, which would be required for a reliable flight system. In view of the many unresolved problems and the resultant risk to NASA's ISPM program schedule, DOE decided, at the beginning of 1979, not to use selenide converters in the ISPM generators. Instead, these generators will employ Si-Ge converters, which were successfully used in earlier flight systems (e.g., on NASA's Voyager mission to Jupiter and Saturn).

Silicon-germanium thermoelements are well characterized, and should therefore present much less risk to the program. On the other hand, since the thermoelectric efficiency of Si-Ge elements (.7.5%) is considerably lower than the promised efficiency of Se elements (.11.3%), they require approximately 50% more heat source modules than originally anticipated. The resultant weight increase is difficult to accommodate on the ISPM spacecraft, which is severely weight limited. This makes it imperative to minimize the weight of all other RTG components; without, of course, impairing the generator's reliability and long-term performance stability required for the mission.

In view of the problems and delays encountered on the selenide RTG program, DOE was most anxious to avoid any programmatic and schedule risks on the Si-Ge RTG program for ISPM. Fairchild was therefore instructed to base its conceptual design as far as possible on proven converter technology. Specifically, this meant the same hot-junction temperature (1000°C), same thermoelectric element design (except for minor dimensional changes, where necessary), and the same thermal insulation system as in the MHW RTGs. It also meant conductive radiator fins, without heat pipes. (Considerable weight reductions are possible if these constraints are removed for future systems).

With the above ground rules, the design of the major system components, including the heat source modules, the insulation system, and the thermoelements and their associated hardware, are essentially outside the control of the RTG designer. Thus, the only decisions left to the designer are how to put those components together to minimize the RTG weight. Principally, these decisions include the arrangement for supporting the modular heat source stack and applying the required preloads, the mechanical layout of the thermoelectric converter and the electrical layout of its series-parallel network, the design of the RTG housing and end caps, and the choice of the cold-junction temperature and the design of the radiator fins. These items form the subject of this paper. Section II describes the major components and the topology of the RTG design evolved, and Section III describes the parametric optimization studies performed and their results.

DESIGN DESCRIPTION

The heat source modules under development at LASL have a thermal power level of 250 watts each. As illustrated in Figure 2, each module contains four cylindrical 62.5-watt $^{238}\text{PuO}_2$ fuel pellets clad in iridium shells. Each capsule is surrounded by a double shell of pyrolytic graphite. The four assemblies are contained in thick-walled impact shells and aeroshells made of a graphitic material consisting of a carbon-bonded, three-dimensional, carbon-fiber structure. The purpose of the above components is to prevent fuel release under all credible abort conditions, before, during, or after launch.

Fig. 2 -
TYPICAL HEAT SOURCE MODULE

The rationale of the heat source module design and the extensive test program to demonstrate its safety will not be discussed here. The present paper deals not with the heat source module itself, but with its use in a practical RTG design.

The heat source module depicted in Figure 2 has typical dimensions of 3.6x3.6x2 inches and weighs a little over 3 lbs. The exact values are still uncertain, since the design dimensions have not yet been finalized at the time this is written.

The basic building block of the thermoelectric converter is the Si-Ge unicouple depicted in Figure 3. This is essentially identical to the units built by RCA for GE's MHW RTGs, which were successfully used on NASA's Voyager mission. It contains two rectangular Si-Ge legs, bonded to a Si-Mo hot shoe, which is about 0.9-inch square. The hot shoe serves as the couple's hot junction and heat receiver. The thermoelectric legs are wrapped with tightly-wound quartz yarn, to provide electrical isolation between the couples and the metallic radiation shields described below.

Fig. 3 - THERMOELECTRIC ELEMENT (UNICOUPLER)

The cold ends of the Si-Ge legs are bonded successively to flexible copper straps (for electrical connections to other couples), a ceramic wafer (for thermal conduction and electrical isolation between the couple and the generator housing), and a flexible heat shunt (to provide a compliant heat path to the housing). The couples and their fabrication have been described in detail in earlier publications [1], and their performance and performance stability are well characterized.

The thermal insulation between the thermoelectric couples is also identical to that used in the MHW RTGs [2]. It consists of about 60 layers of 0.0003"-thick molybdenum foil, alternating with a similar number of open-weave quartz-cloth spacers. The insulation package is about 0.70 inch thick. An appreciably lighter RTG could have been designed by using a thinner insulation package and shorter thermoelements, but this would have involved too great a departure from proven MHW technology, and was, therefore, rejected for the ISPM system.

To provide the necessary unicouple penetrations through the thermal insulation package, square holes are punched through each layer at appropriate locations. In the Fairchild design, the insulation layers are then wound into a cylindrical configuration, with the square holes carefully registered by means of an assembly fixture, to permit insertion of the quartz-wrapped unicouples. After insertion, the flexible copper straps of neighboring couples are connected to form the desired series-parallel network. As in the MHW RTG, a mounting screw with a C-ring seal is used to mount each unicouple to the inside of the generator's beryllium housing.

As shown in Figure 4, there is a ring of sixteen couples around the circumference of the RTG, arranged in four groups of four parallel couples each. Series connections between groups are made between rings, in the generator's axial direction. The hot shoes of the couple rings form an essentially cylindrical cavity in which the stack of square heat source modules is located. The RTG's housing, like the insulation package, is also cylindrical. A square or octagonal housing circumference might permit a lower RTG weight, but would be harder to fabricate, particularly in beryllium.

Fig. 4 - RTG CROSS-SECTION

In space, the generator is vented to vacuum, and heat transfer from the heat source stack to the hot shoes is by radiation only. As seen, there is no contact between the hot shoes and the heat source. A typical heat source stack consists of about 18 individual modules, and is over 36 inches long. With the module stack supported only at its ends, a very high axial preload--of the order of 10,000 lbs--is required to keep the stack from falling apart under the influence of lateral launch loads. On the other hand, to retain the full safety benefit of the modular design concept, the heat source support structure must allow disassembly of the module stack if the RTG housing burns off during reentry. This presents a difficult design problem, much more so than for the case of a monolithic heat source like MHW. The above problem and Fairchild's suggested solution are discussed in detail in a companion paper published in these proceedings [3].

The suggested support structure is illustrated in Figure 5. At each end of the heat source stack, the axial and lateral support forces are applied by eight cylindrical buttons, which penetrate through the multi-foil insulation system. The support buttons are made of pyrolytic graphite, to reduce parasitic heat losses. The forces are transmitted to the end caps of the generator housing through stiff Belleville springs.

The end cap shown in Figure 5 is the result of extensive structural analyses and design iterations, which succeeded in reducing its weight in beryllium to less than 1 lb. Its main structural members are the two very stiff, I-shaped cross beams, which resist the bending moments produced by the high axial loads at the four corners of the heat source's end face. The design permits final adjustment of the axial and radial spring loads (in the sealed RTG) at the launch site, as illustrated and explained in Reference [3]. The modular heat source stack and its structural support system were modeled for NASTRAN analysis. The results of that analysis confirmed the design's adequacy under prescribed static and dynamic launch loads [3].

Fig. 5 - HEAT SOURCE SUPPORT STRUCTURE

Figure 6 presents a cutaway view of one end of the assembled RTG. It shows the stacked heat source modules, their end support structure, and the end cap bolted to the housing flange, with the joint sealed by means of a C-ring. Part of the heat source stack has been cut away, to display the hot shoes at the opposite side of the converter cavity. The figure also shows the side and end insulations. Both contain 60 layers of Mo multifoil, but the end insulation blanket is much thinner because it uses ZrO_2 particles instead of quartz cloth to separate the foils.

Fig. 6 - CUT-AWAY VIEW OF ASSEMBLED RTG

As indicated, each end of the heat source stack extends about one inch beyond the last ring of hot shoes. This is beneficial, because the heat from about half a module at each end of the heat source is available to compensate for thermal losses through the structural supports and end insulation. Thus, a more uniform axial temperature profile is obtained than if the hot shoes covered the entire length of the heat source stack.

As shown in Figure 6, the unicouples are cantilevered from the housing wall, to which they are attached by external mounting screws. As in the MHW RTG, the mounting holes are sealed by individual seal rings. The couples penetrate through the cylindrical insulation package; and the electrical connections are made on the outside of that package to form the desired series-parallel network. Rivets are used to join the series and parallel copper conductors. The parallel leads normally carry no current, but they provide high reliability against open-circuit failure. The arrangement illustrated is for four couples in parallel. An alternative arrangement for three couples in parallel will be described later.

Figure 7 shows a view of the complete RTG, with typical dimensions in inches. There are eight radiating fins around the circumference of the generator housing. Thus, there is one fin for every two columns of unicouples. This reduces the heat rejection path length and the resultant temperature drop to the fin root. The fins are made of beryllium, and have a trapezoidal cross-section. They are slotted at intervals of about 4 inches, to reduce thermal stresses. The outside of the housing and fins was assumed to be coated with iron titanate, which was used on the MHW RTG and has an emissivity of 0.85.

Fig. 7 - TYPICAL RTG DIMENSIONS

As shown, the RTG has mounting pads and ball bushings at the bottom of alternate radiator fins. Thus, the generator is mounted to the spacecraft in a cantilevered fashion, with the RTG axis perpendicular to the spacecraft spin axis, as illustrated in Figure 8.

Fig. 8 -

RTG MOUNTED ON SPACECRAFT

The two spacecraft, each with its own RTG, are mounted on the IUS upper stage in a piggy-back arrangement. As depicted in Figure 9, the whole assembly fits within the shuttle bay's payload envelope, with each RTG mounted to its spacecraft in its fixed flight position. Therefore, mechanisms for RTG deployment after launch will not be needed.

Fig. 9 - RTG's, SPACECRAFT, AND IUS IN SHUTTLE BAY

DESIGN OPTIMIZATION

The preceding section described the topology and major components of the RTG in general terms. This section describes a computer program which, for a given set of constraints and fixed input parameters, finds the combination of variable parameters which minimizes the generator weight for a specified power level. The section also presents the results of the optimization for the power range of interest.

The fixed input parameters include the thermoelectric hot-junction temperature (1000°C); the length (1.23 in) and weight* (0.0207 lb) of each unicouple and of its associated mounting hardware (0.0066 lb); the hot shoe size (0.9×0.9 in), and the minimum gap between the hot shoes and the heat source corners (0.125 in); the thickness (0.70 in) and density (0.021 lb/cu in) of the multifoil insulation*; the thermal conductance of the pyrolytic graphite buttons used to support the heat source, and the physical properties of the housing and radiator materials (beryllium, unless otherwise stated); the temperature drop between the unicouple's cold junction and its mounting point on the RTG housing (30°C)*; the assumed minimum fabricable housing wall thickness (0.060 in); the number of radiator fins (8) and the minimum practical fin thickness (0.020 in); the weight (0.008 gm/cm²) and emissivity (0.85) of the RTG's surface coating; the temperature-dependent Seebeck coefficient, resistivity, and thermal conductivity of Si-Ge; the prescribed output voltage (28v) of the ISPM generator; and the weight (3.32 lbs), diagonal dimension (5.27 in), and height (2.03 in) of the heat source module. The above parameters are held fixed during the optimization process.

*based on MHW information, with suitable adjustments

The heat source module weight and size cited above, which represent the latest design option under test at LASL, were used as the basis for most of the results presented below. Since the module's size and weight may change, depending on the outcome of LASL's ongoing safety tests, the effect of different heat source options on generator weight is examined at the end of the paper.

The optimization program consists of four main parts: The first varies the variable parameters over specified ranges. The second calculates the power output and RTG weight for any specific set of variable parameters. The third part examines the results of the thousands of cases analyzed in Parts 1 and 2 and picks the lightest design for any given power level. And the fourth part of the program generates graphic and tabular output to display the results of the optimization.

The variable RTG parameters investigated include the number of 250-watt heat source modules (14 to 20), the number of 16-couple rings (22 to 40), the number of couples in parallel (3 or 4) and implicitly the number of couples in series (128 to 192), and the radiator fin length (0 to 12 inches) and fin base (0.020 to 0.180 inch).

Given a set of those parameters, the program first computes the length and thermal power of the heat source. It then calculates the heat losses through its support structure and through the multifoil insulation and along its joints with the thermocouples. The remainder is the heat flow to the couples.

Since the number of couples in series and the RTG output voltage have been specified for a specific design point, the couple voltage is implicitly fixed. Note that in most cases this will not be the voltage which maximizes the couple's efficiency. The program described here seeks to minimize the RTG's weight, not to maximize its efficiency.

From the known couple voltage, the program next uses an initial estimate of the cold-junction temperature to compute the thermoelectric efficiency of the couples. Once the heat flow to the couples and their efficiency are determined, the RTG's power output, system efficiency, and heat rejection rate are known. Also known are the RTG's internal impedance and therefore the lateral dimensions of the thermocouple legs. For the range of interest, the latter are within a few percent of the corresponding values in the MIN design.

Next, the program performs a structural analysis to determine the axial preload required to hold the heat source together during launch, the weight of the end caps needed to sustain that preload, and the housing wall thickness needed to survive the shuttle and IUS launch loads.

In computing the required housing wall, the program checks three criteria: the tensile stress due to the preload combined with the lateral launch loads on the shuttle; ditto for the IUS launch; and the stability of the cantilevered housing wall under bending moments caused by lateral launch loads. There is an optional, fourth criterion: a check for the wall thickness required to keep the cylindrical housing from collapsing when the external pressure exceeds the internal pressure by one atmosphere. But this criterion was generally not applied, because it was assumed that the generator would be stored, shipped, and launched while filled with more than one atmosphere of inert gas; and that it would only be tested under vacuum with the whole RTG contained in a vacuum chamber.

After determining the minimum wall thickness required to satisfy all of the above three (or four) structural criteria with appropriate safety margins, the program chooses that thickness or the assumed lower fabrication limit for the wall material, which-

ever is greater. It then proceeds to analyze the heat rejection system, using the previously computed heat loss rate through the insulation and heat rejection rate of the thermoelectric couples.

The problem of designing a minimum-weight radiator fin for a given heat flow rate and root temperature has been solved elsewhere [4,5] for various fin configurations and constraints. But those solutions are only valid for fins with an unobstructed view of space, and without internal radiation interchange. They do not apply to the present case, where the view to space of the housing and fins is severely blocked, and where radiative interchange between those components is an important heat transfer mechanism. With radiative interchange, the fin design optimizes at different lengths and thicknesses.

The thermal analysis in the program is carried out for a 45-degree segment of the housing enclosed by two radiator fins. The fins are of trapezoidal cross-section, tapering from the fin root thickness under consideration to a 0.020-inch fin tip thickness. The housing segment is divided into ten nodes, and each fin consists of 0 to 40 nodes (of approximately 0.3 inch), depending on the fin length under consideration. The thermoelectric heat rejection rate is applied at two housing wall nodes which represent the uncouple mounting points.

For the above model of 10 to 90 nodes, the program computes thermal conductances between adjacent nodes, and radiation view factors between all node pairs which face each other. From these view factors, it then computes radiation interchange factors, which take account of mutual reflections between nodes. The resultant matrix of interchange factors, containing as many as 90x90 terms for the maximum fin length, is then collapsed to one quarter that size, by taking advantage of the symmetry plane midway between the two fins; and the temperatures of the remaining 5 to 45 nodes are computed by conventional matrix inversion methods.

Of particular interest are the fin root temperature, the fin tip temperature, and the housing temperature at the uncouple mounting point. The latter is used to compute the cold-junction temperature, based on a 30°C cold-stack temperature drop. If the result does not agree with the previously assumed cold-junction temperature within prescribed convergence limits, the thermoelectric performance is recomputed for the new cold-junction temperature, and the entire thermal analysis is repeated. This iterative process continues until convergence is achieved.

Once a consistent solution has been found, the RTG weight for the specific set of input variables can be computed. Let us illustrate this for a typical case consisting of 18 heat source modules, 34 rings of 16 couples each, 4 couples in parallel and 136 in series. A fin length of 4.32 inches and root thickness of 0.092 inch. For this case, the heat source stack is 36.5 inches long, the converter cavity formed by the hot shoes has a diameter of 5.52 inches, the generator housing has an outer diameter of 7.98 inches and a required wall thickness of 0.060 inch, and the RTG has an overall length (including end caps) of 40.7 inches and a tip-to-tip span of 16.6 inches.

As illustrated in Figure 10, the converged solution gives a cold-junction temperature of 308°C. The housing wall temperature varies from 278°C at the uncouple mounting points to 265°C at the fin roots, and the fin tip temperature is 190°C. As will be seen, such low fin tip temperatures, with correspondingly low fin effectiveness values, are typical for optimized radiator designs. Typical temperatures for the fuel capsule and heat source surface are also shown in Figure 10.

Fig. 10 - TYPICAL RTG OPERATING TEMPERATURES

For the temperatures and dimensions given above, the heat losses consist of 168 watts through the heat source supports, 36 watts through the end foils, 94 watts through the side foils, and 132 watts through the insulation joints. Thus, of the heat source's thermal power of 4500 watts, a total of 430 watts is lost, giving a thermal efficiency of 90.4% and leaving 4070 watts as the heat input to the thermoelectric elements. The total heat rejection rate of the RTG is 4208 watts.

At the 1000°C hot-junction temperature and the 308°C cold-junction temperature, the maximum efficiency of the Si-Ge couples is 7.22% at an output voltage of 0.190 volts. But since the 136 couples in series must produce the specified output of 28 volts, the actual couple voltage for this case is 0.206 volts. At that voltage, the thermoelectric efficiency is reduced to 7.17%.

Thus, for this specific case, the RTG has an output of 292 watts (10.4 amps at 28 volts), giving a system efficiency of 6.49%. Its internal impedance is 1.80 ohms, and the external load impedance is 2.69 ohms.

The weight breakdown for the illustrated case is summarized in Table I. As expected, the heat source stack is by far the heaviest component, constituting 58% of the total weight. The basis of most other component weights has been discussed before. Whenever possible, they are scaled from weights of corresponding MHW components or from other GE information.

One rather surprising item is the relatively high weight of the thermal insulation system. At 10.9 lbs, which is 10.5% of the total system weight, it is almost as heavy as the T/E converter, and substantially heavier than the RTG housing and the radiation fins.

The other item worth noting is the insulation support frame. This has been included in all our calculations because it was part of GE's MHW design. However, it is not clear to the authors that this support frame performs any necessary function after the converter is assembled. The support frame in the MHW RTG is not connected to the housing, but floats on the inside of the multifoil package. The foil package itself, as stated by GE [2], is essentially supported by the thermoelements. The weight of the support frame is appreciable, and its elimination would reduce the RTG's weight and improve its specific power. However, before it could be eliminated, tests to confirm that it is not needed would first have to be performed.

For every combination of number of heat source modules, number of converter rings, and number of couples in parallel and series, the program repeats the above design point analysis for different fin lengths and root thicknesses, to find the variation of RTG power and weight. Typical results are illustrated

Table I: WEIGHT SUMMARY (LBS)
FOR ILLUSTRATIVE RTG DESIGN (292W BOL)

HEAT SOURCE MODULE:		59.8
Fuel	24.21	
Iridium Capsules	8.57	
Graphite Shells	26.98	
HEAT SOURCE SUPPORTS:		1.0
Moly Cross-Pieces	0.50	
Pyrographite Buttons	0.13	
Springs, etc.	0.40	
T/E CONVERTER:		13.3
Unicouples	11.60	
Electrical Straps	1.08	
Rivets & Washers	0.63	
INSULATION:		13.0
Alumina Spacers	0.61	
Side Insulation	10.33	
End Insulation	0.59	
Support Frame	1.50	
T/E MOUNTING:		3.9
Nut Plates	1.51	
Mounting Flats	0.83	
C-Seals	0.12	
Mounting Screws	1.46	
RTG ENVELOPE:		11.2
Housing	3.88	
End Caps	2.00	
Radiator Fins	4.80	
Mounting Pads & Bushings	0.50	
EXTERNALS:		1.7
Power Connectors	0.25	
Emissive Coating	0.50	
Gas Management System	0.29	
Pressure Relief Device	0.69	
RTG TOTAL		104.0

in Figure 11, for the case of 18 heat source modules, 34 rings of 16 couples, 4 couples in parallel and 136 in series, and for fin lengths from 0 to 10 inches and root thicknesses from 0.020 inch to 0.140 inch. The fins are trapezoidal, with a fin tip thickness of 0.020 inch. The results are displayed as weight-versus-power curves, with the dotted diagonals showing lines of constant specific power. Each curve is for a constant fin root thickness, and shows the variation of RTG weight and power as the fin length is varied. The left ends of all curves converge at a common point, which represents the case of zero fin length.

Fig. 11 - EFFECT OF FIN LENGTH AND ROOT THICKNESS ON RTG WITH 18 H.S. MODULES, 34 RINGS OF COUPLES, 4 IN PARALLEL

As can be seen, the specific power of the RTG decreases for short fins and thin fins, because of increasing cold-junction temperature and reduced conversion efficiency; and it decreases for long fins and fat fins, because of increasing radiator weight. For every power level, there is an optimum combination of fin length and root thickness which minimizes the RTG weight. This is illustrated by the curve in Figure 12, which represents the envelope of the family of curves shown in Figure 11.

Fig. 12 - OPTIMUM FIN DIMENSIONS FOR RTG WITH 18 H.S. MODULES, 34 RINGS OF COUPLES, 4 IN PARALLEL

The tick marks in the above figure indicate the optimum fin length and root thickness at each point and the resultant cold-junction temperature*. The corresponding system efficiencies are shown at the top of the figure. The left end of the curve, and of similar curves in subsequent figures, represents the case of a fin-less RTG housing. For the set of parameters examined in Figure 12, it can be seen that the specific power has a broad maximum at a fin length of about 4 inches and a root thickness of 0.080 inch. This corresponds to a radiator weight of 3.2 lbs, a cold-junction temperature of about 314°C, and a system efficiency of 6.44%. Considerably higher efficiencies can be achieved with longer fins, but at much higher weights. For example, Figure 12 shows that at a fin length of 8 inches the optimum fin root thickness is 0.162 inch, which yields a cold-junction temperature of 276°C and a system efficiency of 6.73%, but results in a radiator weight of 14.0 lbs for the eight fins.

For the case of 18 heat source modules, 4 couples in parallel, and optimized radiator fins, Figure 13 illustrates the effect of varying the number of 16-couple rings on the weight-versus-power curves. Varying the number of couple rings affects the total weight of the couples. Since the output voltage and the number of couples in parallel is fixed, it also affects the voltage per couple. Excessive deviation from the optimum couple voltage lowers the conversion efficiency, and therefore the output power. Below a certain number of couple rings, this effect outweighs the weight reduction due to the reduced number of couples, leading to reduced power-to-weight ratios. Therefore, as can be seen, there is an optimum number of couple rings, though generally not the same optimum for all power levels.

*a somewhat lighter design is possible by using curved instead of tapered profiles, but this was not used because of its relatively small benefit and because of concern about its fabricability.

Fig. 13 - EFFECT OF NUMBER OF 16-COUPLE RINGS ON RTG WITH 18 HEAT SOURCE MODULES

The number of couple rings is limited by the maximum number which can be accommodated in the available converter length without risk of contact between adjacent hot shoes. In general, the maximum number of rings considered in our study was two rings per heat source module. This limit could, of course, be avoided by using smaller hot shoes, but this would have violated DOE's instructions to Fairchild.

The additional effect of varying the number of heat source modules is illustrated in Figure 14. The figure presents curves of RTG weight versus power for 14 through 20 heat source modules and various numbers of couple rings (up to the geometric limit). The figure represents thousands of individual design points. In the curves shown, dashed lines are used to represent cases where the lateral dimensions of the couple legs is less than 100% or more than 110% of the corresponding dimension in the MHW design. As can be seen, most points in the region of interest fall within these limits.

Fig. 14 - RTG WEIGHT-VERSUS-POWER CURVES FOR VARIOUS NUMBERS OF H.S. MODULES AND CONVERTER RINGS

Part 3 of Fairchild's computer program is designed to find and plot the maximum-specific-power envelope from a large data set, such as the one illustrated in Figure 14. For a given power level, it examines every curve in the data set point-by-point, finds the corresponding RTG weight by interpolation, and determines what number of heat source modules and number of converter rings gives the lowest weight. The results are presented graphically and in tabular form.

Figure 15 shows curves of minimum RTG weight versus power for each number of heat source modules, from 14 through 20 (discarding all data points for which the leg dimensions do not fall within 100 to 110% of the corresponding MHW dimensions). As can be seen, for every power level there is an optimum number of heat source modules which minimizes the RTG weight. In selecting a design, it may be desirable to deviate from this optimum; e.g., to use a smaller-than-optimum number of heat source modules to reduce fuel inventory and/or RTG length. Figure 15 shows the weight penalty for this kind of design trade-off.

Fig. 15 - ENVELOPE OF RTG WEIGHT-VERSUS-POWER CURVES FOR EACH NUMBER OF HEAT SOURCE MODULES

As seen, the minimum specific power decreases at the lower power levels. To some extent this is to be expected, since end losses and end section weights become more significant in smaller RTGs. But there is another important reason for the lower specific power levels: As the number of heat source modules is reduced, the number of couple rings which can be accommodated goes down. As the maximum number of couple rings drops significantly below its optimum value, the efficiency goes down, which lowers the specific power.

Thus, below a certain power level, a lighter RTG design can be obtained by using 3 instead of 4 couples in parallel. This increases the couples in series and reduces the voltage per couple. This effect is illustrated in Figure 16. The curve labeled 4-4 represents the master envelope of all the curves shown

Fig. 16 - ENVELOPES OF RTG WEIGHT-VERSUS-POWER CURVES FOR 3 AND 4 COUPLES IN PARALLEL

in Figure 15; i.e., the lightest solutions with 4 couples in parallel. Only data points in which the thermoelectric legs' lateral dimensions fall between 100% and 110% of the MHW values have been included.

The dashed curve labeled 3-3' represents the lightest solutions for 3 couples in parallel, before the above geometric limit is applied. As can be seen, without that limit the 3-in-parallel design gives consistently lower RTG weights, over the entire power range. Fewer couples, with fatter legs, yield a lighter generator. But most of the points on the dashed curve represent designs in which the lateral leg dimensions exceed the MHW value by more than 10%, which violates our ground rule. When these points are discarded, the solid curve labeled 3-3 is obtained. This represents the lightest solutions with 3 parallel couples satisfying the geometric ground rule. The imposition of that ground rule accounts for the very jagged appearance of this and some later envelopes.

Figure 16 shows that, even with the geometric constraint, the 3-in-parallel design is superior for power levels up to about 280 watts. Only above that power are lighter RTGs obtained with 4 couples in parallel. The superiority of the 3-in-parallel design over much of the power range was unexpected, because that design is not well adapted to the basic arrangement of 16 couples per ring, since 16 is not a multiple of 3. As illustrated in Figure 17, with a 4-in-parallel circuit, any number of couple rings can be used to optimize the RTG design; but with a 3-in-parallel arrangement, the number of couple rings must itself be a multiple of 3. Because of this constraint, the three-in-parallel design cannot be optimized as well as the 4-in-parallel case.

Fig. 17 - ILLUSTRATIVE NETWORK ARRANGEMENTS

4 COUPLES IN PARALLEL 3 COUPLES IN PARALLEL

For each power level shown, the envelope of the two solid curves in Figure 16 represents the minimum RTG weight for the set of fixed parameters listed at the beginning of this section. As seen, for most of the power range the lightest system yields a specific power between 2.7 and 2.9 watts/lb. The variable design parameters which produce these minimum weights are summarized in Table II. For each power level from 220 watts to 330 watts, the table gives the optimum number of heat source modules, number of 16-couple rings, number of couples in parallel and series, optimum fin length and root thickness; the resultant cold-junction, fin-root, and fin-tip temperatures; the lateral dimensions of the thermoelectric legs compared to those in the MHW RTG; the thermal, thermoelectric, and system efficiencies; the RTG weight and specific power.

Table II: OPTIMUM DESIGN PARAMETERS

BOL POWER, W	OPTIMUM DES.			FIN		TEMP., °C				% EFFICIENCY		RTG WEIGHT, LB.	SPEC. POWER, W/LB.		
	H.S. MODULES	RINGS	PARALLEL SERIES	LENGTH (IN.)	ROOT (MIL)	COLD JUNCTION	FIN ROOT	FIN TIP	TIE LEG WIDTH REL. TO MHW	THERMAL	THERMOELECTRIC SYSTEM				
220	14	27	3	144	2.66	46	331	290	225	1.05	89	7.03	6.29	79.5	2.77
222	14	27	3	144	3.29	57	323	281	208	1.05	89	7.10	6.34	80.0	2.77
224	14	27	3	144	4.32	66	314	272	183	1.04	88	7.20	6.40	80.9	2.77
226	14	27	3	144	4.75	92	306	262	180	1.03	88	7.27	6.46	82.0	2.76
228	14	27	3	144	5.56	114	296	253	168	1.02	89	7.30	6.51	83.6	2.73
230	15	30	3	160	0.92	20	359	320	293	1.05	89	6.85	6.13	84.2	2.73
232	15	30	3	160	1.32	24	350	311	271	1.04	89	6.90	6.19	84.3	2.75
234	15	30	3	160	1.85	32	341	301	250	1.03	89	7.00	6.24	84.6	2.77
236	15	27	3	144	2.25	43	336	294	237	1.10	89	7.00	6.29	83.8	2.81
238	15	27	3	144	3.09	44	328	286	209	1.09	89	7.10	6.35	84.3	2.82
240	15	27	3	144	3.68	57	321	278	196	1.09	90	7.10	6.40	84.9	2.83
242	15	27	3	144	4.32	72	313	270	184	1.08	89	7.20	6.45	85.8	2.82
244	15	27	3	144	4.92	92	304	261	176	1.07	89	7.30	6.51	86.9	2.81
246	15	27	3	144	5.56	115	296	252	168	1.06	89	7.30	6.56	88.5	2.78
248	16	30	3	160	1.18	20	354	314	277	1.08	89	6.89	6.20	88.8	2.79
250	16	30	3	160	1.74	24	346	306	250	1.07	90	6.90	6.25	89.0	2.81
252	16	30	3	160	2.08	43	337	296	245	1.07	89	7.00	6.30	89.4	2.82
254	16	30	3	160	2.98	46	328	286	214	1.06	89	7.08	6.35	90.0	2.82
256	16	30	3	160	3.70	62	319	276	197	1.05	90	7.10	6.40	90.8	2.82
258	16	30	3	160	4.32	86	309	266	188	1.04	89	7.20	6.45	92.1	2.80
260	16	30	3	160	5.23	111	299	255	174	1.03	90	7.20	6.50	93.9	2.77
262	17	33	3	176	1.84	26	344	303	247	1.05	90	6.80	6.16	94.7	2.77
264	17	33	3	176	2.54	43	333	292	228	1.04	90	6.90	6.21	95.3	2.77
266	16	27	3	144	6.58	134	287	243	153	1.09	89	7.40	6.65	95.6	2.78
268	17	34	4	136	3.07	46	327	285	210	1.05	90	7.00	6.31	96.3	2.78
270	17	30	3	160	2.47	43	333	292	230	1.10	90	7.00	6.35	94.2	2.87
272	17	30	3	160	3.09	53	325	283	212	1.09	90	7.10	6.40	94.8	2.87
274	17	30	3	160	3.70	72	316	274	200	1.08	90	7.10	6.45	95.8	2.86
276	17	30	3	160	4.47	92	307	264	186	1.08	90	7.20	6.49	97.2	2.84
278	17	30	3	160	5.48	111	297	254	169	1.07	90	7.20	6.54	99.1	2.81
280	18	33	3	176	2.16	39	337	296	240	1.07	90	6.85	6.22	99.6	2.81
282	18	35	4	140	1.85	34	340	300	251	1.07	90	6.90	6.27	100.3	2.81
284	18	34	4	136	2.73	46	330	289	222	1.08	90	6.94	6.31	100.7	2.82
286	18	34	4	136	3.09	53	325	283	212	1.08	90	7.00	6.36	101.1	2.83
288	18	35	4	140	3.09	56	324	282	213	1.05	90	7.10	6.40	101.5	2.84
290	18	35	4	140	3.77	63	318	276	196	1.05	90	7.11	6.44	102.2	2.84
292	18	35	4	140	4.32	73	312	269	185	1.04	90	7.20	6.49	103.1	2.83
294	18	35	4	140	4.63	92	306	263	183	1.04	90	7.20	6.53	104.1	2.82
296	19	36	4	144	1.23	21	352	312	274	1.09	91	6.80	6.23	104.7	2.83
298	19	33	3	176	2.55	46	332	290	229	1.10	90	6.90	6.27	104.6	2.85
300	19	35	4	140	2.08	39	338	297	243	1.10	91	6.94	6.32	105.0	2.86
302	19	35	4	140	2.47	45	333	291	231	1.09	90	7.00	6.36	105.4	2.86
304	19	35	4	140	3.03	46	327	286	212	1.09	91	7.00	6.40	105.8	2.87
306	19	35	4	140	3.36	57	322	280	206	1.08	90	7.10	6.44	106.3	2.88
308	19	35	4	140	3.70	69	316	274	200	1.08	91	7.11	6.48	107.0	2.88
310	19	35	4	140	4.09	83	311	266	194	1.07	90	7.20	6.53	107.9	2.87
312	19	35	4	140	4.77	92	305	262	179	1.07	91	7.20	6.57	109.1	2.86
314	20	38	4	152	1.21	21	352	313	276	1.08	91	6.90	6.28	110.0	2.85
316	20	37	4	148	1.68	24	346	306	253	1.10	91	6.90	6.32	109.8	2.88
318	20	37	4	148	1.85	33	341	300	250	1.09	90	7.00	6.36	110.1	2.89
320	20	37	4	148	2.42	39	335	294	230	1.09	91	7.00	6.40	110.5	2.90
322	20	37	4	148	2.78	45	329	288	220	1.08	91	7.05	6.44	111.0	2.90
324	20	37	4	148	3.09	56	323	282	213	1.08	91	7.10	6.48	111.5	2.91
326	20	37	4	148	3.83	63	317	275	194	1.07	90	7.20	6.52	112.3	2.90
328	20	37	4	148	4.32	75	311	268	185	1.07	91	7.20	6.56	113.3	2.90
330	20	37	4	148	4.76	92	305	262	180	1.06	90	7.27	6.60	114.5	2.88

For the power range from 220 to 330 watts BOL, the optimum design calls for: 14 to 20 heat source modules, with corresponding lengths of 28.4 to 40.5 inches for the heat source stack, 26.4 to 38.5 inches for the radiator fins and 32.5 to 44.6 inches for the total RTG; 432 to 592 thermoelectric couples, arranged in 27 to 37 rings; 136 to 176 couples in series, with

3 in parallel at lower power levels and 4 in parallel at the upper end of the power range; and fin lengths ranging from 0.92 inch to 6.58 inches, with corresponding root thicknesses of 20 to 134 mils, and RTG spans from 9.8 to 21.1 inches. These design optima yield temperatures of 296 to 359°C for the cold junctions, 252 to 320°C for the fin roots, and 153 to 293°C for the fin tips; thermoelectric legs 2 to 10% thicker than the MHW design; thermal efficiencies of 86 to 91%, thermoelectric efficiencies of 6.8 to 7.3%, and system efficiencies of 6.1 to 6.6%. The minimum RTG weights range from 79.5 to 114.5 lbs, and the maximum specific power generally lies between 2.7 and 2.9 watts per lb.

There are a number of interesting tendencies to be noted in Table II. The fact that 3-in-parallel circuits give lower weights than 4-in-parallel networks over much of the power range has already been mentioned. The optimum number of couple rings is generally lower than the geometric limit of two rings per heat source module. The optimum radiator has relatively short fins, even though this leads to high cold-junction temperatures and low efficiencies. The benefits obtained by longer fins do not compensate for the resultant radiator weight increase. Similarly, the optimum fins have relatively thin roots, even though this leads to large fin temperature drops and low fin effectiveness. Particularly noteworthy is the narrow range of the maximum specific power over the wide power range examined.

The lower curve in Figure 18 is the envelope of the two curves shown in Figure 16; i.e., the minimum-RTG-weight envelope for the stated assumptions. The upper curve shows a similar weight envelope for the case of two half-power RTGs instead of one full-power generator. As would be expected, for any given power level the two-RTG case is appreciably heavier than the single generator. This is partly because of the weight and heat losses of the additional end sections; and partly because, in the two-RTG case, the number of heat source modules can only be varied in steps of two, which explains the deep scallops in the curve for two RTGs.

Fig. 18 - COMPARISON OF ONE FULL-POWER RTG WITH TWO HALF-POWER RTGS

The weight difference shown in Figure 18 represents only the difference in actual RTG weights. It does not include any additional structural support weight required for the two-RTG case. Nor does it take note of the overall length increase (about 6 inches) which would result if the two RTGs were stacked end-to-end.

Next, let us examine the sensitivity of the RTG weight to the heat source module design parameters, comparing the heat source design analyzed thus far

with two other design options tested earlier at LASL. In the order of increasing weight, these will be referred to as Option A, B, and C. (Option B, the heat source module design used in the preceding analysis, is depicted in Figure 19).

Fig. 19 - HEAT SOURCE MODULE DESIGN, OPTION B

All three designs use identical fuel pellets, iridium capsules, and double pyrolytic graphite shells. The main difference between them is in the design of the 3D-graphite impact shells and aeroshells. In the Option A and B designs, two fuel capsules are contained in a single impact shell with a 0.150-inch side wall. The end walls and the wall between the two capsules is 0.150 inch thick in Option A, and 0.230 inch in Option B. The Option C design uses a separate impact shell for each fuel capsule, with wall thicknesses of 0.200 inch. All three designs use a minimum aeroshell thickness of 0.200 inch. But in the Option A and B designs the aeroshell is hollowed out, and there is no gap between adjacent impact shells; while in the third design the aeroshell is not hollowed out, except for the necessary fuel cavities, which are separated by a minimum 0.050 inch of 3D graphite. The three modules have respective heights of 2.02, 2.03, and 2.11 inches; diagonal dimensions of 5.06, 5.27, and 5.46 inches; weights of 3.15, 3.32, and 3.61 lbs; and specific powers of 79.5, 75.3, and 69.3 thermal watts per lb.

Fig. 20 - SENSITIVITY OF RTG WEIGHT TO HEAT SOURCE MODULE DESIGN

The effect of these heat source design differences is illustrated in Figure 20. The Option B curve is the envelope of the two curves shown earlier in Figure 16. The other curves represent the corresponding envelopes for Options A and C. As can be seen, variations in the heat source design have an appreciable effect on RTG weight.

For the above heat source options, Figure 21 shows a weight comparison of the ISPM RTG with the MHW RTGs used in previous missions. Based on measured MHW degradation rates, the present ISPM power specification of 250 watts at the end of mission (EOM) requires a BOL power of 292 watts. For that BOL power, Figure 21 shows that the RTG described in this paper would weigh 100, 103, and 108 lbs, respectively, for heat source options A, B, and C. By comparison, meeting the ISPM goal of 250 watts EOM would require two MHW generators, weighing a total of 166 lbs.

Fig. 21 - COMPARISON OF RTG WEIGHT WITH MHW

It is recognized, of course, that we are comparing an actual-hardware design with a paper design which could grow significantly during its development. But the paper design, except for the heat source, is based largely on the same components as the hardware design. And the weight difference between them is so large that the design described here could grow substantially and still be a much lighter solution for ISPM than the MHW option.

Finally, let us briefly examine the effect of the choice of housing and radiator material on RTG weight. All the results presented thus far have been for beryllium. At typical wall temperatures of 250 to 300°C, the mechanical strength of aluminum and magnesium alloys is so low as to make these materials unattractive from a weight viewpoint.

But the ISPM RTGs will in any case have to be cooled during the shuttle launch phase. Cooling will be by a circulating water loop, delivering the reject heat to a water boiler which discharges the resultant steam outside the shuttle. Therefore, it would be possible to maintain the RTG housing at a relatively

low temperature, e.g., below 100°C, during the shuttle launch phase. This would gain the benefit of high material strength when it is most needed, i.e., when launch loads are most severe.

It is true, of course, that the RTG must also survive the IUS launch loads, after the spacecraft has left the shuttle and its auxiliary cooling system. Therefore, during IUS firings, the RTG housing would be at its normal operating temperature of 250-300°C, and hence aluminum or magnesium would be in a weakened condition. But this may not be too serious, because the IUS launch loads are less than half as high as the shuttle launch loads [3].

Figure 22 presents a weight comparison of RTGs with beryllium, aluminum (2219-T851), and magnesium (HM31A) housings and fins. The different densities, strengths, and thermal conductivities of these materials were taken into account. The analysis assumed transverse launch loads of 40G at 90°C for the shuttle, and 20G at 300°C for the IUS.

Fig. 22 - EFFECT OF HOUSING AND FIN MATERIAL ON RTG WEIGHT

For the above assumptions, Figure 22 shows that magnesium gives almost the same RTG weight as beryllium. Aluminum gives appreciably higher weights, but these could be brought much closer to those for the other alloys if, because of its superior machinability, the minimum housing wall thickness could be reduced from 0.060 to 0.040 inch. Whether such a thin-walled housing could be fabricated is still under investigation, as is the question whether aluminum or magnesium could be used without excessive creep deformation during launch. In view of the difficulty and cost of fabricating and joining beryllium parts, the use of the other alloys seems well worth further exploration.

CONCLUSIONS

For the basic design topology described in Section II, and for a stipulated set of assumptions, ground rules, and constraints, the optimization program described above can rapidly determine the parameters which yield the lightest RTG design for a given power level. The program can easily be adapted to different sets of assumptions.

The program was applied to parametric analyses of ISPM RTGs, with results leading to the following conclusions:

1. Even though the heat source thermal power can only be varied in steps of 250 watts, the RTG output power is infinitely variable, by adjusting the other design parameters.
2. With optimized designs, the maximum specific power varies very little ($\pm 3\%$) over the wide power range investigated (210 to 330 watts).
3. RTG weights tend to minimize at relatively high cold-junction temperatures, with relatively short and thin radiator fins, even though efficiencies could be increased by using heavier radiators.
4. Over much of the power range investigated, a circuit with three couples in parallel gives a lighter RTG than one with four in parallel.
5. Over the power range investigated, two half-power RTGs would be considerably heavier than one full-power RTG.
6. RTG weights are quite sensitive to differences between the heat source module designs under current study.
7. Even the most conservative heat source module design under current consideration yields a much lighter power system for ISPM than would the previously used MHW RTGs.
8. For a shuttle-launched space mission, where auxiliary cooling must in any case be provided while the RTGs are inside the shuttle bay, the use of aluminum or magnesium alloys for the RTG housing and fins may be feasible.
9. The use of these alloys instead of beryllium would increase RTG weights only slightly, and should substantially reduce fabrication problems and costs.

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