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LOW COST HIGH PERFORMANCE GENERATOR TECHNOLOGY PROGRAM

LCHPG Design Study
June 1975

PREPARED FOR NUCLEAR RESEARCH AND APPLICATIONS DIVISION, ENERGY RESEARCH AND DEVELOPMENT
ADMINISTRATION UNDER ERDA CONTRACT E(49-15)-3075 (FORMERLY SNSO-3)

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**LOW COST HIGH PERFORMANCE
GENERATOR TECHNOLOGY PROGRAM**

DESIGN STUDY

VOLUME 2

TES-SNSO-3-25

June 1975

**Prepared for
U. S. Energy Research and
Development Administration
Under Contract SNSO-3**

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FOREWORD

This document was prepared for the United States Energy Research and Development Administration (Division of Advanced Nuclear Energy Systems) by Teledyne Energy Systems under Contract SNSO-3.* This is Volume 2, "Low Cost High Performance Generator System Design Study," which describes the design studies performed during the period January 1975 through June 1975.

The following organizations contributed data for heat source and thermoelectric design.

Minnesota Mining and Manufacturing Company

Los Alamos Scientific Laboratory

Oak Ridge National Laboratory

*Contract Number Revised to E(49-15)-3075

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I. ABSTRACT

This report documents the systems studies directed towards up-rating the performance of an RTG using selenide thermoelectrics and a heat source with improved safety. The resulting generator design, designated LCHPG, exhibits conversion efficiency of greater than 10%, a specific power of 3 watts/lb., and a cost of \$6,000/W(e). In the course of system analyses, the significant development activities required to achieve this performance by the 1980 time period are identified.

Previous LCHPG studies were constrained by temperature and envelope configuration limits as well as data-limited analysis techniques. By updating the analysis methods and allowing some additional flexibility of operating conditions, an improved design is obtained. Specifically, the operational hot junction temperature is explored to 1000°C leading to selection of a 900°C design point in the recommended design. With the cold junction temperature reduced to 135°C and the use of an improved Fiberfrax insulation, an overall conversion efficiency of 10.1% is predicted for the generator using current standard selenide thermoelectrics.

The additional performance advantages obtained by applying multi-foil insulation to the evacuated thermopile and, further, the use of improved thermoelectrics material properties are evaluated also. Multi-foil shown to enable about a 1% efficiency increase (to about 11%) while the improved selenides can yield an additional 3% efficiency increment and specific power approaching 4 watts/lb.

A heat source configuration with about 30% less weight for a given safety characteristic is obtained by incorporating additional test data and improving the impact response model. Application of MHW derived impact test data to a model which contains better distribution of impact energy among the containment and fuel members yielded a higher

failure velocity than previous estimates. The newer calculations predicted a reduction in clad thickness from earlier LCHPG capsule designs to maintain adequate impact response characteristics and yield a weight reduction. In addition, the effects of other accident environments on the selection of an impact safety index are discussed.

The development efforts required to achieve the predicted performance of the LCHPG are apparent in the discussion of results. A detailed description of the development program can be found in Volume 1, Program Plan, TES-SNSO-3-24.

II. INTRODUCTION

The systems design study presented herein represents the status of work performed on the LCHPG program to date. This present study has been broadened in scope to enable achievement of RTG designs yielding ERDA/SNS stated goals of over 11% efficiency and over 3 watts(e)/pound with the use of currently available selenide thermoelectrics. Most significantly, temperature constraints have been relaxed in this study and more innovative design has been introduced, particularly in the cold end hardware and isolation hot frame areas.

More specifically, this study was directed to the design and performance/safety assessment of a 500 watt PuO_2 fueled selenide RTG with respect to the following variables:

1. Hot junction temperature (800°C - 1000°C).
2. Cold junction temperature (125°C - 175°C).
3. Type of thermal insulation (Min-K, Fiberfrax, Moly-Astroquartz foil).
4. Converter cover gas (vacuum, xenon).
5. Thermoelement spacing and spring pressure.
6. Fuel form (PPO and shards).
7. Reentry insulator (ZrO_2 , pyrolytic graphite).
8. Capsule wall thickness.
9. Heat shield wall thickness.
10. Heat source volume (up to 1500 in.^3).
11. Fuel pellet L/D.
12. Capsule arrangement within each heat shield.
13. Number of heat source modules.

Additionally, further study was given to the heat source design technique which strongly imposes safety on the RTG conceptual design process. Improvements in the impact model were generated that include the direct use of experimental data from the MHW program, the energy absorption contribution of the fuel, a PuO_2 source term analysis, the effects of four different impact environments (two launch pad and two post-reentry), and a revised and broadened definition of a nuclear safety index. Improvement was also introduced into the aerothermal analysis through the use of a three-dimensional model. Work was also initialized in the important area of reentry thermal stress. This activity is significant in that, heat source safety is being considered early as an integral part of the RTG design effort.

Additional work remains to be done before the design synthesis technique is complete. Not only are there other environments that require investigation such as the various fires and the subject of capsule recovery following reentry, but there is the subject of safety criteria that is evolving and requires further definition.

The following sections of this report discuss the parametric study performed on the 500 watt RTG, heat source design technique and application, the reference RTG as recommended for further development and design, and recommendations for further work, in that order.

III. GENERATOR DESIGN EVALUATION

The scope of the LCHPG design study is presented in Figure III-1. Because of the emphasis on high efficiency (11%), systems using multi-foil insulation with consequent very low heat loss were investigated in addition to the more conventional fibrous insulation approach. The other study parameters affecting efficiency, as well as generator size and weight, are operating temperature range, and converter gas fill. The improved selenide thermoelectric materials were added in the course of these studies since the achievement of 11% efficiency was found to be marginal and it was desired to show growth potential for post 1980 applications. Other design variable studies which affect weight and size are the heat rejection surface configuration and the thermoelectric spring pressure. The radiation surface was examined for two and four finned arrays and a lower thermoelectric spring pressure of 60 psi for each leg was evaluated in addition to the 150/300 psi current recommended values.

A. CONCEPT DEFINITIONS

1. General Design Description

As stated previously, two classes of RTG systems were evaluated; one utilizing fibrous insulation, the other a multi-foil insulation. Each was studied with current standard and improved selenide thermoelectrics in conventional spaced element arrays. For improvement in thermal efficiency, a close-packed thermoelectric array of current standard thermoelectrics was also evaluated with the multi-foil insulation. The comparison of alternate design features was based on a 500 watt(e) RTG.

All systems were designed around a rectangular cross section heat source, described in Section IV. Between the heat source and the converter section is an isola-

FIGURE III-1. 500 WATT GENERATOR STUDY*

*See Table IV-6 for scope of heat source study.

tion hot frame (IHF) which is discussed in further detail in the next two subsections. The thermoelectric modules are positioned on two opposite sides of the IHF, (see Appendix A for more detail) the remaining two sides containing thickened insulation for maximizing thermal efficiency. Light weight flexible sleeve cold end hardware is used to supply the required load to the thermoelectrics, while providing a low thermal resistance for waste heat rejection. Heat is rejected from the housing and the four mutually perpendicular fins that are planar extensions of the rectangular housing sides.

To enhance heat transfer to the fins, heat pipes are attached to the external housing surface at each of the two sides where the thermoelectrics are located and outward along the two adjacent fins to the fin tips. The heat pipes are soldered into saddles integrally machined in the housing and fins.

Each fin is dynamically stiffened with ribs which extend across a housing side and out to the tip of the fin. Static rib doublers are located adjacent to each fin in the housing region for the purpose of reacting spring loads and the loads resulting from the internal gas pressure. After the radioisotopic heat source is installed, the ends of the housing structure are seal welded to the end covers. External surfaces of the generator are coated to enhance radiative heat transfer.

Other features of the housing include power feedthrough and gas venting provisions. A hermetically sealed electrical receptacle, used to carry the generated power through the sealed generator housing, is installed near one end of the RTG. A controlled leak device is installed in one end cover. The purpose of this vent is to prevent excessive fuel decay helium gas pressure buildup in the heat source cavity. Appendix A presents a more detailed description of the housing, fins, ribs and heat pipes.

2. Fibrous Insulation Concepts

The fibrous insulation design concepts (Figure III-2) employ the load bearing

- LEGEND
- 1 HEAT SOURCE MIX JUE
 - 2 ISOLATION HOT FRAME
 - 3 THERMOELECTRIC MODULE
 - 4 COLD END MOUNTING
 - 5 FIN MOUNTING TIE TRUCK TIME
 - 6 MOUNTING LUGS
 - 7 ELECTRICAL FEEDTHRU
 - 8 FUELING, COMPLIANT SEAL BELLOW
 - 9 THERMAL INSULATION BULK TYPE
 - 10 FUELING LOAD RING
 - 11 END COVER
 - 12 END COVER FUELING END
 - 13 FILTER VENT (TBD)
 - 14 BI METAL SEAL RING

FIGURE III-2

SECTION A-A

CURRENT STANDARD SELENIDES FIBROUS CONVERTER INSULATION
 $T_{\text{hot}} = 900^{\circ}\text{C}$, $T_{\text{cold}} = 150^{\circ}\text{C}$, $T_{\text{rad}} = 120^{\circ}\text{C}$, $P = 500 \text{ watts(e)}$

characteristics of such insulations. This permits the use of a unique isolation hot frame, the membrane type IHF, constructed of thin sheet metal. It is joined at its ends to the outer housing structure through compliant bellows that are weld sealed to the IHF and to a bi-metal seal ring welded in the housing ends. The membrane IHF is not structurally self-supporting, but relies upon being sandwiched between the heat source from within and insulation from without.

The membrane IHF is assembled around a graphite block or an electrical heater that is dimensionally equivalent to the heat source stack. This block remains in the heat source cavity at all times during subsequent operations. To fuel the generator, the block is pressed out of one end of the generator by the heat source which is pressed in from the opposite end.

The IHF walls are under an inward directed pressure from the thermoelectric module cold end spring loads which are supported by the heat source cavity contents. This same load causes a frictional resistance to the insertion and removal of heat source blocks. The fueling forces are accommodated in the fueling tool by compression in a pneumatic actuator and tension in the wall structure and bolts which attach the fueling tool to the RTG.

The ends of the heat source compartment, inside the IHF bellows, between the housing end covers and the end heat source modules are fitted with load bearing thermal insulation. These insulation plugs are made oversize in thickness and compressed by the housing end covers when they are installed. This technique is used to structurally support the heat source, which is otherwise unrestrained in the longitudinal direction within the generator housing. The cold end hardware springs and thermal insulation support the heat source in the lateral directions.

3. Multi-foil Insulation Concepts

Since most of the concepts were studied with a vacuum in the converter, the highly efficient multi-foil type insulation was applicable. However, since multi-foil insulation is not capable of supporting significant load, a self-supporting IHF structure is necessary. Additionally, supporting structure must be included within the insulation between the IHF and the housing on the non-module sides to prevent movement of the IHF during dynamic environments. In the module area, multi-foil insulation is electrically insulated from the thermoelectrics. Figure III-3 presents a multi-foil insulation design concept.

The IHF structure is a square section box with one inch thick sides, a membrane sheet metal bottom, a sheet metal transition bulkhead to the bellows and a bellows with bi-metal transition ring for the housing seal attachment at the other end. The one inch thick side walls are a composite structure composed of an inner skin, outer skin and a separating corrugation. The outer skin and corrugation are TZM sheet with the corrugation convolutes oriented across the side width. The inner skin is molybdenum sheet and is the pressure wall of the structure with all junctures to it sealed. Figure A-1 in Appendix A shows a section of this IHF. Each inner face of this IHF is fitted with two longitudinal strips of platinum rhodium. These strips, approximately .020 in. thick and .25 in. wide, are spacers to prevent the graphite heat shields from being in direct contact with the molybdenum inner faces, thus avoiding carbon embrittlement of these surfaces and consequent loss of the gas seal integrity.

In order that the IHF with its contained heat source modules be adequately supported dynamically within the housing, half inch long small spool shaped plugs are located on the non-module sides and extend through holes in the multi-foil. There are sixteen plugs per side, made of ZrO_2 ceramic that fit between the IHF outer face and the housing inner face. The two thermoelectric module sides of the IHF are restrained by the thermo-

LEGEND

- 1 HEAT SOURCE MODULE
- 2 ISOLATION HOT FRAME
- 3 COLD END HANDBRAKE
- 4 HERMIELECTRIC MODULE
- 5 FIN HOUSING
- 6 END COVER
- 7 END GILD FILM NL
- 8 FILLING OMI LIANT CAL BELOW
- 9 MULTIFOIL THERMAL INSULATION
- 10 LOAD BEARING THERMAL INSULATION
- 11 EN METAL SEAL PIN
- 12 SIDE LOAD SUPPORTS
- 13 MOUNTING LUGS
- 14 ELECTRICAL FEEDTHROUGH
- 15 HEAT SOURCE VENT (THT)
- 16 SPACER STRIPS (R_h)

electric module spring loads, and the ends by the end load bearing fibrous insulation as in the fibrous insulation concepts.

One half inch thickness of multi-foil insulation is used on the two non-thermoelectric module sides between the IHF and the housing. On the thermoelectric module sides, outside the edges of the modules, wire helix retainers are placed between the outer insulation surfaces and the housing surface to keep the insulation in place against the IHF. This is necessary because the thermoelectric module plus cold end hardware thickness is much greater than the one-half inch multi-foil thickness and filling, this excess space with multi-foil insulation would incur a significant weight penalty.

Fueling is a simpler matter with this design concept. It is fueled and de-fueled from one end and does not require the pressing operation as does the fibrous insulation concept.

The housing and fin designs, as described earlier, maintain the same arrangement, except for dimensions, as does the fibrous insulation concept. Multi-foil designs have larger housing cross section dimensions but smaller axial dimensions due to smaller fuel inventory requirements.

4. Rectangular RTG Selection

The shape of the RTG housing is strongly dependent on the heat source configuration, and since the rectangular cross section heat source exhibits desirably high drag the rectangular housing naturally follows. Additionally, this arrangement permits the use of thicker insulation on the two non-thermoelectric module sides and it allows bearing of the simple planar thermoelectric element array against the cold end hardware without a transition section (i.e. module bar). Also, insulation of the heat source near the square edges is accomplished without structural complexity.

Other significant reasons for using planar surfaces relate to lower fabrication

and assembly costs for the generator internal components as well as ease of fabrication and attachment of heat pipes to the housing.

Although it may be argued that a cylindrical shape can be used to good advantage, the reasons for a rectangular housing are listed in Table III-1.

TABLE III-1
BASIS FOR RECTANGULAR HOUSING

- Maximum compatibility with square heat source - no transition pieces or extra corner losses.
- Application of thicker insulation on non-module locations for increased RTG efficiency.
- Good geometric compatibility with cold end hardware - no transition pieces.
- Easily fabricated, minimum weight.
- Thin wall easier to achieve with planar surface.
- Low cost machining of housing/fins.
- Enhanced ability to achieve low resistance heat pipe bond.
- Easily fabricated planar assembly tooling.
- No housing penetrations or internal auxiliary RTG structure required to aid assembly - only two longitudinal welds needed at assembly.

Thus, despite former exclusive experience with cylindrical configurations, the recommended approach is rectangular, a concept which will enable simple fabrication and which will satisfy the desirable features of modularity as well.

B. PERFORMANCE CHARACTERISTICS: FIBROUS INSULATION CONCEPTS

To determine performance characteristics of fibrous insulation concepts, several arrangements were studied. First, two insulation types were evaluated: Min-K TE 1400 and Fiberfrax HiFi 660-H. The specific attributes of these insulations are discussed in Appendix A-3. For the Min-K system, the fuel cavity volume was varied between 530 in.³ and 1500 in.³ to determine the performance/weight/safety relationship. The Fiberfrax was investigated for the 800 in.³ cavity for comparison. Other sensitivity studies included use of an inert cover gas in the thermopile and reduction of cold end hardware spring pressure. To show system improvement with advanced technology thermoelectric arrangements, designs employing the improved selenide thermoelectrics were created. Housing variations included a two-fin RTG concept for comparison with the basic four-fin designs.

To provide a common basis for comparison, the performance studies of this section were at fixed hot and cold junction temperatures of 900°C and 150°C, respectively. The IHF was taken to operate 30°C higher than the hot junction (Ref. III-1). Based on separate thermal evaluation of the flexible sleeve, the housing/radiator was held at temperatures 30°C cooler than that of the cold junction. In accord with use of the Z-93 coating (zinc oxide base), the radiator was sized to reject heat to a 350°R (195°K) sink.

Total weight, thermal inventory, thermal efficiency, system efficiency, and specific power results are presented in Table III-2. It is seen that a specific power of as high as 3.6 W/lb was obtained for the 530 in.³ cavity* while the largest cavity

*See Table IV-6 and Section IV-D for further information on heat source characteristics.

TABLE III-2
FIBROUS INSULATION CONCEPTS PERFORMANCE SUMMARY

$$T_H = 900^{\circ}\text{C}, \quad T_C = 150^{\circ}\text{C}$$

<u>Insulation</u> ⁽¹⁾		T/E	Fuel Cavity Volume (in. ³)	<u>Heat Losses (watts)</u>					<u>Envelope</u> ⁽²⁾ <u>Dimensions</u>		Total ⁽²⁾ Weight (lbs)	Specific Power (w/lb)	System Efficiency (%)
Module	Non-Module			Module	Around Modules	Blank Sides	Ends	Bellows	Height (in.)	Span (in.)			
Min-K	Min-K	C. S.	530	353	67	145	30	87	45.5	55.0	138	3.6	10.4
Min-K	Min-K	C. S.	800	365	95	169	40	112	45.5	56.8	161	3.1	10.2
Min-K	Min-K	C. S.	1000	353	151	179	45	120	46.0	56.9	178	2.8	10.0
Min-K	Min-K	C. S.	1500	353	214	218	65	144	47.7	56.3	221	2.3	9.8
Fiberfrax	Fiberfrax	C. S.	800	462	119	209	45	112	45.5	58.3	160	3.1	9.8
Fiberfrax ⁽³⁾	Fiberfrax	C. S.	800	547	140	247	45	112	45.5	60.1	166	3.0	9.6
Min-K	Min-K	I	618	302	70	169	45	113	36.5	54.7	127	3.9	12.9
Min-K ⁽⁴⁾	Min-K	C. S.	800	365	95	169	40	112	45.5	54.0	154	3.3	10.2
Min-K ⁽⁵⁾	Min-K	C. S.	800	365	95	169	40	112	45.5	87.6	163	3.1	10.2

-
- (1) Min-K end plugs used on all designs
 (2) See Appendix A for component weights and dimensions
 (3) Xenon gas in converter
 (4) Thermopile spring pressure = 60 psi/element
 (5) Two-fin RTG

yielded 2.3 W/lb. The respective system efficiencies are 10.4% and 9.8%. The use of Fiberfrax insulation (with vacuum in the converter) for the 800 in.³ cavity lowers efficiency by about 0.4% (9.8% system efficiency as compared to 10.2% for Min-K) and necessitates a small increase in heat rejection area.

Fiberfrax insulation with xenon gas fill was evaluated since a gas may be needed in the converter to suppress sublimation of the selenide thermoelectrics at the higher operating temperatures. With xenon, system efficiency drops from 9.8% to 9.6%, and RTG weight and span both increase slightly because of greater thermal losses.

Reducing cold end hardware spring pressure from 300/150 psi to 60/60 psi (N-element/P-element) effects a 4% savings in total RTG weight and an increase in specific power from 3.1 to 3.3 W/lb. This improvement results from a reduction in radiator rib weight. The 60/60 psi spring pressure is representative of Pioneer/Viking pressures and may be applicable for a selenide system. The results are given here to indicate the influence of this parameter.

Use of improved selenide thermoelectrics with their higher thermoelectric efficiency (see Appendix A-2) yields a significant improvement in system performance as compared to the current standard designs; specific power increases from 3.1 to 3.9 W/lb., system efficiency increases from 10.2 to 12.9% and the housing envelope drops from 45 in. x 58 in. to 36 in. x 55 in.

The two-fin RTG is similar to the four-fin RTG in weight and system efficiency, but fin span increases 54%, from 56.8 in. to 87.6 in. with only two fins. This large span would not generally be recommended unless required by a particular spacecraft envelope constraint.

The fibrous insulation, current standard design (800 in.³) was studied with shard fuel form also. It was found that shards could be packaged in the same heat source module

system, with the weight and performance remaining the same as with the PPO fuel. Nuclear safety implications of shards fuel form are discussed in Section IV-D.

C. PERFORMANCE CHARACTERISTICS - MULTIFOIL INSULATION CONCEPTS

The molybdenum-astroquartz multifoil insulation concepts using current standard thermoelectrics were evaluated for a single heat source cavity volume. Since less fuel inventory is required because of higher system efficiency, a 727 in.³ fuel cavity volume (20 heat source modules compared to 22 modules for the fibrous concept) was used. Where the multifoil insulation was used within the thermoelectric module, each thermoelectric element was considered to be wrapped with 20 mil thick Fiberfrax Hi-Fi 660-H insulation to electrically insulate the elements from the metal foils.

The performance results presented in Table III-3 and Figure III-6 show a significant increase in system efficiency with the use of the multifoil insulation, about 1% absolute efficiency. A small increase in weight was the result of the larger IHF which is required to support thermopile spring loads and heat source cavity gas pressure. The principal improvement is realized in the module where with only 0.3 inch thick insulation, the highly efficient multifoil insulation (Table III-3) reduces module losses by about 370 watts within the thermoelectric modules alone. Overall, the multifoil system requires 590 watts less thermal inventory than the all fibrous (Fiberfrax) concept.

The advantage of multifoil insulation in the module alone is illustrated by comparison with the case where Fiberfrax insulation was used in the module and multifoil used elsewhere. Here the system efficiency was reduced to nearly that of the fibrous system.

The close-packed thermoelectric array concept, or eggcrate design, represents a considerable departure from the conventional design concepts. This thermoelectric module arrangement is so compact that it covers but a small fraction of the available

TABLE III-3
MULTI-FOIL CONCEPTS PERFORMANCE SUMMARY
 $T_H = 900^\circ\text{C}$, $T_C = 150^\circ\text{C}$

Insulation ⁽¹⁾		Heat Source Cavity Volume (in. ³)	Heat Losses (watts) ⁽⁴⁾					Envelope Dimensions		Total ⁽²⁾ Weight (lbs.)	Specific Power (w/lb)	System Efficiency (%)
Module	Non-module		Module	Around Modules	Blank Side	Ends	Bellows	Height (in.)	Span (in.)			
Mutli-foil	Multi-foil (current standard)	727	89	34	132	47	56	42.3	53.7	169	3.0	11.1
Fibrous ⁽³⁾ Spacers	Multi-foil (current standard)	727	19	65	125	43	56	42.3	50.4	148	3.4	11.3
Fiberfrax	Multi-foil (current standard)	727	462	34	132	47	56	42.3	57.8	170	2.9	10.3
Multi-foil	Multi-foil (improved)	559	73	21	109	46	56	33.1	50.9	131	3.8	14.3

⁽¹⁾ Min-K end plugs used on all designs.

⁽²⁾ See Appendix A for component weights.

⁽³⁾ Thermoelectric eggcrate design.

⁽⁴⁾ See Ref. III-1

area on the IHF (see Figure III-4). System efficiency and specific power are 11.3% and 3.4 W/lb, respectively. It must be mentioned, though, that a cold end hardware design employing 60 psi spring pressure and the use of 0.2 inch thermoelements contributed to the weight savings of 21 lbs over that of the multi-foil design with conventional modules. For these benefits, the eggcrate design would require an ambitious development program.

To show additional system potential, a multi-foil system was evaluated with improved selenide properties. An efficiency of 14.3% is realized with 3.8 W/lb specific power. This performance encourages the continued development of selenide systems.

As with the fibrous concepts, use of shards fuel form does not change the system efficiency and weight characteristics. However, the self supporting isolation hot frame design causes heat source temperatures of approximately 40 °C higher than those in the fibrous designs (Ref. III-6).

D. OPERATIONAL TEMPERATURE STUDIES AND PERFORMANCE COMPARISON FOR FIBROUS AND MULTI-FOIL RTG SYSTEMS

The system, thermal and thermoelectric efficiencies for the reference fibrous and multi-foil concepts are presented in Figures III-5 and III-6 as a function of hot and cold junction temperature. The incorporation of weight and heat source capsule temperature yields the performance summaries of Figures III-7 and III-8. Inspection of these results indicates that the fibrous design offers higher specific power and lower heat source capsule temperatures while the multi-foil design offers higher system efficiency. Specifically, over the range of hot junction/cold junction temperatures considered, specific power, system efficiency and heat source capsule temperatures are as follows:

LEGEND

- 1 HEAT SOURCE MODULE
- 2 ISOLATION HOT FRAME STRUCTURE
- 3 COLD END HARDWARE
- 4 THERMOELECTRIC MODULE
- 5 FIN-HOUSING SIDE STRUCTURE
- 6 END COVER
- 7 END COVER FUELING END
- 8 FUELING COMPLIANT SEAL BELLON'S
- 9 MULTIFOIL THERMAL INSULATION
- 10 LOAD BEARING THERMAL INSULATION
- 11 BI METAL SEAL
- 12 SIDE LOAD SUPPORTS
- 13 MOUNTING LUGS
- 14 ELECTRICAL FEEDTHROUGH
- 15 HEAT SOURCE VENT
- 16 SPACER STRIPS (P.P.H.)

FIGURE III-4

SECTION B-B

CURRENT STANDARD SELENIDES (CLOSE PACKED) MULTIFOIL CONVERTER INSULATION

 $T_{\text{hot}} = 900^{\circ}\text{C}$, $T_{\text{cold}} = 150^{\circ}\text{C}$, $T_{\text{rad}} = 120^{\circ}\text{C}$, $P = 500 \text{ watts(e)}$

FIGURE III - 5
FIBROUS CONCEPT EFFICIENCY
MIN-K, CURRENT STANDARD SELENIDES

FIGURE III-6
MULTIFOIL CONCEPT EFFICIENCY
MULTIFOIL INSULATION AND CURRENT STANDARD SELENIDES

0.3 IN. MULTIFOIL INSULATION IN VACUUM IN T/E MODULES
 0.5 IN. MULTIFOIL INSULATION IN VACUUM ON SIDES
 2.0 IN. MIN-K (HELIUM) ON FUELING END
 1.5 IN. MIN-K IN VACUUM ON NON FUELING END
 FUEL CAVITY DIMENSIONS: 4.5" X 1.5" X 35.9"
 ISOLATION HOT FRAME 1.0 IN. THICK CONSTRUCTION
 MULTIFOIL TYPE: MOLYBDENUM/ASTROQUARTZ @ 100 FOILS/INCH

FIGURE III-7
PERFORMANCE SUMMARY
FIBROUS INSULATION-CURRENT STD. SELENIDES
800 IN³ HEAT SOURCE CAVITY VOLUME / PPO FUEL
500 W (e) LCHPG

FIGURE III - 8
PERFORMANCE SUMMARY
MULTIFOIL - CURRENT STD. SELENIDES
727 IN³ HEAT SOURCE CAVITY VOLUME/PP0 FUEL
500 W(e) LCHPG

	<u>Fibrous Design</u>	<u>Multi-foil Design</u>
Specific power (W/lb)	2.7 - 3.3	2.6 - 3.17
System efficiency (%)	9.1 - 11.1	9.7 - 12.4
Heat source capsule temp. (°C)	925 - 1125	980 - 1160

It is evident that the multi-foil insulation offers the potential of about 1% higher efficiency, but about 40°C higher heat source capsule temperature because of the self supporting double skin and corrugation isolation hot frame.

Despite the potential efficiency advantage of the multi-foil system, it is recommended that the fibrous design be developed since:

- a. It has a higher success probability of achieving the predicted system efficiency.
- b. It is less expensive to develop.
- c. An earlier completion date for an ETG is possible.
- d. It offers lower heat source capsule operating temperatures.

Further, the results of Figure III-7 suggest that the recommended configuration be designed to operate at a hot junction temperature of 900°C and a cold junction temperature of 135°C. This design point allows generator performance of better than 10% efficiency and 3 watts/lb while maintaining steady state heat source capsule temperature at a reasonable value (<1050°C).

IV. HEAT SOURCE DESIGN

A. INTRODUCTION

The objective of this study was the formulation of an RTG design which exhibits adequate nuclear safety while satisfying weight and efficiency goals. A practical basis for the assessment of system safety was provided by the Heat Source Safety Design Specification (HSSDS - Ref. IV-1) which implies an overall requirement in the form

$$PQ = \text{constant}$$

Here P is the net probability of releasing radioactive material to the atmosphere and Q is the average number of curies released as an airborne source term. (Under the assumption of linear injury response, the expected number of injuries is directly proportional to the airborne source Q.) According to this criterion, accident events which are likely to produce a large source must have correspondingly smaller occurrence probability. To achieve a satisfactory system design, a systematic design procedure is implied, that of (a) performing an initial assessment of accident environments to determine a relative ranking according to their product PQ, and (b) proportioning design effort accordingly.

Quantative specifications of probabilities and source terms are generally subject to large uncertainties (factors which approach three or more), and significance would not be attached to differences much smaller than an order of magnitude. Therefore, it is appropriate to use a logarithmic risk factor, $R = \log_{10} PQ$, rather than PQ directly. Since safety is the antithesis of risk, we define a safety index, S, as

$$S = -\log_{10} PQ.$$

An increase in S by one unit reflects a system which is safer by an order of magnitude.

Prior analysis has resulted in a classification of accidents according to a primary or secondary rating based on their relative PQ values. As currently estimated, primary environments are those which determine the major features of the heat source design. A design which meets the dictates of the primary environments would not be significantly altered by considering secondary environments. Table IV-1 lists the accident environments according to the primary-secondary classification. For more detailed environment discussions see Ref. IV-2, IV-3, and IV-4.

Heat source design selection during the current study was based mainly on the ability to survive impact. Both launch pad and reentry events were considered. Prior to impact consideration, a candidate design must have first demonstrated the ability to survive the reentry thermal environment. For the launch pad accident, the use of high strength graphites (3-D) must provide thermal protection to the fuel capsule from the solid propellant fire environment after impact.

As used in this chapter, the term "heat source" refers specifically to the unit heat source or heat source module, several of which may occupy the RTG heat source cavity. This unit is the reentering or impacting body. Although the RTG housing, thermoelectrics, and insulation may provide some degree of protection for specific environments, safety assessment during this study is based on the performance of the heat source alone.

The heat source consists of four major components as listed in Table IV-2. Their principal functions are concerned with limiting the potential airborne release of radioactive material.

The fuel forms are PPO, the pressed sintered form and shards, a potentially less costly form.

The noble metal alloy encapsulation material (clad), Pt-3008, provides reasonable

TABLE IV-1
ACCIDENT ENVIRONMENTS

A. Launch Complex

Primary

Impact

Solid propellant fire

Secondary

Blast overpressure

Fireball

Fragments

Residual liquid fire

B. In-flight

Primary

Atmospheric reentry:

Thermal stress

Critical temperatures

Graphite ablation (recession)

Impact

Post impact integrity, recovery

TABLE IV-2
MAJOR HEAT SOURCE MODULE COMPONENTS

<u>Component</u>	<u>Design Function</u>	<u>Reference Material</u>
1. Fuel/Fuel Form	<ul style="list-style-type: none"> • Minimize fines production (impact) and fuel vapor (thermal environment). 	PPO, shards
2. Clad	<ul style="list-style-type: none"> • Impact containment (energy absorption). • Post impact containment (oxidation and integrity after impact). 	Pt-3008
3. Insulation	<ul style="list-style-type: none"> • Limit reentry temperatures. • Limit pre-impact cooling. 	Pyrolytic graphite
4. Heat Shield	<ul style="list-style-type: none"> • Survive launch pad blast, impact. * • Provide thermal protection for launch pad, reentry. • Limit ablation. • Absorb impact energy. 	3-D graphite or equivalent

*Blast protection also offered by basic LCHPG housing, thermoelectric materials and insulations and the Isolation Hot Frame.

strength and good oxidation resistance. Clad temperatures for small (~ 25 watt thermal) fuel capsules in air have been computed to be in the 400-600 °C range (Ref. IV-5). A buried capsule may result in higher clad temperatures but in no case will the temperature exceed 950 °C. Calculated metal loss for similar Pt-Rh-W alloys based on measured corrosion rates at 1000 °C are less than 2×10^{-4} inch/year in air. (Ref. IV-6).

The insulator serves to moderate the effects of severe changing thermal environments. Three candidate materials ZrO_2 (solid), ZrO_2 foam, and pyrolytic graphite, were considered. The solid ZrO_2 and ZrO_2 foam designs provide the required thermal protection, but their use entails a severe weight penalty or lack of structural strength.

A specific heat shield graphite has not been identified yet. The term 3-D is intended to apply to graphite composed of an ordered pattern of graphite fibers impregnated with graphite and/or structures of the graphanol type. Strength and relatively high density (~ 1.6 g/cc or greater) are important requirements.

Succeeding sections of this chapter provide a description of the heat source design techniques and results of the current study. Section B describes the design synthesis, evaluation and selection technique and includes a definition of the safety index. Section C describes the application of the technique and presents results which show the influence of the design variables. Specific heat source designs and performance characteristics are found in Section D. The chapter concludes with a discussion of the significance of the results.

B. DESIGN TECHNIQUE

This section presents the technique used for heat source design synthesis, evaluation and selection. The major elements of the process are shown in Figure IV-1.

FIGURE IV-1. SELECTION TECHNIQUE

1. Design Constraints and Guidelines

Candidate heat source designs were subjected to various constraints and selection guidelines based on materials' properties and analysis results. These constraints are summarized in Table IV-3.

TABLE IV-3
DESIGN CONSTRAINTS AND GUIDELINES

1. Fuel

PPO - Fuel Pellet $L/D > 0.4$, ~ 25 watts/fuel pellet
and shards or PuO_2 granules

2. Clad Material (Pt-3008)

Pt - Graphite Eutectic (1840°C)

3. Insulator

Pyrolytic Graphite (0.1 inch thick) with gaps

4. Heat Shield

3-D Graphite

Minimum of 0.3 inch exterior graphite

Minimum of 0.06 inch between capsule cavities

Low graphite recession

Although fuel capsule length to diameter ratios (L/D) and thermal inventory are design variables, guidelines have been provided by LASL with respect to PPO pellets. Specifically, individual fuel pellets are to have an L/D of at least 0.4 and are restricted to small thermal inventories - in the neighborhood of 25 watts/pellet.

The fuel clad material, Pt-3008, experiences a graphite-platinum eutectic at

approximately 1840°C. Although partial clad melt during a severe thermal environment such as prompt reentry would probably not result in fuel release at altitude, the capsule integrity may be compromised to the point that failure on impact is highly probable. Excessive ablative graphite loss could also lead to clad melt.

Another temperature related characteristic of the clad is grain growth, known to have an adverse effect on capsule survivability. Although presently under study at LASL and ORNL, specific grain growth data for Pt-3008 was not available for the current study.

Thermal analysis (Ref. IV-7) has demonstrated the need for a thermal insulation. For a high velocity, shallow prompt reentry flight path angle, the low ballistic (15 lb/ft²) designs exhibited maximum clad temperatures of approximately 1600°C and 2200°C with and without the insulator, respectively. The uninsulated condition exceeds the graphite-platinum eutectic or Pt-3008 melt temperature of 2000°C. Hence, the combination of 0.1 inch pyrolytic graphite with gaps which completely surround the fuel capsule has been included in all designs analyzed. In addition to its suitable thermal properties, pyrolytic graphite was chosen for its good weight and strength properties.

Thermal stress analysis suggested the selection of a minimum thickness of 0.3 inch exterior graphite (minimum dimensions from exterior of pyrolytic graphite to exterior surface of block). This is based on comparative studies of stress as a function of shield thickness for a selected severe reentry environment. See Appendix B-1 for details.

Heat shield manufacturing tolerances requires a minimum thickness radially between capsule cavities (space containing capsules and pyrolytic insulator). The minimum is estimated to be 60 mils for 3-D graphite.

2. Design Variables

Design parameters which were varied during the synthesis process are as follows:

Fuel capsule thermal inventory

Fuel pellet L/D

Clad thickness

Exterior graphite thickness

Heat source cavity volume, dimensions.

The first four parameters have a direct relationship on the impact safety index (as defined below). The heat source cavity volume has an indirect effect in that it affects design freedom, i. e. the number and variety of module designs.

Current studies were limited to a heat source cavity length of about 40 inches. This selection was based on preliminary RTG configurational-weight studies. This length provides overall RTG dimensions which are consistent with objectives of minimum weight and reasonable compactness.

3. Heat Source Module Synthesis

A major tool employed in the heat source synthesis process was the computer program SAFE-HS (Systematic Approach For Evaluating Heat Sources), described in Figure IV-2. This program provides a systematic algorithm for generation and evaluation of the many possible module designs within the specified input heat source cavity and subject to a particular set of the other design variables. For further information regarding the module "construction" algorithm see Ref. IV-8.

For a given input, (cavity dimensions, exterior thermal shield thickness, gap dimensions, clad thickness, fuel pellet dimensions and thermal inventory), the synthesis technique generates designs which can be physically accommodated within the cavity volume. The program orients the capsule axes parallel to or perpendicular to

FIGURE IV-2. SAFE-HS PROGRAM

the long axis of the cavity. Having generated a particular module design, the program then evaluates its weight, critical reentry characteristics and impact safety index (defined below).

A given run generally produces a large number of designs which may vary from a single heat source module containing all of the fuel capsules to the other extreme of a single capsule in each of many modules. Excluded from the output are those designs which exhibit a safety index less than the minimum specified.

4. The Safety Index

A safety index for the impact event was formulated consistent with the PQ criterion definition presented in Section A. The index formulation is an integral part of the SAFE-HS program and is one of the factors used to evaluate each module concept produced by the synthesis program. The elements of this index are summarized in Table IV-4. A detailed description with equations is provided in Ref. IV-9.

The safety index considers all "hard" targets for land impact. The targets, environments and associated accident probabilities are identified in Table IV-5. Impact velocity calculations use a subsonic drag coefficient of 1.2 (referenced to the heat source broad face area) as determined from drop test data for rectangular parallelepiped and related geometries (see Ref. IV-2). A relatively large standard deviation is assigned to failure velocity, ($\sigma_f = 0.2 \bar{v}_f$), because of uncertainties of the applicability of the experimental FSA data to the LCHPG design.

Clad and graphite stress-strain properties, target failure velocity factors, accident probabilities, source term tables and adjustment factors are provided to the SAFE-HS program as input. Having determined the physical aspects of each heat source design and an impact temperature (see 5 below), a safety index for the design is calculated.

TABLE IV-4

IMPACT SAFETY INDEX FORMULATION

A. The Safety Index

$$S.I. = -\log_{10} \left(I \sum_{j=1}^4 p_{a,j} p_{f,j} \bar{Q}_j \right)$$

where

- I is the total RTG fuel inventory (curies)
- p_a is the cumulative accident probability
- p_f is the single capsule failure probability
- $\bar{Q}$ is the average fractional source released given failure (curies of fines/curie of fuel)

The summation is over four impact events (see Table IV-5).

B. Failure Probability

$$p_f = \frac{1}{2\pi} \int_{-\infty}^{\bar{x}} e^{-\frac{u^2}{2}} du$$

$$\bar{x} = \frac{\bar{v}_f - \bar{v}_i}{\sigma} \quad \sigma = \sqrt{\sigma_i^2 + \sigma_f^2}$$

v_i is the impact velocity. Normally distributed with a standard deviation of $\sigma_i = 0.067 \bar{v}_i$. A function of the heat source weight, reference heat shield area and drag coefficient.

v_f is the impact failure velocity. Normally distributed with a standard deviation of $\sigma_f = 0.20 \bar{v}_f$. A function of the impact target, the impact temperature dependant clad elongation and ultimate stress, graphite elongation and ultimate stress, fuel capsule weight to area ratio. Includes fuel energy absorption. Normalized using MHW-FSA impact test data.

TABLE IV-4 (Continued)

C. Source Term

Fines production proportional to v_i^2 , exterior graphite thickness

Fines released proportioned to $\left[\left(\frac{v_i}{v_f}\right)^2 - 1\right]$ given failure, i.e., $v_i > v_f$

Determined by Monte Carlo analysis

Normalized using MHW - FSA impact test data. Values adjusted according to reentry (heat pulsed) or launch pad accident (no "heat pulse").

TABLE IV-5
IMPACT TARGETS AND ASSOCIATED PROBABILITIES

Index (j)	Environment	Impact Target	Accident Probability, p_{aj}
1	Launch complex	Concrete	Launch pad accident (10^{-3}) x probability of impacting concrete (0.6) = 6×10^{-4} . (1)
2	Launch complex	Steel	Launch pad accident (10^{-3}) x probability of impacting steel (10^{-2}) = 10^{-5} . (1)
3	Reentry following orbital decay	Granite	Orbital decay reentry (10^{-2}) x probability of impacting land (0.3) x probability of hard target (10^{-1}) = 3×10^{-4} .
4	Prompt reentry, high velocity, shallow angle	Granite	Prompt reentry (10^{-3}) x probability of impacting land (0.3) x probability of hard target (10^{-1}) = 3×10^{-5} .

(1) Based on Viking analysis: Total probability of launch pad accident is third order. Conditional impact probabilities, (concrete and steel) from Monte Carlo analysis of most probable accident.

5. Reentry Aerothermal Analysis

Computation of the safety index requires an evaluation of the impact temperature. Additionally, an acceptable design must exhibit satisfactory maximum clad temperatures and low graphite ablation. Thermal analysis was performed which relates the impact temperature, maximum clad temperature and graphite recession to three parameters, hypersonic ballistic coefficient, convective stagnation point heating factor and exterior thermal shield thickness. Results of this study are given in Appendix B-2. The data is input to SAFE-HS in tabular form.

Thermal analysis was performed using the TAP-3 thermal analyzer program and a 3-dimensional base configuration model. The use of the 3-D model and other thermal model changes have produced results which are more realistic than the previous study (Ref. IV-2). The analysis assumes stable motion with one of the faces of the heat source perpendicular to the velocity vector.

Prompt reentry analysis was performed for conditions specified by the HSSDS (Ref. IV-1): that of 44,000 ft/sec., flight path angle of -10° at 400,000 ft altitude. With respect to maximum clad temperatures and graphite recession, this is the most severe environment.

The SAFE-HS program evaluates the ballistic coefficient⁽¹⁾ and convective heating factor for each face (broad face and side-on) of the heat source (see Ref. IV-2). Tabular input data is then interpolated to evaluate the impact temperatures, peak clad temperatures and graphite recession. Impact temperatures are used in the evaluation of failure velocities (part of the safety index formulation).

⁽¹⁾ Hypersonic continuum drag coefficient of 1.66, appropriate to flat plate geometry, is used.

6. Design Selection

Each iteration of the SAFE-HS program is based on one specific set of the design variables, but it may yield a large number of designs. Preliminary analysis of the output is performed to reduce the number of designs to a relatively few candidates, those which exhibit the highest values of safety index, acceptable peak clad temperatures and graphite recession characteristics, low weight, and acceptable capsule arrangements. These few designs are subjected to further evaluation which considers other design aspects such as number of piece parts required and ease of fueling. (As an example, designs involving one capsule per heat source module were eliminated.) The result is selection of one design per iteration.

The design selection process does not end with the SAFE-HS formulation. Further modifications are cited in Section D.

C. APPLICATION OF THE TECHNIQUE

The objective of the design synthesis and evaluations technique is the selection of a reference heat source design, one which is consistent with the specific power, efficiency and safety goals. An intermediate step in the process is the characterization of safety and weight with respect to the influence of the individual design variables. Specific values of the design parameters examined during this study are as given in Table IV-6. The exterior system weights (RTG minus total heat source weight) listed in the table are for the fibrous insulation concept with current standard selenides and hot and cold junction temperatures of 900/150 °C.

As previously stated, application of the design synthesis technique produced a large number of designs. Some of these initial results are as shown in Figure IV-3,

TABLE IV-6
DESIGN PARAMETER VALUES

Heat Source Cavity

<u>Volume (in. ³)</u>	<u>Cavity Dimensions (inches)</u>	<u>Exterior System⁽¹⁾ Weight (lb)</u>
530	3.67 x 3.67 x 39.5	78
800	4.5 x 4.5 x 39.5	86
1000	5.0 x 5.0 x 40.0	91
1500	6.0 x 6.0 x 41.7	104

Fuel pellet L/D	1.0 for bulk of study. Variations up to about 2 were used to insure that all "good" designs were found.
Clad thickness	0.020 in. for bulk of study. Varied to a maximum of 0.050 in. for 1000 in. ³ volume.
Exterior graphite thickness	0.3 in. for bulk of study. Varied to a maximum of 0.5 in. for 1500 in. ³ design.
Capsule thermal inventory	Basic study 25 watts and 100 watts for 800 in. ³ volume. 75 watts for 530 in. ³ volume (see text).

(1) Total RTG weight (for fibrous insulation RTG at 900/150 °C) minus the total heat source weight.

which shows the total RTG weight as a function of safety index. The number of heat source designs produced varied with cavity volume; less than 10 for the smaller cavity volumes to about 100 for the largest (1500 in.³). Figure IV-3 gives indications of the envelope and trend weight-safety relationships as shown by the dotted line.

After determining the more attractive configurations in terms of weight and safety, the design evaluation process was continued. Final selection of a heat source for each cavity volume was based on other design considerations such as RTG fueling, heat transfer to the thermoelectrics, minimization of "plugs" (required in the heat shield to load the capsules and insulator), and ease of "coring" (elimination of excess graphite to reduce weight). As an example, the plug considerations led to rejection of designs with capsule axis parallel to the shortest dimension of the module although this orientation showed a small advantage in safety index over the corresponding design with perpendicular orientation. (See discussion of nine capsule designs, Section D below.)

Results of the system weight vs. safety are depicted on Figure IV-4. The lines through points 1a, 2 and 3a represent designs for the 800, 1000 and 1500 in.³ volumes, respectively. These designs feature heat source modules with nine fuel capsules of 25 watts each uniformly distributed within the graphite heat shield with a minimum exterior graphite thickness of 0.3 inch. (Physical aspects of these designs are given in Section D below). The intersections of each of the lines with line A is the result produced by SAFE-HS runs (no coring) directly, that is, a solid graphite matrix exists between fuel capsules. Some weight savings and increased safety is produced by removing some of this internal graphite ("coring") and line B denotes the practical limits to such improvement. The weight reduction yields lower ballistic coefficient with corresponding reduction in temperatures (impact and maximum) and lower impact velocity.

FIGURE IV-3
SYSTEM WEIGHT VERSUS SAFETY INDEX
SAMPLING OF DESIGNS PRODUCED BY SAFE-HS PROGRAM

FIGURE IV-4
WEIGHT - SAFETY ANALYSIS

Some additional improvement in safety is gained for a given volume by increasing the exterior thermal shield thickness, that is, closer spacing of fuel capsules within the module. For the 1500 in.³ volume, a nine-capsule (25 watts each) design with 0.5 in. exterior graphite thickness shows an increased safety index (point 3b) because the increased graphite thickness provides better impact protection.

It should be noted that an increase in capsule clad thickness produces negligible increase in safety with large increase in weight. This is illustrated by the dashed line connecting points 1a, 1b and 1c. Point 1a is the .020 inch thickness case while points 1b and 1c are the result of increasing the clad thickness to 0.035 and 0.050 inch, respectively.

To illustrate the effect of a larger fuel capsule, point 1d characterizes a design for the 800 in.³ cavity with 100 watt capsules. It is seen that the risk is increased by an order of magnitude as compared to the 25 watt capsule design (1a).

For cavity volumes smaller than 800 in.³, larger fuel capsules are required to accommodate the total fuel inventory requirements. Point 4 features a 530 in.³ volume in which 25 watt capsules cannot fit because of the minimum spacing and exterior graphite thickness constraints. Here a 75 watt capsule with .020 in. clad and 0.3 in. minimum external graphite yields a lower weight system with reduced safety index.

The overall result of the parameter variation analysis is shown as line E. This line represents the approximate envelope of the "best" systems. As shown, an increase in safety index from 5 to 6 costs about 30 lb. in RTG weight. Another order of magnitude decrease in risk (safety index from 6 to 7), yields an additional increase of about 80 lbs. Within the confines of these results, an upper limit of 6 on the safety index would appear to be indicated. The basis for a lower limit is discussed in the next section.

D. SPECIFIC DESIGNS

This section presents the physical and safety aspects of specific designs which are a result of the selection process. Specific recommendations are made regarding the study "reference design" based on a selection criterion and the physical aspects of the designs. A comparison is made with existing RTG systems.

Physical aspects of the design selected for the 800 in.³ volume are as shown in Figure IV-5. The 4.5 inch square dimension is that of the heat source cavity. Twenty-two modules, stacked with respect to the third dimension (1.796 in.) constitute the total heat source for the RTG. Pyrolytic graphite insulator and fuel capsules are loaded into three cylindrical cavities machined from opposite edges. The pyrolytic insulator is formed by cylindrical sleeves which extend nearly the length of the cavity and mate with end caps providing complete thermal protection from any exterior surface. Within a cavity, the capsules are separated by two cylindrical graphite space sleeves. As shown, the bulk of the space between capsules (axially) is void. This void constitutes the "coring" of design parts 1a, 2, and 3a (Figure IV-4). This space is available for inclusion of low density "crush-up" material to provide additional impact protection should subsequent studies show it desirable. Graphite felt is used at the interior end of the cavity. This material takes up slack due to tolerance and provides for a tight design. The cavity plugs are 3-D graphite.

Designs for the 1000 and 1500 in.³ volumes (points 2, 3a - Figure IV-4) are essentially the same as that of Figure IV-5 with respect to internal arrangement. The major difference is the heat shield external block dimensions. Again, the square dimension corresponds to the heat source cavity dimension, the edge or smaller dimension being aligned with the long dimension of the heat source cavity.

FIGURE IV-5

HEAT SOURCE MODULE
PPO FUEL 219 W
FOR 800 IN.³ CAVITY

A summary of results for the three designs* is given in Tables IV-7, 8 and 9. In these tables, $T_{c_{max}}$ is the peak clad temperature during reentry, and R is the leading face graphite recession. All other symbols are as previously defined. The notation "broad" and "edge" refer to stable reentry with the broad or large dimension face (of dimensions 4.5 x 4.5 for the 800 in.³ design) perpendicular to the velocity vector or the small or "edge" face (4.5 x 1.796 face - 800 in.³ design) perpendicular to the velocity.

Individual components for the safety index equation are given only for the reentry impact conditions. Launch pad impacts do not contribute significantly to the index. A combination of reduced accident probabilities, reduced impact velocities leading to lower failure velocities, and reduced source terms produce this result. (The ratio of the launch pad to reentry source term used in the analysis was 0.0143.) The reentry case assumes characteristics of a heat source after atmospheric reentry. The non heat pulsed sample is assumed characteristic of a launch complex impact. However, even without the source term difference (i.e., use of the reentry source for both impact and launch complex), no significant contribution to the safety index would have been shown to result from launch pad impacts. Typically, for concrete impact, the conditional failure probability, p_f , was a factor of 14 lower than that for orbital decay. The source term was down an additional factor of 5 (from reduced impact velocity resulting in less fines production). The steel impact is reduced to insignificance by a component of accident probability alone - that of 1 to 100 chances of impacting steel.

Another significant result is that the conditional failure probability for the prompt impact is only about 25% higher than that for impact following an orbital decay reentry, and the source terms are about the same. Hence, the factor of 10 difference in the accident probability is governing and hard target impact following an orbital decay reentry emerges as the important impact situation.

*800, 1000, 1500 in.³ sizes

TABLE IV-7

HEAT SOURCE MODULE SAFETY SUMMARY

Heat source volume (in. ³)	800
Fuel form	PPO
Module dimensions (in.)	4.5 x 4.5 x 1.796
Capsules/module	9
Number of modules	22
Module weight (lb)	3.484
Safety index	6.0

Reentry:

<u>Face</u>	Ballistic Coefficient (lb/ft ²)	Heating Factor	Orbital Decay		Prompt	
			T _{c max.} (°C)	R (in.)	T _{c max.} (°C)	R(in.)
Broad	15	1.30	1310	0.032	1550	0.037
Edge	37	1.87	1590	0.087	1790	0.157

Impact:

	T _i (°C)	$\bar{v}_i$ (ft/sec)	$\bar{v}_f$ (ft/sec)	p _f	p _a	$\bar{Q}$ curies/curie	p _f p _a $\bar{Q}$
Reentry - orbit decay	690	132	282	4.5×10^{-3}	3×10^{-4}	4.8×10^{-6}	6.5×10^{-12}
Reentry - prompt	740	132	273	5.6×10^{-3}	3×10^{-5}	4.8×10^{-6}	8.1×10^{-13}

TABLE IV-8
HEAT SOURCE MODULE SAFETY SUMMARY

Heat source volume (in. ³)	1000
Fuel form	PPO
Module dimensions (in.)	5 x 5 x 1.818
Capsules/modules	9
Number of modules	22
Module weight (lb)	3.984
Safety index	6.1

<u>Reentry:</u>			<u>Orbital Decay</u>		<u>Prompt</u>		
	Ballistic Coefficient (lb/ft ²)	Heating Factor	T _c max. (°C)	R (in.)	T _c max. (°C)	R (in.)	
<u>Face</u>							
Broad	14	1.24	1290	0.028	1520	0.032	
Edge	37	1.84	1590	0.086	1790	0.157	
<u>Impact:</u>							
	T _i (°C)	$\bar{v}_i$ (ft/sec)	$\bar{v}_f$ (ft/sec)	p _f	p _a	$\overline{Q}$ curies/curie	p _f p _a $\overline{Q}$
Reentry - orbit decay	670	126	285	3.2 x 10 ⁻³	3 x 10 ⁻⁴	4.8 x 10 ⁻⁶	4.6 x 10 ⁻¹²
Reentry - prompt	720	126	277	4.0 x 10 ⁻³	3 x 10 ⁻⁵	4.6 x 10 ⁻⁶	5.5 x 10 ⁻¹³

TABLE IV-9

HEAT SOURCE MODULE SAFETY SUMMARY

Heat source volume (in. ³)	1500
Fuel form	PPO
Module dimensions (in.)	6.0 x 6.0 x 1.812
Capsules/modules	9
Number of modules	23
Module weight (lb)	5.098
Safety index	6.3

Reentry:

Face	Ballistic Coefficient (lb/ft ²)	Heating Factor	Orbital Decay		Prompt	
			T _{c max.} (°C)	R (in.)	T _{c max.} (°C)	R (in.)
Broad	12	1.13	1250	0.022	1480	0.024
Edge	41	1.81	1600	0.088	1800	0.163

Impact:

	T _i (°C)	$\bar{v}_i$ (ft/sec)	$\bar{v}_f$ (ft/sec)	p _f	p _a	$\bar{Q}$ curies/curie	p _f p _a $\bar{Q}$
Reentry - orbit decay	650	119	288	2.0 x 10 ⁻³	3 x 10 ⁻⁴	4.6 x 10 ⁻⁶	2.8 x 10 ⁻¹²
Reentry - prompt	690	119	280	2.5 x 10 ⁻³	3 x 10 ⁻⁵	4.4 x 10 ⁻⁶	3.3 x 10 ⁻¹³

For orbital decay reentry, maximum clad temperatures for the edge stable flight are about 250°C below the 1840°C platinum-graphite eutectic. Prior error analysis, (Ref. IV-10) performed to establish the uncertainty of computed peak clad temperatures by considering all significant factors, results in a standard deviation of about 75°C. Assuming a normal distribution, the conditional probability of exceeding the eutectic temperature is less than 10^{-3} . This result is less than the computed p_f for impact. Recession is also seen to be low; the leading face suffers a graphite loss of less than one-third the protection thickness. Hence, with respect to clad melt and graphite recession, orbital decay reentries represent no problem.

Significantly high maximum clad temperatures occur only for the prompt reentry and the edge stable flight. Since reentry motion analysis to the rectangular parallelepiped geometry has not been performed, we have no quantitative basis for assessing the relative probabilities of the assumed motion. The specific reentry analyzed (44000 fps, $\gamma = 10^\circ$), however, was seen to occur for the postulated Pioneer misorientation accident. For that case, thermal analysis showed that high clad temperatures occurred only in a localized region of the V, γ space for the accident. Outside this region peak clad temperatures were at least 200°C lower. Accident analysis showed that this local region of V, γ had about a 1 in 20 probability of occurrence. In light of this, the high temperature result does not appear to present a problem.

In addition to the basic studies which employed PPO fuel, the use of shards was investigated. A design for the 800 in.³ volume is given in Figure IV-6 and Table IV-10. The physical layout is the same as for the PPO design of Figure IV-5. Both designs have 0.020 inch clad thickness and 0.3 inch exterior graphite. The reduced fuel power density (3.0 watts/cm³ for shards as compared to 4.115 watts/cm³ for PPO) requires a larger capsule. The increase in clad weight for the shard design is, however, offset

TABLE IV-10
HEAT SOURCE MODULE SAFETY SUMMARY

Heat source volume (in. ³)	800
Fuel form	Shards
Module dimensions (in.)	4.5 x 4.5 x 1.796
Capsules/module	9
Number of modules	22
Module weight (lb)	3.479
Safety index *	6.1

Reentry:

<u>Face</u>	Ballistic Coefficient (lb/ft ²)	Heating Factor	<u>Orbital Decay</u>		<u>Prompt</u>	
			T _c (°C) max	R (in.)	T _c (°C) max	R (in.)
Broad	15	1.30	1310	0.032	1550	0.038
Edge	37	1.87	1590	0.087	1790	0.157

Impact:*

	T _i (°C)	$\bar{v}_i$ (ft/sec)	$\bar{v}_f$ ft/sec	p _f	p _a	$\bar{Q}$ curies/curie	p _f p _a $\bar{Q}$
Reentry - orbit decay	690	132	304	2.6×10^{-3}	3×10^{-4}	5.5×10^{-6}	4.3×10^{-12}
Reentry - prompt	740	132	295	3.3×10^{-3}	3×10^{-5}	5.2×10^{-6}	5.1×10^{-13}

*Impact evaluation for pressed, sintered fuel. Applicability of model to shards not known.

by the decrease in 3-D graphite weight. The resultant is the same total weights of both the PPO and shards design (Tables IV-7 and IV-10 are essentially the same). The reentry results are the same. Conditional failure probabilities for the shards design are roughly a factor of two lower (owing to a decreased capsule weight to area ratio and resulting higher failure velocity). It is important to recall that the impact failure prediction technique is normalized to the MHW FSA data - a configuration which employs the PPO fuel form. The applicability of the technique to a fuel capsule containing shards is not known. Two specific areas require additional investigation, capsule failure mechanisms and fines production for granular fuels.

The design selected for the 530 in.³ volume is as shown in Figure IV-7. This design features three 75 watt capsules (25 watt pellets) with a clad thickness of 0.020 inch and a minimum thermal shield thickness of 0.3 inch.

The selection of the reference design requires quantifying the safety selection criterion. To accomplish this task, all accident environments were considered in terms of their contributions to the overall safety index - i.e., an index which is generalized to include all environments. However, in the final analysis, one failure mode may dominate. Current assessment points to heat shield thermal stress failure as the limiting case. The consequence of heat shield failure during reentry is release of large amounts of fuel to the upper atmosphere. Estimates for the Viking capsule of the fraction of total fuel released to the atmosphere subsequent to a heat shield failure were first order (10^{-1}). Applying this factor to the LCHPG design with its 10^5 curie inventory yields a total source of 10^4 curies, a value which is four orders of magnitude higher than the impact total source of the designs presented herein. The accident probability

FIGURE IV-7

HEAT SOURCE MODULE FOR 530 IN.³ VOLUME
PPO FUEL, 225 W (THREE 75 W CAPSULES)

related to high thermal stress conditions is about 10^{-4} . (The thermal environment required to produce high stresses occur by a combination of prompt reentry and heat source release from a parent configuration of prompt reentry and heat source release at low altitudes of 100,000 to 200,000 ft.) Since proper heat source design should yield conditional failure probabilities of 10^{-4} or smaller, combination of these factors yields a lower limit of the safety index for thermal stress of 4. This provides the basis for design selection for other environments. That is, choose a design with an index such that the particular environment would not add significantly to the risk. Even in final safety assessment, uncertainties are high, and significance is equated to about an order of magnitude or in terms of the safety index, and increase of one. Since the minimum acceptable value of the index for thermal stress failure is about four, the impact index should be about five.

The heat source module which is recommended as the reference design is the nine capsule module requiring an 800 in.³ cavity. (See Figure IV-5, Table IV-7 PPO fuel form and Figure IV-6, Table IV-10.) This design has an impact safety index of 6 (the upper limit for selection established by the weight safety study). To explain the recommendation, we re-examine the physical aspects of the best design found during this study which just meets the selection criterion of five on the safety index. This is the configuration for the 530 in.³ cavity (Figure IV-6, Table IV-11). This design has essentially no potential for modifications; capsule inventories are restricted to at least 75 watts, and the use of shards fuel form is precluded. (Because of the lower power density, the recommended design is about the smallest which allows the use of this fuel form.) Additionally, the 530 in.³ case is compacted to the point that there is no inter-capsule impact protection. The recommended design avoids these restrictions and allows for modifications which may be necessary as the development process continues.

TABLE IV-11
HEAT SOURCE MODULE SAFETY SUMMARY

Heat source volume (in. ³)	530
Fuel form	PPO
Module dimensions (in.)	3.67 x 3.67 x 1.80
Capsules/module	3 (75 watts(t) each)
Number of modules	22
Module weight (lb)	2.78
Safety index	5.3

Reentry:

<u>Face</u>	Ballistic Coefficient (lb/ft ²)	Heating Factor	Orbital Decay		Prompt	
			T _c max (°C)	R (in.)	T _c max (°C)	R (in.)
Broad	18	1.44	1360	.041	1600	.054
Edge	37	1.90	1590	.087	1800	.159

Impact:

	T _i	$\bar{v}_i$ (ft/sec)	$\bar{v}_f$ ft/sec	p _f	p _a	$\bar{Q}$ curies/curie	p _f p _a $\bar{Q}$ (curie)
Reentry - orbit decay	730	144	252	1.9×10^{-2}	3×10^{-4}	5.4×10^{-6}	3.1×10^{-11}
Reentry - prompt	780	144	243	2.4×10^{-2}	3×10^{-5}	5.4×10^{-6}	3.9×10^{-12}

To measure the effectiveness of the LCHPG concept with respect to impact capability, a comparison is presented with existing systems. The MHW for LES 8 and 9 has been estimated to have a 50% probability of 3 out of 24 PICS failing given impact on hard rock. The source term derived from FSA test data is 3.2×10^{-6} curies of fines released per curie inventory. The source term which would have been assigned by the technique employed in this study is a factor of 5 higher or 1.6×10^{-5} curies/curie. This difference is a function of the uncertainties associated with the impact and failure velocities. (For details see Ref. IV-9). Using the LCHPG accident probability and total curie loading, and the nominal source term, one estimates MHW safety index as $-\log_{10} (3 \times 10^{-4} \times 0.5 \times 3/24 \times 3.2 \times 10^{-6} \times 1.5 \times 10^5) \approx 5.0$. If the source term factor, 3.2×10^{-6} is replaced by 1.6×10^{-5} as discussed above, the safety index is 4.3. Thus, compared to the MHW, the LCHPG reference design has the potential for improved impact protection by a factor ranging from 10 to 50 depending on the source term uncertainty. During Phase II of the LCHPG program, heat source materials and components will be developed and tested to support this analytical prediction of heat source safety performance.

The SNAP 19 Viking⁽¹⁾ has about a 50/50 chance of surviving a hard target impact following reentry. The prompt impact failure source term was estimated at 10^{-6} curie of fines released/curie inventory. The above remarks regarding the source term also apply. Strict application of the LCHPG technique would use a source term of about 5 times this value. Hence, the Viking index would be 4 or 4.6, depending on the choice of source term. This is a factor of 25 to 100 less than that of the LCHPG reference design.

⁽¹⁾NOTE: Viking impact results are presented here for purposes of comparison only. Viking reentry impact protection was negated by the capsule oxidation characteristics. Release of fuel due to oxidation was estimated to occur shortly (5 to 20 days) after impact.

V. RECOMMENDED RTG DESIGN

The RTG design recommended for immediate development contains a heat source cavity of 800 in.³ enclosed in a membrane IHF. It employs fibrous insulation and operates at a hot junction temperature of 900°C and a cold junction temperature of 135°C. The fibrous system is chosen primarily because of low development risk and acceptable efficiency (> 10%). Further, the fibrous system is 5% lighter than that with multifoil and enables fuel capsule temperature 40 C° cooler through the membrane IHF application. Specific characteristics are contained in Tables V-1 through V-4, and a trimetric view is given in Figure V-1.

The system efficiency of 10.4% shown in Table V-2 assumes the use of Min-K type conductivity material (e. g. , Min-K TE 1400) in the converter section of the generator. The use of Fiberfrax Hi Fi 660-H paper with currently measured thermal conductivity yields a slightly lower system efficiency of about 10.1%. With either insulation, system specific power at the 500 watt level is about 3 watts/lb. Complete weight details are given in Table V-3.

The highest capsule temperature is calculated to be 1025°C a value believed well within the capability of noble metal clad materials. Other component temperatures summarized in Table V-2 likewise can be accommodated by the appropriate choice of materials.

The selection of the 800 in.³ heat source cavity volume was discussed in Section IV-D. A summary of that design from a safety viewpoint is given in Table V-4. Heat source design sketches are given in Figures IV-5 (PPO) and IV-6 (Shards). Of particular significance is the low ballistic coefficient, the correspondingly low impact velocity and peak capsule temperatures and an overall safety measure superior to present technology heat sources.

LCHPG REFERENCE DESIGN

FIGURE V-1

500 W(e) LCHPG Modular Generator

TABLE V-1
CONFIGURATION/MATERIAL SUMMARY

<u>Item</u>	<u>Material</u>	<u>Configuration</u>
Fuel	PPO or Shards	Right circular cylinder, 24.3 watts/pellet, L/D = 1
Encapsulating Member	Pt-3008	Right circular cylinder, 0.020 in. thick, 9 per heat shield
Heat Shield	Graphite, 3D Type	Parallelepiped, 22 per RTG
Reentry Insulator	Pyrolytic Graphite	Sleeve plus end caps; surrounds group of 3 capsules
Compliant Pad	Graphite Felt	Disc
Spacer for Capsules	POCO Graphite	Tube; axially separates capsules
Isolation Hot Frame	Pt-30 Rh	Square can open at ends
Bellows	Pt-30 Rh	Circular, 0.010 in. thick
Hot Frame Flanges	TZM at fueling end; Pt-30 Rh at other end	Transition pieces from square can to circular bellows
Seal Ring	Al /st Steel	Transition piece from bellows to housing; explosively bonded
Hot Side Dielectric	BeO	Either sprayed on straps or separate sheet
Hot Straps	Molybdenum	Sculptured; 0.060 in. thick
Thermoelectric P-Leg	Copper-Silver Selenide, Partitioned	Length = 0.3 inch Diameter = 0.227 inch No. = 634
Thermoelectric N-Leg	Current Standard Gadolinium Selenide	Length = 0.3 inch Diameter = 0.297 inch No. = 634
Cold Straps	Copper	0.010 in. thick; 4 string parallel circuit
Cold End Hardware	Aluminum, BeO Dielectric, 17-7 PH Steel Spring	Flexible sleeve type; 20 one-mil wraps; helical, round wire compression type spring
Converter Insulation	Min-K TE 1400 or Fiberfrax HiFi 660-H Paper	Min-K 20 lbs/ft ³ ; Fiberfrax 12 lbs/ft ³

TABLE V-1 (Continued)

<u>Item</u>	<u>Material</u>	<u>Configuration</u>
End Plug Insulation	Min-K TE 1400 or Min-K 2000	2 inches thick
Housing	Aluminum	Rectangular section, 0.050 in. , min.
Fins	Aluminum	0.020 in. , 27.8" x 39.5", 4 per RTG
Heat Pipes	Copper/Water	Internal axial grooves; 44 per RTG; 0.297 in. O. D.
Housing Rib	Magnesium	2.3 inches high; 0.080" thick
Fin Rib	Magnesium	Tapered from root-to-tip; 0.035" thick
Electrical Receptacle (power output)	TBD	Hermetically sealed to housing
Emissive Coating	ZnO + K ₂ SiO ₄ binder (IITRI Z-93)	0.005" thick
Controlled Leak Device	TBD	Located in one end cover
Heat Source Cavity Gas Fill	Helium	Few psi up to 1/2 atm.
Converter Gas Fill	Vacuum	Leak-tight

TABLE V-2
PERFORMANCE SUMMARY

OUTPUT POWER, BOL (watts(e))	500
LOAD VOLTAGE (volts)	28
SPECIFIC POWER, BOL (watts/lb)	3.0
STEADY STATE TEMPERATURES, BOL (°C)	
Sink	-79
Radiator	105
Cold Junction	135
Hot Junction	900
Isolation Hot Frame	930
Heat Shield, Outer Surface	935
Heat Shield, Maximum	970
Pyrolytic Sleeve, Maximum	1015
Outboard Capsules	1015
Inboard Capsules	1025
EFFICIENCIES, BOL (%)	
Overall	10.4
Thermoelectric	12.4
Thermal	83.9
FUEL INVENTORY (watts(t))	4807

TABLE V-3
SIZE/WEIGHT SUMMARY

<u>Component</u>	<u>Weight (pounds)</u>
Isolation Hot Frame	10.4
T/E Elements	5.2
Thermopile Hardware (hot straps, cold straps, dielectric, cold end hardware)	12.9
Insulation	13.1
Heat Pipes	8.4
Heat Pipe Saddles	3.0
Fin (Web)	8.5
Housing/Covers	11.9
Housing Ribs	5.0
Fin Ribs	7.8
Emissive Coating	3.8
Heat Shield & Reentry Insulators, PPO/Shards	40.8/39.6
Encapsulating Members, PPO/Shards	7.7/9.4
Fuel, PPO/Shards	26.7/27.0
Total, PPO/Shards	165.2/166.0
RTG Height (inches)	45.5
Fin Tip-to-Tip, diagonally (inches)	63.7
Housing Cross Section (inches)	7.0 x 7.6

TABLE V-4
HEAT SOURCE SAFETY SUMMARY
REFERENCE DESIGN

Heat Source Volume (in. ³)	800
Fuel Form	PPO, Shards
Module Dimensions (in.)	4.5 x 4.5 x 1.796
Capsules/Module	9
Number of Modules	22
Module Weight (lb)	3.42
Total Inventory (curies)	1.46 x 10 ⁵

1. Reentry

Face	Ballistic Coefficient (lb/ft ²)	Heating Factor	Orbital Decay		Prompt	
			Maximum Capsule Temperature (°C)	Graphite Recession (in.)	Maximum Capsule Temperature (°C)	Graphite Recession (in.)
Broad	15	1.30	1310	0.031	1540	0.037
Edge	37	1.87	1580	0.086	1790	0.155

2. Impact

	Impact Temperature (°C)	Mean Impact Velocity (ft/sec)	Mean Failure Velocity (ft/sec)	Capsule Failure Probability	Accident Probability	Fractional Released Fines (curies/curie)	Σ/I (curie ⁻¹)
Orbital Decay Reentry	690	130	285	4 x 10 ⁻³	3 x 10 ⁻⁴	4.8 x 10 ⁻⁶	5.8 x 10 ⁻¹²
Prompt Reentry	740	130	275	5 x 10 ⁻³	3 x 10 ⁻⁵	4.7 x 10 ⁻⁶	0.7 x 10 ⁻¹²

$$\text{Safety Index} = \left[-\log_{10} \Sigma \right] = 6.0$$

VI. RTG FLIGHT SYSTEM COST

The projected cost of a Low Cost High Performance Generator for a 1980 space mission application is \$6118 per electrical watt. This includes fuel and associated fabrication and encapsulation costs, system engineering, fabrication and assembly, quality control and acceptance tests. A summary of the major cost items is tabulated in Table VI-1. Current radioisotope thermoelectric space power supplies are priced at approximately \$30,000 per watt electrical. Thus, in the 1980 time frame, the selection of an LCHPG system will result in a factor of five cost saving compared to today's technology.

The cost reductions in radioisotope thermoelectric space power are achieved through the use of the selenide materials which exhibit greater efficiencies than current thermoelectric materials, the projected reduction in PuO_2 fuel costs for the 1980's and the economic engineering design and fabrication of reliable and modular converter and heat source components.

The development program costs for the LCHPG system are described in report TES-SNSO-3-24, Program Plan.

TABLE VI-1COST OF A PRODUCTION 500 W(e) LCHPG

• Converter 500 W(e)	\$	\$/W(t)	\$/W(e)
Thermoelectric Modules ⁽¹⁾	400,000		800
System Fabrication, Assembly and Quality Control	250,000		500
System Engineering, Safety and Reliability	55,000		110
System Test	80,000		160
Other Direct Charges	15,000		30
	<hr/> 800,000	-	<hr/> 1,600
• Heat Source			
Vent Materials & Manufacturing	96,140	20	
Capsule Material & Manufacturing	134,596	28	
Heat Shield Material & Manufacturing	240,350	50	
	<hr/> 471,086	<hr/> 98	<hr/> 942
• Fuel (4807 W(t))			
Basic Fuel ⁽¹⁾	1,163,294	242	
Fuel Fabrication ⁽¹⁾	298,034	62	
Recovery ⁽¹⁾	81,719	17	
Encapsulation & Assembly of Heat Source ⁽¹⁾	182,666	38	
	<hr/> 1,725,713	<hr/> 359	<hr/> 3,451
	<hr/> <hr/> 2,996,799	<hr/> <hr/> -	<hr/> <hr/> 5,993

(1) Reference TES-SNSO-3-24, Program Plan
Appendices for cost details.

VII. CONCLUSIONS AND RECOMMENDATIONS

This latest phase of the LCHPG system technology program has yielded an RTG with significant improvement in conversion efficiency and weight characteristics. By relaxing the operational temperature constraints and employing improved thermal insulation properties, an overall efficiency of 10% is predicted for the near-term selenide generator. The higher efficiency, together with design innovation has yielded reduced weight without loss of safety so that an RTG with specific power of at least 3 watts/lb is a practical goal.

The increased efficiency of the recommended RTG design is obtained from the increased operational temperature difference across the thermoelectrics (900° to 135°C compared to previous 800°C to 200°C range) and the use of insulation with lower thermal conductivity. An improved version of fiberfrax and its judicious application was found to yield a thermal efficiency of about 81% enabling an overall system efficiency of 10.1%.

The weight reduction effected by the increased conversion efficiency is mainly in heat source and thermopile, amounting to about 20 pounds total. A lower fuel inventory and encapsulation yields a 10 pound reduction from corresponding components in the previous design (Ref. IV-2). The thermoelectric material and associated hardware weight savings is about 10 pounds also. Although the heat rejection requirement is reduced, a weight savings is not reflected in housing and fin components because of minimum structural requirements and some small refinement of design.

Of large importance is the additional weight reduction of 30 pounds obtained from design innovation. Most of this, 20 pounds, arises from a modified heat source configuration. Development of an improved calculational model for evaluating fuel capsule response to impact environments enabled specification of a smaller clad thickness (.020 in. compared to .060 in.) without the requirement for much more graphite. The remaining 10 pounds

savings results primarily from application of the flexible sleeve cold end hardware which obviates the necessity for relatively massive conductive and structural components of the previously used square follower. A small additional weight reduction (~ 5 pounds) is afforded by the membrane isolation hot frame, but its main importance is a significantly reduced temperature difference from the thermoelectric hot shoe to the heat source module surface (about 40 C° reduction).

In order to achieve the performance predictions of the recommended LCHPG, timely development of the following components is required:

- (1) Heat Source Components
- (2) Thermoelectric Array
- (3) Thermal Insulation
- (4) Isolation Hot Frame
- (5) Flexible Sleeve Cold End Hardware

These items, together with the appropriate generator assembly processes, represent the significant areas for accomplishing the high performance, low cost goals of the LCHPG program. The fuel form, clad and heat shield materials require development to define operational capability. The source term and impact response affecting characteristics of PPO and shard fuels must be defined as applicable to the current safety design philosophy. Likewise, the clad and heat shield materials must be characterized. The thermoelectric material production, electrode contacting and applicable insulation are well recognized development areas, but must receive intensive attention if the promise of selenides is to be realized. Although of lesser challenge to technology, the isolation hot frame should be studied early to achieve the structural and thermal advantage which reflects into the heat source and thermopile designs. The flexible sleeve development is in progress currently and continuation of this effort at an accelerated pace can yield a system

advantage from performance as well as fabrication simplicity viewpoint.

Not to be overlooked in the development process is the continued formulation of analytical techniques for evaluating materials response data and thereby enable predictions of performance to avoid dilution of resources. Through the promulgation of coordinated analytical and test efforts, the probability of compliance with program objectives can be maximized.

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(1) Concept numbers are referred to in references; they are:

- Concept 1 - Fibrous
- Concept 2 - Multifoil
- Concept 3 - Multifoil - Eggcrate Design
- Concept 4 - Fibrous - Improved Selenides
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APPENDIX A
CONVERTER COMPONENT EVALUATION

1. Isolation Hot Frame

The primary functions of the Isolation Hot Frame (IHF) are to:

- provide a hermetic gas seal between the heat source and the thermoelectric sections of the generator;
- be capable of receiving fuel;
- be mechanically load bearing;
- provide a minimal thermal impedance between heat source and thermopile.

The heat source section operates under a helium atmosphere, typically 5 to 15 psi pressure, via a controlled leak to space. The thermoelectric section operates under high vacuum conditions, typically $< 1 \times 10^{-7}$ torr.

Two IHF design concepts are subjects of this study; the self supporting type for use with multifoil insulation and the membrane type applicable to load bearing type fibrous insulation.

a. Self Supporting Isolation Hot Frame Description

This design approach, utilized in prior LCHPG studies, requires that the IHF exhibit structural capability for reacting the high thermoelectric element spring loads (300 psi on the N-leg and 150 psi on the P-leg) such that deflection of the IHF wall (elastic plus creep) at operational temperatures is restricted to a maximum of .010 in. for some specified finite time. Within this time interval, fueling or de-fueling can be accomplished.

The "self supporting" IHF configuration, also referred to as a double skin and corrugation IHF, is a square section box with one inch thick sides and a membrane sheet metal bottom. The open end is attached to the housing fueling port opening via a metal bellows which is attached to the IHF by means of a sheet metal transition bulkhead. The one inch thick side walls are composite structures composed of an inner skin, outer skin and a separating corrugation. Consideration of environmental and design parameters (see Ref. A-1, Table 1) leads to use of the molybdenum alloy TZM for all structural members but the primary gas or isolation barrier (inner face). Here low carbon molybdenum is used to prevent the embrittlement of the inner face by CO attack and propagation of gas through wall defects. Also, "doubler" or sacrificial strips of Pt-30 Rh are brazed onto the heat source side of the inner face to prevent direct carburization of the molybdenum inner face and the associated formation of brittle cracks.

b. Membrane Type Isolation Hot Frame Description

The membrane type IHF was conceived as a measure to reduce heat source capsule temperatures and to save weight by reducing IHF and overall system sizes and reducing thermal losses. Pertinent features of this design concept are:

- (1) The gas barrier is provided by a thin rectangular metal can, open at both ends, which conforms to the dimensions of the heat source module stack and which is supported by the module stack, or a dimensionally equivalent graphite mandrel or heater block. The sealed transitions from the open ends of the membrane can to the top and bottom ports (openings for fuel or heater insertion and withdrawal, respectively) are accomplished by means of bellows. The membrane can is attached to stiffened flanges at each end; the bellows effect the transitions from these flanges to the top and bottom housing ports.

- (2) Thermoelectric spring loads are reacted by either the graphite mandrel, heater block, or heat source module stack. The membrane can walls are supported externally (areas not covered by thermopile hot shoes) by the insulation which reacts the heat source cavity gas pressure.
- (3) Fueling or de-fueling is accomplished by forced insertion through the top port, or fueling end of the IHF. The displaced component (e.g., fuel stack, heater block or mandrel) is ejected through the lower port. The insertion force is applied via an auxiliary tool which is attached to the stiffened flange at the top end of the rectangular membrane can.
- (4) The membrane IHF is constructed of Pt-30 Rh with a TZM loading flange. The bellows are fabricated of Pt-30 Rh. The bellows-IHF transition flange is Pt-30 Rh at the non-loading end of the IHF.

c. IHF Study Results

The self-supporting IHF walls are sized as beams under the combined axial and transverse loads imposed by the heat source cavity gas pressure and thermoelement spring pressure (Ref. A-3). Reference A-8 presents a comprehensive tabulation of the loads imposed by the thermoelement spring pressure. The membrane IHF walls are sized for the tension stresses imposed by the fuel-loading condition (Ref. A-2).

Dimensions of a section of the self-supporting IHF are shown in Figure A-1; the membrane IHF is .015 in. thick.

FIGURE A-1

DOUBLE SKIN AND CORRUGATION IHF SECTION

Weights and dimensions for the double skin/corrugation and membrane type IHF's for various LCHPG design configurations and hot frame temperatures are presented in Tables A-1 and A-2, respectively. Table A-2 presents hot frame weights for coefficients of static friction (μ) of .1 and .5. The remainder of this report presents weights for $\mu = .1$. If the coefficient of static friction lies somewhere between .1 and .5, it is clear from Table A-2 that the change in membrane IHF weight will have little impact on LCHPG total weight and specific power.

2. Thermopile

a. Thermoelectrics

This report addresses the study of current standard and improved selenide thermoelectrics. Current standard selenides will be the available thermoelectric material for applications prior to about 1980. Improved, "or potential future," selenides will be available for subsequent applications.

Design constraints applied in this study were:

- power output - 500 watts
- parallel-series strings - 4
- thermoelement length - .3 inch
- (eggcrate length - .2 inch)
- sink temperature - 350°R
- load voltage - 28 volts

Variables to be investigated were:

- hot junction temperature - 800, 900, 1000 °C
- cold junction temperature - 125, 150, 175 °C
- heat source cavity volume - 530, 800, 1000, 1500 in.³
- concept - (selenides, insulation, thermoelectric converter, IHF, etc.) (Ref. A-11)

TABLE A-1
DOUBLE SKIN/CORRUGATION IHF
500 W(e) LCHPG

Design Configuration	T/E ⁽¹⁾	Heat Source Cavity Volume (in. ³)	Hot Junction Temperature (°C)	Hot Frame Temperature (°C)	Inside Width (in.)	Height (in.)	Weight of Frame Box (lbs)	Weight of Separators (lbs)	Weight of End Wall (lbs)	Weight of End Frame (lbs)	Weight of Bellows (lbs)	Total IHF Weight (lbs)	References
Multifoil	C.S.	727	800	885	4.5	36.8	18.70	.75	.11	.09	.72	20.37	A-3, A-6, A-7
Multifoil	C.S.	727	900	974	4.5	36.8	18.70	.75	.11	.09	.72	20.37	A-3, A-6, A-7
Multifoil	C.S.	727	1000	1065	4.5	36.8	18.70	.75	.11	.09	.72	20.37	A-3, A-6, A-7
T/E Eggcrate, Multifoil	C.S.	618	900	973	4.5	36.8	18.70	.75	.11	.09	.72	20.37	A-3, A-6, A-7
Multifoil	I	559	900	976	4.5	36.8	14.03	.72	.11	.09	.56	15.51	A-4, A-6, A-7

(1) C.S. = Current standard selenides
I = Improved selenides

TABLE A-2
MEMBRANE IHF
500 W(e) LCHPG

Design Configuration	T/E ⁽¹⁾	Heat Source Cavity Volume (in ³)	Hot Junction Temperature (°C)	Hot Frame Temperature (°C)	Inside Width (in)	Height (in)	Coefficient of Static Friction	Weight of Hot Frame Box (lbs)	Weight of Loading Frame (lbs)	Weight of End Frame (lbs)	Weight of Bellows (lbs)	Total IHF Weight (lbs)	References
Fibrous	C S	530	900	930	3 67	39 5	1(5)	5 20(7 00)	1 44(2 87)	09(14)	1 00(1 34)	7 73(11 35)	A-7, III-2
Fibrous	C S	800	800	830	4 5	39 5	1(5)	7 00(10 19)	2 05(2 84)	11(11)	1 32(1 32)	10 48(14 46)	A-7, A-5
Fibrous	C S	800	900	930	4 5	39 5	1(5)	7 00(9 80)	2 00(2 82)	11(11)	1 32(1 32)	10 43(14 05)	A-7, A-5
Fibrous	C S	800	1000	1030	4 5	39 5	1(5)	7 00(12 06)	2 00(2 64)	11(11)	1 32(1 32)	10 40(16 13)	A-7, A-5
Fibrous	C S	1000	900	930	5 0	40 0	1(5)	7 64(9 17)	2 78(3 75)	18(18)	1 74(1 74)	12 31(14 85)	A-7, A-5
Fibrous	C S	1500	900	930	6 0	41 67	1(5)	9 56(9 56)	3 74(4 99)	23(23)	2 46(2 46)	15 99(17 24)	A-7, A-5
Fibrous	I	618	900	930	4 5	30 5	1	5 25	2 12	09	1 43	8 89	A-7, A-4

(1) C S = Current standard selenide
I Improved selenides

In addition to thermoelectric elements, the thermopile includes hot side dielectric, hot straps and cold straps. The hot side dielectrics are made of BeO and electrically insulate each thermoelectric couple from the isolation hot frame. The N and P elements are electrically connected to form couples by means of the hot straps which are constructed of molybdenum. The cold straps, made of copper, connect groups of four couples to each other to complete the electrical hookup of four strings in parallel.

Figure A-2 shows a typical thermoelectric module array and a thermoelectric couple with associated dimensions and hardware. Thermoelectric module dimensions and other parameters are found in detail in Ref. A-8. Weights of the thermoelectric elements, numbers of couples per RTG and thermoelement diameters are presented here in Table A-3 (Ref. A-11, Table 1); weights of the hot side dielectrics, hot straps and cold straps are included in the thermopile hardware weights presented as part b of this section.

The thermoelectric "eggcrate" design, shown in Figure A-3, has thermoelectric elements placed next to each other but electrically insulated by a .010 in. thick separator. This separator initially resembles an eggcrate, into which the N and P elements are placed; hence, the term "eggcrate."

The proper couple electrical connections are made by flame spraying between or over separators to achieve the hookup of four strings in parallel. The main advantage of the eggcrate design is that it minimizes thermoelectric module heat losses, serving to improve system efficiency over that obtained with the conventional thermoelectric module design.

FIGURE A-2

THERMOELECTRIC COUPLE

TABLE A-3

THERMOELECTRIC ELEMENT WEIGHTS

500 Watt(e) RTG; 28V, 4 Strings in Parallel

Design Configuration	T _{hot} (°C)	T _{cold} (°C)	Thermoelectric Type	No. Couples per RTG	D _N (in.)	D _P (in.)	Element Length (in.)	Thermoelectric Weight (in.)
Fibrous or multifoil	800	125	Current standard	762	.301	.229	.3	6.41
	800	150		757	.312	.252	.3	7.20
	800	175		750	.323	.277	.3	7.86
	900	125		635	.296	.223	.3	5.20
	900	150		634	.299	.233	.3	5.23
	900	175		632	.306	.247	.3	6.57
	1000	125		553	.288	.206	.3	4.04
	1000	150		551	.288	.214	.3	4.15
	1000	175		449	.293	.227	.3	4.41
	900	150	Improved	563	.255	.224	.3	3.85
T/E eggcrate design, multi-foil	900	150	Current standard	634	.299	.233	.2	2.38

Length = 0.2 inches
 Power output = 31.25 W(e) @
 $T_{\text{hot}}/T_{\text{cold}} = 900/150^{\circ}\text{C}$
 Voltage = 28 V
 4 strings in parallel

FIGURE A-3

THERMOELECTRIC EGGCRATE DESIGN

Figures A-4 and A-5 (Refs. A-10 and A-11, respectively) present thermoelectric efficiency (data from 3M) as a function of hot and cold junction temperature for current standard and improved selenides, respectively. This information is significant because it gives an indication of the upper limit of LCHPG system efficiency. At a nominal $T_{\text{hot}}/T_{\text{cold}}$ combination of 900/150°C, 100% thermal efficiency (no heat losses) gives system efficiencies of 12.1% and 15.65% for current standard and improved selenides, respectively.

b. Thermopile Hardware

Thermopile hardware includes the cold end and hot end hardware. The cold end hardware is discussed in this section; thermopile hardware weights are also presented here.

The function of the cold end hardware is to provide a good thermal path between the thermoelectric cold junction and the RTG housing, and to provide a compressive load to the thermoelectric elements and electrodes. In order to minimize total system weight, a tradeoff between the weight and temperature difference associated with candidate thermal designs is required.

Two cold end hardware (CEH) designs are presented in this report. The primary design is the flexible sleeve, being developed by Teledyne Energy Systems. This design is shown in Figure A-6, along with associated materials and dimensions. The aluminum sleeve provides a good thermal path and its flexibility absorbs any misalignments due to small structural shifts or dimensional tolerances. For further discussion, see TES-SNSO-3-26, LCHPG Component Development (Ref. A-12). The backup CEH design is the square follower type, a development of the 3M Company. Figure A-7 depicts the square follower CEH, which employs an assemblage of springs, screw-springs,

FIGURE A-4
TPM EFFICIENCY
CURRENT STANDARD
0% EXTRANEIOUS RESISTANCE

ACTUAL DATA POINTS ARE CIRCLED

FIGURE A-5
T/E EFFICIENCY
IMPROVED SELENIDES
0% EXTRANEOUS RESISTANCE

ACTUAL DATA POINTS
 ARE CIRCLED

FLEXIBLE SLEEVE
COLD END HARDWARE

*May be reduced to 0.550

FIGURE A-6

FLEXIBLE SLEEVE COLD END HARDWARE

FIGURE A-7

SQUARE FOLLOWER COLD END HARDWARE

followers, rails and copper balls to provide the good thermal path between the thermoelectric cold end and RTG housing. For further information, see the 3M Company letter of August 28, 1974 (Ref. A-13).

The temperature drop across both CEH designs is approximately 30°C. Table A-4 presents a weight comparison of the two designs. Since the thermal impedance of both are approximately equal, the lighter flexible sleeve design (by 10.4 lbs per RTG) was chosen for the reference RTG design.

Table A-5 presents thermopile hardware weights (Ref. A-11, Table 1; Ref. A-13) for all design configurations and hot junction/cold junction temperature combinations studied.

The thermopile hardware weight of the eggcrate design (Concept 3, see Thermopile Section of Appendix A) is so low since there is not a one to one correspondence between thermoelectric elements and flexible sleeve/spring CEH units. Due to the extremely compact thermoelectric eggcrate module design there are only 28 CEH units per 80 thermoelectric elements. All other design concepts require one CEH unit per thermoelectric element.

3. Insulation

The function of the LCHPG insulation is to minimize parasitic heat losses with the least amount of insulation weight. This, in turn, maximizes system efficiency and specific power ($\frac{W(e)}{lb}$).

A number of different insulations were studied, the three main ones being Min-K, Fiberfrax and Multifoil. Fiberfrax and Min-K are both fibrous type insulations. Multifoil is a moly/quartz layered insulation (approximately $100 \frac{\text{foil layers}}{\text{inch}}$) with alternating layers of .0005 in. moly foil and .005 in. quartz cloth. In evaluating these insulations for application in the LCHPG, their conductivities, densities and load bearing capabilities

TABLE A-4
COLD END HARDWARE WEIGHT COMPARISON (LBS.)

	<u>Square Follower</u>	<u>Flexible Sleeve</u>
Follower or sleeve (aluminum)	0.0096	0.00220
Rails (aluminum)	0.0024	-
Springs (steel)	0.0082	0.00577
Balls (copper)	0.0053	-
Cap (aluminum)	-	0.00072
Electrical Isolator	-	0.00052
Total/Couple	<hr/> 0.0255	<hr/> 0.00921
Total/RTG*	16.2	5.8

*500 watts(e); 634 couples to produce 28 volts with 4 parallel strings.

TABLE A-5
THERMOPILE HARDWARE WEIGHTS (500 WATT RTG)

Concept	T _{hot} (°C)	T _{cold} (°C)	Thermoelectric Type	Hot Side Dielectric Weight (lbs)	Hot Strap/ Partitions/ Gimbals Weight (lbs)	Cold Strap Weight (lbs)	Cold End Hardware Weight (lbs)	Thermopile Hardware Weight (lbs)
Fibrous or multifoil	800	125	Current standard					
	800	150		2.1	6.7	1.1	8.0	17.9
	800	175						
	900	125						
	900	150		1.5	4.9	0.8	5.8	13.0
	900	175						
	1000	125						
	1000	150		1.2	3.8	0.6	4.5	10.1
	1000	175	Current standard					
	900	150	Improved	1.2	4.1	0.7	4.9	10.9
T/E eggcrate design	900	150	Current standard	0.6	1.8	0.3	2.2	4.9

must be examined. The minimum conductivity-density ($k\rho$) product is an indication of a good insulator (lightest, minimizes heat losses). However, a low $k\rho$ product may be offset by poor load bearing capability or other system considerations.

Table A-6 (refs. A-15, A-16, III-4) presents an insulation comparison of candidate insulations. It should be noted that Min-K can be machined to shape and is load bearing. Fiberfrax 660-H paper is cut to shape and stacked in layers. Where compressive loads are required, such as the side of the generator, the thickness of the stack in the free (non-compressed) state is greater than the thickness of the space that it occupies so that it is under compressive load in the assembled generator. Multifoil insulations are non-load bearing.

Both moly/quartz multifoil and Fiberfrax LoCon blanket have low and nearly identical $k\rho$ products (in vacuum). However, to achieve similar heat losses in LCHPG applications, the Fiberfrax thickness must be approximately ten times that of multifoil. Thus, if heat losses are the same for both insulations, the Fiberfrax necessitates a larger housing with a resultant weight penalty. Thus, on the basis of $k\rho$ product alone, multifoil (in vacuum) is the preferred insulation. However, being non-load bearing, it requires:

- (a) the use of the heavier self-supporting isolation hot frame instead of the lighter membrane IHF;
- (b) the use of ZrO_2 spacers between the housing and IHF on the two non-module sides to prevent shifting of the IHF under dynamic loading; these spacers increase heat losses.

These disadvantages of non-load bearing insulation are ample reasons to consider load bearing Min-K TE or Fiberfrax 600-H paper. Further disadvantages of multifoil are its potential for oxidation with associated conductivity degradation, thermal leakage at

TABLE A-6
INSULATION COMPARISON

Material	ρ - Density (lbs/ft ³)	Max. Temp. (°F)	k-Conductivity @ 1000°F (Btu/hr-ft-°F)		k ρ Product Btu-lb hr-ft ⁴ -°F		Comments
			<u>Air</u>	<u>Vacuum</u>	<u>Air</u>	<u>Vacuum</u>	
Min-K TE	20	1600	.021	.015	.42	.30	Moldable, machine-able, load bearing.
Moly/quartz multi-foil (100 foil layers/inch)	58	2000	.027	.002	1.57	.12	Non-load bearing, requires electrical insulation.
Fiberfrax Hi Fi 660-H paper	12	2300	.029	.019 (est.)	.35	.23	Cut to size and stacked; load bearing.
Fiberfrax LoCon blanket	6	2300	.065	.019	.39	.12	"Cleanest product"
Fiberfrax 970-H paper	10	2300	.048		.48		.020", .040", .080" thicknesses.
Fiberfrax bulk fiber	6	2300	.076		.46		
	12	2300	.060		.72		
Fiberfrax H bulk fiber blanket	6	2600	.067	.020	.402	.12	

joints and its potential for causing electrical shorts in the RTG thermopile. However, it is also recognized that Min-K may be incompatible with the thermoelectric selenides. An altered Min-K fabrication process might alleviate this problem. If not, Fiberfrax would be the insulation of choice. A number of different LCHPG cases were studied using Min-K, multifoil and Fiberfrax 660-H paper, individually and in various combinations. In all cases, however, Min-K was used as the end plug insulation because of the required high load bearing capability. Results of this comparative study are presented in Section III of this report.

Insulation weights for all LCHPG cases studied are presented in Table A-7 (Ref. A-11).

4. Housing/Radiator

The primary functions of the LCHPG housing/radiator system are to:

- radiate the required reject heat to space
- react the thermoelectric cold end hardware loads
- maintain gas seal
- support the cantilevered fins with heat pipes
- provide means of attachment to spacecraft
- provide suitable micrometeoroid protection

Figure A-8 shows a typical LCHPG housing/radiator with associated materials. Heat is rejected from the housing and the four mutually perpendicular fins. Each fin is a planar extension of one of the housing sides and is .020 in. thick, the minimum fabrication thickness. The housing and fins are constructed of Al 6061-T6 which offers high thermal conductivity, low density, high strength characteristics, good availability, low cost and easy attachment of the electrical receptacle and hot frame transition ring. This four-finned design requires an envelope of about two-thirds that of an RTG with two fins. (Ref. A-18).

TABLE A-7
INSULATION WEIGHTS
500 WATT LCHPG

Design Configuration	T/E*	Heat Source Cavity Vol. (in. ³)	T _{hot} (°C)	T _{cold} (°C)	Insulation Type	Module Insulation Weight (lbs)	Module Periphery Insulation Weight (lbs)	Ends and Blank Sides Insulation Weight (lbs)	Total Insulation Weight (lbs)
Membrane IHF	C. S.	530	900	150	Min-K	.841	1.866	8.47	11.18
		800	800	125	Min-K	.931	2.066	9.38	12.38
			800	150		.931	1.928		12.24
			800	175		.892	1.928		12.20
			900	125		.788	2.879		13.05
			900	150		.743	3.003		13.13
			900	175		.743	2.934		13.06
			1000	125		.663	3.553		13.60
			1000	150		.660	3.553		13.59
		800	1000	175		.660	3.512	9.38	13.55
		1000	900	150		.743	3.664	10.21	14.62
		1500	900	150	Min-K	.743	5.179	12.74	18.67
Membrane IHF		800	900	150	Fiberfrax 660-H	.461	1.91	7.37	9.74
Double Skin IHF		727	800	125	Multifoil	2.39	4.10	13.22	19.71
			800	150		2.37	3.91		19.51
			800	175		2.31	3.78		19.31
			900	125		2.02	5.05		20.28
			900	150		2.00	5.02		20.24
			900	175		1.95	5.02		20.19
			1000	125		1.77	5.76		27.74
			1000	150		1.75	5.76		20.72
			1000	175	Multifoil	1.66	5.86	13.22	20.73
Double Skin IHF			900	150	Fiberfrax 660-H**	.385	2.78	14.24	17.40
T/E Eggcrate, Membrane IHF	C. S.	727	900	150	Multifoil	.008	8.00	11.59	19.60
Membrane IHF	I	618	900	150	Min-K	.636	2.14	10.59	13.37
Double Skin IHF	I	559	900	150	Multifoil	2.00	10.00	5.50	17.50

* C. S. = Current standard selenides

I = Improved selenides

** Fiberfrax in T/E module, Min-K in ends, multifoil elsewhere

To enhance heat transfer to the fins, L-shaped heat pipes are attached to the external housing surface at each of the two sides where the thermoelectrics are located, and outward along the two adjacent fins to the fin tips. The heat pipes are spaced evenly from the top to bottom of each fin. The copper/water grooved type heat pipes are solder bonded to their attachment saddles which are integral with each housing/fin plate and extend along the full length of each.

A study was performed to evaluate a purely conductive radiator (Ref. A-19), i.e., one without heat pipes, and results reinforced the desirability of a heat pipe augmented radiator. The conductive radiator was found to be at least five times as heavy as the heat pipe assisted system.

Stiffening ribs are situated at each heat pipe location (riveted to an extension of each heat pipe saddle) and are spaced similarly over the non-module sides (sides without heat pipes) of the housing. Each rib is made of magnesium-thorium alloy and consists of two pieces, a dynamic fin rib that extends the full length of the housing/fin plate and a doubler static rib that is riveted to the fin rib and extends only across the width of the housing side. The dynamic fin rib is tapered from the housing/fin juncture point to zero height at the fin tip; its thickness and maximum height are typically .030 and 2.5 in., respectively. The doubler static rib is field riveted to the dynamic fin rib and is elliptically shaped. Its thickness is the total required thickness of the static housing rib minus the dynamic fin rib thickness. A typical doubler static rib thickness and maximum height is, then, .045 and 2.5 in., respectively, so that the total required static housing rib thickness is .075 in.

The objective in the analysis of the housing/radiator system is to determine the lightest system that satisfies its desired functions (viz. rejects required heat, supports LCHPG components). A computer program POWER (Ref. A-20), was written to perform

a combined thermal/structural weight optimization of the housing/radiator system. This optimization is performed with respect to fin efficiency and fin thickness for a given radiator temperature and heat load. Other constraints are: design configuration (fibrous or multifoil), heat source cavity volume, housing dimensions and thermoelectric hot junction and cold junction temperatures. As noted in the thermopile section of this Appendix (A-2), the radiator temperature is approximately 30°C below the cold junction temperature.

In performing the weight optimization, utilization of POWER provides:

- radiative/conductive heat transfer calculations (fin sizing, heat pipe spacing)
- static rib (housing) stress calculations (rib sizing)
- dynamic (fin) rib sizing using sinusoidal and acoustic response techniques
- heat pipe sizing
- component weighing (housing, end covers, ribs, heat pipes and saddles, radiator web, emissive coating)

To size the fins, the amount of heat directly converted to electrical energy (500 W(e)) is subtracted from the thermal inventory (Section III-B) to yield the amount of heat radiated from the housing and fin surfaces of the RTG. The program, POWER, then is used to determine the fin width (root to tip dimension) that is required to radiate this amount of heat.

The heat pipe spacing is basically a function of fin efficiency, fin thickness and radiator temperature. The heat pipe spacings for all combinations of fin efficiencies and fin thicknesses are evaluated and the combination that results in the minimum weight system is chosen as optimum. The optimum fin efficiency varies from approximately 80% to 90%.

Static rib stress calculations are performed to determine the required height and thickness of the static housing rib. POWER enables treatment of the static rib as a pin-ended beam under axial tension and uniform partial transverse loading (Ref. A-21, A-23) which reacts the thermopile spring load and heat source cavity gas pressure. The Mg-Th rib is sized as a function of fin efficiency (rib spacing) and fin thickness.

The dynamic (fin) rib is considered as a cantilevered beam vibrating transversely under a dynamic environment of sinusoidal vibration and sound pressure (Ref. A-21, A-23). This environment is taken to be that provided by the Titan III-C booster where the RTG is mounted directly to a rigid spacecraft. This rib is made of Mg-Th and is also sized as a function of fin efficiency and fin thickness.

The heat pipes are sized for outside diameter based on heat transport data obtained from the Dynatherm Corporation (Ref. A-22). This data is incorporated in POWER to enable determination of heat pipe diameter for each combination of fin efficiency and fin thickness considered. Heat pipe diameter is a direct function of heat load per heat pipe, heat pipe transport length and radiator (heat pipe) temperature. The heat pipes are conservatively sized for a temperature 100°F below radiator temperature (in space) to prevent heat pipe burnout and resultant failure during ground operation.

After sizing all LCHPG components, POWER provides their weights (housing, end covers, end cover flanges, ribs, heat pipes and saddles, radiator web, emissive coating). Included is the basic housing sidewall thickness of .050 in; the thermoelectric module area being .100 in. thick with thermoelectric spring recesses reducing this to .050 in. The zinc oxide emissive coating enhances radiation of heat and reduces the absorption of solar energy. Its weight is solely a function of RTG surface area and the coating density (0.00033 lb/in.^2 , assuming uniform density).

Results of the housing/radiator weight optimization study are presented in Tables A-8 and A-9 (Refs. A-11, A-24, respectively). Table A-8 presents a heat rejection subsystem summary of various dimensions, sizes and parameters. Table A-9 presents a summary of housing/radiator component weights.

TABLE A-8
HEAT REJECTION SUBSYSTEM SUMMARY

Note	Heat Source Cavity Vol.	T _{hot}	T _{cold}	Insulation ⁽¹⁾		T/E	Fin Ht.	Outside Housing Width		Fin Width	Q _{inv}	η _{fin}	Heat Pipe Outside Dia.	No. of L Pipes per Fin Pair	Heat Pipe Spacing	Housing Rib Ht. Module Side	Housing Rib Ht. Non-Module Side	Housing Rib Thick-ness Mod. Side	Housing Rib Thick-ness Non-Mod. Side	Fin Rib Thick-ness
				Module Side	Non-Module Side			Module Side	Non-Module Side											
	in ³	°C	°C				(in)	(in)	(in)	(in)	(W)		(in)		(in)	(in)	(in)	(in)	(in)	(in)
	800	800	125	Min-K	Min-K	C.S	39.5	7.6	7.0	34.78	5155	.85	.334	10.27	3.85	2.41	2.62	.084	.099	.045
	↓	800	150	↓	↓	↓	↓	↓	↓	28.55	5319	.80	.290	9.40	4.20	2.65	2.82	.084	.102	.035
	800	800	175	↓	↓	↓	↓	↓	↓	22.33	5495	.80	.257	10.21	3.87	2.58	2.98	.084	.119	.030
	900	900	125	↓	↓	↓	↓	↓	↓	31.73	4762	.85	.319	10.27	3.85	2.32	2.37	.076	.079	.040
	900	900	150	↓	↓	↓	↓	↓	↓	24.41	4926	.85	.264	11.27	3.50	2.29	2.49	.076	.089	.030
	900	900	175	↓	↓	↓	↓	↓	↓	20.23	5076	.80	.248	10.21	3.87	2.44	2.62	.075	.086	.025
	1000	1000	125	↓	↓	↓	↓	↓	↓	30.20	4566	.85	.311	10.27	3.85	2.32	2.21	.071	.062	.040
	1000	1000	150	↓	↓	↓	↓	↓	↓	22.88	4673	.85	.258	11.27	3.50	2.24	2.24	.072	.071	.030
	800	1000	175	↓	↓	↓	↓	↓	↓	18.99	4831	.80	.243	10.21	3.87	2.43	2.35	.073	.067	.025
	1000	900	150	↓	↓	↓	40.0	8.1	7.5	24.09	4980	.85	.265	11.42	3.50	2.41	2.49	.086	.088	.030
	1500	900	150	↓	↓	↓	41.67	9.1	8.5	23.27	5126	.85	.264	11.90	3.50	2.61	2.51	.102	.094	.030
	727	800	125	Multifoil	Multifoil	↓	36.8	↓	9.0	33.29	4763	.85	.329	9.56	3.85	2.63	2.71	.077	.086	.040
	↓	800	150	↓	↓	↓	↓	↓	↓	25.56	4931	.85	.271	10.50	3.50	2.60	2.90	.081	.103	.030
	↓	800	175	↓	↓	↓	↓	↓	↓	21.38	5131	.80	.255	9.51	3.87	2.80	3.08	.077	.102	.030
	↓	900	125	↓	↓	↓	↓	↓	↓	29.44	4300	.85	.309	9.56	3.85	2.58	2.55	.068	.068	.035
	↓	900	150	↓	↓	↓	↓	↓	↓	22.63	4477	.85	.259	10.50	3.50	2.51	2.58	.070	.079	.030
	↓	900	175	↓	↓	↓	↓	↓	↓	17.45	4642	.85	.232	11.41	3.23	2.46	2.68	.068	.089	.025
	↓	1000	125	↓	↓	↓	↓	↓	↓	25.44	4018	.90	.276	12.13	3.03	2.25	2.30	.067	.071	.030
	↓	1000	150	↓	↓	↓	↓	↓	↓	20.55	4157	.85	.250	10.50	3.50	2.47	2.43	.066	.063	.025
	727	1000	175	↓	↓	↓	↓	↓	↓	15.72	4307	.85	.225	11.41	3.23	2.45	2.44	.068	.073	.020
(2)	800	900	150	Min-K	Min-K	↓	39.5	↓	7.0	39.84	4926	.85	.349	11.27	3.50	2.29	2.49	.076	.089	.050
(3)	↓	↓	150	Min-K	Min-K	↓	↓	↓	↓	24.41	4926	.85	.264	11.27	3.50	2.29	2.49	.076	.089	.030
	↓	↓	150	Fiberfrax	Fiberfrax	↓	↓	↓	↓	25.12	5043	.85	.267	11.27	3.50	2.29	2.49	.076	.089	.030
(4)	↓	↓	150	Fiberfrax	Fiberfrax	↓	↓	↓	↓	27.83	5494	.85	.279	11.27	3.50	2.29	2.49	.076	.089	.035
(5)	800	↓	150	Min-K	Min-K	↓	39.5	↓	7.0	22.94	4926	.90	.248	14.30	2.76	1.64	1.62	.036	.033	.025
	727	↓	↓	Fiberfrax	Fiberfrax	↓	36.8	↓	9.0	24.86	4822	.85	.268	10.50	3.50	2.51	2.58	.070	.079	.030
(6)	727	↓	↓	-	Multifoil	C.S	36.8	↓	7.6	20.95	4371	.90	.241	13.32	2.76	1.76	1.86	.023	.031	.025
	618	↓	↓	Min-K	Min-K	I	30.5	7.5	7.0	23.38	3862	.85	.261	8.71	3.50	2.22	2.11	.072	.063	.030
	559	900	150	Multifoil	Multifoil	I	27.6	7.5	9.0	20.6	3490	.90	.234	9.99	2.76	2.20	2.24	.066	.070	.025

(1) Min-K end plugs used on all designs.

(2) Two-fin RTG.

(3) Shards fuel.

(4) Xenon gas in converter

(5) Thermopile spring pressure = 60 psi.

(6) Thermoelectric eggcrate design.

TABLE A-9
LCHPG WEIGHTS SUMMARY

NOTE	INSULATION ⁽⁵⁾		T/E	H.S. CAVITY VOL. (IN) ^{**3}	T(HOT) DEG(C)	T(COLD) DEG(C)	HEAT PIPES (LBS)	H.P. SADDLE (LBS)	RAD. WEB (LBS)	EMISSIVE COATING (LBS)	HOUSING (LBS)	END COVERS (LBS)	END COVER FLANGE (LBS)	RIBS (LBS)	HEAT REJECTION SYSTEM (LBS)
	MODULE SIDE	NON-MODULE SIDE													
	MIN-K	MIN-K	C.S.	530	900	150	6.70	2.31	7.55	3.25	8.73	.65	.92	10.64	40.75
	MIN-K	MIN-K	C.S.	800	800	125	12.50	3.01	10.77	4.51	10.43	.79	1.16	16.97	61.02
	MIN-K	MIN-K	C.S.	800	800	150	7.52	2.55	8.84	3.80	11.40	.79	1.35	13.74	49.98
	MIN-K	MIN-K	C.S.	800	800	175	5.64	1.98	6.92	3.08	10.92	.79	1.22	13.67	44.22
	MIN-K	MIN-K	C.S.	800	900	125	10.57	3.41	9.82	4.16	10.29	.79	1.16	13.22	53.43
	MIN-K	MIN-K	C.S.	800	900	150	6.90	2.41	7.56	3.32	9.66	.79	1.03	12.30	43.97
	MIN-K	MIN-K	C.S.	800	900	175	5.03	1.77	6.27	2.84	10.61	.79	1.22	9.85	38.36
	MIN-K	MIN-K	C.S.	800	1000	125	9.70	3.18	9.35	3.99	10.13	.79	1.15	11.29	49.57
	MIN-K	MIN-K	C.S.	800	1000	150	6.36	2.23	7.09	3.15	9.48	.79	1.03	10.27	40.39
	MIN-K	MIN-K	C.S.	800	1000	175	4.69	1.65	5.88	2.70	10.43	.79	1.22	8.29	35.64
	MIN-K	MIN-K	C.S.	1000	900	150	7.05	2.47	7.56	3.37	10.30	.89	1.10	13.52	46.24
	MIN-K	MIN-K	C.S.	1500	900	150	7.39	2.58	7.61	3.49	11.79	1.08	1.24	16.56	51.75
(1)	MIN-K	MIN-K	C.S.	800	900	150	8.44	2.55	6.17	2.81	9.66	.79	1.03	13.88	45.32
	F-FRAX	F-FRAX	C.S.	800	900	150	7.17	2.50	7.78	3.40	9.66	.79	1.03	12.49	44.62
(2)	F-FRAX	F-FRAX	C.S.	800	900	150	8.26	2.85	8.62	3.71	9.66	.79	1.03	13.72	48.64
(3)	MIN-K	MIN-K	C.S.	800	900	150	7.69	2.71	7.11	3.15	8.12	.79	.72	6.03	36.31
(4)	MIN-K	MIN-K	C.S.	800	900	150	6.90	2.41	7.56	3.32	9.66	.79	1.03	12.30	43.97
	M-FOIL	M-FOIL	C.S.	727	800	125	10.89	3.44	9.60	4.12	12.84	.98	1.86	14.86	58.59
	M-FOIL	M-FOIL	C.S.	727	800	150	6.91	2.40	7.37	3.29	12.00	.98	1.65	14.84	49.44
	M-FOIL	M-FOIL	C.S.	727	800	175	5.04	1.77	6.17	2.85	13.43	.99	1.96	12.59	44.79
	M-FOIL	M-FOIL	C.S.	727	900	125	8.78	2.88	8.49	3.71	12.71	.98	1.86	11.36	50.76
	M-FOIL	M-FOIL	C.S.	727	900	150	5.90	2.07	6.53	2.98	11.87	.98	1.65	11.17	43.15
	M-FOIL	M-FOIL	C.S.	727	900	175	4.72	1.65	5.04	2.43	11.20	.98	1.49	11.09	38.59
	M-FOIL	M-FOIL	C.S.	727	1000	125	8.17	2.82	7.34	3.28	10.25	.98	1.30	12.15	46.29
	M-FOIL	M-FOIL	C.S.	727	1000	150	5.26	1.85	5.93	2.76	11.69	.98	1.65	8.93	39.04
	M-FOIL	M-FOIL	C.S.	727	1000	175	4.28	1.49	4.54	2.24	11.03	.98	1.49	8.95	34.99
	F-FRAX	F-FRAX	C.S.	727	900	150	6.66	2.32	7.17	3.22	11.87	.98	1.65	11.72	45.59
(6)	M-FOIL	M-FOIL	C.S.	727	900	150	6.49	2.28	6.04	2.76	8.51	.87	1.08	4.92	32.95
	MIN-K	MIN-K	I	618	900	150	5.06	1.77	5.60	2.50	7.65	.78	1.02	7.48	31.85
	M-FOIL	M-FOIL	I	559	900	150	4.05	1.42	4.47	2.10	7.63	.97	1.15	8.39	30.17

(1) 2-FIN RTG

(2) XENON GAS IN CONVERTER

(3) THERMOPILE SPRING PRESSURE = 60 PSI

(4) SHARDS FUEL

(5) MIN-K END PLUGS USED ON ALL DESIGNS

(6) T/E EGGRATE DESIGN

APPENDIX B
SAFETY DATA

1. Thermal Stress

A comparative study was performed on the effect of thermal shield thickness on maximum stress (Ref. B-1). The results of that study and a conclusion regarding minimum allowable thickness are reported herein.

Thermal conditions severe enough to cause high stresses occur upon RTG breakup and release of heat shields at relatively low altitudes. The primary reentering body may range from an RTG to a complete spacecraft/rocket stage, configurations with relatively high ballistic coefficient. A heat source released under these conditions undergoes rapid deceleration, and the associated heating produces severe thermal gradients. The temperature distribution used for this study is typical of those produced by the above sequence of events.

Available techniques limited our analysis to two dimensional geometry with cylindrical symmetry. The heat shield was approximated by a hollow cylinder (see sketch, Figure B-2). A linear temperature variation over the first 0.1 inch thickness was used. The top surface temperature was 6000°F. The thermal shield thickness was varied from 0.2 to 0.4 inches. POCO graphite was assumed. (Reliable properties for 3-D graphites were not available for this study.)

Maximum tensile and compressive stresses as a function of radius are as shown on Figure B-1. Excepting "edge" effects, tensile stresses were found to decrease with increasing thicknesses; compressive stresses exhibit the reverse property.

Consistent with current analysis⁽¹⁾, an effective measure of the probability of failure is provided by the volume integral of the stress. The radial average of the

(1) See, for example, the work of Batdorf and Crose, Ref. B-2.

FIGURE B-1
MAXIMUM TENSILE AND COMPRESSION THERMAL STRESSES AS A FUNCTION OF INNER RING WALL THICKNESS AND RADIAL DISTANCE

MATERIAL - FORD ; ELASTIC-PLASTIC SOLUTION
AXISYMMETRIC MODEL

CONSTANT TEMP. DISTRIBUTION: 0.5 ± 1.5 IN. $T = 1650^\circ F$
 1.8 IN. $T = 6000^\circ F$

○ WALL THICKNESS 0.40 IN.
 □ " " 0.30 IN.
 △ " " 0.20 IN.

FIGURE B-2
AVERAGE MAXIMUM THERMAL STRESS VS. HEAT SHIELD
WALL THICKNESS.

POCO HEAT SHIELD, AXISYMMETRIC MODEL (SAAS III).

ELASTIC-PLASTIC SOLUTION

CONSTANT TEMPERATURE DISTRIBUTION (SIMULATED LOW ALTITUDE/HIGH VELOCITY RELEASE):
 $0 \leq z \leq 1.5 \text{ IN}, T = 1650^\circ \text{F}$; $z = 1.6 \text{ IN}, T = 6000^\circ \text{F}$ FOR ALL r 'S.

$$\bar{\sigma} = \frac{\int_0^R \sigma(r) r dr}{\int_0^R r dr}$$

$R = 1.7 \text{ INCHES}$

maximum stresses as a function of thermal shield thickness is shown in Figure B-2. A significant decrease occurs in tensile stresses as the thickness increases from 0.2 to 0.3 inches. Although compressive stresses increase, they are of less importance since they occur in regions of high temperature where graphites have demonstrated a near plastic behavior and failures occur at values of greater than three times the computed values.

We have tentatively concluded that a heat shield thickness of 0.3 inch would be sufficient to effect the required low failure probability. These studies are, however, only preliminary. Future work will utilize 3 dimensional analysis with 3-D graphite properties.

2. Aerothermal Parametric Results

Reported herein is reentry data derived from aerothermal reentry analysis. Details of the thermal analysis are given in Ref. IV-7. Time dependent stagnation heating rates and other atmospheric flight data are reported in Ref. B-3. The data presented herein are input to the SAFE-HS program in tabular form for use in evaluating the reentry characteristics and safety index for each heat source module design.

Results for two reentry conditions are presented:

- a. Prompt reentry: heat source with a velocity of 44,000 ft/sec and a flight path angle of -10° at 400,000 ft altitude.
- b. Reentry from orbital decay heat source with a velocity of 25,690 ft/sec and a flight path angle of -0.1° at 400,000 ft altitude.

Stable flight is assumed with one face of the rectangular parallelopiped heat source perpendicular to the velocity vector. The independent variables are:

1. Ballistic coefficient = $W/C_D A$ (lb/ft²)

where W = the heat source weight (lb)

C_D = the hypersonic continuum drag coefficient (1.66 for this study)

A = the leading face area (ft²)

2. Heating factor - the ratio of the convective stagnation point heating to that of a reference 12 inch sphere and is given by

$$H. F. = 0.565 \left(\frac{12}{b} \right)^{1/2} \left[1 + \left(\frac{b}{l} \right)^{1/2} \right]^{1/2}$$

where b and l are the face dimensions in inches, b is the smaller of the two.

3. Exterior thermal shield thickness - the minimum distance from the capsule insulator to the exterior surface of the block.

The dependent variables are maximum fuel capsule clad temperature, clad impact temperature and graphite recession.

Maximum clad and clad impact temperatures for the prompt reentry are given in Figure B-3. Temperature results for the orbital decay reentry are presented in Figure B-4. Not shown, is the effect of varying the external thermal shield thickness. For thicknesses of greater than 0.2 inch the response was evaluated to be linear showing a decrease of 29°C per 0.1 inch of added thickness for the peak reentry temperature and an increase of 13°C per 0.1 inch of added thickness for the impact temperatures. This result applies to the prompt and orbital decay reentry conditions.

Leading face graphite recession results are given in Figure B-5. Recession is essentially independent of graphite shield thickness (variation of less than 6 mils for a thickness range of 0.13 to 0.52 inches). Results are for a POCO thermal shield, but are believed to be a good approximation to 3-D graphite with reasonable high density (≥ 1.6 gm/cm³).

FIGURE B-3

MAXIMUM CLAD & IMPACT TEMPERATURES

SUPERORBITAL/SHALLOW (44K/-10°)
 GRAPHITE WALL THICKNESS = 0.26 IN.
 0.1 IN. PYROLYTIC GRAPHITE INSULATOR W/ GAPS
 900°C HOT JUNCTION TEMPERATURE, HELIUM

5/2/75

FIGURE B-4

MAXIMUM CLAD & IMPACT TEMPERATURES

ORBITAL DECAY
GRAPHITE WALL THICKNESS = 0.26 IN.
0.1 IN. PYROLYTIC GRAPHITE INSULATOR W/ GAPS
900°C HOT JUNCTION TEMPERATURE, HELIUM

FIGURE B-5

GRAPHITE RECESSION

GRAPHITE WALL THICKNESS = 0.25 IN.

1-1
5/5/16

Initial temperature conditions at 400,000 ft were for an RTG with a hot junction temperature of 900°C and helium cover gas in the heat source cavity. Steady state capsule clad temperatures for various conditions are given in Table B-1. For hot junction temperatures other than 900°C, helium fill, the maximum clad temperatures for a prompt reentry show a linear variance. The change in peak temperature is roughly half the change in capsule temperature over the values of Table B-1. Thus, for an 800°C hot junction temperature, the peak clad temperatures would be about 50°C less than the values of Figure B-3. Over a 200°C range (800 to 1000°C), the initial heat source temperatures have negligible effects on peak temperatures during on orbital decay reentry and on impact temperatures or graphite recession for either prompt or orbital decay reentry.

TABLE B-1
CAPSULE TEMPERATURE SUMMARY (°C) ⁽¹⁾
PPO/SHARDS

<u>Hot Junction Temperature (°C)</u>	<u>Fibrous Insulation RTG (Membrane IHF)</u>			<u>Multifoil Insulation RTG (Double Skin & Corrugation IHF)</u>		
	<u>Vacuum</u>			<u>Vacuum</u>		
	<u>Helium</u>	<u>$\epsilon = .16^{(2)}$</u>	<u>$\epsilon = .8^{(3)}$</u>	<u>Helium</u>	<u>$\epsilon = .16^{(2)}$</u>	<u>$\epsilon = .8^{(3)}$</u>
800	927/919	1215/1170	1074/1048	981/973	1244/1200	1112/1086
900	1027/1019	1270/1229	1146/1123	1069/1062	1296/1255	1180/1155
1000	1126/1117	1331/1294	1224/1202	1162/1154	1355/1320	1253/1233

-
- (1) Clad temperatures are for the hottest (center) capsule of a 9 capsule array. Other capsules will be about 10°C cooler.
- (2) For an uncoated capsule.
- (3) For a typical coated capsule.