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SNAP 21A - TEN-WATT
DEEP SEA ISOTOPIC
POWER SUPPLY

Contract AT(30-1)3322

MND-3322-7

CLASSIFICATION CANCELLED

DATE 2/27/68

BT

For The Atomic Energy Commission

H. J. Carroll

Chief, Declassification Branch

First Quarterly Progress Report
for Period
March Through May 1964

W. A. McDonald
Engineering Manager

Approved P. J. Dick
P. J. Dick
Program Manager

Published July 1964

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FOREWORD

This is the first quarterly progress report describing technical status of the SNAP 21A development program. It covers the work accomplished during the period from March through May 1964 under Contract AT(30-1)-3322 for the U.S. Atomic Energy Commission.

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SUMMARY

Phase I of the SNAP 21A program was initiated by the U.S. Atomic Energy Commission in March 1964. The objective of this development program is to design and develop a highly efficient and reliable thermoelectric strontium-90 fueled power system suitable for use in deep sea applications at hydrostatic pressures up to 10,000 psi.

The Phase I program will be conducted over a 12-month span time and will culminate in the performance demonstration of a full-scale operating model. Component development tests and preparation of detail engineering drawings and specifications to facilitate manufacture of prototype units having a system output of 10 watts for a minimum of five years unattended operation will be included. A technical report summarizing all efforts undertaken during the program will be prepared at the conclusion of the Phase I development program.

The SNAP 21A power supply is composed of four major subassemblies which are:

- (1) A lower titanium pressure vessel containing uranium-8% molybdenum biological shield and an evacuated thermal insulation assembly.
- (2) A tantalum-10% tungsten fuel capsule containing 28.5 kilocuries of strontium-90 and a mechanically attached uranium-8% molybdenum biological shield cover plate.
- (3) A segmented, lead telluride-bismuth telluride thermoelectric system of modular construction.
- (4) An upper titanium pressure vessel containing a dc-to-dc power conditioner and electrical feedthrough connector.

The total system weight including strontium-90 fuel and shielding is calculated at 490 pounds. The unit is cylindrical in shape with overall dimensions of 19-1/4 inches in length with a 13-3/4-inch diameter across the pressure vessel closure flange. Overall system efficiency will be greater than 6-3/4% including power conditioning to produce 10 watts(e) at 24 volts for five years or more of unattended operation.

During this first quarter, the detailed preliminary design and analysis of the prototype electrically heated unit was completed.

The thermoelectric aging tests were run during the first quarter to determine the most promising thermoelement bonding arrangements. The results have facilitated selection of material combinations to be

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tested in a six-couple thermoelectric module test series. A thermoelectric couple test box has been designed and a test procedure has been written which will allow the accumulation of the most useful data from this test series. These tests will also be programmed to experimentally evaluate operation of the thermoelement materials above 1000° F hot junction temperatures in the latter part of the test program.

Preliminary breadboard tests on the dc-to-dc power conditioner have shown an overall efficiency of 87.8%. This value will be increased to 90% by component matching. A method of conditioner regulation to maintain essentially constant hot junction temperature is under study.

The safety effort was also initiated during this quarter. The materials selection from a safety standpoint was reviewed and design criteria is now being defined.

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I. INTRODUCTION

During the first quarter, March through May 1964, of the SNAP 21A development program, the Martin Company devoted major efforts to detail design and analysis of the power system with emphasis on procurement of materials and assemblies for component development tests. Major components of the system, including fuel capsule, pressure vessel, thermal insulation and biological shield, were engineered and released for procurement and manufacture. The thermoelectric system was also analyzed to facilitate design and manufacture of component test assemblies.

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II. GENERATOR DEVELOPMENT*

A. DESIGN CONFIGURATION

SNAP 21A is a 10-watt undersea generator assembly designed to be compact, lightweight and highly efficient. The electrically heated prototype (design) configuration, which was completed during the quarter, consists of four major subassemblies (see Fig. 1). These are:

- (1) Titanium lower pressure vessel with depleted uranium shielding.
- (2) Vacuum insulation system.
- (3) Fuel capsule with the attached top uranium shield plug.
- (4) Titanium upper pressure vessel including power conditioner, segmented thermoelectric module assembly and electrical feedthrough connector.

The pressure vessel is forged and machined Ti-6Al-4V alloy designed to withstand up to 10,000-psi external pressure without collapse or leakage. The vessel is fabricated in two sections which are bolted together with 18 titanium bolts. The parting flange seal is made using two Viton-A O-rings. It is planned that the completed vessel assembly will be hydrostatically pressure tested at the U.S. Naval Research Laboratory up to the 10,000-psi design level during this program.

The lower portion of the pressure vessel contains the biological shielding which is depleted uranium alloyed with 8% molybdenum. The shielding is held in place by eight lugs which mesh with a retaining groove in the lower section of the pressure vessel. The initial shielding in the lower vessel assembly concept was maintained by shrink fitting the titanium on the shielding. However, the question of maintaining the required flange straightness to effect a good seal after heating the titanium for shrink fitting could not be fully resolved without development tests. Therefore, a mechanical holddown scheme was selected. The adequacy of this system is presently under investigation.

The vacuum insulation assembly is then fitted into the center portion of the shield body. This is a highly efficient, load-bearing member from which the fuel capsule assembly is supported. The insulation consists of foils to shield from the radiated heat and fibers to control the conducted heat. The insulation system does require vacuum in the order of 1×10^{-3} mm Hg to function efficiently.

*Contributors: F. Waters, S. Morrison, J. Thomas, V. Trusselo, A. Spamer, W. Fretague.

LEGEND

- A Electrical feedthrough connector
- B Cooling fin
- C Instrumentation space
- D Dc/Dc converter
- E Pressure vessel
- F Indexing pin
- G Viton "O" ring seals
- H Hold-down ring
- I T/E module assembly
- J Vacuum insulation assembly
- K Biological shielding
- L Fuel capsule
- M Fuel cannister
- N Cold junction heat sink

Fig. 1. SNAP 21A-Ten Watt Undersea Generator

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The fuel capsule is tantalum-10 tungsten alloy, a high density, high strength material. This design has not yet been finalized with ORNL because a closure weld penetration of 0.150 inch is required to provide for 300-year protection from corrosion in a seawater environment. A milk-can type fusion weld alone would give 0.050 to 0.070 inch penetration. Therefore, a filler metal welding technique will be developed by ORNL to obtain the required penetration. A review with ORNL indicates this is not a major problem area but some welding development is required.

The fuel capsule assembly includes an attached uranium-molybdenum shield plug to provide complete biological shielding of the heat source. The assembly is held in place by a retaining ring of Min-K insulation preloaded through a ring plate and bolting arrangement which carries the load back into the lower shield body and the lower pressure vessel. This forms a very compact lower system assembly. However, a problem could exist in this area in the ability of the Min-K to withstand dynamic loads and in the response of the Min-K and retaining plate system to random vibrational inputs. This area will be checked experimentally during this Phase I program.

The upper section of the pressure vessel houses and supports the thermoelectric module assembly, instrumentation, the dc-to-dc converter and the electrical feedthrough connectors. The thermoelectric module package is bolted in place, drawing a conical-shaped section of the heat sink into contact with a mating surface on the inside of the pressure vessel. Initially, this was a flat plate attachment; however, to provide additional heat transfer area so a thermoelectric cold junction temperature of 80° F might be maintained, the conical wedge section evolved. Additional heat transfer area was obtained with the addition of 20 externally welded fins. In addition, to assure a good heat transfer contact along the copper wedge to titanium interface, an aluminum grease will be used on the mating surfaces.

Two module configurations are presently under development. These are shown in Figs. 2 and 3. The SNAP 21A primary module design employs a printed circuit arrangement with the thermoelectric couples bonded to the circuit at the cold junction. The module assembly is then evacuated to allow approximately one-half atmosphere pressure to preload the elements through the bellows and bellows diaphragm action. This assembly can be seen in Fig. 2. A possible problem area existed here in that close control must be maintained on the element height or the danger of unloading an element appears. The alternate design, shown in Fig. 3, employs a conventional spring and piston-type arrangement to ensure that compressive loading is maintained on all thermoelements. Instrumentation for the unit will consist of thermocouples and voltage taps in the thermoelectric module along with a pressure transducer to monitor the module pressure. In addition, there will be

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Fig. 2. Printed Circuit Module

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Fig. 3. Spring and Piston-Type Module

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a humidity detector installed to monitor pressure vessel internal moisture conditions. The electrical feedthrough connector will provide leads for all instrumentation feedthroughs plus leads for the generator power output and output regulation. The dc-to-dc converter will be packaged into the envelope shown in Fig. 1.

Final vessel sealing is made at the electrical feedthrough connector after the electrical leads are pulled through and soldered to the connector. This then is bolted in place to the top pressure vessel head. The hermetic seal for this unit also relies upon the use of Viton-A O-rings to effect the seal.

B. PERFORMANCE AND OPERATING CHARACTERISTICS

The SNAP 21A Undersea Power Supply (thermoelectric generator and converter) is designed to produce 10 watts with 24 volts and maintain a thermoelectric efficiency of $\geq 10\%$ and a system efficiency of $\geq 6.75\%$ with a weight ≤ 550 pounds. This will be done by utilization of semiconductor thermoelectric materials for direct energy conversion. By heating one end of the thermocouples and keeping the other end cool, d-c electrical power is generated. The heat in SNAP 21A is produced by the decay of the radioisotope SrTiO_3 , strontium titanate, which is readily available.

In order for the generator to be most efficient, the parasitic heat losses (that heat which does not go through the thermoelements) must be kept to a minimum. This required a carefully coordinated thermal analysis and design effort to determine a configuration of high efficiency and manufacturing producibility. Temperature profiles were established so that:

- (1) Materials data and stress analysis could be related to the proper operating temperatures.
- (2) Heat losses could be determined.
- (3) The design temperature differential over the thermoelectric elements could be checked for the overall system optimization.

a. Analytical techniques and data

For thermoelectric optimization, the VINCE-TOM code for the digital computer was used. The property data used, (Appendix A) consisted of Martin measured points for the PbTe and PbSnTe segments of the elements and Melpar measurements of Melcor materials for the Bi_2Te_3 segments. The figures of merit for the data used analytically

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can be found in Appendix A. The additional data necessary for using the code was contact resistances. From previous experience which showed that resistances measured at room temperature increased at operating temperatures and with operating time, a method of estimating end-of-design life contact resistances was established.

The room temperature value of the hot junction resistance was doubled for temperature effects, while the intermediate segment junction room temperature resistance was increased by a factor of 1-1/2 for the same effect. Then, the resultant values for the hot junction, segment junction and cold junction resistances were all doubled to account for operating time effects.

Certain assumptions were made regarding the thermal analysis. The generator is to operate with a steady state voltage characteristic in ambient water temperatures of 20° to 41° F; therefore, 40° F was chosen analytically as ambient temperature for design point generator operation. The design hot junction temperature at end of life was selected as 950° F.

TABLE 1

SNAP 21A Performance Study
(all items refer to end-of-design life except where noted)

Power to load (watts)	10.8
Dc-to-dc voltage converter efficiency (%)	90
Power from generator (watts)	12
Thermoelectric efficiency (%)	10.2
Heat to elements (watts)	117.6
Parasitic heat loss (watts(t))	
Module insulation (Min-K)	11
Module bellows	7.9
Vacuum insulation	9.7
Vacuum insulation seal bellows	11.7
Fuel block retainer ring (Min-K)	3
Total heat inventory (watts(t))	160.9
Generator efficiency (%)	6.75

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TABLE 1 (continued)

Isotope decay factor	1.13
Thermal inventory beginning of life (watts)	182
Fuel loading beginning of life (kilocycles)	28.5
Electrical characteristics	
Power conditioner	
Output (watts)	10.8
Load (volts)	24
Open circuit voltage	3.5
Generator	
Segmented module	
Output power (watts)	12
Load voltage (volts)	2.0
Open circuit voltage (volts)	3.5
Optimum load resistance (ohms)	0.333
Internal resistance (ohms)	0.251
Couples/module (No.)	16
Thermoelectric element characteristics	
Material	
N leg	PbSnTe BiSeTe
P leg	PbTe BiSbTe
Length (in.)	0.5

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TABLE 1 (continued)

Area (in. ²)	
N leg	0.069
P leg	0.0846
Diameter (in.)	
N leg	0.297
P leg	0.328

C. PERFORMANCE SUMMARY (SEE TABLE 1)

1. Design Point Generator Operation

The SNAP 21A generator is designed to produce 10.8 watts out of the converter after five years at a design hot junction temperature of 950° F. The thermoelectric cold junction will operate at 80° F in 40° F ambient water. The cold end design (with fins) includes enough heat transfer area (skin to water) to limit the drop to 10° F. With these temperatures determined, the thermoelectric efficiency of 10.2% was calculated and a required thermal inventory of 182 watts at beginning of life was established. This produces 12 watts(e) from the thermoelectric generator and 10.8 watts at 24 volts from the system with a 90% efficient dc-to-dc voltage converter. Results from a thermoelectric parametric study with temperature and contact resistance parameters is shown in Appendix A.

2. Off-Design Generator Operation

To produce 12 watts at end of life, there must be an excess fuel inventory at the beginning of life due to the isotope decay. SrTiO_3 has a half life of 28 years which results in an isotope decay factor of 1.13 for five years. This overloading of fuel increases the hot junction temperature to 1050°F at beginning of life for the constant voltage generator. Within the bounds of present knowledge, there is a question as to the ability of the lead telluride thermoelectric couples to operate at temperatures above 1000°F without material sublimation. Therefore, a study is being carried on in the power conditioner design and development area in an attempt to effect hot junction cooling by drawing a higher current with the dc-to-dc converter. In addition, module tests will be conducted during which the hot junction temperature will operate at temperatures above 1000°F in order to observe the effects on the thermoelectric couples and to verify whether a problem which necessitates the

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control of beginning-of-life temperature exists. At the unregulated temperature, the beginning-of-life power out of the elements is 14 watts(e) or 12.6 watts from the converter (see Appendix C). Therefore, although the power will drop if the current is raised at beginning of life for hot junction cooling purposes, there is approximately a 2.6-watt margin over the 10-watt required power level.

3. Thermal Characteristics

a. Parasitic heat losses

There are three components through which the heat is lost parasitically (see Fig. 1). The first component is the thermoelectric module which consists of Min-K 1301 with round holes into which the thermoelectric couples are inserted. Although Min-K has a low thermal conductivity, the relatively large cross sectional area of thermal insulation results in a heat loss of 11 watts. The thin walled stainless steel bellows which surrounds the module and is thermally in parallel with the thermoelements loses about 7.9 watts by conduction. The second component is a ring of Min-K which holds down the bottom portion of the generator against an upward load. Through this ring approximately three watts are lost. The other component is the vacuum super insulation system which surrounds the fuel capsule, channeling the heat to the thermoelements. This package is being supplied by the Linde Company, a division of Union Carbide. Using the thermal conductivity data supplied by the Linde Company, a loss of 9.7 watts was calculated through the foil insulation itself. This loss is a combination of radiation, for which many layers of copper foils insulate, and conduction, insulated by interface layers of quartz fibers. The insulation system is evacuated and sealed by a thin stainless steel bellows. Heat will be conducted through the bellows and radiated away. Radiation will carry heat up the channel between the bellows and the Min-K ring. In taking a conservative approach, calculations showed a heat loss of 11.7 watts through the bellows, making the total system loss 21.4 watts.

Those separate losses give an overall generator parasitic loss of 43.3 watts which corresponds to a thermal efficiency of 73.1%.

b. Temperature profile

With a design point hot junction temperature of 950° F, the hot side profile can then be determined (see Table 2). No temperature problem exists on the hot end at end of design life or beginning of life structurally; therefore, it is sufficient to simply show the values. However, the cold end presented somewhat of a problem. To reach the design objective of > 10.0% thermoelectric efficiency with a 950° F hot junction temperature, the cold junction temperature must be maintained at 80° F. There is good conduction through the copper cold junction heat sink plate but due to the

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placement of the power conditioner assembly directly above the cold plate the copper heat sink titanium pressure vessel interface originally had a relatively small contact area. As a result, the possibility of a large temperature drop across this interface existed. To increase this interface area, a cone-wedge extension of the copper plate was added along with aluminum grease to facilitate conduction in the gap, reducing the temperature drop to 40° F. Heat transfer through the titanium pressure vessel dome and from the dome to the water was poor because of the relatively poor conductivity of titanium coupled with a relatively small area. Since the use of titanium is advantageous for its strength-to-weight ratio, adequate heat transfer can be obtained by increasing the transfer area with the addition of 20 shaped fins. The addition of fins assures a cold junction temperature of 80° F in ambient water of 40° F. Development tests will be conducted to determine the need for cooling fins.

TABLE 2

SNAP 21A Temperature Profile--Hot Side End of Design Life

	<u>(°F)</u>
Hot junction	950
Module bellows	1000
Shield plug	
Surface	1155
At tantalum interface	1195
Tantalum fuel capsule	
Outside surface	1215
Inside surface	1227
Fuel centerline	1327

4. Weight

Since there is a limiting weight specification of 550 pounds placed on the SNAP 21A generator, this area has been carefully controlled. In the initial design developmental stage, the weight increased above the limit, the bulk of it being in the shielding. By a substitution of tantalum-10 tungsten for Hastelloy C in the fuel capsule, 1/4 inch on the diameter of the shielding could be removed because of the higher density of the Tantung which serves as more shielding in itself. In

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addition, some shield shaping was possible in the lower area where the radiation dose rate was comparatively low. These measures greatly reduced the weight of this component. The copper cold junction heat sink plate was an area in which some weight reduction was also possible. However, it was desirable to maintain a fairly large volume of copper for heat transfer purposes since its high thermal conductivity results in low temperature drops through it and thus lower thermoelectric cold junction temperatures. Some additional reduction has been effected by cutting the pressure vessel to minimum thickness. This is still structurally acceptable. A detailed weight breakdown is given in Table 3 and shows the system weight to be 489.5 pounds.

TABLE 3
Weight Breakdown

	<u>(lb)</u>
Fuel	2.68
Fuel capsule	21.40
Shield	285.00
Cold plate	19.40
Module	10.00
Pressure vessel	125.00
Converter	2.00
Insulation	2.50
Bolts, nuts, washers, O-rings, electrical feedthrough connector	<u>21.50</u>
Total	489.5

5. Structural Characteristics

The design criteria for the SNAP 21A unit is that structural integrity must be maintained in a seawater environment with accompanying pressures up to 10,000 psi. In addition, in the event of seal leakage, fuel containment must be maintained for a period of 300 years in the same environment. This made it necessary to design the fuel capsule, as well as the outer pressure vessel, to withstand pressures up to 10^4 psi.

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The main areas of structural discontinuity that were analyzed are shown in Fig. 4. The results of the analysis, along with allowable stress values, can be found in Table 4. The detailed structural analyses may be found in Appendix C.

1. Fuel capsule lower plate
2. Fuel capsule upper plate
3. Pressure vessel upper flange
4. Pressure vessel shield interface

Fig. 4. Structural Discontinuity Areas

6. Shielding Analysis

The shielding analysis work performed during the past quarter for the SNAP 21A generator consisted of determining the thicknesses of depleted uranium shields for various capsule materials and determining dose rates at the surface of the rounded corner of the shield.

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Maximum Stress Levels

<u>Location</u>	<u>Surface</u>	<u>Calculated Stress (psi)</u>	<u>Allowable Stress* (psi)</u>
1. Fuel capsule	Inside corner		
	Axial	-84,400	158,000
	Circumferential	-40,180	
	Outside corner		
	Axial	39,800	
	Circumferential	7,580	
1. End plate	Center of plate		
	Top surface	-37,540	158,000
	Radial and circumferential		
2. Fuel capsule	Inside corner		
	Axial	-89,000	158,000
	Circumferential	-42,650	
	Outside corner		
	Axial	44,400	
	Circumferential	7,860	
2. End plate	Center of plate		
	Top surface	-48,280	158,000
	Center of plate		
3. Pressure vessel upper flange	Inside corner		
	Axial	-78,470	120,000
	Circumferential	-35,260	
	Outside corner		
	Axial	40,730	
	Circumferential	19,940	
4. Pressure vessel	Outside radius	-2,070**	120,000
4. U-8% Mo shield	Inside radius	-28,900	110,000

*Yield strength values

**Based on 0.005-inch shrink fit between shield and vessel. Stresses incurred using mechanical holddown scheme instead of shrink fit are under investigation.

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The safety criteria used in determining shield thicknesses was, according to ICC shipping regulations, that thickness required to maintain the following:

- (1) Ten milliroentgens per hour or less at one meter from the generator axis.
- (2) Two hundred milliroentgens per hour or less at the surface of the generator.

The surface dose rate of 200 mr/hr proved to be the controlling criterion for thicknesses.

The capsule materials considered were Hastelloy C, tantalum and platinum. Figure 5 shows the variation with shield thicknesses of dose rates at the surface of the generator and at one meter from the center of the fuel. The dose rates were calculated by the use of the generalized shielding code developed at Martin. This code integrates dose rates from an array of point sources and uses buildup factors to approximate the contribution of scattered radiation to total dose rate. The bremsstrahlung spectrum for strontium-90 was that developed in Ref. 1.

Table 5 shows the thicknesses of depleted uranium required to obtain dose rates of 200 mr/hr at the generator surface for the different capsule materials. Table 6 shows the effect on dose rate of the use of tantalum instead of Hastelloy C as the capsule material. The values in Table 6 are for a Hastelloy C capsule shielded with the thicknesses of uranium determined for a tantalum capsule. These values also show that the surface dose rate is about a factor of two higher than allowed by shipping regulations, even though the dose rates at one meter are allowable.

TABLE 5

Thicknesses of Uranium Shield Required to Obtain 200 mr/hr
at Surface

	Hastelloy-C (in.)	<u>Capsule Material</u>	
		Tantalum (in.)	Platinum (in.)
Top	2.0	1.75	1.6
Bottom	2.5	2.2	2.0
Side	2.6	2.4	2.2

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Fig. 5. Variation of Dose Rates with Shield Thicknesses for Different Capsule Materials

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TABLE 6

Dose Rate Using Hastelloy C Capsule with Thicknesses of Uranium Same as for Tantalum Capsule

	At Surface Dose Rates (mr/hr)	1 Meter from Generator Axis
Top	420	9.6
Bottom	430	10.0
Side	380	7.8

To reduce shielding weight, the bottom corners of the shield and housing were rounded. The generalized shielding code was used to determine surface dose rates at the rounded corner. The variation of dose rate with corner radius is shown in Fig. 6. Side and bottom shield thicknesses were those from Table 5 for the tantalum capsule. A sketch showing the dose point locations is also shown on Fig. 6. A radius of 3-7/8 inches will give a dose rate of 200 mr/hr.

7. Materials Compatability Effort

A materials review was held during the quarter and the following possible problems were discussed.

The fuel capsule originally had a side closure weld requiring a full penetration depth of 0.496 inch. To weld the high melting point Tantalum (5516° F) at this thickness, a multipass weld with filler metal was required. The heat input required to make that weld was felt to be sufficient to melt the stainless steel liner around the fuel and possibly the fuel itself. This area has been redesigned to permit a top closure weld of 0.156 inch penetration. The heat input required for this weld should be considerably reduced from the 0.496 inch weld depth.

The possible air oxidation of the U-8% Mo during thermoelectric module change was considered. To determine the magnitude of this problem, a preliminary test was run wherein three wafers (1/16- by 3/4-inch diameters) of U-8% Mo were heated in air at 1000° F for periods of 5, 20 and 60 minutes and the corresponding weight changes were determined. The oxide which formed spalled freely from the exposed surfaces of the specimens during cooling to room temperature. The weight losses listed were determined after a final brushing to remove nonadherent oxide:

Fig. 6. Dose Rates at Surface of Rounded Corner

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Test Time (min.)	Weight Loss (original weight: 7 gm) (gm)
5	0.039
20	0.103
60	0.132

These data show by the decrease in the apparent oxidation rate that while at the test temperature the oxide becomes somewhat protective.

A third area of concern was the compatibility of the U-8% Mo shield with the stainless steel of the Linde Insulation Package. The area of concern here is that the U-8% Mo will leach the nickel from the Type 321 Stainless Steel, forming a low melting point eutectic and weakening the steel to a point of uselessness.

To investigate this problem, wafer specimens of U-8% Mo were placed in tight pressure contact with the following materials: iron, Type 321 Stainless Steel and nickel. The samples were heated for two hours at 1300° F and then for two hours at 1700° F. The tests were performed in an argon atmosphere with the specimens encased in thin tantalum foil to minimize oxidation effects. At 1300° F, all specimens exhibited a tendency to stick, indicating a long time diffusion bond would probably occur. When heated to 1700° F all of the material combinations reacted, the U-8% Mo showing evidences of reaction completely through the test piece in each case. The reactions were most severe in the case of nickel and progressively less severe in the case of stainless steel and iron. A subsequent test was run with the U-8% Mo and iron separated by tantalum. These specimens were clamped together by molybdenum bolts. No evidences of sticking (bonding) were observed at either temperature.

To protect the stainless steel from the U-8% Mo shield and to protect the U-8% Mo against oxidation, more tests were planned using a surface coating of flame-sprayed aluminum oxide on the shielding. More shielding material has been ordered for these tests.

The possible problem of the U-8% Mo shield growing because of thermal cycling was also checked during the quarter. A piece of the shielding material 3/4 inch long and 3/4 inch in diameter was cycled five times between 80° and 150° F and five times between 1000° and 1300° F. It was checked for size before and after each cycle. No detectable change in dimensions was noted.

In addition, the possible gamma radiation problem was checked with the National Lead Company, Albany, New York. It was stated that no

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gamma radiation problem exists with the material. There is only a neutron irradiation problem. It was also pointed out that both the U-8% Mo and the U-10% Mo alloy have the gamma phase stabilized down to room temperature. However, on thermal cycling around 1000° F but below the gamma transformation temperature for long periods of time, a precipitation of the U-Mo compound in a delta uranium matrix will occur. This involves a volume change but the change is not as great as the alpha to beta change in unalloyed uranium.

Another area of investigation was the corrosion resistance of the titanium pressure vessel material. The Crucible Steel Company was contacted and they stated that the corrosion resistance of the Ti-6 Al-4 V alloy in a seawater environment was as good as unalloyed titanium if the stress levels in the material remain below the yield stress.

The last area of investigation during the past quarter was the possible galvanic corrosion of the Tantung alloy. Conflicting statements have been found in the literature about this material. Therefore, a simple test has been planned using Ti-6 Al-4 V, Ta-10% W, 6061 Al and Type 321 Stainless Steel coupled electrically in seawater to determine whether a problem exists and, if so, the seriousness of the problem.

8. Power System Reliability

An initial study of the SNAP 21A system configuration was made during the quarter and the reliability model of the system determined. This model can be schematically represented by:

This model reflects the design goal of 10 watts minimum continuous output when operated under the ocean for a period which extends for five years from the start of fueling. Presumably the fueling period through the launch period of the model will be of a short time duration. Nevertheless, the system should be analyzed to assure that the environmental conditions experienced prior to the operational portion of the mission have a negligible probability of causing system failure.

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Assuming that the reliability of the generator is a function only of the operational period, the operational reliability model can be represented as:

The preliminary reliability evaluation is now made following the operational reliability model path. The pressure vessel and shielding is designed to a specification in which all the environmental conditions are specified, and sufficient design margins are provided over the design life. In this case, the success probability can be taken as one. The fuel behavior obeys known physical laws and is predictable with time, therefore, it too may be assigned a success probability of one.

If the Linde super insulation package fails to meet requirements by a small amount, a small change in fuel loading can compensate for the performance. Therefore, from a reliability standpoint, the system will fail only if gross changes occur in the conductivity of the insulation.

The conductivity of the insulation will vary with gas pressure. The magnitude of the variation is not known at this time. If the conductivity variation with gas pressure and the gas pressure variation with time were known, the system could be analytically evaluated from a reliability standpoint. However, a reliability alternative for evaluating the insulation does exist, although it is costly. That alternative is to order a large number of insulation packages and test them under simulated operating conditions. Then from the initial distribution of the con-

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Fig. 7. System Reliability Versus Unit Failure Rate for System of 16 Thermoelectric Couples for Five Years Operation

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ductivities and their change with time, the probability that the package will provide the necessary insulation characteristics through the operating life can be determined.

The 16-couple module assembly is the next area where the vacuum to end of life is critical. The same comments made for the insulation package are pertinent here, except that instead of a change of conductivity, the parameter of interest is loss of axial load on the thermoelectric elements due to the loss of the gas pressure. The internal pressure is a function of three things:

- (1) Initial vacuum.
- (2) Leak rate of the container.
- (3) Outgassing of materials internal to the container.

Again, if the container cannot be designed to maintain vacuum on the basis of known behavior, then short and long term development testing on the container is necessary to determine the distribution of pressure changes. The thermal insulation and other parasitic heat conductors are assumed to have a thermal conductivity which is time independent under the operating conditions for five years. This hypothesis should be tested. If found invalid, it will be necessary to design the module so that either: (1) the behavior of the parasitic heat conductors at operating stresses is known as a function of time and margins of safety are incorporated into the thermal design of the 16-couple module assembly; or, (2) the distribution of parasitic heat conduction as a function of time should be evaluated on the basis of appropriate tests and the results incorporated into the design. The 16-thermocouples in series connection can tolerate no catastrophic failure without system failure. Since they are series connected, the probability of success of all 16 thermocouples for five years is the probability of success of one thermocouple for five years raised to the sixteenth power. Each of the thermocouples has four different thermoelement materials, at least six different bonds, two shoes and possible additional parts and bonds if barriers are required. The probability of success of an individual thermoelement is the product of the probabilities of success of each material, each bond and both shoes. The probability of success of all 16 thermocouples for five years is the product of all the probabilities of success making up one element all raised to the sixteenth power.

From the graph (Fig. 7), it can be seen that a failure rate of about 0.03%/1000 hr is required to obtain a success probability of 0.8. This represents the failure rates of each of the separate items comprising the unit. It can also be seen that this failure rate has been shown to be achievable in the past.

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Since the elements are connected in series from a reliability standpoint, no open circuit conditions can occur without system failure. However, a certain amount of degradation may be tolerable. The probability of success for the module over five years is the joint probability of no open circuits in five years and the probability that the degradation is not excessive in five years.

From Fig. 7, it can be seen that if the thermocouples were in reliability parallel and some failure was tolerable without system failure, then the system would have a higher reliability for the same thermocouple failure rate; or a higher thermocouple failure rate is permissible for the same system reliability. For the SNAP 21A program, the series arrangement is necessary to achieve minimum working voltage for the dc-to-dc converter of the required efficiency. It appears that power supplies with much greater reliability can be attained for larger power units. There are two reasons for this. First, larger power units permit redundancy which permits catastrophic failures of individual thermocouples without destroying the system. In turn, this provides greater reliability. Second, if the power is large enough so the thermocouples may be electrically series connected to achieve the desired voltage, the dc-to-dc converter may be eliminated. Naturally, the best reliability condition is that which both eliminates the converter and incorporates redundancy. This refers probably to systems of more than 100 watts with a 28-volt output.

The dc-to-dc converter is essentially a conventional piece of electronics hardware. Similar converters have been reliability analyzed by conventional electronics reliability techniques for other nuclear powered thermoelectric generators and a five-year reliability of from 0.85 to 0.90 resulted. This appears to be a realistic success probability range for the SNAP 21A generator, also.

Cable and connector reliability form a series reliability link with the rest of the system. Although underwater cable and connectors are known to be feasible, the reliability of these items on the SNAP 21A cannot be determined.

The only place where redundancy is incorporated is in the external pressure seals. However, the redundancy provision is not completely foolproof, and, therefore, the design should be tested to verify that the seal is initially good.

The internal metals compatibility reliability represents the probability that the success of the mission for five years is not impaired by time dependent changes in the properties of the principal heat path or by time dependent changes in the chemistry of individual materials. Generally, these problems are black or white in the sense that if there is a problem, it is severe enough to be revealed in short term testing.

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For reliability purposes, it will be assumed that prior knowledge and results from testing during Phase 1 will be sufficient to assure a reliability of one for the principal heat path and for materials compatibility during subsequent phases.

This evaluation indicated areas in which reliability evaluation is necessary. The problems indicated can be regarded as risk areas. The success of the power supply for five years is directly proportional to reliability achieved in the design of each of these risk areas.

As a general guide for the establishment of reliability objectives, Fig. 8 is presented. It is applicable to series reliability systems with five-year missions assuming constant failure rates. To use it, first determine the reliability objective desired, i.e., the probability that the system will meet its performance objectives throughout the five-year period without failure. From this, read over to the line and down to determine the system failure rate required to meet this objective. While this graph is strictly applicable only to series reliability systems, it is useful in determining a first estimate of system failure rate objectives. As a generalized conclusion, it can be stated that for probabilities of greater than 0.65 of operating without failure for five years, reliability programs are highly desirable. For probabilities of success in excess of 0.95, stringent reliability control is necessary.

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Fig. 8. Probability That Given System Will Meet or Exceed Performance Requirements at all Times Within Five-Year Mission Versus System Failure Rate

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III. COMPONENT DEVELOPMENT*

A. OBJECTIVE

The objective of the component development effort for the SNAP 21A program is to design the major components of the system and have these components released through detailed engineering drawings and specifications for fabrication and test to prove the performance characteristics. This effort covers the thermoelectric couple and module, the insulation system and the pressure vessel and electrical feedthrough connectors.

During the past quarter, three 6-couple module test boxes were designed and built. Each module box will allow testing of two six-couple thermoelectric modules in series. The test specifications for the module tests were written and the following information will be obtained.

- (1) Operational data at the end-of-life hot junction temperature of 950° F.
- (2) Operational data at temperatures above 1000° F.
- (3) Data on couple material combinations to aid in the selection of the best couple materials.

For the six-couple module test, each element leg is instrumented so that the behavior of individual legs may be studied to see if one leg or the other is consistently good or bad in its behavior. The evaluation of the material combinations tested in the six-couple module boxes will allow a selection of the two best combinations which will be tested further in the 16-couple test series.

The design details of the prototype 16-couple module (printed circuit type) were also completed during the quarter, except for couple material combinations to be studied.

B. THERMOELECTRIC MATERIALS

During the proposal phase of the SNAP 21A system, the initial generator concept called for double or triple segments for the elements with the following materials.

*Contributors: D. Evens, A. Gregory, J. Thomas.

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	<u>N Element</u>	<u>P Element</u>
Cold end	BiSeTe	BiSbTe
Hot end	PbTe or PbSnTe for both	

After evaluation of the available data on Martin formulations of lead telluride and lead tin telluride, it was decided that a two-segment element would be superior to a three-segment element. The reason was that the data showed a second lead telluride or lead tin telluride segment would not add enough to the couple output to compensate for the required increase in the number of junctions and, hence, joint resistances.

On the basis of the figure-of-merit data, lead tin telluride doped with lead iodide was chosen as the N leg hot segment and lead telluride doped with sodium was chosen for the P leg hot segment.

Development of the best combination of materials for the thermoelectric couple was planned in three phases. First, elevated temperature furnace aging tests were to be used to evaluate various combinations of bond and shoe materials. Second, a series of three 6-couple module tests was planned. Each of these three tests will be run in a single module test box which will contain two independent six-couple modules. The outputs of the two modules will be connected in series and couples of each module will be instrumented to permit individual monitoring of the N and P legs. Based on the results of the furnace aging tests, changes will be made in bond and shoe materials among the couples in each of the three module test boxes, with possibly a single change between each of the two 6-couple modules in a single box. This should provide sufficient redundancy to give a good comparison of performance among the different types of couples. Third, six 16-couple pancake modules would be built. Four would use the printed circuit concept and two the springs and piston alternate design. The final design of the operating generator module will be decided upon after this test series. These modules will contain at most two modifications of the thermoelectric couple with the chosen materials configurations based as far as possible on data accumulated during the first two phases of the module development program. The best of these configurations will be chosen for integration with the other components of the generator.

C. INITIAL JUNCTION MATERIAL TESTS

Initial studies of materials for lead telluride to bismuth telluride bonds, lead telluride to shoe bonds and shoes were conducted by furnace aging samples under a purified argon atmosphere with periodic measurements made of room temperature bond and element resistance and

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Seebeck output. This type of test differs from actual generator operation in its effect on the elements in two important respects.

First, elements in a generator are under compressive load while no such load exists in a furnace aging test, and, second, elements in a generator operate under a steady-state temperature distribution while samples in a furnace aging test are submitted to a weekly thermal cycle to permit electrical measurements to be made. Consequently, electrical data obtained from these tests cannot be considered as absolute design data. However, as a comparative test between different types of samples, this test has proven valuable.

In the investigation of the lead telluride to shoe bonds, the following material combinations were investigated.

a. Shoe materials

- (1) FerroVac E--This is the only ferrous material known to be nondoping to lead telluride.
- (2) Type 301 stainless steel--This material was chosen because it has the closest thermal coefficient of expansion match to lead telluride among common ferrous base materials.
- (3) Haynes 25.
- (4) Roll-bonded laminates of FerroVac iron-Type 301 Stainless Steel and nickel-iron-stainless steel. These materials were tested to see if a thin, nondoping, iron cladding could be made to take the more favorable thermal expansion characteristics of Type 301 Stainless Steel.

b. Bond materials

- (1) Nickel--This material was tested on N type elements only and was applied as a plated coating on the shoe, as a roll-bonded coating on the shoe and as loose foil.
- (2) Tin
- (3) GeTe
- (4) SnTe

Thermoelectric materials in both cast and hot pressed forms were investigated. The N type material was $\text{Pb}_{0.85}\text{Sn}_{0.15}\text{Te}$ and the P type material was PbTe. The furnace aging tests were run at 975° F under

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argon with weekly measurement of room temperature junction and element resistivities and Seebeck output between 975° and 115° F.

The following conclusions were drawn from the data obtained from these tests:

- (1) Initial (as-bonded) junction resistivities for junctions made with nickel were generally higher than those made with tin on N type elements.
- (2) The N type elements bonded to Type 301 Stainless Steel with tin have lower junction resistivities and unbond far less frequently than those bonded to iron. This is attributed to the better match in thermal coefficient of expansion between austenitic stainless steel and lead telluride.
- (3) Bonds to cast elements are generally comparable in resistance to bonds to hot pressed elements for the same shoe materials and bond materials.
- (4) GeTe, although it forms some bonds of fairly low initial resistance, it forms a hard bond which causes a large percentage of the samples to crack through the element after one or two thermal cycles. Those samples which do not fail show large increases in junction resistivity with time which are probably due to micro cracks forming in the end of the element. Changes in the amount of bonding material and in the bonding heat treatment have been ineffective in preventing this. SnTe has been little, if any, improvement over GeTe.

In the investigations of lead telluride to bismuth telluride bonds the following low melting alloys were investigated as possible bonding materials.

- (1) Bismuth
- (2) Indium
- (3) Tin
- (4) Cadmium
- (5) 60% Bi--40% Cd
- (6) 80% Bi--20% In.

Samples both with and without an iron diffusion barrier between segments were investigated. The samples were aged at 390° F (the

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approximate operating temperature of the segment interface) in argon with weekly measurements of element and junction resistivity. Seebeck output of the composite was also measured at the beginning of test and at the end of four weeks. The results of these tests showed that the lowest resistance joints were, for the N leg, an iron barrier bonded to the $\text{Pb}_{0.85}\text{Sn}_{0.15}\text{Te}$ segment with tin and to the BiSeTe segment with bismuth; and for the P leg, bismuth solder with no diffusion barrier. The Seebeck outputs of the composites remained constant over the four-week aging period indicating no evidence of deleterious cross-doping between segments.

D. MATERIALS FOR FIRST SIX-COUPLE MODULE TESTS

The test results summarized led to the selection of the materials combination diagrammed in Fig. 9 for the first six-couple module tests. Because of the inconclusiveness of the data as to the ultimate merits of Sn and GeTe for the P leg hot shoe bond, one 6-couple module will be fabricated using each material. Except for this one material difference, the couples in the first two 6-couple module tests will be identical. The fabrication of these couples is presently nearing completion.

Fig. 9. Couple for First Six-Couple Module Tests

Fig. 10. Figure-of-Merit for N-Type Materials

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E. MATERIALS FOR SECOND SIX-COUPLE MODULE TESTS

While aging tests were being run and couples for the first six-couple module tests being fabricated, accurate figure-of-merit measurements were made on the bismuth telluride supplied by Melcor for the cold end segments. These data were not supplied with the material, since it is Melcor practice to supply only room temperature measurements of Seebeck coefficient and resistivity. The Seebeck coefficient and electrical resistivity were measured over the range 25° to 225° C, and the material thermal conductivity was calculated from the measured resistivity and the temperature by the Wiedemann-Franz-Lorenz relation. The resulting figure-of-merit curves are shown in Figs. 10 and 11, along with the figures of merit of the PbTe materials being used on the hot end segments. These graphs show that although BiSbTe makes a real contribution to the P leg, BiSeTe makes only a marginal contribution to the N leg. Consequently, aging test samples were run which consisted of two segments of hot pressed $\text{Pb}_{0.85}\text{Sn}_{0.15}\text{Te}$ at different doping levels. Figure 10 shows that the figure of merit of this composite is at least as good as a $\text{Pb}_{0.85}\text{Sn}_{0.15}\text{Te}$ -BiSeTe composite and may even be better for the particular combination of hot and cold junction temperatures being used in SNAP 21A. Results of the aging tests showed that the junction resistance between the differently doped segments of $\text{Pb}_{0.85}\text{Sn}_{0.15}\text{Te}$ was practically negligible and that the Seebeck output remained constant for four weeks, apparently indicating no need for a diffusion barrier between segments. Therefore, this type of N leg will be used in the second set of six-couple module tests. A diagram of the complete couple is shown in Fig. 12.

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Fig. 11. Figure of Merit for P-Type Materials

Fig. 12. Couple for Second Six-Couple Module Tests

IV. POWER CONDITIONER DEVELOPMENT*

The SNAP 21A system has a 24-volt dc output requirement. This is achieved by taking a two-volt dc input from the thermoelectric package and stepping it up to the 24-volt dc output with a dc-to-dc converter. The converter is used because to obtain 24 volts directly from the thermoelectric package would require a module with too many couples and, hence, a very inefficient system. The converter will lose some power during operation but this package is usually around a 90% efficient system.

The preliminary converter design has been tested during the past quarter and has found to be 87.8% efficient. This efficiency will be increased to 90% in the packaged system by optimum component matching. A study of the concept of varying both current and voltage to maintain a constant power and cool the beginning of life hot junction temperature was started during the quarter. It was decided that the concept is feasible and the unit will be breadboarded in a future quarter to demonstrate this capability.

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V. SAFETY ANALYSIS AND TEST*

The first safety studies of the SNAP 21A unit were initiated late this quarter with the following items being investigated:

- (1) Tantung corrosion and oxidation rates.
- (2) Fuel shielding integrity under 6 g loading.
- (3) Pressure vessel in standard 1700° F fire.

With regard to the SNAP 21A capsule being subjected to a corrosive or oxidizing medium, fuel containment can be guaranteed because:

- (1) The inert atmosphere surrounding the capsule prevents the corrosion or oxidation of the material.
- (2) The inert atmosphere will be guaranteed before resealing the unit after opening.

In the event of the pressure vessel suffering a standard 1700° F fire for one hour, the temperature in the vicinity of the fuel capsule will not exceed 2500° F while the melting point of the Tantung alloy is 5500° F. In addition, the oxidation rate of the alloy at 2500° F is less than 2.43×10^{-4} cm/hr. Therefore, it is obvious that no fuel release will result from the standard one-hour fire.

In a seawater environment, the corrosion rate of Tantung will not exceed 0.001 in./yr. (Actually, from DMIC133, the rate will be less than 0.0005 in./yr.) This value is felt to be very conservative since this is the corrosion rate in a hydrochloric acid solution. Based on this value, the amount of fuel remaining after a 0.156-inch weld penetration is corroded away is approximately 470 curies. (For the 0.0005 in./yr rate, 9.6 curies remain.) Based on complete disappearance of the side walls of the container (0.496 inch thick) the fuel remaining is 0.1 curie. In addition, further containment is offered by the stainless steel capsule liner surrounding the fuel.

The analysis of the fuel capsule and attached shield plug under a 6 g loading condition showed the holddown bolts and the aluminum plate were acceptable and the shielding of the capsule can be guaranteed. The safety analysis can be found in Appendix E.

The safety criteria definition effort was just started during this period. This criteria will be developed during the next quarter.

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APPENDIX A

SNAP 21A THERMOELECTRIC OPTIMIZATION ANALYSIS

The thermoelectric analysis for the module of the SNAP 21A conversion system was performed by a digital program* developed by Martin. The program input includes conversion material properties, element lengths, electrical contact resistivities and hot and cold junction temperatures. The program output includes thermoelectric efficiency and number and size of elements for a desired electrical output power.

The segmented couples in the SNAP 21A module consist of lead telluride and bismuth telluride on the P leg and lead tin telluride and bismuth telluride on the N leg. The couples are designed so as to maximize the efficiency obtained from each element by utilizing each material over its optimum temperature range. This range is determined from figure of merit versus temperature curves (Figs. A-1, A-2). These curves were derived from the thermoelectric data curves (Figs. A-6 through A-14). The temperature range that provides the maximum area under the curve is optimum. The segmented couple arrangement is shown in Fig. A-3. The temperatures at which the figure of merit curves intersect (T_M) giving the optimum overall range were determined to be 385° and 475° F for the N and P elements, respectively. The thermoelement segment lengths were calculated by the relation:

$$\frac{\ell_2}{\ell_1} = \frac{k_2}{k_1} \frac{(T_M - T_C)}{(T_H - T_M)}$$

and

$$\ell_1 + \ell_2 = 0.5 \text{ inch}$$

where

T_C = cold junction temperature = 80° F

T_H = hot junction temperature = 950° F

*"Development of a Thermoelectric Optimization Code for IBM 709,"
MND-P-2372, Martin Company

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Fig. A-1. Figure of Merit for P-Type Thermoelectric Materials

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Fig. A-2. Figure of Merit for N-Type Thermoelectric Materials

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Fig. A-3. Segmented Couple Arrangement

- T_M = segment junction temperature
 = 475° F for P element
 = 385° F for N element
 k_1 = average thermal conductivity of PbTe (or PbSnTe for N leg) for temperature range $T_H - T_M$
 k_2 = average thermal conductivity of Bi_2Te_3 for temperature range $T_M - T_C$
 l_1 = length of Bi_2Te_3 segment
 l_1 = length of PbTe segment (or PbSnTe segment for N leg)

Using this relationship, a parametric study was made with hot junction temperature parameters of 900°, 950° and 1000° F, cold junction temperature parameters of 60°, 80°, and 100° F, unadjusted contact resistivities of 140 and 260 microhms/cm² and contact resistivities adjusted for temperature and life degradation of 420 and 780 microhms/cm² for the N and P elements in both cases, respectively. These parameters and the Seebeck voltage, resistivity and conductivity for the materials were used as code inputs. The resulting output efficiencies are shown in Figs. A-4 and A-5. Table A-I gives the detailed output for the design point.

TABLE A-I

SNAP 21A Module Performance Characteristics

Junction temperatures (° F)	
Hot	950
Cold	80
Contact resistivities ($\mu\Omega/cm^2$)	
N leg	420
P leg	780
Thermoelectric efficiency (%)	10.2

Fig. A-4. Thermoelectric Efficiency Versus Hot Junction Temperature

Fig. A-5. Thermoelectric Efficiency Versus Hot Junction Temperature

Fig. A-6. Seebeck Coefficient

Fig. A-7. Resistivity

Fig. A-8. Thermal Conductivity

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Fig. A-9. Seebeck Coefficient

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Fig. A-10. Resistivity

Fig. A-11. Conductivity

Fig. A-12. Seebeck Coefficient

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Fig. A-13. Resistivity

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Fig. A-14. Conductivity

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TABLE A-I (continued)

Load

Voltage (volts)	2
Current (amp)	6
Resistance	0.333
Open circuit voltage (watts)	3.508
Internal resistance	0.251
Couples (No.)	16
Output power (watts)	12
Diameter (in.)	
N element	0.297
P element	0.328

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APPENDIX B

SNAP 21A GENERATOR THERMAL ANALYSIS--END OF DESIGN LIFE

A. TEMPERATURE PROFILE CALCULATIONS--HOT SIDE

The hot junction temperature of the segmented module is 950° F. A temperature drop through the hot shoe, mica, electrical insulation, module seal bellows and the related interfaces is assumed from experience to be approximately 50° F, making the temperature on the hot side of the module diaphragm 1000° F. The temperature drop across the nitrogen-filled gap between the module and the shield plug is given by the equation:

$$Q = \frac{kA (T_1 - T_2)}{l} + \frac{\sigma A (T_1^4 - T_2^4)}{\frac{1}{e_1} + \frac{1}{e_2} - 1}$$

where

k = 0.035 Btu/hr-ft-° F (thermal conductivity of nitrogen)

A = cross sectional area of gap
= 0.165 ft²

l = length of gap = 0.00833 foot

T_1 = uranium shield plug surface temperature

T_2 = module diaphragm temperature = 1000° F

σ = Stefan-Boltzmann constant

= 0.1714 x 10⁻⁸ Btu/hr-ft-° F

e_1 = emissivity of uranium shield plug = 0.8

e_2 = emissivity of stainless steel bellows plate = 0.6

Q = 440 Btu/hr

Solving by trial and error gives T_1 = 1155° F.

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The temperature drop through the upper large portion of the uranium shield plug is given by the equation:

$$\Delta T = \frac{l Q}{k A}$$

where

l = length of plug portion = 0.133 foot

k = thermal conductivity of uranium
= 22 Btu/hr-ft-° F

A = cross sectional area of upper shield plug
= 0.223 ft²

Q = 469 Btu/hr

Then through the lower portion of the shield plug that surrounds the fuel capsule cap the same equation is used where:

l = length of lower shield plug = 0.075 foot

A = cross sectional area of lower shield plug
= 0.060 ft²

Q = 486 Btu/hr

The temperature drop calculated with these equations is 40° F which results in a shield plug temperature at the fuel capsule interface of 1195° F. Since the capsule is screwed into the shield plug a reasonably tight interface is maintained and a temperature drop of 20° F is assumed. The temperature drop across the fuel capsule top is given by

$$\Delta T = \frac{Q l}{k A}$$

where

Q = 486 Btu/hr

l = thickness of fuel capsule top = 0.8 inch = 0.067 foot

A = cross sectional area of top = 0.067 ft²

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k = thermal conductivity of tantalum
= 40 Btu/hr-ft-° F

This gives a ΔT of 12° F and makes the temperature on the inside of the capsule 1227° F. The temperature drop across a helium-filled gap of approximately 0.010 inch between the capsule and the fuel must be determined and then with the drop through the fuel itself, the fuel centerline temperature may be determined. The temperature drop through the gap is caused by a combination of heat radiation and conduction and is given by:

$$Q = \sigma F A (T_1^4 - T_2^4) + \frac{kA}{l} (T_1 - T_2)$$

where

Q = 486 Btu/hr

F = 0.5

$\tau = 0.1714 \times 10^{-8}$ = Stefan-Boltzmann constant

A = cross sectional area = fuel area (fuel diameter
= 2.56 inch)

k = conductivity of helium at 1200° F
= 0.18 Btu/hr-ft-° F

l = 0.010 inch

T₂ = 1227° F

Trial and error gives a ΔT of approximately 12° F. The final drop through the fuel is given by

$$\Delta T = \frac{Sr^2}{4k}$$

where

S = heat production/unit volume
= 8.6×10^4 Btu/hr-ft³

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r = fuel radius = 1.28 inches

k = thermal conductivity of SrTi_2O_3

= 2.8 Btu/hr-ft-° F

This equation gives an 88° F drop to the center of the fuel, making this centerline temperature 1327° F.

B. COLD END TEMPERATURE PROFILE

The SNAP 21A generator will operate in 28° to 41° F water. For analysis, 40° F was chosen as the ambient temperature. To determine the temperature drop from the generator pressure vessel skin to the water, free convection heat transfer was assumed, and the coefficient is given by

$$h = \frac{N_{NUF}}{l}$$

where

h = heat transfer coefficient

$N_{NUF} = 0.56 (N_{GRF} \cdot N_{PRF})^{0.25}$
(for free convection from horizontal plate facing upward)

N_{GRF} = Grashof number

N_{PRF} = Prandtl number

k = conductivity of water at film temperature

l = length of plate

Since the great majority of the heat is rejected from the pressure vessel dome, this region was considered as a horizontal plate for the best heat transfer coefficient determination and h was calculated as $11.1 (\Delta T)^{1/4}$. The Grashof and Prandtl numbers and the thermal conductivity of water are dependent on the film temperature which is unknown until the generator skin temperature is determined. However, by trial and error within the expected range, values may be established for these variables.

Because of the difficulty in accurately establishing a temperature gradient within the pressure vessel dome and, therefore its effective heat rejector area, a mathematical model (Fig. B-1) was established for the cold end and the pressure vessel was treated as fins that rejected heat from one side only from an area equal to that of the pressure vessel. The bottom half of the pressure vessel was accounted for by long inefficient fins having the same area as the sides of the pressure vessel. The efficiencies of these model fins were determined by the relation

$$\eta = \frac{\tanh(n)}{n}$$

where

$$n = \ell \sqrt{\frac{h}{kY_0}}$$

ℓ = length of fin

h = convective heat transfer coefficient

k = thermal conductivity of titanium

Y_0 = one-half fin root thickness

The efficiencies were calculated to be 40.6% for the pressure vessel dome and 26.3% for the sides. The temperature drop is then given by the equation

$$\Delta T = \frac{Q}{\eta_1 A_1 h_1 + \eta_2 A_2 h_1}$$

where

Q = 476 Btu/hr

η_1 = 40.6% = efficiency of pressure vessel dome

A_1 = area of pressure vessel dome = 1.57 ft²

η_2 = efficiency of pressure vessel sides = 26.3%

A_2 = area of pressure vessel sides = 1.61 ft²

h = 16 Btu/hr-ft-°F = convective heat transfer coefficient

The ΔT is then calculated to be approximately 20° F. The only way to decrease this temperature drop to about 10° F is to increase the heat transfer area. The equation

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Fig. B-1. Cold End Mathematical Model

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$$Q = hA_2 \eta_2 \Delta T$$

where

$$h = 19.76 \text{ Btu/hr-ft-}^\circ\text{F} = \text{convective heat transfer coefficient}$$

$$A_2 = \text{area of pressure vessel sides} = 1.61 \text{ ft}^2$$

$$\eta_2 = \text{efficiency of pressure vessel sides} = 27.6\%$$

$$\Delta T = 10^\circ \text{ F},$$

gives 88 Btu/hr rejected by the sides of the pressure vessel leaving 388 Btu/hr to be dumped from the pressure dome. From the same equation

$$Q = hA_1 \eta_1 \Delta T$$

where

$$h = 19.76 \text{ Btu/hr-ft-}^\circ\text{F} = \text{convective heat transfer coefficient}$$

$$A_1 = \text{area of pressure vessel dome} = 1.57 \text{ ft}^2$$

$$\eta_1 = \text{efficiency of pressure vessel dome} = 43\%$$

$$\Delta T = 10^\circ \text{ F}$$

the heat rejected from the pressure vessel dome is 134 Btu/hr. The additional area to be supplied by the fins on the dome must reject 254 Btu/hr. From the same equation

$$Q = hA_1 \eta_1 \Delta T$$

where

$$Q = 254 \text{ Btu/hr}$$

$$h = 12 \text{ Btu/hr-ft-}^\circ\text{F} = \text{convective heat transfer coefficient for fins}$$

$$\Delta T = 10^\circ \text{ F},$$

it was calculated that the product ηA , the effective fin heat transfer area, must necessarily be 2.12 ft^2 . By trial and error, it was determined that approximately 4 ft^2 with an efficiency of 52% (approximately 2.5-inch fins) would supply the necessary heat transfer area. These fins were then included in the design.

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The temperature drop through the titanium pressure vessel dome is given by

$$\Delta T = \frac{Q \ln(r_2/r_1)}{\ell 2\pi k}$$

where

$$Q = 476 \text{ Btu/hr}$$

$$\ell = 0.571 \text{ foot} = \text{length of mathematical model fin}$$

$$\ln r_2/r_1 = 0.207$$

$$k = \text{thermal conductivity of titanium} = 10 \text{ Btu/hr-ft-}^\circ\text{F}$$

Solution of this equation gives 12.6°F , making the temperature at the titanium-copper cold plate interface 62.6°F . To minimize the interface temperature drop, a cone-wedge cold plate arrangement was designed to allow a large interface contact area in which aluminum grease will be smeared. Through this small space, linear conduction was assumed and the equation

$$\Delta T = \frac{Q\ell}{kA}$$

where

$$Q = 476 \text{ Btu/hr}$$

$$\ell = \text{length of gap} = 0.1 \text{ inch} = 0.0083 \text{ foot}$$

$$k = \text{conductivity of aluminum grease} = 10 \text{ Btu/hr-ft-}^\circ\text{F}$$

$$A = 80\% \text{ of contact area} = 0.164 \text{ ft}^2$$

shows a ΔT of 2.4°F . All the values used included safety factors: 0.1 inch is the maximum gap possibility; the conductivity of aluminum grease is expected to be higher than $10 \text{ Btu/hr-ft-}^\circ\text{F}$; 80% of the contact area was used to allow for the possibility of radiation gaps due to uneven grease distribution. For additional conservatism, this 2.4°F value is doubled, making the temperature on the copper cone 67.4°F . The remaining temperature drop is through the copper configuration. This drop may be found by the relation

$$\Delta T = Q \left[\frac{\ell_1}{kA_1} + \frac{\ln r_2/r_1}{2\pi k \ell_2} + \frac{\ln r_5/r_2}{2\pi k \ell_3} + \frac{\ln r_4/r_3}{2\pi k \ell_4} + \frac{\ell_5}{kA_2} \right]$$

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$$= Q \left\{ \frac{\ell_1}{kA_1} + \frac{\ell_5}{kA_2} + \frac{1}{2\pi k} \left[\frac{\ell \ln r_2/r_1}{\ell_2} + \frac{\ell \ln r_3/r_2}{\ell_3} + \frac{\ell \ln r_4/r_3}{\ell_4} \right] \right\}$$

where

$$Q = 476 \text{ Btu/hr}$$

$$k = \text{thermal conductivity of copper} = 226 \text{ Btu/hr-ft-}^\circ\text{F}$$

$$\ell_j = \text{length of copper sections}$$

$$r_j = \text{radius of copper sections}$$

$$A_1 = \text{cross sectional area of main copper plate section}$$

$$A_2 = \text{representative cross sectional of cone-wedge copper section}$$

The first term refers to the linear heat flow from the module up through the adjacent copper section, the next three terms apply to the radial heat flow through the three copper stepped sections extending radially from this first copper section, and the last term applies to linear heat flow up through the copper wedge. According to this relation, the temperature drop is approximately 12.5° F and the cold junction is then 79.9 or 80° F.

C. HEAT LOSSES (PARASITIC)

The parasitic heat losses from the SNAP 21A generator can be isolated in three major components. The Min-K 1301 insulation in the module, into which the thermocouples are inserted, is in parallel with the thermoelements; within the same temperature gradient, the heat lost through this area is given by

$$Q = k\Delta T \left[\frac{\pi W_1^2}{4} - A_E/\ell \right] = 37.5 \text{ Btu/hr} = 11 \text{ watts}$$

where

$$k = \text{thermal conductivity of Krypton-filled Min-K 1301 at 1/2 atmosphere} = 0.017 \text{ Btu/hr-ft-}^\circ\text{F}$$

$$\Delta T = 870^\circ$$

$$\ell_1 = \text{element length} = \text{Min-K length} = 0.5 \text{ inch}$$

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$$A_E = \text{element area} = 2.46 \text{ in.}^2$$

$$W_1 = \text{diameter of module} = 4.75 \text{ inches}$$

The stainless steel module bellows which encases the module package is a thin stainless steel member also in thermal parallel with the elements. The heat loss here is given by

$$Q = \frac{kA \Delta T}{\ell} r = 27.2 \text{ Btu/hr} = 7.9 \text{ watts}$$

where

$$k = \text{thermal conductivity of stainless steel} = 10 \text{ Btu/hr-ft-}^\circ\text{F}$$

$$A = \text{bellows cross sectional area} = \pi Dt$$

$$D = \text{diameter of bellows} = 5.125 \text{ inches}$$

$$\Delta T = 1020^\circ \text{ F}$$

$$t = \text{bellows thickness} = 0.005 \text{ inch}$$

$$\ell = \text{extended length of bellows} = 3 \text{ inches}$$

$$r = \text{radiation factor} = 1.2$$

The Min-K ring which holds down the shielding is also a path through which heat travels and this loss is given by

$$Q = \pi k \frac{[D_2^2 - D_1^2]}{4\ell} \Delta T = 10.4 \text{ Btu/hr} = 3 \text{ watts}$$

where

$$k = \text{thermal conductivity of dry N}_2\text{-filled Min-K} = 0.02 \text{ Btu/hr-ft-}^\circ\text{F}$$

$$D_2 = \text{outside diameter of Min-K ring}$$

$$D_1 = \text{inside diameter of Min-K ring}$$

$$\ell = \text{length of ring}$$

$$\Delta T = 1000^\circ \text{ F}$$

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The remaining heat loss is found through the Linde superinsulation system. The system may be divided into two parts, insulation and seal bellows, and the insulation may be divided into four configuration parts, one load-bearing and three nonload bearing areas (Fig. B-2). The insulation losses are given by

$$Q = \frac{2\pi k_1 \ell_1 \Delta T_1}{\ln r_2/r_1} + \frac{k_2 A_1 \Delta T_2}{\ell_2} + \frac{2\pi k \ell_3 \Delta T_3}{\ln r_4/r_3} + \frac{k_1 A_2 \Delta T_4}{\ell_4}$$

where

k_1 = thermal conductivity of nonload bearing Linde insulation = 0.00046 Btu/hr-ft-°F

k_2 = thermal conductivity of load-bearing Linde insulation = 0.00190 Btu/hr-ft-°F

r_1, r_3 = inside radii of vertical insulation sections

r_2, r_4 = outside radii of vertical insulation sections

ℓ_1, ℓ_3 = lengths of vertical insulation sections

ℓ_2, ℓ_4 = thickness of horizontal insulation sections

$\left. \begin{array}{l} \Delta T_1 \\ \Delta T_2 \\ \Delta T_3 \\ \Delta T_4 \end{array} \right\}$ For location see Fig. B-2

The heat loss in this area is calculated to be slightly less than 10 watts or about 9.7 watts. (Linde calculations = 11.5 watts.) The critical heat loss within the system is through the stainless steel bellows. The loss is a combination of conduction through the bellows and then radiation from them and radiation from the shield plug. A shim of Min-K has been inserted above the bellows at the top of the radiation path, making a radiation factor of 1.2 sufficient. The equation, therefore, for this heat loss is

$$Q = \frac{kA \Delta T}{\ell} \quad r = 40 \text{ Btu/hr} = 11.7 \text{ watts}$$

where

k = thermal conductivity of stainless steel = 10 Btu/hr-ft-°F

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Fig. B-2. Linde Insulation

A = bellows cross sectional area = πDt

D = 6.625 inches

ΔT = 1000° F

r = radiation factor = 1.2

l = extended length of bellows = 2.6 inches

t = 0.0005 inch

The total heat loss is given in Table B-I. The total heat loss is approximately 43 watts, making the fuel loading at end-of-life about 161 watts (117.6 watts to the thermoelements and 43.34 watts to compensate for the heat losses). The thermal efficiency is 73.1%.

TABLE B-I
Parasitic Heat Losses, End-of-Life

	<u>(watts)</u>
Module insulation	11.0
Module bellows	7.9
Fuel block retainer ring	3.0
Vacuum insulation	9.7
Vacuum insulation seal bellows	11.7
Total	<u>43.3</u>

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APPENDIX C

SNAP 21A OFF-DESIGN OPERATION

There are two particular operating cases which vary significantly from the end-of-design-life characteristics: beginning-of-life generation; and operation in 130° ambient air. Because of the isotope decay factor, an excess fuel loading is necessary so that the required output power is available at the end of life.

The equation

$$Q = K_E (\Delta T) + K_I (\Delta T) + NI \alpha_H T_H$$

is utilized to determine the hot junction temperature at beginning-of-life. K_E is calculated to be 0.0904 watt/° R for end-of-design-life conditions, where

$$Q = \text{heat through module} = 136.5 \text{ watts}$$

$$\Delta T = 870^\circ \text{ R}$$

$$K_I = 0.0194 \text{ watt}/^\circ \text{ R}$$

$$N = \text{number of couples} = 16$$

$$I = \text{current} = 5.97 \text{ amperes}$$

$$\alpha_H = |\alpha_N| + \alpha_P = 548 \mu\text{v}/^\circ \text{ C}$$

$$T_H = 1410^\circ \text{ R}$$

Using the same equation, where

$$Q = Q \text{ through module at end of life x radioisotope decay factor} = 154 \text{ watts}$$

α_H and I are temperature dependent.

The hot junction temperature at beginning of life is determined by trial and error to be 1050° F. With present knowledge, it appears that the thermoelectric material and/or hot junction bonds cannot withstand temperatures above 1000° F without deterioration taking place. Therefore, some means of cooling may be necessary. There are two possible solutions: (1) lower the end-of-design-life hot junction temperature to

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900° F or (2) by means of the power conditioner, draw more current at beginning of life, thus cooling the hot junction at that time. The first alternative is undesirable because the smaller ΔT across the thermoelements lowers the thermoelectric efficiency. The second possibility has been investigated thermally and is now being studied electrically. There are two approaches to determine the necessary current or power to be drawn.

In both cases, an E-I curve (voltage versus current) and a power curve are necessary. The module is designed to produce 1.925 volts and six amperes at end-of-life operation at 950° F hot junction temperature. At beginning-of-design-life, with the higher hot junction temperature, the current increases to approximately 6.6 amperes and the voltage increases to about 2.12 volts for a constant external load. This gives the operating points on both the beginning-of-life and end-of-life E-I curves. To establish two more points on the curves, open circuit and short circuit conditions are determined. On open circuit, the equation

$$Q = K_E (\Delta T) + K_I (\Delta T) + NI \alpha T_H$$

becomes

$$Q = K_E (\Delta T) + K_I (\Delta T)$$

because the last term disappears since $I = 0$. Solving this equation for T for the two cases

(1) where $Q = 136.5$ watts

(2) where $Q = 154$ watts

gives hot junction temperatures of approximately 1500° and 1340° F for the open circuit beginning-of-life and end-of-life hot junction temperatures, respectively. The open circuit voltages are 5.55 and 4.925 volts, respectively. On short circuit conditions, the equation

$$Q = K_E (\Delta T) + K_I (\Delta T) + NI \alpha T_H$$

must be solved by trial and error with I (current) determined by the equation

$$I = E_{oc} / R_L + R_E$$

where E_{oc} is temperature dependent and $R_e = 0$, since there is no external resistance. In this case the temperatures are approximately

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850° and 775° F while the currents are 12.5 and 11.35 amperes at these temperatures. With these points, the E-I curves are established and, subsequently, the power curves, where:

$$\text{power} = \text{current} \times \text{voltage}.$$

This is shown in Fig. C-1.

The first approach to determine the current necessary to maintain the hot junction temperature at 1000° F, again utilizes the equation

$$Q = K_E (\Delta T) + K_I (\Delta T) + N I \alpha T_H$$

where

$$K_E = 0.0904 \text{ watt/}^\circ\text{R}$$

$$K_I = 0.0194 \text{ watt/}^\circ\text{R}$$

$$N = 16$$

$$\alpha_H = 550 \mu\text{v/}^\circ\text{C}$$

$$T_H = 1460^\circ \text{R}$$

$$T = 915^\circ$$

$$Q = 154 \text{ watts.}$$

The current necessary for these conditions is calculated to be 7.51 amperes. Going to the previously established E-I curve, the voltage which the power conditioner must draw to produce this current is about 1.75 volts. The power at this point is 13 watts and, though not optimum, is more than is required. The other method for determining the necessary current is graphic. The above power curve is used to obtain the values to plot a beginning-of-life hot junction temperature versus current curve (Fig. C-2). Then from this curve, the current at 1000° F can be determined and is approximately 7.5 amperes. There is a possibility that the power conditioner could maintain a constant power output over the life of the generator. For this purpose, as the power is held constant, the voltage increases and the current decreases with time. This can be seen graphically by going on a constant power line in Fig. C-1 from the beginning-of-life curve to the end-of-life operational point. These various methods, as has been stated, are

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Fig. C-1. SNAP 21A--Power-Voltage-Current

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Fig. C-2. Hot Junction Temperature Versus Current at Beginning-of-Life

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being investigated. In addition, six-couple module tests will be run at temperatures up to and above 1050° F to ascertain whether a problem exists with the thermoelectric couples at these temperatures.

The remaining off-optimum condition is that encountered in shipping, or 130° F ambient air where preliminary analysis shows a tremendous increase in the cold junction temperature. The segment junctions cannot be operated at temperatures above approximately 300° F and since the junctions are designed to operate at 395° and 475° F at end of life, and somewhat higher at beginning of life when shipping would feasibly take place, there is very little temperature leeway. Short circuiting the generator does not appear to be sufficient to remedy the situation and it is likely that some exterior means of cooling such as refrigeration, water cooling or some type of finned shipping cask will be necessary. The possibility of shipping the upper portion of the generator, which contains the module separate from the lower portion which contains the fuel, has been considered. However, this would require on-the-site gas-filling and sealing of the generator, which is undesirable. In any case, this condition will be investigated further and resolved during the remainder of the program.

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APPENDIX D

SNAP 21A STRUCTURAL ANALYSIS

A. FUEL CAPSULE ANALYSIS

The fuel capsule for the SNAP 21A system must be capable of withstanding 10^4 psi at a temperature of $\approx 40^\circ\text{F}$ (in seawater). The capsule is tantalum-10% tungsten which has a negligible corrosion rate (< 0.0005 in./yr) in seawater (Ref. D-1). The following discontinuity analysis is made to determine the stresses in the vessel under the high hydrostatic loading condition.

For the cylinder, the mean radius is

$$R_m = \frac{R_i + R_o}{2} = 1.69$$

and

$$\frac{R_i}{t} = 2.88.$$

This is technically a thick shell since $R_i/t < 10$; however, Bijlaard and Dohrmann (Ref. D-2) have shown that for $R_i/t = 2$, the difference

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between them and thick shell theory is within 5%. Therefore, the shell strip can be treated as a beam on an elastic foundation with a foundation modulus of:

$$k = \frac{Et}{r_m^2}$$

where

E = Young's modulus (psi)

t = wall thickness (in.)

r_m = mean radius (in.).

The shell characteristic is:

$$\beta = \sqrt[4]{\frac{3(1-\nu^2)}{R_m^2 t^2}} = \frac{1.285}{\sqrt{R_m t}} = 1.4/\text{in.}$$

$$\beta l = 1.4 \times 3.27 = 4.58.$$

This will be treated as a long cylinder since for $\beta l \geq \pi$, the influence coefficients will be within 4% for effects of one edge loading on the other edge.

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For the loading shown, it is assumed that the effect of Q on the flat plate is taken into account in a radial shortening only. The effect on the bending behavior is ignored. This is a good assumption as long as Q/Q_{cr} is a small value. This will be checked after solving for the loads.

For the cylinder, the deflection at the edge is

$$\delta_c = \frac{Q}{2\beta^3 D} - \frac{M}{2\beta^2 D} - \frac{P r_m^2}{E t_c} (1 - \nu/2)$$

where

$$D = \frac{E t_c^3}{12 (1 - \nu^2)}$$

M = edge moment (in. -lb/in.)

Q = edge shear (lb/in.)

P = external pressure (psi)

and the rotation is

$$\theta_c = \frac{Q}{2\beta^2 D} - \frac{M}{\beta D}.$$

For the flat head, the deflection (radial) is given by

$$\delta_P = - \frac{Q r_m}{E t_p} (1 - \nu).$$

The edge rotation is given by

$$\theta_P = \frac{M r_m}{D_p (1 + \nu)} - \frac{P r_m^3}{\delta D_p (1 + \nu)}.$$

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Writing the equations as

$$\delta_c = \delta_{c, Q} Q + \delta_{c, M} M + \delta_{c, P} P \quad (1)$$

$$\theta_c = \omega_{c, Q} Q + \omega_{c, M} M \quad (2)$$

$$\delta_P = \delta_{P, Q} Q \quad (3)$$

$$\theta_P = \omega_{P, M} M + \omega_{P, P} P. \quad (4)$$

At the loaded junction, these equations are solved by equating angles and deflections which yield

$$Q (\delta_{P, Q} - \delta_{c, Q}) = \delta_{c, M} M + \delta_{c, P} P$$

$$Q = \frac{\delta_{c, M} M + \delta_{c, P} P}{(\delta_{P, Q} - \delta_{c, Q})}$$

$$\frac{(\omega_{c, Q}) (\delta_{c, M})}{(\delta_{P, Q} - \delta_{c, Q})} M + \frac{(\omega_{c, Q}) (\delta_{c, P})}{(\delta_{P, Q} - \delta_{c, Q})} P$$

$$+ (\omega_{c, M} - \omega_{P, M}) M = \omega_{P, P} P$$

$$M \left[(\omega_{c, Q}) (\delta_{c, M}) + (\delta_{P, Q} - \delta_{c, Q}) (\omega_{c, M} - \omega_{P, M}) \right] =$$

$$\left[\omega_{P, P} (\delta_{P, Q} - \delta_{c, Q}) - (\omega_{c, Q}) (\delta_{c, P}) \right] P.$$

At the 0.69-inch end plate, evaluating constants give

$$D_c = \frac{21 \times 10^6 \times 0.125}{12 (0.91)} = 0.24 \times 10^6$$

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$$D_P = \frac{21 \times 10^6 \times 0.328}{12 (0.91)} = 0.63 \times 10^6.$$

$$\beta D_c = 0.336 \times 10^6$$

$$\beta^2 D_c = 0.47 \times 10^6$$

$$\beta^3 D_c = 0.659 \times 10^6$$

$$\delta_{c, Q} = 0.76 \times 10^{-6}$$

$$\delta_{c, M} = -1.062 \times 10^{-6}$$

$$\delta_{c, P} = -0.231 \times 10^{-6}$$

$$\delta_{P, Q} = -0.0816 \times 10^{-6}$$

$$\omega_{c, Q} = 1.062 \times 10^{-6}$$

$$\omega_{c, M} = -2.98 \times 10^{-6}$$

$$\omega_{P, M} = 2.065 \times 10^{-6}$$

$$\omega_{P, P} = -0.736 \times 10^{-6}$$

Using these constants and solving for M_1 gives

$$M_1 \left[\begin{array}{l} (1.062)(-1.062) + (-0.0816 \\ -0.76)(-2.98 - 2.065) \\ -0.736(-0.0816 - 0.76) \\ -1.062(-0.231) \end{array} \right] P =$$

$$M_1 = 0.278 P$$

$$Q_1 = \frac{(-1.062)(0.278) + (-0.231)}{(-0.0816 - 0.76)} P$$

$$Q_1 = 0.626 P.$$

At the 0.812-inch end plate, evaluating constants give

$$D_P = 1.03 \times 10^{-6}.$$

All cylinder constants remain the same; the only changes are in the end plate constants

$$\delta_{P, Q} = -0.0694 \times 10^{-6}$$

$$\omega_{P, M} = 1.26 \times 10^{-6}$$

$$\omega_{P, P} = -0.45 \times 10^{-6}.$$

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Solving for M_2 and Q_2 yields

$$M_2 \left[(-1.13) + (-0.0694 - 0.76)(-2.98 - 1.26) \right] = \left[(-0.45)(-0.0694 - 0.76) - (1.062)(-0.231) \right] P$$

$$M_2 = 0.259 P$$

$$Q_2 = \frac{(-1.062)(0.259) + (-0.231)}{-0.0694 - 0.76} P$$

$$Q_2 = 0.611 P.$$

Stress determinations:

The axial bending stress is $\sigma_{x_1} = \pm 6 M/t^2$.

<u>Surfaces</u>	<u>Point A</u>	<u>Point B</u>
Outside	+66,700	+62,100
Inside	-66,700	-62,100

The axial membrane stress is $\sigma_{x_2} = -P/1 - \frac{d_i^2}{d_o^2}$.

Outside	-22,300	-22,300
Inside	-22,300	-22,300

The hoop bending stress is $\sigma_{\phi_1} = \nu \sigma_{x_1}$.

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<u>Surfaces</u>	<u>Point A</u>	<u>Point B</u>
Outside	+20,010	+18,630
Inside	-20,010	-18,630

The hoop compression (membrane) is $\sigma_{\phi_2} = -2 M \frac{\beta^2 R}{t}$.

Outside	-36,850	-34,350
Inside	-36,850	-34,350

The hoop tension (membrane) is $\sigma_{\phi_3} = 2 Q \frac{\beta R}{t}$.

Outside	59,300	57,900
Inside	59,300	57,900

The pressure membrane stresses are $\sigma_{\phi_4} = -P \frac{r_o^2}{r^2} \left(\frac{r_i^2 + r^2}{r_o^2 - r_i^2} \right)$.

Outside	-34,600	-34,600
Inside	-45,100	-45,100

The totals for the axial stresses are as follows.

Outside	+44,400	+39,800
Inside	-89,000	-84,400

The totals for the hoop stresses are as follows.

Outside	+7,860	+7,580
Inside	-42,660	-40,180

The equivalent uniaxial stresses can be found from

$$\sigma_{eq} = \sqrt{\sigma_x^2 + \sigma_{\phi}^2 - \sigma_x \sigma_{\phi}}$$

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From inspection, it can be seen that the equivalent uniaxial stress will be less than either of the maximum values shown. For tantalum-10% tungsten, the yield and ultimate strength values are:

$$\sigma_{\text{yield}} = 90,000 \text{ psi at } 1600^{\circ} \text{ F}$$

$$\sigma_{\text{ultimate}} = 95,000 \text{ psi at } 1600^{\circ} \text{ F.}$$

At room temperature, $\sigma_{\text{yield}} = 158,000 \text{ psi}$.

Since the external pressure loading condition occurs at 40° F , no problem is anticipated for the fuel capsule under this loading. During normal operation ($\approx 1500^{\circ} \text{ F}$), the loads on the capsule will never approach these values. Therefore, the capsule is structurally acceptable.

The stresses in the end plates can be found from

$$\sigma_{\phi_{r=0}} = \sigma_{r_{r=0}} = + \frac{6 M_1}{t_P^2} - \frac{3 P r_m^2}{8 t_P^2} (3 + \nu) - \frac{Q}{t_P} =$$

$$+ \frac{P}{t_P^2} [1.668 - 3.535] - \frac{0.626}{t_P} P$$

$$\sigma_{\phi_{r=0}} = \sigma_{r_{r=0}} = -48,280 \text{ psi} = \text{top surface for } 0.69\text{-inch plate.}$$

For the 0.812-inch plate,

$$\sigma_{\phi_{r=0}} = \sigma_{r_{r=0}} = -37,540 \text{ psi} = \text{top surface for } 0.812\text{-inch plate.}$$

These stresses are low and present no problem.

The maximum deflection of the end plates may be found from

$$y_{r=0} = - \frac{M a^2}{2 D (1 + \nu)} + \frac{P a^4 (5 + \nu)}{64 D (1 + \nu)} = \frac{P a^2}{2 D (1 + \nu)}$$

$$\left[\begin{array}{l} -0.259 M_2 \\ -0.278 M_1 \end{array} + 0.473 \right]$$

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$$y_{r=0} = \left[\begin{array}{l} 0.214 \text{ at } 0.812 \text{ in.} \\ 0.195 \text{ at } 0.69 \text{ in.} \end{array} \right] \frac{0.57 \times 10^{-2}}{t^3}$$

$$y_{r=0} = 0.00338 \text{ inch at } 0.69\text{-inch end}$$

$$y_{r=0} = 0.00228 \text{ inch at } 0.812\text{-inch end.}$$

These are negligible and all this loading will only occur if the outer shell is breached.

The thermal stress may be estimated from

$$\sigma_x = \pm 0.353 E \alpha \frac{\Delta T}{t} \sqrt{r_m t.}$$

For a 100° F/in. axial gradient (Ref. D-3)

$$\sigma_x = \pm 8100 \text{ psi (small contribution)}$$

and for a 50° F radial gradient (Ref. D-3)

$$\sigma_r = \sigma_\theta = E \alpha \frac{\Delta T}{2} = 6250 \text{ psi (small contribution).}$$

B. UPPER PRESSURE VESSEL

The upper housing of the generator tends to give the appearance of being built in at the flange because of the attached fins and the rigidity of the lower forging. Therefore, the effect of the fins is neglected and the following loading situation is assumed:

$$h = 1 \text{ inch}$$

$$r_2 = 3.38 \text{ inches}$$

$$\alpha = 15^\circ$$

$$y_1 = 16.68 \text{ inches}$$

$$r_1 = 4.32 \text{ inches}$$

$$y_2 = 13.1 \text{ inches.}$$

It is also assumed that the ends are far enough apart so that the cone may be considered long, i. e., the end effects are of a local nature and do not influence one another.

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At the lower loaded edge, the condition of the built-in support requires that

$$\theta_1 = 0$$

$$\delta_1 = 0$$

where

$$\theta_1 = \text{edge rotation}$$

$$\delta_1 = \text{edge deflection.}$$

From the discontinuity equations for a cone (NASA-TR-R-103)

$$\theta_1 = \omega_{1,M_1} M_1 + \omega_{1,Q_1} Q_1 + \omega_{1,P} P \quad (1)$$

and

$$\delta_1 = \delta_{1,M_1} M_1 + \delta_{1,Q_1} Q_1 + \delta_{1,P} P \quad (2)$$

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where

$$\omega_{1,M_1} = - \frac{m^3}{E h^2} \sqrt{\frac{2 r_1}{h \cos \alpha}} \Omega_1$$

$$\omega_{1,Q_1} = \frac{m^2 r_1}{E h^2} \Omega_2$$

$$\omega_{1,P} = - \frac{m^2 r_1^2 \tan \alpha}{2 E h^2} \Omega_2 - \frac{3 (1 + \nu) \tan^2 \alpha}{m E} \sqrt{\frac{r_1}{2 h \cos \alpha}} \Omega_1 \\ + \frac{3 r_1 \tan \alpha}{2 E h \cos \alpha}$$

$$\delta_{1,M_1} = \frac{m^2 r_1}{E h^2} \Omega_2$$

$$\delta_{1,Q_1} = - \frac{m r_1}{E h} \sqrt{\frac{2 r_1 \cos \alpha}{h}} \Omega_3$$

$$\delta_{1,P} = \frac{m r_1^2 \sin \alpha}{E h} \sqrt{\frac{r_1}{2 h \cos \alpha}} \Omega_3 + \frac{m^2 r_1 \tan^2 \alpha}{8 E (1 - \nu)} \Omega_2 \\ - \frac{(1 - \nu/2) r_1^2}{E h \cos \alpha}$$

and

$$m = \sqrt[4]{12 (1 - \nu^2)}; \quad \xi = 2 \mu_1 \sqrt{y_1}$$

$$\mu_1 = \frac{m}{\sqrt{h_1 \tan \alpha}}$$

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where y_1 is measured from apex of the cone and the Ω functions depend on ξ .

For $\nu = 0.3$

$$m = 1.818$$

$$\mu_1 = 3.51$$

$$\xi = 2 \times 3.51 \sqrt{16.68}$$

$$\xi = 28.7$$

$$\cos \alpha = 0.966$$

$$\tan \alpha = 0.268$$

$$\sin \alpha = 0.259$$

$$E = 15.5 \times 10^6 \text{ psi.}$$

From Fig. 3 of NASA-TR-R-103 at $\xi = 28.7$

$$\Omega_1 = 0.992$$

$$\Omega_2 = 0.990$$

$$\Omega_3 = 0.969.$$

Then substituting values yield:

$$\omega_{1,M_1} = - \left[\frac{(1.818)^3}{15.5 \times 10^6} \right] \left[\sqrt{\frac{2 \times 4.32}{0.966}} \right] [0.992]$$

$$\omega_{1,M_1} = -11.5 \times 10^{-6}$$

$$\omega_{1,Q_1} = \frac{(1.818)^2 (4.32)}{15.5 \times 10^6} (0.990)$$

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$$\omega_{1,Q_1} = 0.911 \times 10^{-6}$$

$$\begin{aligned}\omega_{1,P} = & - \frac{(1.818)^2 (4.32)^2}{2 \times 15.5 \times 10^6} (0.268) (0.99) \\ & - \frac{3 (1.3) (0.268)^2}{(1.818) 15.5 \times 10^6} \sqrt{\frac{4.32}{2 (0.966)}} (0.992) \\ & + \frac{3 (4.32) (0.268)}{2 (15.5 \times 10^6) (0.966)}\end{aligned}$$

$$\omega_{1,P} = -0.4268 \times 10^{-6}.$$

Equation (1) then becomes

$$0 = -1.15 M_1 + 0.911 Q_1 - 0.4268 P.$$

Now for Eq (2)

$$\delta_{1,M_1} = \frac{(1.818)^2 (4.32)}{15.5 \times 10^6} (0.99)$$

$$\delta_{1,M_1} = 0.911 \times 10^{-6}$$

$$\delta_{1,Q_1} = \frac{(1.818) (4.32)}{15.5 \times 10^6} \sqrt{\frac{2 \times 4.32 \times 0.966}{1}} (0.969)$$

$$\delta_{1,Q_1} = -1.418 \times 10^{-6}$$

$$\delta_{1,P} = \frac{(1.818) (4.32)^2}{15.5 \times 10^6} (0.259) \sqrt{\frac{4.32}{2 \times 0.966}} (0.969)$$

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$$+ \frac{(1.818)^2 (4.32) (0.268)^2}{8 \times 15.5 \times 10^6 \times 0.7} (0.99) - \frac{0.85 (4.32)^2}{15.5 \times 10^6 \times 0.966} =$$

$$- 0.2083 \times 10^{-3}.$$

Therefore, Eq (2) becomes

$$0 = 0.911 M_1 - 1.418 Q_1 - 0.2083 P.$$

Solving yields

$$- 1.15 M_1 + 0.911 Q = 0.4268 P$$

$$1.15 M_1 - 1.79 Q_1 = 0.263 P - 0.879 Q_1 = 0.6898 P$$

$$Q_1 = - 0.785 P$$

$$M_1 = - 0.994 P.$$

The longitudinal stress is given by $\left[\begin{array}{l} \text{inner} \\ \text{outer} \end{array} \right] \begin{array}{l} \text{edges} \end{array} \left[\begin{array}{l} + \\ - \end{array} \right]$ symbol indicates sign of

$$\sigma_{y_1} = - \frac{y_1 \sin \alpha \cos \alpha}{2h} P + \frac{\sin \alpha}{h} Q_1 \mp \frac{6}{h^2} M_1 = -20,900$$

$$+ 2030 \mp 59,600$$

$$\sigma_{y_1} = 40,730 \text{ psi at outside edge}$$

$$\sigma_{y_1} = -78,470 \text{ psi at inside edge.}$$

The circumferential stress is found from:

$$\sigma_{\theta_1} = - \frac{3 \tan^2 \alpha}{4 (1 - \nu)} \left[\frac{\frac{1}{6} m^2 A \mp \nu C \mp 2 G}{C + 2 \nu G} \pm 1 \right] P$$

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$$-\frac{y \tan \alpha}{h} \left[1 + \frac{\sin^2 \alpha}{2} \frac{\pm 6 \frac{1 - \nu^2}{2} A - \frac{\xi^2}{2} B - \nu C}{C + 2 \nu G} \right] P$$
$$-\frac{\sin \alpha}{h} \left(\frac{\pm 6 \frac{1 - \nu^2}{2} A - \frac{\xi^2}{2} B - \nu C}{C + 2 \nu G} \right) Q_1 - \frac{6}{h^2}$$
$$\left(\frac{-\frac{1}{6} m^2 A \pm \nu C \pm 2G}{C + 2 \nu G} \right) M_1$$

where

$$A = \xi (\text{ber}'_2 \xi \text{bei}_2 \xi - \text{bei}'_2 \xi \text{ber}_2 \xi)$$

$$B = (\text{ber}'_2 \xi)^2 + (\text{bei}'_2 \xi)^2$$

$$C = \xi (\text{ber}_2 \xi \text{ber}'_2 \xi + \text{bei}_2 \xi \text{bei}'_2 \xi)$$

$$G = (\text{ber}_2 \xi)^2 + (\text{bei}_2 \xi)^2$$

and the recurrence formulas are:

$$\text{ber}_2 \xi = \frac{2 \text{bei}' \xi}{\xi} - \text{ber} \xi$$

$$\text{bei}_2 \xi = -\frac{2 \text{ber}' \xi}{\xi} - \text{bei} \xi$$

$$\text{ber}'_2 \xi = -\text{ber}' \xi - \frac{2 \text{ber}_2 \xi}{\xi}$$

$$\text{bei}'_2 \xi = -\text{bei}' \xi - \frac{2 \text{bei}_2 \xi}{\xi}$$

Now for $\xi = 28.7$, the value of the argument is large, therefore:

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$$\text{ber } \xi \approx \frac{e^{x/\sqrt{2}}}{\sqrt{2\pi x}} \cos\left(\frac{x}{\sqrt{2}} - \frac{\pi}{8}\right)$$

$$\text{bei } \xi \approx \frac{e^{x/\sqrt{2}}}{\sqrt{2\pi x}} \sin\left(\frac{x}{\sqrt{2}} - \frac{\pi}{8}\right)$$

$$\frac{\text{ber}' \xi}{\text{bei}' \xi} \approx \cot\left(\frac{x}{\sqrt{2}} + \frac{\pi}{8}\right)$$

$$\frac{\text{bei } \xi}{\text{bei}' \xi} \approx \frac{\sqrt{2}}{2} \left(1 - \cot\left(\frac{x}{\sqrt{2}} + \frac{\pi}{8}\right)\right)$$

Evaluating:

$$\text{ber } \xi = 2.44 \times 10^7$$

$$\text{bei } \xi = 4.31 \times 10^7$$

$$\text{ber}' \xi = -1.345 \times 10^7$$

$$\text{bei}' \xi = 4.75 \times 10^7$$

Then:

$$\text{ber}_2 \xi = \frac{2 \times 4.75}{28.7} \times 10^7 - 2.44 \times 10^7 = -2.109 \times 10^7$$

$$\text{bei}_2 \xi = + \frac{2 \times 1.345}{28.7} \times 10^7 - 4.31 \times 10^7 = -4.2162 \times 10^7$$

$$\text{ber}'_2 \xi = + 1.345 \times 10^7 + \frac{2 \times 2.109 \times 10^7}{28.7} = 1.492 \times 10^7$$

$$\text{bei}'_2 \xi = -4.75 \times 10^7 + \frac{2 \times 4.2162 \times 10^7}{28.7} = -4.456 \times 10^7$$

and

$$A = 28.7 (-1.492 \times 4.2162 - 4.456 \times 2.109) \times 10^{14}$$

$$A = -4.51 \times 10^{16}$$

$$B = 0.2205 \times 10^{16}$$

$$C = 4.49 \times 10^{16}$$

$$G = 0.2225 \times 10^{16}$$

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To evaluate the stress constants

$$\left[\frac{\frac{1}{6} (1.818)^2 (-4.51) + (0.3) (4.49) + (2) (0.2225)}{4.49 + (0.6) (0.2225)} \pm 1 \right] = \begin{matrix} +0.073 \text{ inside} \\ -1.151 \text{ outside} \end{matrix}$$

$$\left[\pm 6 \frac{(0.91)}{(1.818)^2} (-4.51) - \frac{28.7^2}{2} (2205) - 1.347 \right] = \begin{matrix} -18.35 \text{ inside} \\ -20.6 \text{ outside} \end{matrix}$$

The stress is then:

$$\sigma_{\theta} = -\frac{3}{4} \frac{(0.268)^2}{0.7} \left[\begin{matrix} +0.073 \\ -1.151 \end{matrix} \right] P - 16.68 (0.268)$$
$$\left[1 + \frac{(0.259)^2}{2} \right] \left(\begin{matrix} -18.35 \\ -20.6 \end{matrix} \right) P - (0.259) \left(\begin{matrix} -18.35 \\ -20.6 \end{matrix} \right) (0.785) P$$
$$- 6 \left(\begin{matrix} 0.927 \\ 0.151 \end{matrix} \right) (0.994) P$$

$$\sigma_{\theta} = \left| \begin{matrix} -0.00562 \\ +0.0885 \end{matrix} \right| P - \left| \begin{matrix} 1.72 \\ 1.385 \end{matrix} \right| P + \left| \begin{matrix} 3.73 \\ 4.19 \end{matrix} \right| P - \left| \begin{matrix} 5.53 \\ 0.9 \end{matrix} \right| P$$

$$\sigma_{\theta} = \begin{matrix} -35,260 \text{ psi inside} \\ +19,940 \text{ psi outside} \end{matrix}$$

From consideration of the equivalent uniaxial stress form, it can be seen that under the loading conditions it is less than 78,500 psi [calculated at inside edge to be 68,000 psi equivalent uniaxial stress].
 $\sigma_{\text{yield}} = 120,000$ psi minimum.

1. Upper Vessel Bolts

The bolt is now assumed to completely fill the bolt hole. Then the neglecting of the ribs, the assumption of the lower vessel rigidity and the fit will make the bolt react all the shear loading. This is a conservative analysis for the bolt.

This shear stress is found from

$$\frac{n\pi d^2}{4} \tau = 2\pi r_1 Q$$

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where

n = number of bolts

d = bolt diameter

r_1 = radius at which Q is acting

τ_b = shear stress in bolt

$$\therefore \tau_b = \frac{2\pi \times 4.32 \times 0.785P}{18 \times \pi/16} = 60,200 \text{ psi}$$

These bolts are cold-rolled titanium with a minimum yield stress of 130,000 psi. Therefore this loading is acceptable based on an allowable shear stress being equal to half the yield stress of the material.

2. Flange Tearout

This stress is found from

$$\sigma_F = \frac{2\pi r_1 Q}{n \times 2 \times m \times t}$$

where

n = number of bolts

m = edge distance

t = flange thickness

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Therefore:

$$\sigma_F = \frac{2\pi r_1 Q}{18 \times 2 \times \frac{1}{2} \times 1.3}$$

$$\sigma_F = 9,100 \text{ psi}$$

therefore, the edge distance is more than sufficient.

3. Crushing of the Bolt

This stress may be found from

$$\sigma_b = \frac{2\pi r_1 Q}{ndt}$$

where d = bolt diameter; all other values have been previously defined.

$$\sigma_b = \frac{2\pi r_1 Q}{18 \times \frac{1}{2} \times 1.3} = 18,200 \text{ psi}$$

There is no problem here, the flange depth is more than sufficient.

It can be shown that for the same loading condition, the stresses existing in the top of the upper housing are smaller than at the lower edge. Therefore, the upper vessel is structurally acceptable.

C. LOWER PRESSURE VESSEL

The lower vessel and shield assembly is now checked. The two members will be a shrink fit with the frictional load between the surfaces balancing the external pressure vertical load component.

This figure illustrates the pertinent dimensions.

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The loading situation is as shown.

Now for vertical equilibrium;

$$2\pi (R_c - t) F l_f = \pi R_c^2 P_o$$

$$2 \times 4.563 F l_f = (5.438)^2 P_o$$

$$F l_f = 3.24 P_o$$

Assume

$$l_f = 8.0 \text{ inches}$$

Then

$$F = 0.405 P_o$$

however, this load may also be found from

$$F = \mu_f P_i$$

$$\therefore \mu_f P_i = 0.405 P_o$$

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Assume the frictional coefficient is equal to 0.4; then the contact pressure P_i required is $P_i = 1.01 P_o$.

To achieve this contact pressure requires a shrink fit which is calculated from differential radial expansions.

The radial deflection δ is calculated from

$$\delta_1 = \frac{b}{E_1} \left[\frac{P_i b^2 - P_o c^2 + c^2 (P_i - P_o)}{c^2 - b^2} + \mu \frac{P_o c^2 - P_i b^2 + c^2 (P_i - P_o)}{c^2 - b^2} \right]$$
$$\delta_1 = \frac{b}{E_1} \left[\frac{P_i (b^2 + c^2) - 2P_o c^2}{c^2 - b^2} + \mu P_i \right]$$
$$\delta_2 = -\frac{b}{E_2} \left[P_i \frac{(b^2 + a^2)}{b^2 - a^2} - \mu P_i \right]$$

Now the total shrink fit is

$$\delta = \delta_1 + |\delta_2|$$

For

$$E_1 = E_2 = E$$

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$$\delta = \frac{b}{E} \left[\frac{P_i (b^2 + c^2) - 2P_o c^2}{c^2 - b^2} + \mu P_i + P_i \frac{(b^2 + a^2)}{(b^2 - a^2)} - \mu P_i \right]$$

and for $1.01 P_o$

$$\delta = \frac{b}{E} P_o \left[\frac{1.01 b^2 - 0.99 c^2}{c^2 - b^2} + \frac{1.01 (b^2 + a^2)}{b^2 - a^2} \right]$$

Now for

$$a = 2 \text{ inches}$$

$$b = 4.563 \text{ inches}$$

$$c = 5.438 \text{ inches}$$

$$E = 15.5 \times 10^6 \text{ psi}$$

$$\delta = \frac{b P_o}{E} \left[\frac{21.0 - 29.2}{29.5 - 20.8} + \frac{25.1}{20.8 - 4} \right]$$

$$\delta = \frac{0.55 b P_o}{E}$$

$$\delta = 0.00162 \text{ inch for } P_o = 10^4 \text{ psi}$$

Now for a shrink fit of 0.005 inch the contact pressure increases to

$$\frac{E \delta}{b} + 6.79 P_o = 7.27 P_i$$

$$P_i = 1.168 \times 10^4 \text{ psi}$$

This is within the Class C fit $[0.002 - 0.005 \text{ inch}]$. For a maximum pressure of $1.168 \times 10^4 \text{ psi}$ the stress in the shield becomes $\sigma_P = 28,900 \text{ psi}$. This compares to a value of 110,000 psi for the yield strength of the material. For $P_o = 10^4 \text{ psi}$, $P_i = 1.168 \times 10^4 \text{ psi}$, the maximum stress in the titanium is only $\sigma = -2,070 \text{ psi}$.

At the transition of the side of the pressure vessel to the bottom, there is a gap shown between the pressure vessel and the U-Mo shield.

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The maximum stress that can develop at this section is checked. It is assumed that the section can be analyzed as a straight beam, and then curved beam corrections will be applied. This is conservative since using strip theory will tend to overestimate the maximum stress. Hence, for a built-in case;

$$M_1 = M_2 = \frac{1}{12} w l^2$$

$$M_{\max} = \frac{1}{12} P \left(\frac{\pi}{2} r \right)^2$$

$$M_{\max} = 1.35 P$$

$$\sigma = \pm \frac{M c}{I} = \pm \frac{6 M}{t^2} = \pm 10.58 P$$

The curved beam correction curve taken from Roark is:

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Fig. D-1. Coefficient of Thermal Expansion

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at

$$R/c = 5.86$$

$$k_i = 1.13$$

$$k_o = 0.88$$

Therefore, correcting for the curved effect at the inside surface:

$$\sigma_i = 1.13 \times 10.58P = -119,500 \text{ psi}$$

and at the outside surface

$$\sigma_o = 93,000 \text{ psi}$$

The direct stress is

$$\sigma = \frac{PR}{t} = -29,400 \text{ psi}$$

The total stress is the

$$\sigma = -148,900 \text{ psi at the inside surface}$$

and

$$\sigma = 63,600 \text{ at the outside surface}$$

Since the local stress is greater than yield, it would be advisable to fill this void area after installation of the shielding with a firm-setting epoxy to prevent this local stress from occurring. From a purely structural standpoint, a little local yielding would not hurt. However, once this occurs it makes the material susceptible to pitting and crevice corrosion. Therefore, in a seawater environment stresses must be maintained below the yield of the material. The coefficient of expansion curve (Fig. D-1) shows that once the shrink fit is made, there will be no problem of the vessels becoming loose due to temperature because the U-Mo grows faster than the titanium vessel.

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APPENDIX E

SAFETY ANALYSIS

The corrosion rate of tungsten is assumed to be the same in sea-water as in hydrochloric acid solution--0.001 in./year. Based on a closure weld penetration of 0.156 inch, the weld would be gone after 156 years. Since the Sr-90 has a half life of 27.8 years, the fuel would have passed

$$\frac{156}{27.8} = 5.6 \text{ half lives}$$

The initial fuel loading is 28,500 curies so after 5.6 half lives there will remain

$$\frac{28,500}{(2)^{5.6}} = 582 \text{ curies}$$

Based on complete disappearance of the side wall of the fuel capsule; (0.496 inch) the time required would be 496 years at which time the Sr-90 would have passed

$$\frac{496}{27.8} = 17.8 \text{ half lives}$$

After passing 17.8 half lives, there will remain

$$\frac{28,500}{(2)^{17.8}} = 0.2 \text{ curie}$$

A. DYNAMIC LOADING

The following analysis assumes that the capsule and the shielding into which the capsule is threaded is decelerated at 6 g. The total weight of the capsule and shield is not greater than 66.2 pounds.

The configuration is as follows:

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The total force acting on the eight bolts through the aluminum into the U-Mo shield is $66.2 \times 6 = 397.2$. The cross section of the bolt is $\left(\frac{0.209}{2}\right)^2 \pi = 0.0109 = 0.0343$. The total cross section is $0.0343 \times 8 = 0.274 \text{ in.}^2$. The applied stress is $\frac{397.2 \text{ lb}}{0.274 \text{ in.}^2} = 1449.6 \text{ psi}$. The tensile yield of the bolt is 3360 psi. Thus, the bolts will not yield in tension.

B. MIN-K RING

The Min-K ring holding the shield plug must take the 6g loading of the shield plug and the fuel capsule (≈ 70 pounds). The deformation of the Min-K is now checked to determine if the shield plug can hit the module under these conditions.

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where

$$Q = \frac{6 \times 70}{2\pi \times 3} = 22.3 \text{ lb/in.}$$

$$\sigma = \frac{Q}{t} = 55.7 \text{ psi}$$

Based on 94 psi causing 5% deformation at 1300° F, the 55.7 psi load will cause:

$$\frac{\delta}{l} = 2.96\%$$

and

$$\delta = 0.0296 \text{ inch}$$

Since the clearance between the module and the shield plug is 0.188 inch, the two will not contact under these conditions. Now the same force is applied to the aluminum ring at its innermost radius. Assuming that the outer edges are simply supported, from Roark's formula for flat plates, uniform loading along inner edge (at inner edge)

$$\text{Max } S = S_t = \frac{-3W}{2\pi m t^2} \left[\frac{2a^2 (m+1)}{a^2 - b^2} \ln a/b + (m-1) \right]$$

where

W = force acting

t = thickness of plate

m = $1/\nu$, ν = Poisson's ratio

S = unit stress at surface of plates = psi

a = outside radius

b = inside radius

W = 397.2 lb $\approx$ 400 pounds

t = 0.125 inch

a = 4.0625 inch

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$$b = 2.75 \text{ inch}$$

$$a/b = 1.477$$

$$m = 1/\nu = 1/0.36 = 2.778$$

$$\begin{aligned} \text{Max } S &= \frac{-3 \times 400}{2 (2.778) (0.125)^2} \frac{16.5}{2 (4.0625)^2 (3.778) (0.390) + 1.778} \\ &= \frac{-3 \times 200 \times 8 \times 8}{2 (2.778)} \left[\frac{(33) (1.473)}{8.94} + 1.778 \right] \\ &= \frac{-600 \times 64}{8.728} (5.437 + 1.778) \\ &= \frac{38,400}{8,728} (7.215) \\ &= 4400 (7.215) = 31,746 \text{ psi} \end{aligned}$$

The maximum stress applied to the aluminum ring is 31,746 psi. The minimum tensile strength of Al-6061 is 42,000 psi. Thus, shielding of the capsule is guaranteed under 6 g loading.

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