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USE OF MODULAR HEAT SOURCE STACK IN RTGs

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ABSTRACT

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 illustrated with an actual RTG design for a specific mission. A detailed analytical model of the RTG and the spacecraft to which it is attached was constructed, and a NAS/RAN analysis was carried out to assess the structural performance of the heat source support scheme under differential thermal expansion and dynamic launch loads, and 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 supporting a long heat source stack without intermediate supports.

INTRODUCTION

The Department of Energy's Advanced Nuclear Systems and Projects Division is sponsoring a program at Los Alamos (LASL) to develop a radioisotope-fueled heat source module for use in space power systems. The basic module contains all provisions necessary to insure safety under all credible abort modes. The design concept under development is called the General Purpose Heat Source (GPHS), since the modules are designed to satisfy a wide variety of power conversion systems, power levels, and space missions.

Concurrently with the above heat source development program, DOE requested Fairchild Space and Electronics Company* to develop conceptual designs for a thermoelectric power system utilizing those heat source modules. This effort is described in detail in a companion paper published in these proceedings [1]. The power system, with a beginning-of-life power output of about 300 watts, is being designed for use on the joint NASA/ESA International Solar-Polar Mission (ISPM) slated for an early-1983 launch. Two spacecraft, one under NASA/JPL management and one under ESA management, are to be launched together.

In the course of carrying out the above 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 on the Multi-Hundred-Watt (MHW) Radioisotope Thermoelectric Generator (RTG). A discussion of those special problems, a proposed method of dealing with them, and a structural analysis to assess its viability, are the subjects of this paper.

DISCUSSION OF PROBLEM

Figure 1 depicts a representative design of the heat source module under development at LASL. Typically, it has overall dimensions of 3.6x3.6x2 inches, weighs about 3 lbs, and has a thermal power output of 250 watts. A detailed discussion of the module's internal components and their functions will not be presented here, since the present paper deals not with

Fig. 1 - TYPICAL HEAT SOURCE MODULE

the heat source modules themselves, but with how to integrate them into a flight-qualified RTG design.

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_dA). 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.

In a typical space power system, the radioisotope heat source would consist of a fairly large number of heat source modules. For example, the RTG which Fairchild is designing for the Solar-Polar mission 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. (The former launches the two spacecraft into parking orbit; the latter boosts them into a long elliptical orbit, past Jupiter, which helps to lift them out of the ecliptic plane into high solar latitudes.) 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.

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. In the Solar-Polar RTG, the silicon-germanium thermoelements are heated by radiation across a vacuum gap. As illustrated by the cross-sectional view of Figure 2, the thermoelectric elements do not touch the heat source stack. 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.

Fig. 2 - RTG CROSS-SECTION

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 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.

PROPOSED SOLUTION

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.

Fig. 3 - HEAT SOURCE SUPPORTS

The heat source support concept is illustrated by the exploded view in Figure 3. 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.

As seen in Figure 3, 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 4 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. 4 - SPRING SUPPORTS AND SHEAR BALLS

The components which spring-load the PG buttons are depicted in Figure 4, 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. Very 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 4, 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 in Figures 5, 6, and 7. The upper end of the heat source support structure is shown in an exploded view in Figure 5 and in an assembled cut-away view in Figure 6. Figure 7 shows a close-up of the support arrangement at one corner of the heat source stack. The multi-foil thermal insulation has been omitted in Figure 5 and earlier figures, to avoid obscuring

other components; but is included in Figures 6 and 7, which also show part of the RTG housing and radiator fins.

The end cap design depicted at the top of Figure 5 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 contained within two very stiff I-shaped cross-beams, to resist the bending moments produced by the axial spring forces.

Fig. 5 - END CAP AND H.S. SUPPORT SCHEME

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 oper-

Fig. 6 - TOP END OF ASSEMBLED RTG

ate 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 7, the desired axial preloads can be adjusted after final RTG assembly and thermal equilibration (e.g., at the launch site) 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 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.

Fig. 7 - LOAD ADJUSTMENT SCHEME FOR SEALED RTG

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 7, these forces are exerted against the rim of the end cap, at the ends of the two cross-beams.

STRUCTURAL ANALYSIS

While the structural adequacy of the above heat source support scheme under anticipated dynamic loads had been confirmed by preliminary static analysis, it was decided to verify it with a detailed dynamic analysis, using prescribed shuttle and IUS launch loads. For this purpose, a finite-element model of the RTG for NASTRAN [2] analysis was constructed. Figure 8 shows a structural model of the RTG end cap, including cross-beams and flanges. The heavy dots designated by upwards and outwards arrows represent the points at which the axial and radial spring forces are applied to the end cap; and those designated by downwards arrows show where the end cap is attached to the housing flange.

To save computer time in the subsequent analyses, a simplified model of an end cap with equivalent structural characteristics was constructed. The two

Fig. 8 - STRUCTURAL-ANALYSIS MODEL OF RTG END CAP

cross-beams are represented by bars of equivalent stiffness. The model is shown at the top and bottom of Figure 9, 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. 9 - STRUCTURAL-ANALYSIS MODEL OF RTG

The heat source stack depicted 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 10 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. 10

- TOP END OF STRUCTURAL-ANALYSIS MODEL OF ASSEMBLED RTG

Finally, Figure 11 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/spaccraft 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.

The structural evaluation of the RTG examined three critical flight conditions: the shuttle launch,

Fig. 11 - STRUCTURAL-ANALYSIS MODEL OF RTG ON SPACECRAFT

the shuttle landing (in case of an aborted flight in which the RTG is returned to earth), and the IUS 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 SIDWAYS BENDING	Z AXIAL VIBRATION
DYNAMIC INPUT:			
1G SINUSOIDAL			
FORWARD (X)	18.25G	4.41G	1.91G
SIDWAYS (Y)	4.45G	11.05G	0.25G
VERTICAL (Z)	3.67G	5.12G	12.03G
8.4 GRMS RANDOM:			
FORWARD (X)	0.29GRMS	-	-
SIDWAYS (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.

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 11. 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 was 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
	(389.9)	H.S. BENDING ABOUT DIAGONAL (S-SHAPED)

Fig. 12 - FIRST RESONANT BOWING MODE (113 HZ)

Fig. 13 - 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 12, 13, and 14.

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. Since the critical RTG resonances are out of phase with the resonances of the spacecraft and the RTG support structure, the RTG should behave as a rigid body from the viewpoint of the spacecraft.

Fig. 14 - 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 VII, 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 VII.

SUMMARY OF CRITICAL RTG STRESSES (PSI)

FLIGHT PHASE			SHUTTLE		IUS
G-LOADS:			40G	15G	14G
LATERAL LOAD			11G	27G	20G
AXIAL LOAD					
TEMPERATURE, °C:			881	881	1067
HEAT SOURCE			100	100	300
HOUSING					
SPRING LOAD, LBS:			2850	2850	1283
AXIAL SPRING			510	510	519
SIDE SPRING					
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	18,000	16,311	15,082	8,036
H. S. END	COMPRESSIVE	15,000	11,392	8,646	5,207
	SHEAR	3,000	5,649	4,604	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			0.087	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.

Finally, Figure 15 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.

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Fig. 15 - RADIAL DEFLECTIONS UNDER STATIC G-LOADS
40G LATERAL (DIAGONAL) PLUS 11G AXIAL

In summary, the detailed analysis described here has demonstrated 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.

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