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JUNE 1, 1979

FSEC-ESD-217-79/164

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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, to be launched in early 1983.

The use of stacked heat source modules in RTGs requires a structural support system which holds the stack together during launch, but allows it to disassemble when the generator housing melts during reentry. This is a much more difficult support problem than in the case of a monolithic heat source. A proposed, light-weight scheme for solving this problem is described and analyzed. A detailed analytical model of the RTG and the spacecraft to which it is attached was constructed, and a NASTRAN analysis was carried out to assess the structural performance of the heat source support scheme under differential thermal expansion and dynamic launch loads, to determine the system's characteristic frequencies, the loads transmitted to the RTG, and the resultant stresses and displacements. The analytical results presented confirm the structural feasibility of the heat source support scheme.

The paper also describes a detailed thermal analysis and 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 MHW RTG, and comparative RTG weights for different housing and radiator fin materials.

I. INTRODUCTION

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 [1]. 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.

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.

*under Contract EN 77-C-02-4281

Fig. 1 - SOLAR POLAR MISSION PLAN

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 (MHW) 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 the course of carrying out the power system design study, it became apparent that the use of a stacked modular heat source entails special design problems not present in the case of the monolithic heat sources used on previous missions, such as the one in the MHW RTG. A discussion of those special problems, a proposed light-weight solution, and a structural analysis to assess its viability will be presented in Section II.

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 rationale, major components, and topology of the RTG design evolved; Section III describes a detailed structural analysis to assess the viability of the proposed heat source support scheme under anticipated dynamic launch loads; and Section IV describes the parametric optimization studies performed and their results.

II. DESIGN DESCRIPTION AND RATIONALE

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 (UNICUPLE)

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 [2], 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 [3]. 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 claimed advantages of a modular heat source (as opposed to a monolithic heat source) include better scalability to a variety of power levels (without need for safety requalification), improved testability in existing safety test facilities, lower weight and smaller size for a given level of safety, and/or better safety margins for a given weight and size. The last two advantages derive primarily from the heat source's lower impact velocity (after reentry), which permits lighter and smaller impact protection members.

The individual heat source modules have a substantially lower impact velocity than a monolithic heat source because of their lower ballistic coefficient ($W/C_d A$). But this advantage only accrues if the modular heat source stack disassembles during reentry. Without disassembly, the impact velocity of a modular heat source would not be significantly lower than that of a monolithic heat source.

To ensure disassembly of the heat source prior to impact, the structural members which hold the modular stack together must come apart during reentry, to release the individual heat source modules. This precludes the use of refractory metal straps for tying the stacked modules together. The use of non-refractory (e.g., superalloy) straps or tie rods also appears impractical, because of inadequate strength at the anticipated operating and launch temperatures.

The required RTG structure must not only support the heat source during launch, but must also prevent the modular stack from falling apart under the influence of the severe lateral launch loads. This latter requirement would not exist in the case of a monolithic heat source.

The RTG for the Solar-Polar mission typically requires a stack of 18 heat source modules, for a total thermal power of 4500 watts at the beginning of life. Thus, a typical heat source stack is over 36 inches long and weighs about 60 lbs.

The axial and lateral loads to which the heat source will be subjected during launch depend on the inputs from the space shuttle and the IUS. In addition, the loads on the heat source stack depend on the amplification and damping characteristics of the spacecraft and RTG structures. None of the above are yet fully defined, but axial and lateral loads of about 40 G are anticipated. (See Section III)

As already explained, the heat source stack cannot be held together by high-temperature straps or tie rods, since they may prevent disassembly during reentry. Rather, the heat source stack must be held together by the RTG's low-temperature housing, which will usually melt during reentry. There are, of course, abort modes (e.g., a sub-orbital launch failure) in which the housing will not melt, and the generator will impact without prior disassembly. But that case does not present a special safety problem, because the ballistic coefficient of the assembled generator is low enough to ensure an adequately low impact velocity. On the other hand, if the generator housing does melt during reentry, the heat source modules are no longer constrained, and the heat source will disassemble as the result of aerodynamic forces. Thus, the heat source modules will have the desired low impact velocity.

The 36-inch-long heat source stack is supported by the generator housing at its ends. From a structural viewpoint, it would be desirable to provide additional lateral supports at intermediate points on the stack, but this would be undesirable from the thermal and mechanical design standpoint. The use of intermediate supports would require lengthening the generator to provide room for the supports, resulting in increased weight; and would result in increased heat losses and reduced efficiency. For these reasons, no intermediate heat source supports were used in Fairchild's Solar-Polar RTG design.

Since the heat source stack is only supported at its ends, a high axial preload is required to keep the stack from falling apart under the influence of lateral launch loads. The analysis presented in Section III indicates that axial preloads of 12,000 lbs will be required to ensure stability during the shuttle launch. This is a much more severe requirement than for the case of a monolithic heat source.

The RTG designer must solve the problem of providing axial and lateral heat source supports, with very high axial preloads, while minimizing the mass of the structure and the resultant heat losses. The solution evolved in the Fairchild study is to use pyrolytic graphite (PG) buttons, backed by Belleville springs, at the four corners of each end of the heat source. A total of 16 PG buttons is used: eight for axial support, and eight for lateral support. The PG buttons penetrate the multifoil thermal insulation, and the Belleville backup springs press against the housing end caps.

The heat source support concept is illustrated by the exploded view in Figure 5. The figure shows only one heat source module, but it is understood that this represents a stack of 18 such modules. As can be seen, there is a light-weight molybdenum cross-piece embracing the four corners of each end module. At the extremities of those cross-pieces are expanded seats, which mate with the pyrolytic graphite support buttons. They make it possible to locate the axial support buttons as far apart as possible, which reduces the preload required to hold the stack together during lateral launch loads. The cross-piece seats also serve to spread the load of the axial support buttons over a larger area, to reduce the compressive stress on the heat source blocks. Those blocks are made of a three-dimensional carbon-bonded carbon-fiber structure, which has considerably less compressive strength than pyrolytic graphite.

Fig. 5 - HEAT SOURCE SUPPORTS

As seen in Figure 5, the side support buttons are much longer than the end buttons. This is necessary because the heat source's side insulation, which contains the thermoelements, is much thicker than the simple multifoil that insulates the ends of the heat source stack. Since the side PG buttons are longer, their diameter has also been increased to provide columnar stability. Because of their large cross-sectional area, the side buttons are conservatively designed, not only with respect to compressive stress but also with respect to shear stress.

Figure 6 shows the two molybdenum cross-pieces and the sixteen PG buttons joined to the heat source stack. Only the end modules of the stack are shown. Also shown are two graphite shear balls, which are present at each interface between adjacent modules. These balls, which are seated in hemispheric cavities, provide a limited interlock function. They aid assembly and supplement friction in resisting lateral shear forces. But they contribute nothing to overcoming the tensile forces due to bending moments. Those tensile forces are resisted only by the compressive loads exerted by the axial PG buttons.

Fig. 6 - SPRING SUPPORTS AND SHEAR BALLS

The components which spring-load the PG buttons are depicted in Figure 6, at two of the eight corners of the heat source stack. As shown, the side-load spring consists of a single Belleville washer. The end spring, which must furnish a much greater force, consists of pairs of two conical washers in parallel.

Belleville springs, which are very stiff but have a short stroke, are ideal for this application. Relatively little travel is required, because the spring loads are applied after the generator is assembled and the heat source and housing have had a chance to equilibrate at their normal temperatures. Thus, the springs do not need to compensate for dimensional tolerance stack-up. The only spring travel required is for the thermal changes which occur after the final preloads are applied at the launch site.

As seen in Figure 6, at the inner end of each Belleville spring is a metal bushing, to distribute the spring force over the end face of the PG button. There is another bushing at the outer end of each spring. These transmit the axial and radial spring loads to the housing end caps.

The structural support and preload application scheme evolved in the Fairchild study is illustrated by the exploded view in Figure 7 and the assembled close-up view shown in Figure 8. The multi-foil thermal insulation has been omitted in Figure 7 and earlier figures, to avoid obscuring other components.

The end cap design depicted at the top of Figure 7 is the result of extensive design iterations and NASTRAN analyses, to find a low-weight solution for the high design loads. If made of beryllium, each end cap weighs less than 1 lb. As can be seen, the end cap has four cup-shaped seats and load-adjustment holes for the axial Belleville springs. These seats are combined within two very stiff I-shaped cross-beams, to resist the bending moments produced by the axial spring forces.

Fig. 7 - HEAT SOURCE SUPPORT STRUCTURE

As illustrated in Figure 8, the end caps' flanges are bolted to mating flanges on the RTG's cylindrical side wall. Gas-tight seals are obtained by means of metallic C-rings. This type of seal has been successfully used on earlier silicon-germanium RTGs (e.g., on the Voyager mission). Si-Ge thermoelements are designed to operate in vacuum, and sealing is only required for temporary retention of an inert cover gas. Once in orbit, the RTG is vented to space.

As shown in Figure 8, the axial preloads can be adjusted to their desired values after final RTG assembly and thermal equilibration while maintaining the generator in a sealed condition. At each of the eight heat source corners, the desired load can be applied through the center of the lock screws, by

Fig. 8 - LOAD ADJUSTMENT SCHEME FOR SEALED RTG

means of compression fixtures using hydraulically or pneumatically actuated pistons (not shown). Thus, dimensional tolerance stack-ups--within limits--are automatically compensated at the time the eight axial loads are applied.

After the desired axial preloads have been applied, the lock screws are tightened and the compression fixtures removed. Throughout this process, the seal at each sliding piston is maintained either by means of a pressure-loaded C-ring, as shown, or by means of a bellows. Since these operations can be carried out in an air environment, final spring force adjustment can be made at the launch site, shortly before launch. Thus, the very high spring forces need not be present during long-term storage, or during shipment to the launch site.

Similarly, the side spring forces can also be adjusted after RTG assembly and thermal equilibration. As shown in Figure 8, these forces are exerted against the rim of the end cap, at the ends of the two cross-beams.

Fig. 9 - CUT-AWAY VIEW OF ASSEMBLED RTG

Figure 9 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.

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 9, the uncouples 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 C-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 connector straps.

Fig. 10 - ILLUSTRATIVE NETWORK ARRANGEMENTS

4 COUPLES IN PARALLEL 3 COUPLES IN PARALLEL

The parallel leads normally carry no current, but provide high reliability against open-circuit failure. The arrangement illustrated is for four couples in parallel. The corresponding circuit is depicted schematically in Figure 10, which also shows an alternative circuit arrangement for three couples in parallel. The effect of these alternatives on RTG weight will be examined in Section IV.

Figure 11 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 uncouples. 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.

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 12 for the NASA/JPL spacecraft.

Fig. 12 - RTG MOUNTED ON SPACECRAFT

In a similar fashion, an identical RTG is mounted on the ESA spacecraft, as illustrated in Figure 13. After the firing of the IUS upper stage is completed, the ESA spacecraft and its support structure separates from the NASA spacecraft, followed by separation of the NASA spacecraft from the IUS.

Fig. 13 - SPACECRAFTS WITH RTGs ON IUS

As depicted in Figure 14, 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. 16 - STRUCTURAL-ANALYSIS MODEL OF RTG END CAP

While the RTGs are within the shuttle bay, auxiliary cooling must be provided for them, because of their high heat rejection rate. Cooling will be by a circulating water loop, delivering the reject heat to a water boiler which discharges the resultant steam outside the shuttle. The RTGs could be cooled directly, by water tubes bonded to their fin roots. This would require disconnection of coolant supply lines in earth orbit. To avoid that complication, the RTGs could be cooled by air convection and radiation to a separate water-cooled shroud, as illustrated in Figure 15.

Fig. 15 - COOLING ARRANGEMENT FOR RTG IN SHUTTLE BAY

With this arrangement, there is no need for breaking coolant lines in orbit, and no danger of coolant droplets lodging on optical or other sensors. The cooling shroud opens in a clamshell fashion, permitting unobstructed removal of the IUS/spacescraft assembly, and its deployment outside the shuttle.

To save computer time in the subsequent analyses, a simplified model of an end cap with equivalent structural characteristics was constructed. The two cross-beams are represented by bars of equivalent stiffness. The model is shown at the top and bottom of Figure 17, which also shows the finite-element models of the RTG's other structural components: the heat source stack, its support springs, and the generator housing.

Fig. 17 - STRUCTURAL-ANALYSIS MODEL OF RTG

The heat source stack depleted consists of 18 modules, which are modeled as hollow boxes with six face plates. As can be seen, the six middle modules and the three at each end of the stack are modeled individually. The other six have been combined into two groups of three. A more detailed model is used

at the stack's ends and center, where the bending moments due to lateral loads are greatest. Similarly, in the RTG housing model, a finer structure is used at the bottom of every other fin, where the cantilevered RTG is attached to the spacecraft support structure. At each corner of the heat source stack are short lines of heavy dots, representing the axial and radial support springs.

Figure 18 shows the top end of the combined analytical model, which represents the structural relationship of the heat source stack, support springs, end caps, and housing in the assembled RTG. Altogether, the RTG model consists of 560 grid points, 816 elements, 3088 static degrees of freedom, and 264 dynamic degrees of freedom.

Fig. 18
- TOP END OF STRUCTURAL-ANALYSIS MODEL OF ASSEMBLED RTG

Finally, Figure 19 shows the model of a representative ISPM spacecraft with a cantilevered RTG attached to its support structure. Such a combined model can be exercised by applying the anticipated axial and lateral launch loads at the IUS/spacraft interface, to determine the various characteristic frequencies of the system, the loads transmitted to the RTG, and the resultant stresses and displacements. The availability of a structural model of the spacecraft makes it unnecessary to guess at the attenuation or amplification factors between the IUS and the RTG. Note that the quadrapod structure extending above the spacecraft supports the ESA spacecraft. A duplicate RTG is mounted on that spacecraft and a similar analysis would have to be performed in order to determine the loads on the RTG mounted on that spacecraft.

Fig. 19-STRUCTURAL-ANALYSIS MODEL OF RTG ON SPACECRAFT

The structural evaluation of the RTG examined three critical flight conditions: the shuttle launch, launch, including the effect of the 70-RPM 3rd-stage spin of the IUS. The prescribed spacecraft-level quasi-static design loads, before safety factors, are summarized in Table I, where the x-direction denotes the longitudinal axis of the orbiter, the IUS, and the spacecraft. The RTG axis lies in the orbiter's yz-plane, close to the z-direction. Therefore, the static and dynamic g-loads in the x and y directions will produce bending of the heat source stack.

Table I. INPUTS FROM IUS TO ISPM SPACECRAFT
(X, Y, Z Directions Refer To Shuttle Axes)

EVENT	LOAD DIRECTION	STEADY G-LOAD	DYNAMIC G-LOAD
SHUTTLE LAUNCH	X	-1.6	+2.5
	Y	0.0	+1.9
	Z	0.0	+6.0
SHUTTLE LANDING	X	0.0	+0.7
	Y	0.0	+1.0
	Z	+1.0	+7.7
IUS LAUNCH	X	-5.5	+1.5
	Y	0.0	+1.0
	Z	0.0	+1.0

Table II presents the results of a dynamic response analysis, both random and sinusoidal, assuming 5% structural damping. The analysis used a detailed model of the ISPM spacecraft and the RTG support structure, and a concentrated RTG mass, located 22.5 degrees off the orbiter's z-axis. The table shows the three resonant frequencies resulting in vibrational inputs to the RTG, the resultant tri-axial g-load amplification factors at the RTG, as well as the overall responses to random vibration (using the 8.4 GRMS input predicted by Boeing).

Table II. DYNAMIC RESPONSE OF CONCENTRATED RTG MASS DUE TO SINUSOIDAL AND RANDOM INPUTS AT IUS/SC INTERFACE
(X, Y, Z Directions Refer To Shuttle Axes)

RESPONSE FREQUENCY:	37.4 Hz	64.5 Hz	126.8 Hz
DIRECTION OF RTG EXCITATION:	X FORWARD BENDING	Y SIDEWAYS BENDING	Z AXIAL VIBRATION
DYNAMIC INPUT:			
1G SINUSOIDAL FORWARD (X)	18.25G	4.41G	1.91G
SIDEWAYS (Y)	4.45G	11.05G	0.25G
VERTICAL (Z)	3.67G	5.12G	12.03G
8.4 GRMS RANDOM:			
FORWARD (X)	0.29GRMS	-	-
SIDEWAYS (Y)	-	8.31GRMS	-
VERTICAL (Z)	-	-	15.72GRMS

JPL has proposed an input level of 1.5G for the sinusoidal tests, subject to specified load limits which are a function of spacecraft appendage weight. Considering the RTG as a 100 lb appendage with 2% structural damping, the specified test load limit is 25G, which would be achieved by notching the input levels in the sinusoidal test of the spacecraft. For a safety factor of 1.4, this results in an ultimate design load of 35G.

The above launch loads, test loads, amplification factors, together with safety factors of 1.4 for the shuttle launch and 1.25 for the IUS free flight, result in the design loads shown in Table III.

B-B
HEAT SOURCE MODULE
END SUPPORT

Table III. ULTIMATE G-LOAD INPUTS TO RTG
(Including Safety Factors)

EVENT	DIRECTION OF LOAD ON RTG	
	LATERAL	AXIAL
FLIGHT PHASE:		
SHUTTLE LAUNCH	10.2	2.7
SHUTTLE LANDING	12.2	1.4
IUS LAUNCH*	8.8	13.1
SINE TEST (1, 5G):		
X (FORWARD)	38.5	4.0
Y (SIDEWAYS)	23.2	0.5
Z (VERTICAL)	10.8	25.3

* INCLUDING EFFECT OF 70 RPM SPIN

As can be seen, the dynamic test conditions are much more severe than the anticipated flight conditions. It was therefore concluded that these should govern the initial RTG design load criteria. This conclusion must be verified by the dynamic analysis of the combined spacecraft and RTG model shown in Figure 19. Based on the results tabulated above, the three ultimate design load criteria shown in Table IV were selected for the structural analysis of the RTG:

Table IV.

ULTIMATE DESIGN LOAD CRITERIA FOR RTG	
•	40g LATERAL PLUS 11g AXIAL AT SHUTTLE LAUNCH TEMPERATURES
•	15g LATERAL PLUS 27g AXIAL AT SHUTTLE LAUNCH TEMPERATURES
•	14g LATERAL PLUS 20g AXIAL AT IUS LAUNCH TEMPERATURES

The last design criterion is 50% higher than the predicted IUS launch loads, since the required transient response analysis has not been done yet. Although the last load combination is less severe than the first two, it must be examined separately for two reasons: The RTG housing is warmer (and therefore weaker) during launch, because of less effective cooling; and the axial spring forces during the IUS launch are substantially lower, due to the differential thermal expansion effects discussed later.

To determine the axial spring forces required to hold the heat source together during the shuttle launch, the detailed structural model of the RTG described earlier was subjected to a combined quasi-static load of 40g lateral and 11g axial. The results of the NASTRAN analysis indicated that, with a load of 500 lbs on each side spring, a load of 2840 lbs on each axial spring is required to maintain a positive compressive stress at all interface locations, to ensure that the module stack will not fall apart under the bending moments produced by the lateral load. Thus, the analysis indicates that a combined axial spring load of about 12,000 lbs is required to survive the shuttle launch; and a similar analysis showed that a combined spring load of about 4,000 lbs is required to survive the IUS launch loads.

The required spring constant of the axial springs is determined by differential thermal expansion effects. Because of RTG temperature changes between the shuttle launch and the IUS launch, there is a change in the relative length of the heat source and housing. As will be shown, these thermal expansions have the effect of increasing the axial spring lengths and relaxing the spring forces. Therefore, the axial spring constants must be low enough to retain sufficient axial load (4000 lbs) for surviving the IUS launch.

The housing temperature depends primarily on the cooling method employed. If water is pumped through coolant tubes attached to the fin roots, the housing temperature will be around 100°C. If the heat trans-

fer from housing is by natural convection of ambient air (e.g., to a water-cooled shroud), a temperature of about 180°C is expected. And if the housing is in a vacuum, and cooled only by radiation, its temperature is around 300°C.

The temperature difference between the housing and the heat source depends primarily on the cover gas inside the RTG. Cover gas will be used to keep the heat source and the thermoelectrics cool during prelaunch storage and shipment, to minimize any degradation; and to prevent the infusion of oxygen when the RTG is surrounded by atmospheric air.

Table V presents approximate values of the heat source surface temperature for three different cooling methods and five different cover gases (plus vacuum).

Table V. EFFECT OF COOLING METHOD AND COVER GAS ON H. S. TEMPERATURES

HOUSING COOLED BY		FORCED CONVECTION	NATURAL CONVECTION	RADIATION
COOLANT HOUSING TEMPERATURE		WATER 100°C	AIR 180°C	VACUUM 300°C
HEAT SOURCE TEMP., °C				
COVER GAS	HELIUM	604	656	745
	NEON	750	811	908
	ARGON	836	899	996
	KRYPTON	865	928	1025
	XENON	881	944	1041
	VACUUM	909	971	1067

NORMAL OPERATION IN SPACE →

The scenario which the RTG will experience from assembly through launch has not been fully defined, but some general principles can be discerned. After the RTG has been assembled and acceptance tested, it may be stored in a dormant condition for about a year. During this period, the RTG may be filled with helium and water cooled, to lower the heat source temperature in order to minimize grain growth of the iridium fuel capsules and silicon sublimation from the thermoelectric elements. But during subsequent shipment to the launch site, natural-convection air cooling seems preferable to water cooling, to reduce the complexity of the required ground support equipment.

It is probable that the least conductive inert gas (xenon) will be used during the shuttle launch, to maximize the electrical power available to the autonomous spacecraft. The IUS launch will take place after the cover gas has been vented, and with the RTG cooled only by radiation.

The temperature variations shown in Table V can change the relative lengths of the heat source and the RTG housing. Although the heat source can experience very large temperature changes, the thermal expansion coefficient of 3D graphite is extremely low. The dominant length changes will be in the housing, because of the much higher expansion coefficient of beryllium or aluminum. The difference in the length changes between the heat source and the housing must be compensated by a similar length change in the axial springs.

For example, consider the case where the RTG is water-cooled and filled with xenon during the shuttle launch, and radiation-cooled and evacuated before the IUS launch. This raises the temperature of the beryllium housing from 100°C to 300°C, causing a length increase of 0.092 inch; and raises the heat source temperature from 881°C to 1067°C, causing a length increase of 0.018 inch. Thus, there is a net differential thermal growth of 0.074 inch, which must be compensated by a reduction in the axial spring forces.

To provide retention of adequate spring load during the IUS launch, a spring constant of 56,500 lbs/inch per axial spring was selected. Analysis (which also took account of the effective spring constants of the heat source stack, the housing, and the end caps) indicated that the thermal expansion effects cited above will reduce the combined axial spring force from 12,000 lbs to 5348 lbs. This is appreciably higher than the 4000 lb axial load which the NASTRAN analysis had indicated as required to hold the heat source stack together during the IUS launch.

The selected spring constant value of 56,500 lbs/inch per axial spring was used in a NASTRAN modal analysis to determine the resonant frequencies of the RTG. Those frequencies, together with the previously determined resonant frequencies of the ISPM spacecraft, are listed and identified in Table VI.

Table VI.

SPACECRAFT AND RTG RESONANCES YIELDED BY MODAL ANALYSIS

RESONANT FREQUENCIES (Hz)		DYNAMIC MODE
SPACECRAFT & RTG SUPPORTS	HEAT SOURCE & RTG HOUSING	
20.25		ESA SUPPORT STRUCTURE
30.35		NASA SPACECRAFT (SHELF AND PAYLOAD)
(37.4)		LATERAL EXCITATION OF RTG BY SPACECRAFT
40.55		NASA SPACECRAFT
(64.5)		LATERAL EXCITATION OF RTG BY SPACECRAFT
	(113.9)	H.S. AND HOUSING BENDING ABOUT X AND Y AXES (SIMPLE BOW)
(126.8)		AXIAL EXCITATION OF RTG BY SPACECRAFT
	218.5	H.S. ROTATION ABOUT ITS AXIS
	(220.2)	H.S. AND HOUSING BENDING ABOUT DIAGONAL AXIS (SIMPLE BOW)
	247.8	H.S. AXIAL VIBRATION
	(388.9)	H.S. BENDING ABOUT DIAGONAL (S-SHAPED)

Fig. 21 - SECOND RESONANT BOWING MODE (220 HZ)

The resonances which are of greatest concern in evaluating the viability of the heat source support scheme are those which represent bowing of the heat source stack, since they could lead to heat source disassembly and RTG destruction. The three lowest such resonances and the resultant RTG deformations (greatly exaggerated to enhance visibility) are depicted in the NASTRAN-generated grid plots shown in Figures 20, 21, and 22.

As can be seen, none of these resonances (113, 220, 389 hertz), nor any of the others listed in Table VI, are near the three resonant frequencies (37, 65, and 127 hertz) at which the spacecraft excites the RTG. The RTG resonances are sufficiently high such that the dynamic response of the RTG to low frequency excitation should be very minimal.

Fig. 20 - FIRST RESONANT BOWING MODE (113 HZ)

Fig. 22 - THIRD RESONANT BOWING MODE (389 HZ)

The stresses of various RTG members under the three critical design load conditions presented earlier are summarized in Table VI, which also gives the allowable stresses for each material. The properties of the 3D graphite used here are based on the latest available data compilation, and an estimate of 0.3 for Poisson's ratio. Additional mechanical-property tests of a complete aeroshell module would be highly desirable.

Table VI
SUMMARY OF CRITICAL RTG STRESSES (PSI)

FLIGHT PHASE		SHUTTLE		TUS	
G-LOADS: LATERAL LOAD AXIAL LOAD		40G 11G	15G 27G	14G 20G	
TEMPERATURE, °C HEAT SOURCE HOUSING		881 100	881 100	1067 300	
SPRING LOAD, LBS: AXIAL SPRING SIDE SPRING		2850 510	2850 510	1283 519	
RTG COMPONENT	TYPE OF STRESS	ALLOWABLE	COMPUTED STRESS		
END BUTTON	COMPRESSIVE	60,000	8,088	7,877	3,855
SIDE BUTTON	COMPRESSIVE	60,000	2,454	1,795	1,720
END CAP	TENSILE	15,000	16,311	15,082	8,036
H.S. END	COMPRESSIVE	15,000	11,392	8,646	5,207
	SHEAR	3,000	5,649	4,404	2,632
H.S. MIDDLE	COMPRESSIVE	15,000	5,333	3,856	2,207
	SHEAR	3,000	2,366	1,689	977
HOUSING	TENSILE	18,000	8,168	5,042	3,349
FIN	TENSILE	18,000	18,542	10,100	7,474
MAX. DEFLECTION AT H.S. MID PLANE, INCH			0.037	0.014	0.013

As can be seen, none of the stresses exceed the allowables, except for the shear stress at the ends of the heat source stack. But this may be an artifact of our analytical model, in which the axial spring forces were concentrated at the corners of the heat source stack. In actual practice, these forces would be spread out over larger areas by the molybdenum cross-pieces, and those areas could be further enlarged by expanding the shoes of these cross-pieces.

Table VII appears to indicate a marginal condition for the fins during shuttle launch, but this is not so. At the shuttle launch temperatures the allowable stress is 21,000 psi, not 18,000 psi. Thus, the computed stress under 40G lateral and 11G axial is well within the allowable limit.

Finally, Figure 23 illustrates the computed static deformation of the heat source stack and the housing under a diagonal lateral load of 40g together with an axial load of 11g. The dashed lines represent the original positions of the edge of the heat source stack and the column of nearest thermocouple hot shoes, with a normal design gap of 0.125 inch. The solid curves depict the corresponding profiles after deformation. As shown, the minimum gap after deformation is 0.087 inch.

In summary, the detailed analysis described in this section demonstrates the structural viability of the basic scheme for supporting a long heat source stack without intermediate supports, under the anticipated static, dynamic, and thermal expansion loads.

IV. DESIGN OPTIMIZATION

Section II 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 minimize the generator weight for a specified power level. The section also presents the results of the optimization for the power range of interest.

Fig. 23 - RADIAL DEFLECTIONS UNDER STATIC G-LOADS:
40G LATERAL (DIAGONAL) PLUS 11G AXIAL

The fixed input parameters include the thermoelectric hot-junction temperature (1000°C); the length (1.23 in) and weight* (0.0207 lb) of each uncouple 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 uncouple'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.

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.

*based on NHV information, with suitable adjustments

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 MHW 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, whichever 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 [6,7] 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 24, 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 24.

Fig. 24 - 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.

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 total heat rejection rate of the RTG is 4208 watts.

The weight breakdown for the illustrated case is summarized in Table VII. 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.

Table VII: 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

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 [3], 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 in Figure 25, 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. 25 - 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 26, which represents the envelope of the family of curves shown in Figure 25.

*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. 26 - OPTIMUM FIN DIMENSIONS FOR ILLUSTRATIVE CASE

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 26, 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 26 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 27 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

Fig. 27 - EFFECT OF NUMBER OF 16-COUPLE RINGS ON RTG WITH 18 HEAT SOURCE MODULES

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.

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 28. 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. 28 - 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 28. 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 29 shows the weight penalty for this kind of design trade-off.

Fig. 29 - ENVELOPE OF RTG WEIGHT-VERSUS-POWER CURVES FOR EACH NUMBER OF HEAT SOURCE MODULES

As seen, the maximum 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 30. The curve labeled 4-4 represents the master envelope of all the curves shown in Figure 29; 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.

Fig. 30 - ENVELOPES OF RTG WEIGHT-VERSUS-POWER CURVES FOR 3 AND 4 COUPLES IN PARALLEL

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 30 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 earlier in Figure 26, 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.

For each power level shown, the envelope of the two solid curves in Figure 30 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 VIII. 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.

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 88 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 VIII. 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.

Table VIII: OPTIMUM DESIGN PARAMETERS

BOL POWER, W	OPTIMUM DES.				FIN LENGTH (IN.)	FIN ROOT (MIL)	TEMP, °C			% EFFICIENCY				RTG WEIGHT, LB.	SPEC. POWER, W/LB.
	H. S. MODULES	RINGS	PARALLEL	SERIES			COLD JUNCTION	FIN ROOT	FIN TIP	TIE LEG WIDTH REL. TO HW	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	339	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	2.29	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	268	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	46	329	288	223	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	7.23	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

Figure 31 shows a comparison of one full-power RTG with two half-power RTGs. The lower curve is the envelope of the two curves shown in Figure 30; i.e., the minimum-RTG-weight envelope for the stated assumptions. The upper curve shows a similar weight envelope for the case of two identical RTGs, each producing half the power at half the voltage.

Fig. 31 - COMPARISON OF ONE FULL-POWER RTG WITH TWO HALF-POWER RTGS

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, for the case of two identical RTGs, the total number of heat source modules can only be varied in steps of two, which explains the deep scallops in the curve for two RTGs.

The weight difference shown in Figure 31 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 32).

Fig. 32 - 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.

The effect of these heat source design differences is illustrated in Figure 33. The Option B curve is the envelope of the two curves shown earlier in Figure 30. 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.

Fig. 33 - EFFECT OF HEAT SOURCE OPTION ON RTG WEIGHT, AND COMPARISON WITH MHW

Figure 33 also 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, the figure 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.

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.

Figure 34 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. 34 - EFFECT OF HOUSING AND FIN MATERIAL ON RTG WEIGHT

For the above assumptions, Figure 34 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.

V. SUMMARY AND CONCLUSIONS

A conceptual RTG design which is substantially lighter than the MHW design using the same thermocouples and insulation was generated. The RTG employs a stack of 18 heat source modules instead of a single, monolithic heat source.

A light-weight support structure for holding the heat source stack together without intermediate supports was designed. A detailed static and dynamic analysis of the RTG confirmed the structural viability of the design under predicted launch loads and thermal expansion effects.

An RTG optimization program was developed which, for the basic design topology and for a stipulated set of assumptions, ground rules, and constraints, 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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ACKNOWLEDGMENT

The authors wish to express their appreciation to T. Kline and P. Larkin of Fairchild Space and Electronics for their contributions to the above work.