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**SNAP 21-B PROGRAM
DEEP SEA RADIOISOTOPE-FUELED
THERMOELECTRIC GENERATOR
POWER SUPPLY SYSTEM**

**QUARTERLY REPORT NO.2
July 1, 1964 to September 30, 1964**

PUBLISHED OCTOBER 1964

PREPARED BY:

T. J. Antolak

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SNAP 21-B PROGRAM
DEEP SEA RADIOISOTOPE-FUELED
THERMOELECTRIC GENERATOR
POWER SUPPLY SYSTEM

By

T. J. Antolak

V. Yanisch, Jr.

ABSTRACT

This deep sea power supply system will utilize thermoelectric semiconductor PbTe alloys in the segmented form. When operated between 1100°F and 100°F these materials will perform at a 10% materials efficiency.

A high temperature vacuum thermal insulation system will be incorporated into this power supply system to reduce overall heat losses and thus increase system efficiency.

A new high strength titanium alloy -721 is being used for the power supply system pressure vessel. This material has a high strength to weight ratio, high compressive yield strength (110,000 psi), and has excellent resistive properties to salt water corrosion.

The isotopic heat source, SR-90 titanate, to be used in this power system will be directly shielded with a depleted uranium (plus 8% Molybdenum) alloy.

The DC-DC converter incorporated into this system will operate at 90% efficiency.

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SNAP 21-B PROGRAM
DEEP SEA RADIOISOTOPE-FUELED
THERMOELECTRIC GENERATOR
POWER SUPPLY SYSTEM

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THERMOELECTRIC GENERATOR
POWER SUPPLY SYSTEM

SUMMARY

A complete system data review is presented in Appendix B, which compares the current program status with the proposed program. Current weight of the system is 507 pounds.

The overall system efficiency is currently 6.07%, however, improvements are being developed to increase the system efficiency to 6.48%.

The design of the thermoelectric couples has been established. The hot junction temperature is 1100°F and the cold junction temperature is nominally 100°F.

Three hundred (300) "P" and "N" legs of the current design have been made and are being processed for testing in modules and prototype generators.

A 6-couple test module has been designed and three of these units are currently on test. All units exhibit steady power output versus time. One module has been tested for 1800 hours.

The mechanical design of the prototype thermoelectric generator is complete and the first unit is being fabricated.

A stainless steel pressure vessel is being fabricated to be used in heat transfer experiments to aid in the final design. A computer program for thermal analysis of the complete power system has been developed to aid in the design of the final prototype power system.

The 1/4 scale titanium pressure vessel was hydrostatically tested from sea level pressure to 12,000 psi. Test results show good correlation between design predictions and experimental measurements. The 1/4 scale model was tested to

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destruction and failed at 17,500 psi. The design of the prototype pressure vessel is complete.

The fuel capsule test program has been submitted for review and approval. The fuel capsule and inner liner designs are complete, and fabrication of the test capsules is progressing.

The final design of the radiation shield will be completed when the interface components are fixed.

An initial experimental DC-DC converter was assembled and exhibited an efficiency of 89.5%.

A preliminary insulation envelope design is being mechanically and thermally analyzed.

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SNAP 21-B PROGRAM
DEEP SEA RADIOISOTOPE-FUELED
THERMOELECTRIC GENERATOR
POWER SUPPLY SYSTEM

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INTRODUCTION

This contract work is being conducted by the 3M Company, Isotope Power Laboratory, Electrical Products Division. The main objectives of this program are to design, develop, fabricate, and test components leading to the manufacture of a 10-watt, 24 volt power supply for deep sea application at 10,000 psi.

The main system design requirements are:

1. 10-watt, 24 volt continuous output.
2. 5-year life.
3. High efficiency.
4. Minimum weight.
5. High reliability.
6. Resist salt water corrosion at 10,000 psi.
7. Sr-90 titanate fuel.
8. Fully shielded device.
9. Removable thermoelectric generator.

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1.0 SNAP 21-B DEEP SEA RADIOISOTOPE-FUELED THERMOELECTRIC POWER SUPPLY SYSTEM

1.1 PLANNED OBJECTIVES

The planned program objectives are:

- to design and fabricate a power supply system having a minimum continuous power output of 10-watts (e) at 24 volts while operating in sea-water at 10,000 psi external pressure for five years.
- to design and fabricate the system with an overall efficiency of 6.75 percent and maximum weight of five hundred fifty (550) pounds.
- to design the system and sub-systems to withstand the applicable environmental conditions as outlined in the contract Technical Provisions and Scope of Work.
- to design the system to be capable of: unattended operation; ease of repair and removal of the thermoelectric generator and electrical power conditioner; and complete containment within a pressure vessel.
- to design the system to be capable of upgrading to output power levels of 20, 60 and 200 watts (e).

1.2 PROGRAM EFFORT AND ACCOMPLISHMENT THIS PERIOD

- A system review was conducted to determine the status of the current design compared to the proposed design. Appendix B is a tabular comparison of all system details.
- The weight and efficiency of the system were analyzed. Current weight is 507 pounds and current predicted efficiency is 6.07%. System improvements, as discussed below, indicate that the efficiency will be increased to 6.48% for the prototype system.
- An analysis of heat transfer mode to the thermoelectric generator from the biological radiation shield was conducted to determine the applicability of radiative heating of the generator. (Appendix C)
- Design of a generator mounting plate and segmented centering ring is in progress. These components comprise the heat rejection path from the thermoelectric generator and provide structural integration of the system within the pressure vessel.

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1.3 TECHNICAL DISCUSSION AND RESULTS

1.3.1 SYSTEM WEIGHT ANALYSIS

System weight was analyzed on a component basis to determine the current design weight for comparison to the proposed weight and to the specified maximum weight of 550 pounds.

TABLE 1. 1
SNAP 21-B SYSTEM WEIGHT COMPARISON

<u>SYSTEM COMPONENT</u>	<u>PROPOSED WT. (LBS.)</u>	<u>CURRENT DESIGN WT. (LBS.)</u>
Pressure Vessel	122	144
Biological Radiation Shield	380	256
Fuel Capsule and Fuel	6.8	5.8
Thermoelectric Generator	9.0	18.6
Mounting Plate and Segmented Ring	---	23.7
Insulation Envelope	4.4	40.0
DC-DC Converter	0.5	5.0
Thermal Radiation Disc	---	3.0
Leads and Instrumentation	---	5.0
Fasteners and Miscellaneous Insulation	---	2.0
	<hr/>	<hr/>
TOTAL	522.7	506.8

The weight differences shown in Table 1.1 are clarified as follows:

- (1) Pressure vessel material was changed to 721 titanium with a lower yield strength, thereby requiring increased wall thickness.
- (2) Biological radiation shield material was changed from tungsten to U-8 MOLY to reduce weight.
- (3) Fuel capsule design resulted in a reduced wall thickness. Fuel loading was redetermined on the basis of ORNL attainable specific power of the fuel.
- (4) The thermoelectric generator cold frame material was changed from aluminum to copper to allow more reliable hermetic sealing methods.
- (5) The mounting plate and segmented ring were not included in the proposed design.
- (6) The insulation envelope was redesigned to be a support component requiring heavier walls and internal support members.

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- (7) A housing for enclosure and potting of the DC-DC converter has been included in the current design.
- (8) Mode of heat transfer to the thermoelectric generator has been changed from conduction to radiation and conduction through air.
- (9) Leads and instrumentation were incorporated in the Phase I prototype as specified, but not proposed.
- (10) Fasteners and miscellaneous insulation are the result of current system design, and were not included in the proposed design.

1.3.2 SYSTEM EFFICIENCY ANALYSIS

Current system design was thermally analyzed to identify each heat requirement area and the magnitude of the heat transferred through each area. The objectives of the analysis were to compare proposed and current predicted efficiencies, and to establish the system fuel loading requirements.

TABLE 1.2

SNAP 21-B SYSTEM HEAT BALANCE, BEGINNING OF LIFE

HEAT REQUIREMENT AREA	MAGNITUDE OF HEAT TRANSFER (WATTS)	
	PROPOSED	CURRENT DESIGN
Thermoelectric legs	135.0	144.0
Thermoelectric Generator		
Internal Insulation	4.6	6.3
Thermoelectric Generator Case	7.9	11.7
Heat Source Suspension Bolts	15.7	0.0
Insulation Envelope	4.6	24.0
Miscellaneous	1.2	1.0
	169.0	187.0
End of Life Efficiency	6.75%	6.07%

Increased heat requirements (Table 1.2) are the result of design changes and more accurate thermal analysis. The cause of each heat requirement change is as follows:

- (1) The proposed thermoelectric generator was designed to be a 42-couple module. Remeasurement of thermoelectric materials parameters on more accurate equipment indicated the thermal conductivity to be approximately 10% higher than previously determined.

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A second generator was designed which required 48-couples with both legs segmented, and the resultant heat requirement increased to 144 watts at the beginning of life.

- (2) The new 48-couple design required additional hot frame area to accomodate the added couples. Increased area required additional generator internal insulation and subsequent additional heat loss.
- (3) Detailed thermal analysis of the generator case contributed to the elimination of a bellows, as proposed, and also showed extraneous heat losses previously undetermined. The increased loss is a result of a straight case (8.8 watts) and parallel heat path losses (2.9 watts).
- (4) Heat source suspension bolts were eliminated in the current design, thereby eliminating their respective loss.
- (5) Load-bearing insulation is specified in the current insulation envelope design. Higher thermal conductivity of the insulation and higher operating temperature for radiative heat transfer are the factors which determine the insulation envelope loss.
- (6) Miscellaneous losses have been confined to the space between the generator case and the insulation envelope on the current design. This void will be insulated with an expected maximum heat loss of 1-watt.

1.3.2.1 SYSTEM IMPROVEMENT FOR EFFICIENCY GAIN

Efficiency and weight are the primary design considerations for the system. Design and development of each system component is directed for continual improvement on these parameters.

To date, two major improvements have been developed.

- (1) Thermoelectric couple design was re-evaluated, and a new thermoelectric generator sub-assembly was conceived which will reduce the required heat to the thermoelectric legs to 138.6 watts. This new design involves materials and leg length changes, but does not alter the geometry of the generator housing or other system components. Details of this improved couple have not yet been fully defined and designed.

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- (2) All power requirements of the thermoelectric generator have been established on the basis of an 85% efficient DC-DC converter. DC-DC converter design and component evaluation has resulted in an experimental converter which exhibited an efficiency of 89.5% without selective components. Indications are that a repeatable efficiency of 90% will be attained on the SNAP 21-B converter.

The net result of a 90% efficient converter would be approximately a 5% reduction in required heat to the generator for a revised generator requirement of 131.7 watts.

Table 1.3 is a tabular summary of the efficiency improvement expected on the prototype system.

TABLE 1.3
EFFICIENCY INCREASE WITH DESIGN IMPROVEMENT

DESIGN CONDITION	REDUCED HEAT REQUIREMENT (WATTS)	REVISED FUEL LOADING (WATTS)	EXPECTED END OF LIFE EFFICIENCY
1. Current Design	0	187.0	6.07%
2. Thermoelectric Couple Improvement	5.4	181.6	6.24%
3. DC-DC Converter Efficiency Improvement	6.9	174.7	6.48%

One prime area for further efficiency improvement is the insulation envelope. The current design heat loss is 24 watts using load bearing insulation and with a higher source temperature. Initial indications are that load bearing insulation will provide marginal mechanical support for the system. An alternate design consisting of "super insulation" and internal support structure is being investigated. Heat losses of the alternate design have been initially determined to be between 20 and 24 watts with conservative analysis techniques.

1.3.3 SYSTEM INTEGRATION

Transfer of heat from the biological radiation shield to the thermoelectric hot frame by a combination of radiation and conduction through air was analyzed. (Appendix C.) The advantage of radiation is that an air gap exists at the generator hot frame which isolates the thermoelectric assembly from other system components, thereby eliminating direct transmission of shock and vibration to the thermoelectric legs. Analysis results show that this mode of heat transfer is feasible; and has been incorporated in the system design. Protection must be provided against possible eutectic formation of the shield and other components and will be accomplished in conjunction with shield oxidation prevention.

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A generator mounting plate and segmented centering ring (Figure 1.1) has been tentatively designed. The generator mounting plate serves three functions:

- (1) It is the mount for the thermoelectric generator.
- (2) It transfers heat from the generator cold frame to the pressure vessel and segmented ring.
- (3) It is the bearing plate to the pressure vessel from the segmented centering ring.

The material selected for this component is 2024 aluminum on the basis of its high thermal conductivity.

The segmented centering ring serves two functions:

- (1) It positions and retains the insulation envelope and heat source within the pressure vessel.
- (2) It provides additional heat transfer area at the pressure vessel walls.

The mounting ring is segmented to allow for the circumferential decrease in pressure vessel diameter in operational cycling from sea level pressure to 10,000 psi. The convex surface of the mounting ring matches a concave surface on the insulation envelope to provide both position and support to the insulation envelope. A convex ring surface is used to allow rotation of each ring segment to accommodate the radial deflection of the pressure vessel wall by external pressure.

A captive-bolt and spring arrangement provides bearing pressure between the mounting plate and mounting ring for system support and thermal contact. Integral extensions on the mounting plate are spring loaded to each ring segment for heat transfer contact.

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Figure 1.1 Assembly, SNAP 21-B Radioisotope - Fueled
Thermoelectric Power Supply System

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2.0 SNAP 21-B THERMOELECTRIC GENERATOR

2.1 PLANNED OBJECTIVES

The planned objectives are:

- To design, develop and fabricate a thermoelectric generator which shall provide a net system power output of 10-watts, direct current, at 24 volts.
- To design, develop and fabricate the thermoelectric generator to provide the specified output continuously for a minimum period of 5 years.
- To design, develop and fabricate a thermoelectric generator with a thermoelectric leg subsystem efficiency of 10.0 percent.
- To design and fabricate the thermoelectric generator for removal and replacement in the system without a hot cell facility and with minimum special equipment.
- To design, develop and fabricate the thermoelectric generator to withstand the environmental conditions as specified in the contract technical provisions.

2.2 PROGRAM EFFORT AND ACCOMPLISHMENT THIS PERIOD

- The dimensions of the thermoelectric legs have been established by a thermal and electrical analysis.
- The nominal hot and cold junction temperatures have been established as 1100⁰ F and 100⁰ F respectively, at beginning of life.
- Approximately 1000 "P" and "N" legs have been produced to establish production processes and supply legs for testing. About 300 "P" and "N" legs of the current design have been made and are being processed for installation in test fixtures and prototype generators.
- Single thermoelectric couple tests have been performed and test results have been analyzed.
- A 6-couple test module has been designed and three of these units are currently on test. One unit has over 1800 hours of steady state power output, and the other units have less time on test but also exhibit steady state output.

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Figure 2.1 SNAP 21-B Thermoelectric Couple Layout

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- The mechanical design of the preliminary prototype thermoelectric generator has been completed. Individual components are in various stages of drafting, development and fabrication.
- Three test generators without thermoelectric couples and associated hardware are being fabricated to evaluate electron beam welding at the joints of the cold and hot frames with the outer thin-wall case.
- An initial test generator (Figure 2.23) utilizing an "O" ring seal for the cold end hermetic seal has been designed to test the electrical characteristics of the generator while electron beam welding is evaluated.
- A thermoelectric generator-pressure vessel thermal test has been designed. A stainless steel pressure vessel is being fabricated for this test.
- A computer program for the complete system thermal analysis has been established to correlate the thermal experiments conducted using the stainless steel vessel with the final system design.

2.3 TECHNICAL DISCUSSION AND RESULTS

2.3.1 THERMOELECTRIC COUPLE DESIGN

Segmented lead telluride thermoelectric couples are operated in series to provide the required power and voltage at the load. The voltage developed by the thermoelectric leg assembly is raised and regulated by a special circuit. Operating temperatures are chosen for optimum average efficiency over the mission life.

2.3.1.1 ELECTRICAL ANALYSIS

The preliminary electrical analysis based on the methods discussed in Appendix A showed that the optimum design for the thermoelectric legs can be obtained when the length-to-area (L/A) ratio is somewhat different for the "N" and the "P" legs. This difference of L/A can be achieved physically by making the legs of different diameter and the same length or of the same diameter and different lengths. Legs of the same diameter and of different lengths are used in the current design for experimental purposes, as shown in Figure 2.1.

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The following design assumptions are aimed at achieving the goals of high efficiency, high reliability and low weight. The design data listed below reflect the material properties and system requirements.

TABLE 2.1
THERMOELECTRIC LEG DESIGN DATA

Hot junction temperature:	$T_h = 1100^{\circ} \text{F}$ at beginning of life (BOL)
Cold junction temperature:	$T_c = 100^{\circ} \text{F} \pm 10^{\circ} \text{F}$ at end of life (EOL)
Thermoelectric "N" leg:	Hot end segmented PbTe, Spec. M 10 Cold end PbTe, Spec. M 13
Thermoelectric "P" leg:	Hot end segmented (PbSn)Te, Spec. M 15 Cold end (Bi-Sb) ₂ Te ₃ , Spec. M 29
Heat transfer, shield to hot junction:	Radiation and conduction through gas
Thermoelectric couple connections:	series
Voltage input to converter:	$V_{\text{BOL}} = 5.95 \text{ v}$ $V_{\text{EOL}} = 5.10 \text{ v}$

Temperature Selection

The selection of the operating points for the thermoelectric generator is based on the evaluation of a number of interrelated factors including the following:

- (1) The power and voltage required
- (2) The temperature limits of the generator thermoelectric materials
- (3) The efficiency of the thermoelectric conversion materials.

In general, the weight of the thermoelectric generator system is directly related to its efficiency, which in turn depends largely on the temperature difference between the hot and cold junctions. However, because the thermoelectric materials have temperature limitations, it is necessary to choose a hot junction temperature which will permit stable performance for the mission lifetime. The known characteristics of lead telluride thermoelectric materials and extensive experience with operating generators indicate that a BOL hot junction

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FUEL CENTERLINE TEMPERATURE - 1346°F								
FUEL WALL TEMPERATURE - 1329°F								
LINER - 1322°F								
RADIATION SHIELD - 1321°F								
RADIATION DISC - 1305°F								
GAP - 0.100" - HEAT TRANSFER BY RADIATION								
THERMOELECTRIC GENERATOR HOT FRAME - 1130°F								
BORON NITRIDE INSULATION								
HOT JUNCTION ELECTRODE								
THERMAL INSULATION	"N" LEG		PbTe-M10 - 1100°F PbTe-M13 - 115°F	THERMAL INSULATION	"P" LEG		(PbSn)Te - M15 - 1100°F (BiSb) ₂ Te ₃ - M29 - 115°F	THERMAL INSULATION
	SOLDER				SOLDER			
	COLD JUNCTION ELECTRODE				COLD JUNCTION ELECTRODE			
	FOLLOWER				FOLLOWER			
	GREASE				GREASE			
COLD FRAME - 110°F								
GENERATOR MOUNTING PLATE - 95°F								
PRESSURE VESSEL - 65°F - 85°F								
OCEAN TEMPERATURE RANGE - 32°F - 41°F								

Figure 2.2 Thermal Path through Typical Thermoelectric Couple

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temperatures of $1100^{\circ}\text{F} \pm 50^{\circ}\text{F}$ is a reasonable choice. This choice is reinforced by the desire to operate the fuel capsule at temperatures where conventional high-temperature, super alloys still provide good strength. With the 1100°F hot junction temperature and the use of radiative heat transfer from the shield to the hot frame, a large clearance can be provided to insure that mechanical separation can be maintained between the heavy (260 pound) heat source and the thermoelectric generator.

The temperature distribution through the thermoelectric generator resulting from the selected hot and cold junction temperatures is shown in Figure 2.2.

Thermoelectric Materials Selection

The thermoelectric materials for this application are chosen to provide the maximum efficiency over the operating temperature range. Since the thermal output of the Sr-90 heat source decreases with time, the generator must be designed to produce the required output with the heat and temperatures available at the end of the five-year operating life. The efficiency of the thermoelectric material is dependent on the thermoelectric properties:

- S = the Seebeck coefficient in microvolts
- ρ = the resistivity in micro-ohm inches
- K = the thermal conductivity in watts/in./hr/ $^{\circ}\text{F}$

These properties in the semi-metals such as PbTe, (PbSn)Te, and (BiSb)₂Te₃, are strongly temperature dependent and vary widely over the temperature range of interest. This temperature dependence is shown graphically in the experimental curves obtained in this program as described in Section 2.3.1.2.

The thermoelectric properties are related to the thermoelectric efficiency as shown by the equation:

$$\eta = \frac{\Delta T \left[1 + Z \left(\frac{T_2 + T_1}{2} \right) \right]^{1/2} - 1}{T \left[1 + Z \left(\frac{T_2 + T_1}{2} \right) \right]^{1/2} + \frac{T_1}{T_2}}$$

Where:

$$Z = \frac{(S_P - S_N)^2}{\left[(K_N \rho_N)^{1/2} + (K_P \rho_P)^{1/2} \right]^2}$$

The subscripts N and P refer to the properties of the "N" and "P" materials respectively.

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When Z is derived from experimental data, the effective value of Z used in the efficiency calculation is the average value of Z from:

$$Z_{(Ave.)} = \frac{1}{\Delta T} \int_{T_1}^{T_2} Z dT$$

For material evaluation, the figure of merit is related to individual materials by:

$$Z = \frac{S^2}{K\rho}$$

Five batches of thermoelectric material have been manufactured and the thermoelectric properties have been measured. Two of these batches were selected for use in this generator. The range of the figure of merit, Z, for some of the actual materials considered is shown in the curves of Figures 2.4 and 2.5.

The actual thermoelectric materials exhibit measured maxima of the figure of merit at different points over the desired temperature range. A reasonably close approximation to an ideal case can be effected by forming each thermoelectric leg from a pair of homogeneous segments of different composition or different doping levels. The two segments are joined so that the place of juncture operates at the optimum temperature. For this generator the "N" type elements are formed of differently-doped segments of PbTe; the hot end of PbTe M10 having a resistivity of 120 micro-ohm inches; the cold end of PbTe M13 having a resistivity of 400 micro-ohm inches. From an examination of the curves of Figure 2.4, it will be seen that the combination of these two materials gives a higher figure of merit over a larger temperature range than can be obtained from the non-segmented leg formed of PbTe, Spec. M 11, as was first proposed. The "P" type elements are formed of two different telluride materials: the hot segment is (PbSn)Te M 15; the cold segment is (BiSb)₂Te₃ M 29. Figures of merit for the "P" leg segments are shown in Figure 2.5.

Series Operation

The desired voltage output can be attained by connecting a limited number of large diameter thermocouples in series and driving a DC-DC converter with the resulting low voltage-high current output. A solid state DC-DC

converter can approach 85% efficiency under the conditions of close matching. The 15% power loss is not adequately reflected by a simple linear increase in weight. The power loss must be compensated for by increased fuel, a larger fuel capsule, and a greater heat output. Figure 2.3 shows the series connection of the thermoelectric leg assembly.

Figure 2.3 Thermoelectric Leg Assembly with a DC-DC Converter
 (Physical and Electrical Schematic)

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2.3.1.2 THERMOELECTRIC LEG DESIGN

Intimately involved with thermoelectric leg design is the combination of material properties, leg size and shape, and experimental results of single couple tests. Testing and results are presented in Section 2.3.2, however, a brief resume' is required here. Initially, two (2) couple configurations were designed and tested. These leg dimensions are presented in Table 2.2. The single couple test results are presented in Table 2.3. The net result of this couple testing is that the power output is low for the initial test couples by 15-20%. Primarily, the power is low because of higher than normal internal resistance of the segmented P-leg.

TABLE 2.2
INITIAL THERMOELECTRIC COUPLE DESIGNS

DESIGN	TYPE LEG	DIA. (IN.)	AREA (IN. ²)	LENGTH (IN.)	L_c/L^*	$L/A^* (IN.^{-1})$
1. (42 Couples Total)	# N	3/16	0.0276	0.470	0.471	17.02
	# P	3/16	0.0276	0.388	0.581	14.06
2. (48 Couples Total)	# N	3/16	0.0276	0.578	0.384	20.94
	# P	3/16	0.0276	0.578	0.659	20.94

* L_c = Length cold segment
L = Length
A = Area

N-Leg, M 10/M 13 Segmented
P-Leg, M 15/M 29 Segmented

TABLE 2.3
TEST RESULTS
SINGLE THERMOELECTRIC COUPLES AND LEGS

DESIGN	TYPE LEG	COUPLE NO.	* T_H	* T_c	* E_o	* R_i	* P_o
1. Measured	N	3	950	105	103.7	0.0162	0.161
			950	100	109.6	0.0172	0.175
1. Measured	P	3	950	111	80.7	0.0169	0.095
			950	110	82.6	0.0104	0.164
2. Measured	N	1	955	100	103.0	0.0173	0.153
			950	100	105.7	0.0185	0.144
2. Measured	P	1	955	95	82.7	0.0215	0.076
			950	100	82.7	0.0151	0.114

* T_H = Temp. °F, hot electrode
 T_c = Temp. °F, cold electrode
 E_o = Open EMF, mv

R_i = Internal Resistance in milliohms
 P_o = Power Output, Watts

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Initial analysis of the test data indicated that several possibilities existed for high resistance of the P-leg; they are:

- (1) Oxidation of hot end pressure contact
- (2) P-leg material, high resistivity
- (3) Fabrication problems when P-leg segmented units are powder-pressed simultaneously.

In an attempt to control items 2 and 3 above, the P-leg segments were pressed separately and then joined with a metallic intersegment disc. Tests indicated this did not completely eliminate the high resistance of the P-legs. Further investigation showed that a computer analysis used to determine leg L/A ratios and dimensions was not properly programmed. For the P-leg computer program the data reduction cards for the proper materials were not used. This occurrence also affected the N-leg design.

Typical material properties for N-type PbTe and P-type alloys are presented in Figures 2.6 through 2.13. The materials selected for the P and N-legs are:

<u>LOCATION</u>	<u>N-TYPE</u>	<u>P-TYPE</u>
Hot Segment	M 10, PbTe 120 micro-ohm-in. resistivity	M 15, (PbSn)Te alloy
Cold Segment	M 13, PbTe 400 micro-ohm-in. resistivity	M 29, (BiSb) ₂ Te ₃ alloy

Using the technique for determining the performance of a thermoelectric couple (Appendix A), the proper couple configuration was calculated to be as summarized in Table 2.4

TABLE 2.4

P AND N-LEG CONFIGURATION

<u>LEG</u>	<u>COLD SEGMENT LENGTH</u>	<u>HOT SEGMENT LENGTH</u>	<u>L_c/L</u>	<u>L/A</u>	<u>L</u>	<u>D</u>
P-Leg	0.203 in.	0.243 in.	0.455	16.15	0.446 in.	3/16 in.
N-Leg	0.230 in.	0.370 in.	0.383	21.73	0.600 in.	3/16 in.

L_c = Length, cold segment
 L = Total Length

A = Area
 D = Diameter

The above listed couple design is currently being manufactured and will be tested next quarter.

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Figure 2.4 Figure of Merit vs. Temperature for N-Type Material (PbTe)

Figure 2.5 Figure of Merit vs. Temperature for P-Type Material

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Figure 2.6 Seebeck Coefficient vs. Temperature for N-Type Material (PbTe)

Figure 2.7 Resistivity vs. Temperature for N-Type Material (PbTe)

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Figure 2.8 Thermal Conductivity vs. Temperature for N-Type Material (PbTe)

Figure 2.9 Figure of Merit vs. Temperature for N-Type Material (PbTe)

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Figure 2.10 Seebeck Coefficient vs. Temperature for P-Type Material

Figure 2.11 Resistivity vs. Temperature for P-Type Material

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Figure 2.12 Thermal Conductivity vs. Temperature for P-Type Material

Figure 2.13 Figure of Merit vs. Temperature for P-Type Material

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2.3.2 THERMOELECTRIC COUPLE TESTING

During the previous quarter a 6-couple test module was designed, built and put into operation to obtain both short and long term performance data to aid in establishing the final design for the Phase I prototype thermoelectric generator.

A cut-away, sectional drawing of the 6-couple test module and its associated instrumentation panel is shown in Figure 2.14. Figure 2.15 is an extended schematic showing the location of the thermocouple temperature monitors. All thermocouples are iron-constantan. The hot electrode and cold electrode thermocouple wires are 0.010 in. in diameter (30 gauge), whereas the heater block thermocouple wires are 0.020 in. in diameter (24 gauge). Figure 2.16 is a schematic of the external electrical circuitry. Figures 2.17, 2.18, and 2.19 are photographs of the various components of the 6-couple test module. Figure 2.20 is a photograph of a typical thermoelectric couple and the various internal hardware components. Figure 2.21 is a photograph of a typical P-leg prior to and after bonding of the individual segments and electrodes.

Three modules have been fabricated and put on test. These modules were designated A-1, A-2 and A-3. Modules A-1 and A-2 contain 0.578 inch long, 0.188 inch diameter, segmented, simultaneously-pressed "N" and "P" legs. Module A-3 contains 0.578 inch long, 0.188 inch diameter, segmented, individually pressed and bonded "P" legs, with the "N" leg the same as in A-1 and A-2. Table 2.5 is a summary of the "N" and "P" leg data. All three modules were designated to operate with 1100° F hot electrode temperatures and 115° F cold electrode temperatures. These are the beginning-of-life thermal conditions.

TABLE 2.5

LEG DATA: 6-COUPLE MODULES (3M37 A-1, A-2, A-3)

SEGMENT	MATERIAL	LENGTH-L (INCH)	DIAMETER-D (INCH)	AREA-A (INCH ²)	$\frac{L_c}{L}$	$\frac{L}{A}$
Cold	M 13	0.222	0.188	0.0276	-	-
"N" Hot	M 10	0.356	0.188	0.0276	-	-
Total	M 10/M 13	0.578	0.188	0.0276	0.383	20.9
Cold	M 29	0.381	0.188	0.0276	-	-
"P" Hot	M 15	0.197	0.188	0.0276	-	-
Total	M 15/M 29	0.578	0.188	0.0276	0.660	20.9

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Table 2.6 contains the computer predicted module performance data for both the beginning and end of life thermal conditions.

TABLE 2.6
PREDICTED MODULE PERFORMANCE DATA
(BEGINNING AND END OF LIFE CONDITIONS)

T_H ($^{\circ}\text{F}$)	- (Hot electrode temperature)	1100	950
T_C ($^{\circ}\text{F}$)	- (Cold electrode temperature)	115	100
E_O (Volt)	- (Open circuit voltage)	1.37	1.16
E_L (Volt)	- (Closed circuit voltage)	0.69	0.58
R_G (m-ohm)	- (Module internal resistance)	298	251
I_L (amp)	- (Load current)	2.30	2.31
P_O (watt)	- (Output power)	1.57	1.34

Table 2.7 contains selected (typical) performance data for each of the three modules currently on test. Portions of the data from Tables 2.6 and 2.7 are combined in Table 2.8 for comparative purposes.

Examination of the data in Table 2.8 shows that all three modules are developing the predicted open and closed circuit voltages (the 1 to 2 percent deviation is well within the permissible variation due to a combination of computer tolerance and experimental error). However, the internal resistances of all three modules are high, and as such, do not fall within the permissible variation. As a result of the high internal resistances, the load currents are reduced, thus causing a reduction in the power output.

To locate and identify the cause of the high internal resistance, the performances of the individual "N" and "P" legs of the number five couples in the modules were determined using the iron wires of the hot and cold electrode thermocouples as shown in Figure 2.15. This data, along with the computer predicted beginning-of-life leg performance data, is shown in Table 2.9.

Examination of this data shows that the individual "N" and "P" leg voltages are very close to the predicted values. One would naturally expect this to be the case since the combined 6-couple voltages are about as expected. The "N" leg resistances also appear to be very close to the predicted values, and in fact, are

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somewhat lower than predicted in modules A-2 and A-3. This experimental observation is not an indication of a large experimental error but is made possible by the fact that the computer prediction is based on an average of data taken from batch samples. The cause of the high module resistances is seen to be associated with a high "P" leg resistance. The "P" leg resistances vary from 24 percent high in module A-1 to 49 percent high in module A-2.

It was suspected that the "P" leg resistances were caused by an unusually high material resistivity. The room temperature resistivities of the "P" legs used in the number five couples of all three modules were determined from resistance-length data accumulated prior to their installation into the modules.

TABLE 2.7
6-COUPLE MODULE PERFORMANCE DATA

A-1 *							
DATE	T _H (°F)	T _C (°F)	E _O (VOLT)	E _L (VOLT)	I _L (AMP)	P _O (WATT)	R (M-OHM)
8-4	1097	116	1.31	0.66	1.96	1.29	0.342
8-5	1097	115	1.31	0.66	1.96	1.29	0.342
8-11	1097	115	1.32	0.66	1.90	1.27	0.343
8-19	1100	115	1.35	0.68	1.92	1.30	0.351
8-24	1100	115	1.35	0.68	1.93	1.31	0.347
8-25	1101	116	1.35	0.68	1.92	1.31	0.349
9-18	-	117	1.33	0.68	2.00	1.37	0.322
9-21	-	117	1.33	0.68	1.99	1.37	0.324
9-22	-	118	1.33	0.68	1.99	1.37	0.324

* Begin test 7-28-64 (1440 hours as of 9-25-64)

A-2 *							
8-19	1103	116	1.33	0.67	1.93	1.29	0.345
8-20	1102	116	1.34	0.67	1.94	1.29	0.347
9-21	-	116	1.33	0.68	1.89	1.29	0.342
9-22	-	118	1.33	0.68	1.89	1.29	0.342

* Begin test 8-11-64 (1024 hours as of 9-25-64)

A-3 *							
9-15	1098	108	1.34	0.66	1.95	1.28	0.346
9-30	1105	114	1.35	0.67	1.97	1.32	0.345

* Begin test 9-15-64 (264 hours as of 9-25-64)

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TABLE 2.8

PREDICTED AND ACTUAL MODULE PERFORMANCE DATA

COMPUTER PREDICTION		EXPERIMENTAL		
		A-1	A-2	A-3
T_H ($^{\circ}\text{F}$)	1100	1100	1100	1105
T_C ($^{\circ}\text{F}$)	115	117	117	114
E_O (Volts)	1.37	1.33 (2% low)	1.33 (2% low)	1.35 (1% low)
E_L (Volts)	0.69	0.68	0.68	0.67
R_G (m-ohm)	298	323 (8% high)	342 (15% high)	345 (16% high)
I_L (amp)	2.30	1.99	1.89	1.97
P_O (watt)	1.57	1.37 (13% low)	1.29 (18% low)	1.32 (19% low)

TABLE 2.9

PREDICTED AND ACTUAL LEG PERFORMANCE DATA

COMPUTER PREDICTION				EXPERIMENTAL								
				A-1			A-2			A-3		
	N	P	Couple	N	P	Couple	N	P	Couple	N	P	Couple
T_H ($^{\circ}\text{F}$)	1100	1100	1100	1100	1100	1100	1100	1100	1100	1105	1105	1105
T_C ($^{\circ}\text{F}$)	115	115	115	119	117	118	120	112	116	111	117	114
E_O (Volt)	0.127	0.102	0.229	0.123	0.097	0.220	0.124	0.098	0.220	0.126	0.097	0.223
E_L (Volt)	-	-	-	0.076	0.035	0.110	0.081	0.027	0.108	0.088	0.033	0.121
R_G (m-ohm)	23.2	25.0	48.2	23.4	31.1	54.5	22.8	37.2	60.0	19.0	32.0	51.0
I_L (amp)	-	-	-	1.99	1.99	1.99	1.91	1.91	1.91	1.97	1.97	1.97
P_O (watt)	-	-	-	0.152	0.070	0.222	0.155	0.052	0.207	0.173	0.065	0.238

This analysis revealed that the resistivities of the "P" legs in modules A-1 and A-2 were about as expected; however, the resistivity of the single "P" leg checked for module A-3 was considerably greater than had been anticipated.

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As was discussed in Section 2.3.1.2, closer quality control checking during manufacture of the P-leg and a re-design of the thermoelectric couple configuration is expected to alleviate the high couple resistance and thus increase power output.

Although the power output is not as high as expected for the first 6-couple modules, the power output vs. time curves are quite flat as shown in Figure 2.22. Module 3M-37-A-1 test data is presented in this figure and, except for normal fluctuations caused by temperature variations, the open circuit voltage, power output, and internal resistance have continued at a steady level for about 1800 hours (75 days).

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Figure 2.15 6-Couple Module Instrumentation

Figure 2.16 6-Couple Module External Electrical Circuit

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Figure 2.17 6-Couple Test Module Components

Figure 2.18 Basic 6-Couple Module Assembly

Figure 2.19 6-Couple Module Assembly Less Case

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Figure 2.20 Typical Thermoelectric Couple

Figure 2.21 P-Leg Assembly

Figure 2.22 Electrical Properties vs. Time, 6-Couple Module

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Figure 2.23 48-Couple Thermoelectric Generator Layout

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2.3.3 THERMOELECTRIC GENERATOR DESIGN

The mechanical design of the preliminary thermoelectric generator has been completed (Figure 2.23). The generator consists of 48 thermoelectric couples, series connected through Armco iron hot electrodes and ball-type copper cold electrodes with 16 gauge copper leads. The thermoelectric couples utilize pressure contacts at the hot electrode provided by a spring and follower arrangement from the cold electrode into the cold frame. The cold frame is tellurium copper; the hot frame and generator case are 347 stainless steel. Hot side electrical insulation is boron nitride and internal insulation is powdered Min-K 1301. The hermetically sealed generator will be back-filled with Xenon gas.

Three test generators without thermoelectric legs are being fabricated to evaluate electron beam welding of the hot and cold frames to the generator case. These units will be used to establish the welding procedure, generator processing techniques, and to determine the integrity of the hermetic seal.

An initial test generator with an "O"-ring seal at the cold frame-case joint is being fabricated for complete module testing of the generator electrical characteristics. This unit may be electron beam welded with minimum alteration if the welding evaluation progresses ahead of schedule.

2.3.3.1 THERMOELECTRIC GENERATOR CASE DESIGN

The thermoelectric generator case connects the hot frame to the cold frame as a structural member, and maintains a hermetic seal for the thermoelectric leg sub-assembly. The design criteria established for this component are:

- (1) to minimize extraneous heat losses;
- (2) to provide structural strength; and
- (3) to maintain a hermetic seal.

347 stainless steel was selected as the case material on the basis of high temperature strength and low thermal conductivity.

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It is apparent that heat losses can be reduced by decreasing the wall thickness. However, restriction of fabrication techniques and allowable stresses on the case indicate that 0.005 inch is the minimum thickness which may be used.

Heat losses were analyzed using the formula:

$$Q = \frac{KA\Delta T}{L}$$

Where: Q = Heat transferred

K = Thermal conductivity

A = Cross-sectional area of heat transfer path

L = Length of heat transfer path

The heat transfer by conduction through a 0.005 inch thick wall is 8.8 watts.

The total heat loss through the case must consider a parallel heat transfer path to the generator cold frame from the thin wall case. By electrical analogy, the heat loss from this side wall to the cold frame has been calculated to be 2.9 watts. The total heat loss due to the generator case is 11.7 watts.

Structural strength of the generator case was analyzed to account for conditions of operation and processing after assembly. The tensile stresses incurred during operation of a generator with a 0.005 inch thick case would be:

- 3,300 psi tensile stress from internal pressure build up along the axis from higher internal temperatures;
- 5,500 psi tensile stress along the axis of the generator from the springs providing the pressure contacts for the thermoelectric legs;
- for a total of 8,800 psi tensile stress along the axis.

The creep strength of 347 stainless steel is approximately 12,000 psi at design temperature.

The stresses caused by pressure build-up were based on the Perfect Gas Law:

$$\frac{P_1 V_1 = M R T_1}{P_2 V_2 = M R T_2}$$

Where: P = Pressure
V = Volume
M = Mass of the gas under consideration
R = Gas constant
T = Temperature (absolute)

An average internal temperature of 530°F and a constant gas pressure within the thermoelectric generator was assumed. No allowance was made for any space taken up by the generator insulation which fills all the voids within the case. Expansion of the generator insulation with respect to gas (and the lesser volume of actual gas because of the insulation) may reduce the stress values shown.

Processing of the generator after assembly would collapse the case without reinforcing ribs based on the following formulas for buckling pressure for "end held circular but not constrained vessel."

$$P_{\text{Critical}} = 0.87 \frac{E t^2}{L R} \sqrt[4]{\left[\frac{1}{(1 - \mu^2)} \right]^3 \frac{t^2}{R^2}} \quad (3)$$

Where: μ = Poissons ratio
L = Length of short cylinder or distance between reinforcement ribs on a long cylinder

If Poisson's ratio = 0.3, then:

$$P_{\text{cr}} = \frac{0.87 (t/R)^2}{(L/R) (R/t)^{1/2}} \quad (4)$$

For: E = 29 (10)⁶
and t = 0.005 inch
and μ = 0.3;
P_{cr} = 5.3 psi

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This calculation does not consider any support given by the generator insulation. Processing of this generator will be performed with a technique which will control the differential pressure and prevent collapse of the case.

After processing of the generator, no apparent structural limitations exist to limit the thin wall case design. Therefore, reinforcing ribs or heavier walls are unnecessary.

2.3.3.2 HOT SIDE ELECTRICAL INSULATION

With a generator design with two flat planes making up the hot and cold sides of the generator, an inherent problem is encountered with diametrical differential expansion. This different expansion of the hot and cold frames is caused by the difference in temperature at operating conditions and the different coefficient of expansions. The differential expansion would cause the thermoelectric legs to tilt or cock which can contribute to poor thermoelectric generator performance relative to the high efficiency and environmental requirements of the SNAP 21-B system. To correct this situation, Boron Nitride was investigated for the hot side electrical insulation disc between the hot electrodes and the hot frame.

The low coefficient of expansion ($4.48 (10)^{-6}$ in./in./ $^{\circ}\text{F}$ along the direction parallel to pressing direction, and $0.49 (10)^{-6}$ in./in./ $^{\circ}\text{F}$ perpendicular to pressing direction) would mean approximately 0.001 inch growth on the radius of the boron nitride disc at operating temperatures, if the radial direction is perpendicular to the pressing direction. The calculation is:

$$\begin{aligned} \text{Elongation on the} \\ \text{radius} &= (\text{C.O.E.}) (\Delta T) (\text{RADIUS}) \\ \text{of boron nitride} &= \frac{(0.49) (10)^{-6}}{^{\circ}\text{F}} (1030^{\circ}\text{F}) (1.85 \text{ in.}) \\ &= 0.00093 \text{ inch} \end{aligned}$$

Where: C.O.E. = the coefficient of expansion

ΔT = Temperature difference from room to operating conditions

RADIUS = Radius from centerline to outermost leg

Therefore: a boron nitride hot side insulation disc results in improved conditions with respect to leg tilting.

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For example, without a boron nitride disc between the hot electrode and the hot frame, the hot electrodes the greatest distance from the centerline of the generator would move approximately 0.014 inch outward. This is caused by the expansion of the hot frame at higher temperatures. Less than 0.001 inch expansion (radially) would be experienced on the cold frame. As a result, cocking of the legs would occur, causing plastic deformation of the legs at the hot side.

With boron nitride, cold and hot end expansions are very nearly identical, resulting in parallel leg movement.

Temperature drop across the boron nitride disc is considered so as not to require higher source temperatures and additional fuel. As shown by the following calculations, the temperature drop is acceptable.

The thermal conductivity is

$$\frac{9.33 \text{ BTU}}{\text{hr. } ^\circ\text{F-ft.}} \quad \text{at } 1311^\circ\text{F when parallel to the pressing axis;}$$

$$\text{and } \frac{18.66 \text{ BTU}}{\text{hr. } ^\circ\text{F-ft.}} \quad \text{when perpendicular to the pressing axis of the boron nitride.}$$

Therefore, from the equation:

$$Q = \frac{KA \Delta T}{L}$$

$$\text{Where: } Q = 158 \text{ watts} = 539 \text{ BTU/hr.}$$

$$K = \text{Thermal conductivity} = \frac{9.33 \text{ BTU}}{\text{hr. } ^\circ\text{F-ft.}}$$

$$A = \text{Area of disc} = 0.077 \text{ ft.}^2$$

$$\Delta T = \text{Temperature drop in degrees F.}$$

$$L = \text{Thickness of disc}$$

$$\text{The drop across } 1/16 \text{ inch boron nitride} = \Delta T = -3.9^\circ\text{F}$$

Tensile stress of the boron nitride disc in a radial direction was studied to determine its mechanical stability in the generator. This tensile stress would be induced by the springs providing leg contact against the hot electrodes. The hot electrodes are forced against the boron nitride, which is then forced against the hot frame of the thermoelectric generator. The hot frame expands 0.014 inch radially in the region around the outer radius of the boron nitride disc. Therefore, the expanding hot frame would tend to pull the disc radially outward along with it. A section through the center of the boron nitride disc was analyzed, and tensile stresses were approximately one-half of the ultimate allowable stress at operating temperature.

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2.3.3.3 COLD FRAME

The functions of the generator cold frame are:

- (1) to provide a framework for various parts of the generator;
- (2) to dissipate heat from the thermoelectric legs to the heat transfer plate out to the pressure vessel and surrounding water;
- (3) to provide a structural base to fasten the case of the generator which joins the hot and cold frames together;
- (4) to form part of the enclosure that maintains a hermetic seal for the generator.

Aluminum is used as a cold frame in thermoelectric generators because of its light weight and good thermal conductivity with respect to weight. However, copper has been selected as the cold frame material since fastening of the generator case can be by welding rather than soldering. A weld joint will provide increased strength and more reliability in maintaining a hermetic seal.

A tellurium copper cold frame has been designed and will weigh approximately 15-18 pounds in comparison with 6 pounds for aluminum. The thermal conductivity of tellurium copper at 68°F is 209 BTU/hr./°F/ft. in comparison to 2024 aluminum at 77°F of 109 BTU/hr./°F/ft. This difference will be advantageous for the cold end heat transfer.

2.3.3.4 BALL AND SOCKET DESIGN ON THERMOELECTRIC COUPLES

A ball and socket arrangement is utilized on the cold end of the thermoelectric couples. The advantage of a ball and socket versus a solid or square configuration is that if the legs tilt or pivot with respect to the cold frame, the legs will not fracture or lose electrical contact. This can be beneficial under unusual shock conditions. Temperature drop expected across the ball and socket joint is 4.5°F (Appendix D).

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2.3.4 THERMOELECTRIC GENERATOR HEAT SINK ANALYSIS AND TESTING

2.3.4.1 PRESSURE VESSEL THERMAL TEST

During the previous quarter a test program was initiated with the objective of experimentally determining the thermal feasibility of the current pressure vessel and cold plate materials, configurations, and joining techniques. In addition to demonstrating the feasibility, the test results are expected to confirm various theoretical considerations of the heat-transfer involved and to help locate and resolve thermal trouble areas which may not have been considered during the initial design work.

Work has been started on fabrication of a full-scale stainless steel vessel which will be electrically heated and submerged in a 40°F infinite water bath. The vessel and thermoelectric generator mounting plate will be fully thermocouple instrumented to obtain radial temperature profiles for various cold plate centerline temperatures. Heat will be supplied to the mounting plate in a manner which will simulate the heat rejection of the thermoelectric generator.

Test vessel material was selected as CF-8C (AISI type 347) stainless steel because of its ease of fabrication and low thermal conductivity. Although the prototype pressure vessel will be constructed of titanium alloy which has thermal property values similar to, but different from, those of stainless steel, it is expected that a reasonable estimate can be made of its thermal environment through validation of the theoretical analysis. This will be done by comparing the theoretical thermal predictions based on stainless steel property values with the experimental results. Once a confidence level has been established for the theoretical approaches, titanium property values will be substituted to estimate the operational thermal environment.

The stainless steel vessel is being manufactured from a casting with the critical heat transfer surfaces being machined.

2.3.4.2 THERMAL ANALYSIS

Computer programs for thermal analysis have been combined to form a general purpose thermal analysis design tool for cylindrically shaped thermoelectric generators. This composite program is capable of performing all calculations necessary to transform the SNAP 21-B system into a thermal network consisting of

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conductivities and nodes and to solve for the temperatures at all established nodes given the operating boundary values (temperature of the environment and heat supplied) of the system. The program has operated successfully on two sample test sets of data indicating that it is essentially debugged. The presently designed generator heat sink configuration has been translated to input data to the program. The following is a description of the program operation by means of one of the sample test sets.

(1) Input to the Program

Figure 2-24 shows a half cross section of a cylindrically shaped object consisting of several materials. For each material, the following information is required as input to the program.

A. For each rectangular cross section

- a. The X and Y coordinates of the upper left corner ($X = 0$, $Y = 0$ is a reference point to which all other dimensions are referenced.)
- b. The length and width.
- c. The conductivity of the material represented (BTU/hr. -ft. - $^{\circ}$ F)
- d. The number of nodes to be used along the length and width. (Optional - if nothing specified, then nine nodes are assumed for the rectangle.)

B. Boundary Temperature Planes

- a. Temperature ($^{\circ}$ F).
- b. X and Y coordinates of the edges of the circular plane.

C. Heat Inputs

- a. Quantity of heat (BTU/hr.).
- b. X and Y coordinates of the point at which the heat is to be supplied.

All dimensions are in inches.

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All of the input information is shown on Figure 2.24. This information was punched onto computer punch cards and run with the thermal analysis program on a CDC 1604 computer.

(2) Operation of the Program

After reading the input data, the program translates the geometrical configuration to a network consisting of thermal conductances and node connections.

Figure 2.25 shows the network generated by the program for the sample set of test data. The nodes are numbered from top to bottom starting with the points closest to the center of the generator and working outward radially. The conductance between two nodes is determined in the following manner:

- A. The length between nodes is calculated (L_{ij}),
- B. An appropriate area for the material or materials between the nodes is calculated (A_{ij}).
- C. The conductivity of the material or materials is found by scanning the input data (k_{ij}).
- D. The conductance between the two nodes is calculated as $C_{ij} = K_{ij}A_{ij}/L_{ij}$ BTU/hr. °F.

The temperatures in this thermal network are then determined by the computer as follows:

- E. An equation is "written" for each of the N nodes in the network according to Kirchoff's law - the sum of heat flows into a point is equal to zero. This results in N equations with N unknowns.
- F. These equations are solved by the method of substitution, which is to repetitively eliminate unknowns and equations until only one equation with one unknown remains. This unknown is then calculated and used in the preceding equation to determine another unknown, until all temperatures and boundary flows have been calculated. Heat flow between any two nodes is then calculated as $Q_{ij} = C_{ij}(T_i - T_j)$.

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(3) Computer Program Output

The computer program output is a printed list of:

- (1) the X, Y coordinates of each network node;
- (2) the node number assigned to each network node;
- (3) all nodes directly adjacent to a given network node;
- (4) the flows in the network branches between adjacent nodes and a given node;
- (5) the flows out of all network nodes for which the temperature or flow was specified as input data; and
- (6) the temperature at each node of the network.

(4) Sample Analysis

A large number of the temperatures and flows as computed or as specified in an initial test problem are listed at their appropriate node locations in Figure 2.26. Noting the conductivities specified for each rectangle for the test problem the output is justified as follows:

Conductivity of rectangle ① *	=	0.062 BTU/ft. hr. °F
Conductivity of rectangle ②	=	100.0 BTU/ft. hr. °F
Conductivity of rectangle ③	=	1.0 BTU/ft. hr. °F
Conductivity of rectangle ④	=	1.58 BTU/ft. hr. °F
Conductivity of rectangle ⑤	=	0.128 BTU/ft. hr. °F
Conductivity of rectangle ⑥	=	2.0 BTU/ft. hr. °F

* See Figure 2.26 for rectangle numbering.

Rectangle Number ①, being of low conductivity, is in essence an insulator in comparison to the other rectangles. The injection of 1000 BTU/hr into this insulator would cause a hot spot (64,290°F) at the injection point (see ① of Figure 2.26). The temperatures of adjacent nodes, however, would be expected to be much less in magnitude (12,040 and 15,750°F) due to the flow distribution (787.8 and 212.0 BTU/hr away from the injection point

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and the low thermal conductivity of rectangle ①. The temperatures at all nodes in rectangle ② are essentially equal due to the high conductivity of its material.

Referring to Figure 2.26, the summation of flows out of the system equals the input flow (i. e., $182 + 334 + 0 + 143 + 181 + 6.9 + 6.0 + 147 \cong 1000$ BTU/hr). The largest flow is out of rectangle ④ since this is the shortest flow path from the injection point. The flow of zero out of the lower right hand corner node of rectangle ④ (see ③ of Figure 2.26) occurs because the specified temperature for the node above the corner node (see ③ of Figure 2.26) for that to the left of the corner node (see ④ of Figure 2.26) and for the corner node itself are all 1100°F . Thus, since ΔT is zero for all flow paths into the corner node, the sum of the flows into the corner node equals zero, necessitating a zero flow out of it. Since rectangle ⑥ is highly conductive with respect to rectangle ⑤, almost all flow out of the highly conductive rectangle ② is ejected out of the only exit node of rectangle ⑥, which is the lower right hand node of rectangle ⑤ (147 BTU/hr., see ⑤ of Figure 2.26). Due to the low conductivity of rectangle ⑤, the flows out of its exit paths (6.9 and 6.0 BTU/hr.) are small in magnitude.

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ONE HALF CROSS SECTION OF A CYLINDRICAL THERMOELECTRIC GENERATOR

Figure 2.24 Thermal Analysis: Program Input

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Figure 2.25 Thermal Analysis: Program-Generated Network

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3.0 SNAP 21-B PRESSURE VESSEL

3.1 PLANNED OBJECTIVES

The planned objectives are:

- to design and fabricate a pressure vessel to completely contain the system under operating conditions for a minimum five year period.
- to design and fabricate a pressure vessel to withstand the corrosive conditions of the operational environment and to incur no leakage during placement and operation of the system for its design life.
- to design and fabricate a pressure vessel with provisions for external instrumentation and power leads capable of being permanently sealed.
- to design and fabricate a pressure vessel to withstand all other environmental conditions as specified in the contract Technical Provisions.

3.2 PROGRAM EFFORT AND ACCOMPLISHMENT THIS PERIOD

- The 1/4 scale model of the prototype pressure vessel was hydrostatically tested from sea level pressure to 12,000 psi. Test results indicated good correlation between design predictions and experimental measurements.
- The 1/4 scale model of the prototype pressure vessel was successfully tested to destruction to validate empirical design. Failure pressure was 17,500 psi.
- An investigation of prototype pressure vessel fabrication was completed. Fabrication modes, vendor qualifications, material suppliers and quality requirements have been established.
- Prototype pressure vessel design was completed.
- An electrical feed-through receptacle has been selected. A stainless steel receptacle is being procured to test the integrity of its O-ring seal.

3.3 TECHNICAL DISCUSSION AND RESULTS

3.3.1 PRESSURE VESSEL MODEL TESTING

A series of hydrostatic pressure tests was conducted on a 1/4 scale model of the prototype pressure vessel. These tests were conducted to validate the empirically-based mechanical design of the prototype pressure

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Figure 3.1 Pressure Vessel Component Parts

Figure 3.2 Assembled Pressure Vessel

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Figure 3.3 Strain Gage Instrumentation

Figure 3.4 Data Recorders and Pressurization System

vessel. In particular, a correlation between the empirically-based design and experimental observations was desired to permit accurate predictions of the prototype pressure vessel's heat sink flange, cylinder, and penetration radial deflections; shortening of the cylinder and lower hemisphere, rotation of the heat sink flange, and failure pressure. For complete test validation, the model vessel was machined from Titanium 721 bar stock and contained two 1/4 scale, Viton O rings. Figure 3.1 is a photograph of the vessel's component parts, and Figure 3.2 is a photograph of the assembled vessel. Tests were conducted at David Taylor Model Basin, Washington, D. C.

3.3.1.1 TEST IDENTIFICATION

The hydrostatic pressure tests that were conducted are categorized according to desired information as:

(1) Deflection

Local strain (inches of deflection per inch) versus external loading pressure (psi) was determined at selected locations on the interior and exterior of the model pressure vessel. Seventy-six active and six passive Budd Metalfilm type C6-1x1-32 strain gages were used to detect the local strains, and two Gilmore model 114, 48-channel plotters were used to record the strain data. Photographs of the instrumentation system are shown in Figures 3.3 and 3.4. Figure 3.5 is a pressure vessel schematic showing the location of the strain gages.

This series of tests was designed to determine the manner in which the vessel would bend or distort when subjected to various degrees of external pressure. In particular, the distortion profile at the 10,000 psi operating pressure was desired for design purposes. Three deflection tests were conducted: 0-5,000 psi; 0-10,000 psi; and 0-12,000 psi. The loading schedules for these three tests were as follows:

0-5,000 psi

Pressure increments of 500 psi were used throughout the test. Upon application of each pressure increment, a two-minute waiting period was observed before the strain data was recorded. This was to allow the system to come to static equilibrium.

0-10,000 psi

A 500 psi pressure increment was used between 0 and 8,000 psi and a 200 psi increment was used between 8,000 and 10,000 psi. Waiting periods of two minutes were observed between 0 and 8,000 psi and one minute between 8,000 and 10,000 psi.

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TABLE 3.1

STRAIN SENSITIVITIES (IN/IN/PSI)
 SNAP 21-B 1/4 SCALE MODEL PRESSURE VESSEL

SENSITIVITY (IN/IN/PSI)				SENSITIVITY (IN/IN/PSI)			
GAGE NO.	5,000	10,000	12,000	GAGE NO.	5,000	10,000	12,000
501	-0.25	-0.25	-0.27	601	-0.48	-0.50	-0.46
502	-0.21	-0.19	-0.19	602	-0.25	-0.23	-0.24
503	-0.37	-0.37	-0.38	603	-0.20	-0.21	-0.19
504	-0.27	-0.26	-0.26	604	-0.27	-0.27	-0.26
505	-0.34	-0.34	-0.34	605	-0.20	-0.21	-0.21
506	-0.29	-0.28	-0.28	606	-0.21	-0.21	-0.21
507	-0.32	-0.32	-0.32	607	-0.21	-0.24	-0.23
508	-0.32	-0.30	-0.30	608	-0.25	-0.25	-0.26
				609	-0.25	-0.25	-0.25
510	-0.26	-0.25	-0.25	610	-0.18	-0.19	-0.18
				611	-0.07	-0.13	-0.16
512	-0.26	-0.26	-0.24	612	-0.02	-0.08	-0.06
513	-0.65	-0.64	-0.64	613	-0.10	-0.11	-0.10
514	-0.21	-0.20	-0.20	614	-0.13	-0.13	-0.14
				615	+0.03	+0.02	+0.07
517	-0.14	-0.15	-0.15	616	-0.14	-0.15	-0.12
518	-0.28	-0.26	-0.25	617	+0.03	+0.05	+0.04
519	-0.08	-0.07	-0.07	618	-0.13	-0.14	-0.14
520	-0.34	-0.31	-0.28	619	-0.12	-0.13	-0.13
521	-0.08	-0.06	-0.06	620	-0.23	-0.24	-0.24
522	-0.23	-0.22	-0.21	621	-0.11	-0.12	-0.12
523	-0.05	-0.04	-0.04	622	-0.25	-0.26	-0.26
524	-0.38	-0.36	-0.36	623	-0.09	-0.10	-0.10
525	-0.07	-0.05	-0.06	624	-0.28	-0.27	-0.26
526	-0.37	-0.37	-0.36	625	-0.09	-0.10	-0.11
527	-0.07	-0.05	-0.06	626	-0.26	-0.27	-0.26
528	-0.36	-0.35	-0.35	627	-0.07	-0.07	-0.07
529	-0.03	-0.04	-0.03	628	-0.25	-0.26	-0.25
				629	-0.04	-0.05	-0.05
				630	-0.24	-0.24	-0.24
				631	-0.24	-0.24	-0.23
				632	-0.17	-0.19	-0.19
				633	-0.20	-0.20	-0.20
				634	-0.24	-0.24	-0.24
				635	-0.22	-0.23	-0.22
				636	-0.19	-0.19	-0.18
				637	-0.23	-0.24	-0.24
				638	-0.24	-0.25	-0.25
				639	-0.23	-0.23	-0.22
				640	-0.27	-0.27	-0.27
				641	-0.23	-0.25	-0.24
				642	-0.23	-0.24	-0.24
543	-0.31	-0.29	-0.30	643	NOT OPERATING		
544	-0.29	-0.29	-0.29	644	-0.21	-0.21	-0.20
701	+0.15	+0.18	+0.17				
702	-0.51	-0.50	-0.50				
703	+0.17	+0.17	+0.17				
704	-0.52	-0.50	-0.51				

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0-12,000 psi

A 500 psi pressure increment was used between 0 and 10,000 psi and a 100 psi increment was used between 10,000 and 12,000 psi. Waiting periods of two minutes were observed between 0 and 10,000 psi and one minute between 10,000 and 12,000 psi.

(2) Creep

A short-term creep test was conducted to determine how the pressure vessel's deflection profile would change over a period of time when the vessel was subjected to a constant external pressure. The same instrumentation as described above was used to monitor the creep strains. Creep data were taken every 15 minutes for four hours with an external pressure of 12,000 psi.

(3) Failure

To determine the correlation between the empirically-determined failure pressure and that actually realized in practice, the vessel was subjected to increasing external pressures until failure occurred. All strain gage instrumentation was removed prior to this test since the strain data obtained beyond the yield point of the material would have little or no value for design or analysis purposes. In addition, the presence of the numerous strain gage wires would have caused a problem in sealing the testing chamber at the anticipated failure pressure.

The loading schedule for this test was 100 psi pressure increments from 12,000 psi to the failure pressure. The vessel was loaded from 0 to 12,000 psi in about 12 minutes. A 30-second stabilization period was observed between application of the 100 psi pressure increments.

3.3.1.2 TEST RESULTS

The following is a summary of the results obtained from the above-described tests:

(1) Deflection

The strain sensitivities (strain per pound of external pressure) determined from the 5,000, 10,000 and 12,000 psi pressure loadings were nearly identical for any given strain gage. The individual gage sensitivities are shown in Table 3.1. In this table a minus sign indicates a compressive strain and a plus sign indicates a tensile strain. The close agreement in strain sensitivities for a given gage under the three loading conditions is an indication of good instrumentation techniques and provides reliable, trustworthy data.

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Examination of the strain sensitivity data of Table 3.1 reveals two regions of concern from the standpoint of exceedingly high compressive strains (stresses) at the operating pressure of 10,000 psi. These areas are the inside of the penetration (gages 702 and 704) and the inside of the upper hemisphere near the heat sink flange (gage 513). It should be noted that gages 702 and 704 detect circumferential strains whereas gage 513 detects longitudinal strains.

To determine the bi-axial stresses associated with the strain sensitivities, the following equations are applicable:

$$S_{\text{Cir}} = \frac{E}{1 - \mu^2} \left[\epsilon_{\text{Cir}} + \mu \epsilon_{\text{Long}} \right] P \quad (1)$$

$$S_{\text{Long}} = \frac{E}{1 - \mu^2} \left[\epsilon_{\text{Long}} + \mu \epsilon_{\text{Cir}} \right] P \quad (2)$$

where: S_{Cir} = Circumferential stress (psi)
 S_{Long} = Longitudinal stress (psi)
 E = Modulus of Elasticity (psi)
 μ = Poissons' Ratio
 ϵ_{Cir} = Circumferential strain sensitivity (in/in/psi)
 ϵ_{Long} = Longitudinal strain sensitivity (in/in/psi)
 P = External loading pressure (psi)

The "maximum" stresses in the areas in question were calculated for the operational external pressure ($P = 10,000$ psi) assuming $E = 18 \times 10^6$ psi and $\mu = 0.30$ (typical values for Titanium 721). The maximum stress in the area of the penetration (gages 702 and 704) was a circumferential compressive stress of 91,000 psi. Although this stress was much larger than those in the remainder of the vessel (except the other area in question), it was well below the compressive yield strength of the 721 alloy (110,000 - 120,000 psi). The maximum stress in the area of the heat sink flange (gages 513 and 514) was a longitudinal compressive stress of 138,000 psi. This stress was greater than the compressive yield strength of the material.

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To determine the deflections associated with the strain sensitivities, the following equations are applicable:

$$\delta \text{ Rad} = \epsilon_{\text{Cir}} r P \quad (3)$$

$$\delta \text{ Long} = \epsilon_{\text{Long}} L P \quad (4)$$

where: $\delta \text{ Rad}$ = radial deflection

$\delta \text{ Long}$ = longitudinal deflection

r = radius

L = length

and ϵ_{Cir} , ϵ_{Long} and P are as defined for equations (1) and (2).

At an external loading pressure of 10,000 psi, the full scale prototype pressure vessel's radial deflections are expected to be as shown in Table 3.2. These deflections are based on the experimentally determined strain sensitivities and the dimensions of the prototype pressure vessel. The expected deflections in the critical design areas are as follows:

- the inside of the electrical feed-through penetration will deflect radially inward about 0.006 inches.
- the heat sink flange will deflect radially inward about 0.013 inches.
- the inside of the cylinder will deflect radially inward about 0.023 inches along its entire length except near the heat sink flange, where the deflection will be about 0.016 inches.
- the cylinder will shorten about 0.004 inches and the lower hemisphere about 0.019 inches resulting in a longitudinal shortening of the composite cylinder and lower hemisphere of about 0.023 inches.

TABLE 3.2

PREDICTED PROTOTYPE PRESSURE VESSEL RADIAL DEFLECTIONS
(BASED ON EXPERIMENTAL RESULTS OF A 1/4-SCALE MODEL TEST)

(P = 10,000 PSI)

<u>LOCATION (GAGE NO.)</u>	<u>DEFLECTION (INCHES)</u>	<u>LOCATION (GAGE NO.)</u>	<u>DEFLECTION (INCHES)</u>
704	0.006	520	0.019
502	0.012	522	0.014
504	0.017	524	0.023
506	0.018	526	0.023
508	0.020	528	0.022
510	0.013	530	0.022
512	0.013	543	0.020
514	0.013	544	0.019
518	0.016		

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Based on the strain sensitivities of gages 510 and 512, it appears that there will be very little or no rotation of the heat sink flange. If there were rotation, the upper and lower portions of the flange would be in different degrees of compression at any given external pressure (the strain sensitivities would not be the same).

(2) Creep

Examination of the creep data revealed that most of the strain gages indicated little or no change in strain after a one-hour "settling out" period. The only area of concern was, again, the inside of the top hemisphere near the heat sink flange (gage 513). Although this area tended to creep throughout the four-hour test, the creep rate was about 0.000020 inches per inch per hour during the last one-half hour of the test and was asymptotically approaching zero. This slow creep is not significant to influence either the failure pressure or the design details.

(3) Failure

The initial design was based on a failure pressure of 15,000 psi and a material compressive yield strength of 110,000 psi. The 721 alloy used to fabricate the 1/4 scale pressure vessel was mechanically tested and had a compressive yield strength of about 120,000 psi. Due to the increased strength, the predicted failure pressure was revised to approximately 17,000 psi. The vessel failed at 17,500 psi. Figure 3.6 is a photograph of the pressure vessel taken after failure showing the location and mode of the fracture.

Figure 3.6 Pressure Vessel Fracture

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3.3.1.3 TEST CONCLUSIONS

The conclusions drawn from the results of the three tests are as follows:

(1) Deflection

Preliminary analysis of the test results shows that all portions of the pressure vessel deformed very nearly as predicted. The only high stress area of concern was on the inside of the upper hemisphere near the heat sink flange. In this area unexpected high longitudinal compressive stresses were observed; however, it appears that these stresses were very localized, and as such, redistributed by the mechanism of cold flow. Since these stresses were highly localized, there was little chance of a premature failure at this location. This fact was demonstrated when the pressure vessel was loaded to failure and failure occurred elsewhere.

(2) Creep

It appears that creep will be no problem after allowance is made for the initial deformation of the vessel (the deformations observed when loading the vessel from 0 to 10,000 psi). The creep observed in this test is denoted "short-term" creep as opposed to a "long-term," extremely slow, creep observed for most metallic materials. The effects of long-term creep cannot be predicted from this experiment, however, the material was selected on the basis of negligible exhibited long-term creep.

(3) Failure

Failure of the vessel at 17,500 psi indicated there should be no difficulty in achieving the desired operating pressure of 10,000 psi with the prototype pressure vessel. As designed, the vessel could have failed at any location at the predicted failure pressure. The location of the failure was in the lower hemisphere and there were no unexpectedly high stresses recorded in the failure area prior to the destruction test.

The close agreement between the predicted and experimentally determined deflections and failure pressure indicated confidence in the predicted performances for design purposes. It appears that the critical deflections will be about as predicted and that there should be no difficulty in achieving the desired operating pressure.

3.3.2 PROTOTYPE PRESSURE VESSEL FABRICATION

Titanium 721 was selected as the alloy to be used for the SNAP-21B deep submergence pressure vessel. This alloy was selected because of its:

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- (1) ability to absorb a large localized strain without propagating a crack (if the case is accidentally dropped or struck by debris, it will not crack open). This characteristic is achieved from a favorable combination of strength and toughness.
- (2) excellent weldability (100 percent joint efficiency).
- (3) low creep rate (no creep observed when tested at 80 percent of its compressive yield strength).
- (4) apparent immunity from delayed cracking phenomenon.

An investigation was conducted to determine the best method of vessel fabrication in terms of cost, time and quality. Chemical, mechanical and fabrication characteristics of Titanium 721 were analyzed, yielding two practical methods of fabrication.

- (1) a two-piece construction consisting of two press forgings or extrusions (one forging would be the top hemisphere and would contain an integral flange and penetration; the other would be the combined cylinder and lower hemisphere). The forging operations would be followed by rough and final machining.
- (2) a multi-component construction consisting of pressed or spun hemispheres, a plate stock flange and penetration, and a forged cylinder. The forming operations would be followed by rough machining, welding and final machining.

Each of the above methods of fabrication has its own peculiar advantages and disadvantages; however, a characteristic of the 721 alloy poses a common disadvantage to both methods. This characteristic is that the alloy achieves its desired compressive, tensile, and impact strength by a crucial combination of heat treating and mechanical working. Due to the relative newness of the 721 alloy, this heat treating-work operation has been commercially developed only for plate products. This means that all forgings, extrusions, etc., must develop the desired properties in the respective forming operation, and to date, these techniques are essentially untried or unproven.

Initially, a rolled and welded cylinder was considered which would permit the multi-component construction to be fabricated entirely from plate products, thus guaranteeing a vessel which would have the desired mechanical properties. After careful analysis, however, it was decided that the longitudinal weld or welds in the cylinder would be undesirable from the standpoint of acting as stiffeners (the strength and toughness of the weld is known

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to be greater than that of the parent metal). This would cause a non-uniform circumferential inward deflection under high external pressure loading conditions.

Further investigation of Titanium fabricators indicated that the desired mechanical properties could be attained in forging, and that fabricators would guarantee the required properties.

A two-component forging method of fabrication was selected to fabricate the pressure vessel. The following are the selection criteria:

(1) Titanium 721 can be welded efficiently, however, a non-welded vessel would be superior because:

- (a) a more homogeneous structure would exist (the weld zone would necessarily have mechanical properties different from those of the parent metal).
- (b) the possibility of welding defects is eliminated.

Note: Titanium 721 was partially selected on the basis of its weldability. Use of this advantage has not been precluded, since upgraded systems will undoubtedly require internal weldments to the pressure vessel.

(2) Closer production control can be exercised during the pressure vessel fabrication since:

- (a) the number of components is minimum.
- (b) a single vendor can fabricate the entire vessel.

(3) Testing and quality checks are reduced in proportion to the reduced number of components.

(4) At a quantity level of three or more, the forging method becomes very economical as compared to welding, due to the distribution of the initial die costs.

To insure the procurement of a structurally-sound pressure vessel that would possess all of the required mechanical properties, a set of technical specifications was drafted to control and determine the quality of the finished product (Appendix E).

3.3.3 PRESSURE VESSEL DESIGN

The pressure vessel design was completed with provisions for mounting of the DC-DC converter, selection of O-ring seal material, and selection of the electrical feed-through fitting (Figure 1.1).

The mounting location for the DC-DC converter is on the opening (penetration) for the feed-through connector inside of the hemisphere. This mounting provides:

- (1) a flat heat transfer surface to the pressure vessel;
- (2) a heat sink remote from the generator heat sink area for dissipation of converter and regulator heat while maintaining electronic components at reliable operating temperatures; and
- (3) easy access to the DC-DC converter during assembly and servicing.

To allow for penetration deflection at operating pressures, a captive-bolt and spring arrangement is used to mount the DC-DC converter.

O-ring seal material was selected as Viton (Minnesota Rubber compound 514J), a fluoroelastomer capable of long-term exposure to atmosphere and seawater over the complete range of specified ambient temperatures, and elevated temperatures experienced by the operating system.

An electrical feed-through fitting has been selected to be made of Titanium alloy 721 to prevent galvanic corrosion between the pressure vessel and the fitting. The fitting is designed to withstand 10,000 psi external pressure with a minimum safety factor of 1.5. A single O-ring seal will be used between the fitting and the vessel. Currently, a stainless steel model of the fitting is being procured to test the integrity of the seal; and evaluation of glass sealing to Titanium 721 is in progress.

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4.0 SNAP 21-B FUEL CAPSULE

4.1 PLANNED OBJECTIVES

The planned objectives are:

- to design and fabricate a fuel capsule capable of complete containment of the fuel under all conditions of intended use, credible accidents, and for a minimum period of 300 years.
- to design and fabricate a fuel capsule for ease of encapsulation of strontium-90, titanate fuel at a government facility.
- to verify the fuel capsule design by conducting a test program using the prototype capsules.

4.2 PROGRAM EFFORT AND ACCOMPLISHMENT THIS PERIOD

- Fuel capsule and inner liner designs are complete. Fabrication of prototype liners has been initiated.
- Fuel capsule drawings and a tentative welding schedule have been forwarded to Oak Ridge National Laboratory for review.
- Fabrication of the fuel capsule end plates has been initiated.
- The fuel capsule test program was completed and submitted for approval. A tentative schedule for testing operations was also submitted.
- Design of a protective exterior capsule liner is in progress.

4.3 TECHNICAL DISCUSSION AND RESULTS

4.3.1 FUEL CAPSULE AND INNER LINER DESIGN

The fuel capsule is designed for 187 watts generation with a power density of 0.8 watts/cc of fuel. Once the required fuel volume of 234 cc was determined, the fuel pellet was designed to give an L/D ratio of one. The pellet dimensions are 2.630 inches in diameter by 2.630 inches long, with a weight of 1.8 pounds.

For ease of handling and decontamination, ORNL suggested that the pellet be double encapsulated. A thin inner capsule (Figure 4.1) of 316L stainless steel with an 0.018 inch wall is used for this purpose. It is 2.688 inches

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Figure 4.1 SNAP 21-B Fuel Capsule and Liner Assembly

in diameter, 2.732 inches long, and weighs less than 1/5 of a pound. The capsule has 0.010 inches clearance on the radius and 0.015 inches clearance on each end for ease of encapsulation

The outer capsule (Figure 4.1) will be constructed to withstand all the required safety tests. It will be constructed of 0.250 inch wall Hastelloy "C" alloy. The capsule has an outside diameter of 3.198 inches and a length of 3.332 inches. The inner capsule slips inside of the outer capsule with the end plates fitting flush against the inner capsule. The end plates of the outer capsule are welded in place in a helium atmosphere. The estimated weight of the capsule and liner is 4.0 pounds, and the completed, fueled capsule weight is 5.8 pounds.

4.3.2 FUEL CAPSULE TEMPERATURE DETERMINATION

To assure interface fits during assembly, an analysis was made of the fuel capsule temperature rise after fueling. The analysis indicated that thermal expansion of the capsule at 700° F will not hinder its insertion in the biological radiation shield. The following is the analytical method.

Assumptions:

- (1) the fuel capsule is loaded with 187 watts of strontium-90 fuel.
- (2) the capsule is surrounded by still air.
- (3) heat transfer is by free convection to air and radiation to surroundings.

A heat balance on the capsule then yields for steady state conditions:

$$Q_{\text{fuel}} = Q_{\text{convections}} + Q_{\text{radiation}} \quad (1)$$

$$Q_{\text{convection}} = f(N_u, G_r) \text{ capsule} \quad (2)$$

$$Q_{\text{radiation}} = f(\epsilon, A, T) \text{ capsule} \quad (3)$$

Where: Q = Heat flow (BTU/hr)
 N_u = Nusselt Number (dimensionless)
 G_r = Grashof's Number (dimensionless)
 ϵ = Emissivity (dimensionless)
 A = Heat transfer area
 T = Temperature (°R)

Assuming the fuel capsule to be small compared to the surroundings, the heat loss through radiation is:

$$Q_{\text{radiation}} = \sigma A_{\text{capsule}} \left[T_{\text{surface}}^4 - T_{\text{surroundings}}^4 \right] \quad (4)^{\text{(Ref 1)}}$$

Where σ = Stefan-Boltzman constant

The heat loss through free convection varies as the geometrical attitude of the capsule varies. Therefore, three different situations were assumed.

- (1) The capsule was assumed to be vertically oriented. The heat loss can then be evaluated using the following expression and equation (6).

$$N_u = 0.378 (Gr)^{1/4} \quad (5)^{\text{(Ref 2)}}$$

- (2) The capsule was assumed to be horizontally oriented. The heat loss can be evaluated using the above expression and letting $X = 2.5 d$.

Where: X = characteristic length (ft.)
 d = capsule diameter (ft.)

- (3) For ease of calculation and a good approximation, the capsule was assumed to be a sphere of equal area and the heat loss was evaluated. ^(Ref 3)

Sample calculations: Case 1

$$Q/A = \phi = 1970 \text{ Btu/hr-ft}^2$$

$$\phi = (h \Delta T)_{\text{convection}} + \left[\sigma \epsilon (T_1^4 - T_2^4) \right]_{\text{radiation}} \quad (6)$$

Where: h = film coefficient of heat transfer

Assume $\Delta T = 300^\circ \text{ F}$

$$N_u = \frac{hX}{k} = 0.378 (Gr)^{1/4} = 16.4 \quad (5)$$

$$h = 1.2 \text{ Btu/hr-ft}^2 \text{ } ^\circ \text{ F}$$

$$\phi = 1.2 (300) + 0.173 \times 10^{-8} \times 0.4 \times 3.96 \times 10^{11}$$

$$\phi = 641 \text{ Btu/hr-ft}^2$$

Therefore: The calculated heat flux is lower than the actual and the assumed temperature difference is too low. Iterating by assuming a new ΔT results in the correct answer.

Using the procedure outlined, the surface temperature of the fuel capsule at rest is predicted to reach a maximum temperature of 700° F.

4.3.3 FUEL CAPSULE ELEVATED TEMPERATURE EVALUATION

Use of radiative rather than conductive heat transfer from the biological shield to the thermoelectric generator will require the capsule to operate at approximately 1350° F rather than 1175° F. An analysis was conducted to determine the structural strength of the capsule at the elevated temperature.

The principal stresses evaluated in the fuel capsule design are:

Collapsing stress

Shear stress on the end plate

Rupture stress on the end plate

The formulas used to determine the stresses are:

$$\text{Collapsing stress} \quad S_c = \frac{2P_o r_o^2}{r_o^2 - r_i^2} \quad (7) \text{(Ref 4)}$$

$$\text{Shear stress} \quad S_s = \frac{P_o r_i}{2t} \quad (8) \text{(Ref 5)}$$

$$\text{Rupture stress} \quad S_r = \frac{3P_o r_i^2}{8t^2} (1 + \mu) \quad (9) \text{(Ref 5)}$$

Where: P_o = external pressure

r_o = outside radius

r_i = inside radius

t = thickness

μ = Poisson's ratio

The ultimate tensile strength of Hastelloy C was assumed as:

115,000 psi at room temperature

80,000 psi at 1400° F

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The predicted stresses at an operating pressure of 10,000 psi are:

	STRESS (PSI)	PREDICTED SAFETY FACTOR		ACTUAL SAFETY FACTOR
		<u>HOT</u>	<u>COLD</u>	<u>COLD</u>
Collapse	-69,300	1.15	1.66	2.37
Shear	26,900	1.49	1.92	No Shear Failure
Rupture	35,000	2.26	3.25	20

The actual fuel capsule safety factors were determined from the results of the prototype fuel capsule pressure test discussed previously in quarterly report MMM(3321)-3.

Elevated temperature operation of the fuel capsule will induce an exothermic eutectic formation between the Hastelloy "C" fuel capsule material and the Uranium-8 Molybdenum biological shield material. To prevent this eutectic formation, a double-cup outer liner (Figure 4.1) has been designed. This liner provides physical separation of the capsule and shield. Titanium has been selected as the outer liner material since it is compatible with both uranium and Hastelloy "C." Design of this fuel capsule component is in progress.

4.3.4 FUEL CAPSULE TEST PROGRAM

A proposed fuel capsule test program was drafted and submitted for approval. The purpose of the test program is to verify the fuel capsule design and to insure the integrity of the fuel capsule for all credible conditions of transport and use. Following is a tabular summary of the test program and a description of each of the proposed tests.

References:

1. "Introduction to Heat Transfer," 3rd Edition, Brown and Marco - McGraw Hill.
2. "Heat and Mass Transfer," 2nd Edition, Eckert and Drake - McGraw Hill.
3. "Laminar Free Convections from a Sphere," Chiang, Ossin and Tien - ASME 63-WA-103.
4. "Design of Machine Elements," 2nd Edition, Spotts - Prentice-Hall.
5. "Mechanical Engineering Handbook," 6th Edition, Marks - McGraw Hill.

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TABLE 4.1

SNAP 21-B FUEL CAPSULE TESTING SUMMARY

TEST	NUMBER OF CAPSULES TO BE TESTED	DESCRIPTION	METHOD OF ANALYSIS
Hydrostatic Pressure	4	The fuel capsule will be closed at both ends and inserted into an isostatic hydraulic press. The external pressure will be raised until the capsule ruptures or until the press limit of 29,500 psi is reached.	Visual and metallographic examinations of failure. Metallographic examinations of weld structure if no failure occurs. Redesign and retest if results so indicate.
Fire	2	Capsule is placed in a furnace and subjected to conditions as outlined in the two-hour fire test ASTM E 119-61.	Visual examination for failure. Hydrostatic pressure check for failure. (Weigh for detection of fluid intrusions.) Helium leak check.
Thermal Shock	2	Capsule is heated to maximum fire test conditions (1850° F) and is held until equilibrium is reached. The capsule is then removed and dropped into 50° F water.	Visual examination for stress cracking. Hydrostatic pressure check for failure. Helium leak check.
Vibration	1	Capsule will be tested for a one-hour period. Maximum loading will be 16 g's with frequency varying from 10 - 10,000 c/sec.	Visual examination for failure. Helium leak check.
Impact (on Granite Surface)	15	Facilities at the Sandia Corp. Laboratory, New Mexico, will be used for impact testing of the capsules. Impact orientations will be transverse, head on, 45° to head on, end on, and 45° to end on. Three capsules at each orientation for a total of 15 capsules.	Visual examinations. Helium leak check. Hydrostatic pressure test. Metallographic examinations of failure.
TOTAL NUMBER OF TEST CAPSULES	24		

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Figure 5.1 SNAP 21-B Shield and Plug Assembly

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5.0 SNAP 21-B BIOLOGICAL RADIATION SHIELD

5.1 PLANNED OBJECTIVES

The planned objectives are:

- to design and fabricate a biological radiation shield to limit the nuclear radiation level to 200 mr/hr. at the system surface and 10 mr/hr. at one meter from the system surface.
- to design the radiation shield to permit ease of repair or replacement of the thermoelectric sub-system without exceeding the maximum permissible radiation exposure limits as established by the AEC; and without the use or need of hot cell facilities.

5.2 PROGRAM EFFORT AND ACCOMPLISHMENT THIS PERIOD

- The shield design (Figure 5.1) was sufficiently completed for quotation purposes. The final design will be completed as soon as all interfaces are defined.

5.3 TECHNICAL DISCUSSION AND RESULTS

The biological radiation shield has been designed to limit the system radiation level to 200 mr/hr. at the surface of the pressure vessel when the fuel capsule is loaded with 30,000 curies of strontium-90 titanate. Figure 5.1 shows the shield as submitted for fabrication quotations and is essentially the design which will be used in the SNAP-21 system. Shield design modifications are not expected, but actual fabrication of the shield will not begin until the shield-insulation interface and shield-generator interface are completely defined.

6.0 SNAP 21-B DC-DC CONVERTER

6.1 PLANNED OBJECTIVES

The planned objectives are:

- to develop a regulated power conditioner (DC-DC converter) to provide 10 watts minimum power output at 24 volts with an external rated load of 57.6 ohms.
- to develop a DC-DC converter with a minimum efficiency of 85% and a five-year life.

6.2 PROGRAM EFFORT AND ACCOMPLISHMENT THIS PERIOD

- A study of principal converter losses was conducted to identify the best areas for efficiency improvement.
- An initial experimental DC-DC converter was assembled and exhibited an efficiency of 89.5%.
- Reliability is being assured through conservative design and procurement of known reliability components.

6.3 TECHNICAL DISCUSSION AND RESULTS

6.3.1 STUDY OF PRINCIPAL DC-DC CONVERTER LOSSES

Previous converters which had operating levels similar to those required for the SNAP 21-B generator exhibited 84% efficiency when connected in their circuits. Since these converters operated between 5 volts input and 15 volts output, and used the series configuration (Appendix F), their basic efficiency was only about 78%. If these converters were operated between 5 volts input and 25 volts output as is required for this program, they would exhibit an efficiency of only 81.25%.

To identify the best areas for efficiency improvement, a study of the principal losses was conducted. This study indicated that the first area for improvement should be reduction of rectifier losses. Typical silicon rectifiers drop about one (1) volt. This voltage drop represents a power loss of about 4%.

If the silicon rectifiers are replaced with transistors to produce a driven rectifier (Figures 6.1 and 6.2), the rectifier loss is reduced to about 0.1 volt, which is about 0.4%. To this voltage loss must be added the drive power of about 0.4%. The net gain in efficiency is, therefore, about 3.2% through the use of the driven rectifiers. These rectifiers will be incorporated in the SNAP 21-B DC-DC converter.

Figure 6.1 DC-DC Converter: Standard Royer Circuit

The next most logical area for efficiency improvement is reduction of the switching losses. Switching losses can be reduced in at least four ways:

- (1) selection of transistors with a good trade-off between saturation voltage and switching speed.
- (2) the operating frequency can be reduced until the transformer losses start to increase faster than switching losses decrease.
- (3) the starting bias supply can be chosen so that the transistors operate near maximum current gain during starting.
- (4) a separate oscillator can be used to drive the converter.

Figure 6.2 DC-DC Converter: Standard Royer Circuits with Driven Rectifiers Q_3 and Q_4

Tests were conducted on seven of the most promising transistor families. The switching times of four of the transistor families were twice as long as the originally selected transistor. The switching time of one transistor was $1/5$ as long as that of the original transistor, but its saturation voltage was too high. One transistor family had a switching time $1/3$ as long as that of the original transistor and also had lower saturation voltage. This last transistor family is the one that has been selected for the main switching transistors (Q_1 and Q_2 of Figure 6.2).

Further switching speed tests indicated that the original driven rectifier transistors (Q_3 and Q_4 of Figure 6.2) were the best choice.

The optimum operating frequency was calculated to be between 125 cps and 250 cps for the original transistors. For the high-speed transistors, the optimum operating frequency was determined to be about 250 cps. The

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optimum operating frequency depends not only on the transistor switching speed but also on the transformer core material. If either is changed, the optimum operating frequency would change.

The action of the DC-DC converter depends on converting the DC input voltage to a square wave voltage. This square wave voltage is stepped up to a higher voltage by the turns ratio of the transformer. The higher voltage is rectified to convert it back to DC. The resulting output voltage is higher than the input voltage.

When the square wave voltage goes from positive to negative, the converter is said to switch from one state to the other. This change of state is started when the transformer core "saturates." Saturation of the core causes one pair of transistors, say Q_1 and Q_3 of Figure 6.2, to turn off and in turn causes the other pair of transistors, say Q_2 and Q_4 to turn on. This state will continue until the core saturates in the opposite direction. Saturation of the core will cause Q_2 and Q_4 to turn off and Q_1 and Q_3 to turn on. This action goes through a complete cycle 250 times per second.

The switching of the DC-DC converter from one state to the other depends on the oscillations that can be produced by the circuit when the transformer is saturated (Appendix F). These oscillations can be promoted by operating the transistors at a current level which gives maximum current gain. This is considered to be the optimum starting bias level. This amount of starting bias will give the best possible starting and will result in the least amount of power loss during switching. The SNAP 21-B converter has been designed to take advantage of this fact.

A converter circuit which is driven from a separate oscillator (Figure 6.3) would seem to have a theoretical advantage over a standard converter circuit (Figure 6.2). The advantage is a reduction in losses during the switching interval along with a possible reduction in transformer losses. Laboratory pulse generators were used along with the present converter circuit to simulate a driving oscillator (Figure 6.3). This technique should be capable of simulating the most refined driving oscillators, but only a 0.2% improvement in efficiency was noted. The principal reason for the lack of adequate improvement is that the present switching losses are only a small percent of the total losses. An additional factor is that the transformer losses do not vary greatly with changes in operating frequency when the operating voltage is held constant.

The final area for possible efficiency improvement is in the transformer. Considerable work has been done in the past at 3M on formulas for calculation of the proper core sizes and for calculation of the transformer

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Figure 6.3 DC-DC Converter: Driven Oscillator Circuit

losses. These formulas were used for this converter to investigate the efficiencies of various transformer designs without having to construct and test the transformers. Among those cores having adequate dimensions, a variation of 3% in efficiency was calculated. Due to the extremely lengthy calculations involved, the study was necessarily limited to tape-wound cores. Calculations of the efficiencies of the most promising cores indicate that the maximum efficiency point for the present core material has been reached.

Investigation of other core materials for the DC-DC converter would necessitate an extended development effort at considerable expense. However, system upgrading to higher output power levels will require transformer core studies requiring extensive calculations. A computer analysis based on the presently-developed formulas would expedite calculations and permit the analysis of a greater number of transformer cores for the upgraded systems.

6.3.2 EXPERIMENTAL DC-DC CONVERTER

Figure 6.4 is a graph of the efficiency of the first experimental converter (Figure 6.5). The peak efficiency at 7 watts was 0.25% higher than the efficiency at the operating power level of 8 watts (see "Series Configuration," Appendix F). This data agrees quite well with the calculated data and was a good confirmation of the design equations.

Figure 6.4 Variation of DC-DC Converter Efficiency with Output Power. Basic Circuit Only.

It should be noted that these converters will operate from a thermoelectric generator which has a low voltage tap for the starting bias. This technique results in an efficiency improvement of about 4% in this converter circuit (Figure 6.2).

The first experimental converter exhibited an efficiency of 82.5% as designed. The addition of speed-up capacitors (also a part of the design) increased the efficiency to 84%. These efficiencies were for the converter alone. When connected in the series configuration, the efficiencies would be 85.3% and 86.4% respectively. Neither the optimum transformer nor the high-speed transistors were available at the time that this test was conducted.

Figure 6.5 Experimental DC-DC Converter

Addition of the optimum transformer and the high-speed transistors, along with reduction of operating frequency to 250 cps, resulted in an efficiency of 89.5%. Although the efficiency was observed on a converter that did not use specially-prepared or selected components, a variation in efficiency of $\pm 2\%$ must be assigned to this figure until further investigations confirm its repeatability. This wide range is necessary because the observed efficiency is higher than the calculations indicate. If necessary, it would probably be possible to select components to maintain 90% efficiency. In addition to the above uncertainty, the regulator will absorb from 0.5% to 1 % of the power output and reliability considerations for the transformer may decrease efficiency slightly. The exact additional losses cannot be determined until after the transformer manufacturers respond to the formal inquiry which has been sent to them.

6.3.3 DC-DC CONVERTER RELIABILITY

Basic reliability of the converter design is assured by 3M design techniques (Appendix F). These techniques take into account the maximum expected variations in the component parameters. One parameter that was

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Figure 6.6 Evaluation of Drift in V_{BE} for a Medium Power DC-DC Converter

considered was the base-to-emitter voltage. Extrapolation of some data taken by transistor manufacturers under conditions of 110°C (232°F) storage indicated an increase of 10% per 1000 hours. If linear extrapolation is used, an increase of 440% at 5 years would result. Since allowing for this much change in the base-to-emitter voltage would reduce the efficiency, the transistor manufacturer was contacted. The manufacturer produced life test data which showed that the base-to-emitter voltage begins to level off after 2000 hours. This data went out to only 5000 hours, but other data which might be used to estimate changes in the base-to-emitter voltage indicates that base-to-emitter voltage may start to reduce after 8000 hours. This data was carried out to 10,000 hours.

These life tests were carried out at 110°C , whereas the SNAP 21-B circuit will operate below 70°C (150°F). The chemical processes that cause changes in the transistor characteristics are accelerated by a factor of two when the temperature is increased by 6°C (10.8°F) to 10°C (18°F). If the worst factor, 10°C , is used, then the aging process at 70°C is only 1/16 of that at 110°C . Assuming that this extrapolation is valid, the leveling off of the base-to-emitter voltage would occur at about 32,000 hours and would remain constant until about

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128,000 hours. This is much longer than the 5-year life of 44,000 hours. Figure 6.6 is a graph illustrating the changes. The time axis extrapolation was on the basis of a doubling in degradation for each 10°C increase in temperature. Since the actual degradation seems to double at a rate closer to every 5°C , the extrapolated time scale is conservative. Other effects could invalidate the extrapolation at periods beyond 9 years; however, the extrapolation to 5 years is considered valid.

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7.0 SNAP 21-B INSULATION ENVELOPE

7.1 PLANNED OBJECTIVES

The planned objectives are:

- to develop an insulation envelope with the thermal conductivity of the insulation to be not greater than 0.0005 BTU/hr-ft-°F during operation.
- to develop an insulation envelope with a five-year life to withstand the environmental conditions of the system as outlined in the technical provisions.

7.2 PROGRAM EFFORT AND ACCOMPLISHMENT THIS PERIOD

- A subcontract was negotiated with the Linde Division of Union Carbide Corporation to develop the SNAP-21B insulation envelope.
- Boundary conditions were established and a preliminary insulation envelope design is currently being mechanically and thermally analyzed.

7.3 TECHNICAL DISCUSSION AND RESULTS

A preliminary insulation envelope design (Figure 7.1) was conceived for purposes of establishing boundary conditions of the envelope. The following conditions were established:

- (1) The expected external shell temperature is approximately 50°F to 68°F.
- (2) The heat source temperature profile ranges from 1310°F to 1323°F.
- (3) The generator case parallel to the insulation bellows is in thermal gradient from 1100°F to 100°F.
- (4) The insulation envelope must withstand the system environmental conditions with consistent thermal performance while supporting the radiation shield.
- (5) Motion of the radiation shield is restricted to 0.005 in. both diametrically and axially.
- (6) A condition of potential eutectic formation exists between the radiation shield and the insulation envelope inner shell.

The insulation envelope concept (Figure 7.1) shows a stepped bellows at the access opening for the thermoelectric generator. A stepped bellows is expected to reduce heat losses from the generator case to the bellows

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and allow thermal insulation to be inserted in the void space between the generator and bellows.

A eutectic formation will occur between U-8 MOLY (radiation shield material) and metals of Group VIII of the periodic table at temperatures between 1337°F and 1364°F and with chromium at higher temperatures.

Although the radiation shield temperature profile is less than 1337°F during operation, protection against this eutectic formation will be provided to allow for temperature excursions during assembly, generator replacement, and transportation.

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NOTES:
1. THESE DIMENSIONS ARE REFERENCED TO HEAT SOURCE DRAWING #PD-37-1057
ACTUAL DIMENSIONS SHOULD BE SUCH THAT THE HEAT SOURCE WILL FIT INTO INSULATION SYSTEM CAVITY SO THAT RELATIVE MOVEMENT ALONG "X" AND "Y" AXIS WILL NOT EXCEED .005 AT 70° F

Figure 7.1 SNAP 21-B Insulation Envelope Profile

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APPENDIX A

THERMOELECTRIC MATERIALS ELECTRICAL AND MECHANICAL PERFORMANCE.

1. THERMOELECTRIC COUPLE PERFORMANCE

a. Determination of Thermoelectric Properties

The thermoelectric properties are the Seebeck Coefficient S , the electrical resistivity ρ , and the thermal conductivity k , which are functions of the temperature T . These properties cannot be measured directly but are determined by finding the dependence of measurable quantities on an independent variable, usually the temperature. The thermoelectric properties are defined by definite integrals where the experimental variables appear as the limits on the integral. It will be convenient to first treat a general case.

$$\phi(\alpha) = \int_{u_0(\alpha)}^{u_1(\alpha)} f(x, \alpha) dx \quad (1)$$

Now apply "Leibniz' Rule for Differentiation Under the Integral Sign" to equation (1) and obtain:

$$\frac{d\phi}{d\alpha} = f(u_1, \alpha) \frac{du_1}{d\alpha} - f(u_0, \alpha) \frac{du_0}{d\alpha} + \int_{u_0(\alpha)}^{u_1(\alpha)} \left[\frac{\partial f(x, \alpha)}{\partial \alpha} \right] dx \quad (2)$$

These mathematical procedures can now be applied to the determination of the thermoelectric properties.

- Seebeck Coefficient

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The voltage E developed across a thermoelectric leg having a Seebeck Coefficient S , operating between a cold junction temperature T_c and a hot junction temperature T_h is given by

$$E = \int_{T_c}^{T_h} S(T) dT \quad (3)$$

Now apply equation (2) to equation (3) below:

$$\frac{dE}{dT_h} = S(T_h) \frac{dT_c}{dT_h} - S(T_c) \frac{dT_c}{dT_h} + \int_{T_c}^{T_h} \left[\frac{\partial S(T)}{\partial T_h} \right] dT \quad (4)$$

Since S is a property of the system $\frac{\partial S(T)}{\partial T_h} = 0$ and Equation (4) becomes

$$S(T_h) = \frac{dE}{dT_h} + S(T_c) \frac{dT_c}{dT_h} \quad (5)$$

The measurement technique used is to measure the voltage E while varying the junction temperatures T_h and T_c and applying Equation (5) to the experimental data.

It is often possible experimentally to hold the cold junction temperature constant in which case $\frac{dT_c}{dT_h} = 0$, and Equation (5) reduces to

$$S(T_h) = \frac{dE}{dT_h} \quad (6)$$

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● Thermal Conductivity

It will be assumed that the heat flow is one dimensional. Convective heat losses can be effectively eliminated by providing a high vacuum environment. Although radiative heat losses exist, it has been shown that in a plane perpendicular to the cylindrical axis the temperature on the surface of the sample is not significantly different than the temperature in the center.

For one dimensional heat flow, the heat flow Q is given by

$$Q = k(T) A \frac{dT}{dx} \quad (7)$$

where A is the cross sectional area.

If the leg has a length L , boundary conditions can be defined as

$$\begin{aligned} T &= T_c \quad \text{at} \quad X = 0 \\ \text{and} \quad T &= T_h \quad \text{at} \quad X = L \end{aligned}$$

Equation (7) can be written as

$$Q dx = k(T) A dT$$

and integrating both sides

$$\int_0^L Q dx = \int_{T_c}^{T_h} k(T) A dT$$

Since Q is independent of x

$$Q \frac{L}{A} = \int_{T_c}^{T_h} k(T) dT \quad (8)$$

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Now apply Equation (2) to (8) below:

$$\frac{L}{A} \frac{dQ}{dT_h} = k(T_h) \frac{dT_c}{dT_h} - k(T_c) \frac{dT_c}{dT_h} + \int_{T_c}^{T_h} \left[\frac{\partial k(T)}{\partial T_h} \right] dT \quad (9)$$

Since $k(T)$ is a property of the system $\frac{\partial k(T)}{\partial T_h} = 0$, Equation (9) becomes

$$k(T_h) = \frac{L}{A} \frac{dQ}{dT_h} + k(T_c) \frac{dT_c}{dT_h} \quad (10)$$

The experimental procedure is to measure the heat conducted across the cold junction of the leg, Q_m , into a calorimeter. But Q in Equation (10) is the heat conducted across the hot junction, so radiation losses from the sides of the element must be taken into account. The heat radiated from a differential element, dx , radiating to a black body at temperature T_a , is given by

$$dQ_r = \pi D \epsilon \sigma (T^4 - T_a^4) dx \quad (11)$$

where ϵ is the emissivity

and D is the diameter

The independent variable in (11) can be changed from x to T by

$$dx = \frac{dx}{dT} dT$$

But from (7)

$$\frac{dx}{dT} = \frac{k(T)}{Q} A \quad (12)$$

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Substituting (12) into (11)

$$dQ_r = \frac{\pi D \epsilon \sigma}{Q} A k(T) (T^4 - T_a^4) dT$$

or

$$Q_r = \frac{\pi D \epsilon \sigma}{Q} A \int_{T_c}^{T_h} k(T) (T^4 - T_a^4) dT \quad (13)$$

The procedure is to calculate $k(T)$ from (10), then use this result to find Q_r from (13). Then find a new $k(T)$ from (10) using $Q = Q_m + Q_r$. Convergence between (10) and (13) will occur rapidly since in general Q_r will be small compared to Q_m .

● Electrical Resistivity

The resistance R of a leg of length L is given by

$$R = \frac{1}{A} \int_0^L \rho(T) dx \quad (14)$$

If a temperature difference exists across the leg then equation (14) cannot be used to determine $\rho(T)$ unless the explicit relation between T and X is known. The independent variable in (14) can be changed from X to T to give

$$R = \frac{1}{A} \int_{T_c}^{T_h} \frac{\rho(T)}{\frac{dT}{dx}} dT \quad (15)$$

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By substituting Equation (12) into (15)

$$R = \int_{T_c}^{T_h} \frac{\rho(T)k(T)}{Q} dT. \quad (16)$$

Now apply (2) to (16) and get

$$\frac{dR}{dT_h} = \frac{\rho(T_h)k(T_h)}{Q} \frac{dT_h}{dT_c} - \frac{\rho(T_c)k(T_c)}{Q} \frac{dT_c}{dT_h} + \int_{T_c}^{T_h} \frac{\partial}{\partial T_h} \left[\frac{\rho(T)k(T)}{Q} \right] dT. \quad (17)$$

Expanding the third term of (17) we get:

$$\int_{T_c}^{T_h} \frac{\partial}{\partial T_h} \left[\frac{\rho(T)k(T)}{Q} \right] dT = \int_{T_c}^{T_h} \left\{ \frac{Q \left[\rho(T) \frac{\partial k(T)}{\partial T_h} + k(T) \frac{\partial \rho(T)}{\partial T_h} - \rho(T)k(T) \frac{\partial Q}{\partial T_h} \right]}{Q^2} \right\} dT.$$

But since: $\frac{\partial k(T)}{\partial T_h} = \frac{\partial \rho(T)}{\partial T_h} = 0$, the third term reduces to:

$$- \int_{T_c}^{T_h} \frac{\rho(T)k(T)}{Q^2} \frac{\partial Q}{\partial T_h} dT.$$

Since Q and $\frac{\partial Q}{\partial T_h}$ are independent of T , the third term becomes:

$$- \frac{\frac{\partial Q}{\partial T_h}}{Q} \int_{T_c}^{T_h} \frac{\rho(T)k(T)}{Q} dT.$$

Substituting from (16), the third term becomes $-\frac{\frac{\partial Q}{\partial T_h}}{Q} R$,
so Equation (17) becomes:

$$\frac{dR}{dT_h} = \frac{\rho(T_h)k(T_h)}{Q} - \left[\frac{\rho(T_c)k(T_c)}{Q} \right] \frac{dT_c}{dT_h} - \frac{\frac{\partial Q}{\partial T_h}}{Q} R. \quad (18)$$

Solving (18) for $\rho(T_h)$ and using (10) for $k(T_h)$: $\rho(T_h) =$

$$\frac{Q}{\frac{L}{A} \frac{dQ}{dT_h} + k(T_c) \frac{dT_c}{dT_h}} \left[\frac{dR}{dT_h} + \frac{\rho(T_c)k(T_c)}{Q} \frac{dT_c}{dT_h} + \frac{R}{Q} \frac{dQ}{dT_h} \right] \quad (19)$$

The Q in Equation (19) is the heat flowing past the hot junction so the measured heat through the cold junction must be corrected for radiation as in the previous section.

Equations (3) through (19) describe a procedure for determining all the thermoelectric properties in a single in-gradient experiment. In addition, the measured quantities E , R and Q completely describe the performance of the thermoelectric leg as a function of the junction temperatures.

b. Calculation of the Performance of a Thermoelectric Leg

The object of this section is to present a method for predicting the performance of a thermoelectric leg from the material properties. In addition to one dimensional heat flow, it is necessary to assume, further, that the temperature distribution in the leg does not change between open circuit and closed circuit (i.e. matched load) conditions for fixed junction temperatures. This has been verified experimentally to be a good approximation for present day thermoelectric materials.

Consider a thermoelectric leg of length L , cross sectional area A , having properties $S(T)$, $\rho(T)$, and $k(T)$ operating between a heat source at temperature T_h and a heat sink at temperature T_c . Assume the leg to be in a non-convective environment and that radiation losses can be calculated as above. Define an axial coordinate X , such that,

$$T = T_c \text{ at } X = 0$$

$$T = T_h \text{ at } X = L$$

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For one dimensional heat flow, the conducted heat Q_c , is given by

$$Q_c = k(T) A \frac{dT}{dx} \quad (20)$$

Neither Q_c nor A depend on either T or X so the variables can be separated and equation (20) can be integrated

$$\frac{Q_c}{A} \int_0^L dx = \int_{T_c}^{T_h} k(T) dT$$

or

$$Q_c \frac{L}{A} = \int_{T_c}^{T_h} k(T) dT \quad (21)$$

The open circuit or Seebeck voltage E , is given by

$$E = \int_{T_c}^{T_h} S(T) dT \quad (22)$$

The resistance of the leg R , is given by

$$R = \frac{1}{A} \int_0^L \rho(T) dx \quad (23)$$

In order to perform the integration in (23) the independent variable is first changed from X to T .

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$$R = \frac{1}{A} \int_{T_c}^{T_h} \rho(T) \frac{dx}{dT} dT \quad (24)$$

$$\text{from (20)} \quad \frac{dx}{dT} = \frac{k(T)A}{Q_c} \quad (25)$$

So (24) becomes

$$R = \int_{T_c}^{T_h} \frac{\rho(T)k(T)}{Q_c} dT \quad (26)$$

And since Q_c is independent of T

$$RQ_c = \int_{T_c}^{T_h} \rho(T) k(T) dT \quad (27)$$

It was shown that the properties can be measured as a function of the temperature. Using the "Method of Least Squares", these properties can be expressed explicitly as power series of the temperature. When this has been done the right hand side of Equations (21), (22) and (27) can be calculated directly. For example, suppose that

$$S(T) = A_0 + A_1 T + A_2 T^2 + \dots A_n T^n$$

$$\text{Then } \int_{T_c}^{T_h} S(T) dT = A_0(T_h - T_c) + \frac{A_1}{2} (T_h^2 - T_c^2) + \dots \frac{A_n}{n+1} (T_h^{n+1} - T_c^{n+1})$$

Thus Q_c is found from (21), E from (22) and R from (27).

The only variable remaining is the way the leg is loaded, which will specify the current. For the general case, let the current be I .

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The power P delivered to the load is

$$P = I(E - IR) \quad (28)$$

The Peltier heat absorbed at the hot junction, Q_{p_h} , is

$$Q_{p_h} = S(T_h) T_h I \quad (29)$$

The Peltier heat rejected at the cold junction, Q_{p_c} , is

$$Q_{p_c} = S(T_c) T_c I \quad (30)$$

The Joulean heat generated in the leg, Q_j , is

$$Q_j = I^2 R \quad (31)$$

The Thompson heat generated (or absorbed) in the leg, Q_t , is

$$Q_t = I \int_{T_c}^{T_h} T \frac{dS}{dT} dT \quad (32)$$

The efficiency η , is given by the ratio of the power delivered to the load to the power required at the hot junction, which is

$$\eta = \frac{I(E - IR)}{Q_c + Q_{p_h}} \quad (33)$$

The heat that must be dissipated at the cold junction, Q_d , is

$$Q_d = Q_c + Q_{p_c} \quad (34)$$

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Notice that neither the Joulean nor the Thompson heats appear in (34). This is because it was assumed that the temperature distribution is unchanged between open and closed circuit conditions, so these heat terms are included in Q_c .

In practice, the leg is often operated at matched load conditions so as to obtain maximum power, in which case the matched load current I_{ML} , is

$$I_{ML} = \frac{E}{2R} \quad (35)$$

and the quantities defined in (28) through (34) can be found using this particular current.

● Segmented Legs

When operating a thermoelectric leg across a large temperature difference it is often desirable to place two or more thermoelectric materials thermally in series so as to obtain the best properties in a given temperature range. In segmentation one knows at what temperature the segmenting junction should be, but in order to construct such a leg, the lengths of each segment must be known. The method described above can answer this question, and in addition can describe the performance of the entire leg.

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Consider a thermoelectric leg consisting of N segments operating under the conditions previously described. Now look at the i^{th} segment, which has a length l_i and cross sectional area A_i (each segment must have uniform cross sectional area, but each of the N segments need not have the same cross sectional area), and operates between temperatures T_{i-1} and T_i .

$$E_i = \int_{T_{i-1}}^{T_i} S_i(T) dT \quad (36)$$

$$Q_i l_i = A_i \int_{T_{i-1}}^{T_i} k_i(T) dT \quad (37)$$

(Notice that Q_c is the same for all segments)

$$\text{and } R_i Q_i = \int_{T_{i-1}}^{T_i} i(T) k_i(T) dT \quad (38)$$

A similar set of equations can be written for each of the N segments, and as before, the right hand side of (36), (37) and (38) can be calculated directly. The length of the entire leg L , containing the N segments is

$$L = \sum_{i=1}^N l_i \quad (39)$$

Now add together all equations of type (37) and get

$$Q_c = \frac{1}{L} \sum_{i=1}^N \int_{T_{i-1}}^{T_i} A_i k_i(T) dT \quad (40)$$

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Since Q_c is now known it can be substituted into (37) to determine the l_i and into (38) to determine the R_i . For the entire leg, it follows that

$$E = \sum_{i=1}^N E_i \quad (41)$$

and

$$R = \sum_{i=1}^N R_i \quad (42)$$

Thus the length of each segment is known and since E , R , and Q_c are known for the entire leg, the other performance parameters for the leg can be calculated as before.

c. Comparison with Experiment

● Measurement of Thermoelectric Properties.

The principal advantage of the measurement technique described in Section 4-B, 1-a, is that all of the thermoelectric properties can be measured simultaneously. There are, however, other methods of measuring each of the properties in separate experiments.

The most common method of measuring the Seebeck Coefficient is to place the leg between a variable temperature heat source and a fixed temperature heat sink. Under these conditions Equation (6) applies and

$$S(T_h) = \frac{dE}{dT_n}$$

The most common method of measuring the electrical resistivity is to maintain the entire sample at a fixed temperature. Under these conditions equation (14) becomes

$$R = \frac{L}{A} \rho(T)$$

or $\rho(T) = RA/L$; (since $\rho(T)$ is independent of x)

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The experiment can be repeated at several different temperatures to get ρ as a function of T .

The accuracy with which thermoelectric performance can be predicted from property measurements and analysis attests to the reliability of the foregoing procedures.

● Performance of Thermoelectric Legs

The good agreement between predicted and measured performance of a thermoelectric leg will be illustrated in this section. The properties used in the calculation were measured by both of the techniques described above.

Figures A-1, A-2 and A-3 shown predicted and measured values of E vs T_h , R vs T_h , and Q vs T_h respectively of an N type thermoelectric leg which has a diameter of .375 inches and a length of .225 inches. The agreement between measured and predicted values seems quite acceptable. Differences can arise through experimental errors, or to the fact that these thermoelectric materials are doped semiconductors whose properties are very dependent on the chemical composition, and some compositional variation is to be expected between batches.

Figures A-4 and A-5 show predicted and measured values of E vs T_h and R vs T_h for a model 3M-19 thermoelectric generator which contains 80 thermocouples connected electrically in series and thermally in parallel. The agreement between predicted and measured values of the voltage is quite good. The generator resistance requires some additional comment. This resistance is made up of several parts: the resistance of the individual thermoelectric legs, the resistance of hot junction and cold junction electrodes, the resistance of lead wires and terminals, and contact resistance between the thermoelectric

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Figure A-2 Calculated and Measured Voltage vs Hot Junction Temperature

Figure A-1 Calculated and Measured Voltage vs Hot Junction Temperature

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Figure A-3 Calculated and Measured Heat Flux vs. Hot Junction Temperature

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Figure A-5 Calculated and Measured Voltage vs. Hot Junction Temperature (Complete Generator)

Figure A-4 Calculated and Measured Resistance vs. Hot Junction Temperature (Complete Generator)

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legs and the electrodes. Better agreement between measured and predicted values could be achieved by merely adjusting one of the resistance parameters other than the thermoelectric leg resistance. The most important result is the fact that the temperature dependence of the generator resistance has been accurately predicted. This result together with Equation (27) allows the conclusion that the temperature dependence of the thermal conductivity is also correct for if this were not so, the correct resistance could not be predicted as a function of the temperature.

2. THERMOELECTRIC COUPLE MECHANICAL DESIGN

The literature on thermoelectricity is predominantly devoted to the electrical and thermal properties of materials. On the other hand, very little information is available concerning the mechanical properties of these same materials. This omission has resulted in some unfortunate mechanical failures.

The high thermal gradients that are imposed in a thermoelectric structure may produce stresses that result in fracture of the usually friable thermoelectric materials. It is the purpose of this discussion to point out principle stress areas that are encountered in the design of thermoelectric devices, and to suggest possible corrective measures. Unfortunately, empirical analysis can not be as quantitatively applied to the mechanical problems as it may be to the electrical and thermal design. This is due, in no small measure, to the fact that mechanical characteristics of thermoelectric materials are more variable, at the present state of the art, than the thermal and electrical characteristics.

In thermoelectric material dynamic thermal or mechanical loads may result in highly localized stresses. This characteristic is intrinsic to good thermoelectric material, in that they have a low lattice thermal conductivity, which by definition presents a high impedance to thermal or accoustical phonons so that stress gradients will be necessarily higher than may be expected for metallic materials under similar dynamic load. Unfortunately, this problem does

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not lend itself to a tractable analytical solution. The procedure to be avoided is the lumping of the dynamic problems with the other problems that shall be discussed later. Trial and error method, at the present level of mechanical design is unavoidable; but analysis can be great value in the expediting the final result.

The most important mechanical difficulties encountered in the design of thermoelectric devices are the result of thermal expansion. It is necessary to identify the principle "thermal expansion" induced stresses in order to establish a procedure for mechanical design. Some pertinent physical properties data are given in Figures A-6 and A-7.

An important stress area that is subject to thermal expansion induced stresses is where the thermoelectric member is joined to a metallic connecting member. Ideally, the material selected for the metallic contacting member would have the same thermal expansion coefficient as the thermoelectric material. The requirement that the metallic connector be chemically inert with respect to the thermoelectric material limits the number of choices available. If the metallic member has a thermal expansion coefficient less than the thermoelectric element the stress in the element will be compressive and less objectional. Also, the stress due to the thermal expansion difference will be highest at the periphery of the thermoelectric member and will be proportional to the diameter of the element. This means that some diameter will be within the stress limitations of the thermoelectric element.

Another mechanical reaction that may occur at the metal to thermoelectric member junction, less obvious than the simple thermal expansion difference referred to above, is the stress due to a thermal gradient. When all members are at the same temperature there is no relative distortion and the metallic member and the thermoelectric element, if joined, would be stress free. However, when a thermal gradient is imposed, there is a distortion due to differential

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Figure A-6 Stress-Strain Relationship for Equilibrium Annealed Lead Telluride

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Figure A-7 Creep Characteristics of Lead Telluride at High Temperature

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thermal expansion. The metal contact member is less distorted than the element, which results because of the difference in the thermal conductivity. For the same heat flux, the thermal gradient in the metal is much less than in the thermoelectric member, so that the metal will distort less than the thermoelectric member. Obviously, if the two members were to be intimately joined, a stress would exist. The nature of the stresses will be discussed later using the assumption that the metallic member is an undistorted plane surface restricting the bending of the thermoelectric member.

Another stress condition that is apparent due to thermal expansion is a dimensional change in operation of the hot junction electrode without a corresponding change in the cold junction electrode. Consider an elementary case where the cold electrodes are held rigid and the hot electrodes are allowed to expand. The deflection of the hot junction electrode must equal the deflection of the hot ends of the elements.

$$d(T_n - T_c)l_1 - \frac{Fl_1}{(A)(E)} = 2 \frac{F}{A_2 G} l_2$$

α = coeff. thermal expansion

G = Modulus of rigidity

F = Compressive force in hot junction electrode

E = Modulus of elasticity of hot junction electrode

to solve for F Assume $G \approx \frac{E_2}{2}$

E_2 = Modulus of elasticity of thermoelectric member

l_1 = Spacing of legs

l_2 = Length of legs

$$\text{then (2) } F = \frac{\alpha(T_n - T_c) l_1}{\frac{4l_2}{A_2 E_2} + \frac{l_1}{E_1}}$$

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The stresses from the bending moment are greatest at the cold junction and will be tensile on the side of the elements closest together. Stress analysis of actual design with lead telluride type alloy show that unless flexibility is incorporated in the mechanical system, thermoelectric member fracture is extremely probable. There are many ways in which flexibility may be incorporated in designs. A thermal gradient stress factor is the result of restraining movement at the hot and cold junction areas by the soldering or welding of the thermoelectric member to a metallic strap. An example from Gatewood, "Thermal Stresses" McGraw Hill, New York, 1957, will serve to illustrate the nature of the problem. For the plate shown, and with a thermal gradient in the Y direction, the stresses away from the edges are:

$$\sigma_x = \sigma_z = \frac{E T(y)}{1 - \nu} + \frac{1}{2c(1 - \nu)} \int_{-c}^c \alpha E T(y) dy + \int_{-c}^c \alpha E T(y) Y dY \quad (43)$$

E = Young's Modulus

σ = Stress

ν = Poisson's Ratio

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This equation is valid for the completely unstrained case. If the plate is restrained in compression along the x and z axes, the correct stress may be obtained by dropping the second term. If the plate is restrained in bending (i. e. not allowed to warp) the correct stress is obtained by dropping the third term. Tensile stresses will have a positive sign and compressive stresses a negative sign. For the purpose of obtaining an approximation we may assume that the temperature function is linear so that

$$T(y) = \frac{\Delta T}{2} (1 + Y/C) \quad (44)$$

If we further assume Poisson's ratio, ν is zero and substitute Equation (44) into Equation (43) and evaluate the integral.

$$\sigma_x = -\frac{\alpha E \Delta T}{2} (1 + Y/C) + \frac{\alpha E \Delta T}{2} + \frac{\alpha E \Delta T Y}{2C} = 0 \quad (45)$$

From Equation (45) it is seen that for an unrestrained plate with a linear temperature profile the stresses are all zero. This result is completely general. Any isotropic, unrestrained body of any shape has no thermal stresses whenever the temperature varies according to the linear law.

In a thermoelectric device utilizing bonded junctions, the elements are kept from bending at the hot and cold junction areas, but are not restrained in the x and z direction, Equation (45) becomes

$$\sigma_x = -\frac{\alpha E \Delta T}{2} (Y/C) \quad (46)$$

The elements will thus be in compression for $Y > 0$, (i. e., near the hot end) and in tension for $Y < 0$ (i. e., near the cold end). The elements would like to

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bow in a convex manner at the hot end and a concave manner at the cold end when a thermoelectric element is prevented from doing so by bonding to the connecting metal the above stress is produced.

For lead telluride, the thermal gradient is not linear, but is steeper at the hot end. By expressing the thermal gradient in polynomial form it may be shown that this characteristic has the beneficial result of reducing the tensile stress at the cold junction.

For the purpose of demonstration it is convenient to restrict the consideration to a one dimensional problem. Consider a thermoelectric element as the thin strip shown:

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The member tends to bend so that the hot surface is convex. In actuality, the forces are distributed more uniformly over the ends of the element. The shear force diagram for the member is given below.

There is therefore a shear force $F/2$ acting in the plane perpendicular to the x -axis in the Y direction over the entire beam. The magnitude of F necessary to keep the element from bending can be calculated from the expression for the bending moment caused by the temperature profile.

$$M = b \int_{-c}^c E T(Y) Y dY \quad (47)$$

Knowing the equation for the temperature profile will give a solution for the bending moment and therefore a solution for the shear stresses. One shear stress will be in the plane perpendicular to x in the Y direction and an equal shear stress in the plane perpendicular to y acting in X direction. From the theory of beams it can be said that the stresses are distributed non-uniformly over the beam according to:

$$S_{xy} = S_{yx} = \frac{V}{2I} (C^2 - Y^2) \quad (48)$$

V = total shear force

I = moment of interior

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It can be seen from Equation (48) that the shear forces will be maximum at $Y = 0$ and be zero at $Y = \pm C$.

These shear forces, if calculated for practical designs, are usually not as serious as the stresses discussed earlier but they are high enough to warrant detailed examination.

From the foregoing discussion, the mechanical disadvantages of firmly bonding or welding the hot junction to a metallic joining strap are obvious. Nearly all the important stress areas can be minimized by the use of a pressure at the hot junction, rather than a bonded joint.

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REF ID: A62113

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3M Research Report

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APPENDIX B
COMPARISON OF SYSTEM DESIGN DETAILS
AS PROPOSED VS. CURRENT DESIGN

GENERATOR SYSTEM	AS PROPOSED	CURRENT DESIGN
Design life, years	Five (5) years	Same
Weight, total system, lbs.	515 lbs.	507 lbs.
System dimensions, ft.	1.37 ft. dia x 2.125 ft. long	1.32 ft dia. x 2.32 ft. long
Nominal thermal input, watts (th)	Beginning of life - 169 watts	Beginning of life - 187 watts
Nominal electrical output, watts (e)	10 watts to load	Same
Design voltage, volts	24 volts (from converter)	Same
Design current, amps	0.417 amps (from converter)	Same
FUEL LOADING		
Total weight of fuel/generator, gm	805	810
Total weight of fuel/capsule, gm	805	810
Total amount of fuel, curies	26.5×10^3 (@ 0.71 w/cc)	30×10^3 (@ 0.80 w/cc)
FUEL CAPSULE DATA		
Capsule material	Hastelloy-C	Same
Liner material (if used)	Not used	Stainless Steel (316)
Overall capsule dimensions, inches	3-1/4 dia x 3.54	3.20 dia x 3.34
Number of capsules/generator	1	Same
Internal volume of capsule, in ³	13.3	15.6
Void volume of capsule, in ³	Nominal (0)	Encapsulation clearance
Capsule design life, years	10 ⁴	Same
Wall thickness, sides, in.	3/8	1/4

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FUEL CAPSULE DATA (Continued)	AS PROPOSED	CURRENT DESIGN
Wall thickness, ends, inches	3/8 (0.47 weld end)	1/4 (3/8 weld end)
Weight of capsule, lbs.	5.15	4.0 with liner
Capsule melting temperature, °F	2318° - 2381° F	Same
Thermal conductivity of capsule material BTU/hr-ft-°F	9.9	Same
Strength of material Tensile, psi	88 x 10 ³ at 1200° F	Same
Shear, psi	29 x 10 ³ at 1200° F	Same
Closure method	Heliarc weld, 80% penetration	Heliarc weld
THERMOELECTRIC CONVERSION DATA		
Couples/generator	42	48
Couples/module	42	48
Modules/generator	1	1
Resistance, ohms	1.32 ohms (end of life)	Same
Type of circuitry	All elements in series	Same
THERMOELEMENTS		
Thermoelectric material	N-Type-PbTe P-Type-(BiSb) ₂ Te ₃ and PbTe-SnTe	N-Type-PbTe Segmented Same
Doping, a/o		
Fabrication method	Powder pressed	Same
Length/area, inches	N-Type-17.4 P-Type-13.9	N-Type-21.73 P-Type-16.15
Average operating temperature, °F	T _h = 1100° F T _c = 100° F	Same
Hot junction bonds (material, thickness, bonding method)	Pressure contact	Same
Hot shoes (material, dimensions)	Iron, 0.50 in. long x 0.310 in. wide x 0.062 in. thick	Iron, 0.548 in. long x 0.248 in. wide x 0.060 in. thick

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THERMOELEMENTS (Continued)	AS PROPOSED	CURRENT DESIGN
Cold junction bonds (material, thickness, bonding method)	Tin: 0.005 in. thick, soldered	Same
Cold shoes (materials, dimensions)	Copper, D = 3/16 in., t = 0.030 in.	Copper, D = 3/16 in. length (N) = 0.176 in. (P) = 0.330 in.
Electrical leads and connections	Copper wire - 16 ga.	Same
Dimensions, in.	N-Type: L = 0.48 in. D = 3/16 in. P-Type: L = 0.48 in. D = 0.210 in.	N-Type: L = 0.60 in. D = 3/16 in. P-Type: L = 0.446 in. D = 3/16 in.

THERMAL PROFILE

Maximum fuel temperature, ° F	1173° F	1346° F
Average centerline temperature, ° F	1155° F	1338° F
Fuel capsule surface temperature, ° F	1130° F	1333° F
Hot junction temperature, ° F	1100° F	Same
Cold junction temperature, ° F	110° F	Same
Generator surface temperature, ° F	1100° F at hot junction	Same
Total thermal input, watts (th)	169 watts	187 watts
Heat to T/E, watts (th)	135 watts	144 watts
Heat loss through insulation and structure, watts (th)	34 watts	43 watts

THERMAL INSULATION

Type	Linde Super Insulation	Same
Composition	Stainless steel and quartz	Same
Operating pressure, psi	1.9×10^{-4} psi	Same
Weight, lbs.	4.4 lbs.	40.0 lbs.
Volume, ft ³	0.256	0.675
Thermal conductivity, BTU/hr-ft-° F	0.0005	Same

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GENERATOR HOUSING	AS PROPOSED	CURRENT DESIGN
Material	Aluminum and stainless steel	Copper and stainless steel
Dimensions, ft.	0.88 ft. dia x 0.21 ft. thick	0.376 ft. dia x 0.308 ft. thick
Wall thickness	SS bellows - 0.005 in. thick SS hot plate - 1/8 in. thick Aluminum cold ring - 1.75 in. thick	SS wall - 0.005 in. th SS hot plate - 1/4 in. Aluminum cold ring - 0.835 in. thick
Coatings	None	None
Structural supports	None	None
Seals (Types, materials, dimensions)	Stainless to stainless weld and stainless to aluminum solder	Stainless to stainless weld and stainless to copper weld
Weight, lbs.	9	18.6
SHIELDING		
Material	W-5 Kennametal Tungsten Alloy	Depleted U-8 Uranium - 8 Moly
Weight, lbs.	380	256
Specifications	mr/hr at surface of generator - 200	Same
Dimensions, ft.	0.771 dia x 0.812 L	0.716 dia x 0.730 L
PRESSURE VESSEL		
Material	Titanium alloy (13V-11Cr-3 Al)	Titanium alloy 7 Al-2Cb-1Ta)
Seals (Type, materials, dimensions)	Double O Ring 14 in dia x 1/4 in. thick	Same
Weight, lbs.	122	144
Specifications	Withstand 10,000 psi external pressure	Same
Dimensions, ft.	1.37 ft dia x 2.125 ft long	1.32 ft dia x 2.09 ft long
Corrosion properties	0.0001 in./yr Titanium	Same
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DC-DC CONVERTERS	AS PROPOSED	CURRENT DESIGN
Physical location of converter	On pressure vessel	Same
Radiation intensity to converter, mr/hr	100 mr/hour	Same
Range of input voltage variation	4.97 - 4.87 volts	Same
Circuit type	Royer	Modified Royer
Converters/system	1	1
Principal components	Transistors, diodes, transformer, resistor	Same
Weight, lbs.	0.5 lb.	Approx. 5.0 lbs, in- cluding mounts
Dimensions, inches	5 in. x 3 in. x 1-1/2 in.	8 in. OD x 3 in. ID x 4 in. high
Impedance, ohms	1.97 ohms	Same
Transistor material	Germanium	Same
Output power, watts (e)	10.1 watts (end of life)	Same
Output voltage, volts	24 volts	Same
Output current, amps	0.42 (end of life)	Same
Transformer core material	50% NiFe (Orthonol)	Same
Input power, watts (e)	12.0 (end of life)	11.6 (end of life)
Input voltage, volts	4.87 (end of life)	Same
Input current, amps	2.47 (end of life)	Same
Oscillating frequency, cps	800	250
Voltage regulation, percent	$\pm 3 \%$	Same
Ripple, mv	100 mv	Same
Generator-load mismatch ration	44	Same
Operating temperature, $^{\circ}\text{F} \pm ^{\circ}\text{F}$	$45^{\circ}\text{F} \pm 5^{\circ}\text{F}$	$70^{\circ}\text{F} \pm 5^{\circ}\text{F}$

APPENDIX C

HEAT TRANSFER FROM THE RADIATION SHIELD TO THE THERMOELECTRIC GENERATOR

The heating of the hot frame of the thermoelectric generator can be accomplished by radiation and conduction through air or by conduction only. This heat transfer is from the biological radiation shield to the hot frame. Introduction of an air gap between the hot frame and shield will necessitate radiative heat transfer with inherent additional heat losses because of a higher source temperature. The advantages of an air gap are:

1. No direct connection exists between the radiation shield and the generator, thereby minimizing shock which could be transmitted from the shield.
2. No movement of the thermoelectric leg assemblies occurs due to compression of the pressure vessel.
3. Higher reliability is attained because the hot and cold frame do not shift or move with respect to each other except through differential thermal expansion.

A compromise of thermal efficiency for system integrity and reliability is the net result of utilizing an air gap and radiative heat transfer.

Prior to acceptance of heat transfer by radiation and conduction through air for the SNAP 21-B system, an analysis is required to establish the system fuel loading and identify pertinent design parameters associated with radiative heat transfer.

The following analysis was performed for different thickness air gaps to determine the air gap which could be used and to determine component temperatures for design purposes.

1. Heat requirements to the generator were previously determined to be 165 watts (th) at beginning of life.

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2. The heat transfer formula used to determine the heat transfer across the gap between the radiation disc (emitter on biological shield) and the hot frame is:

$$Q = \underbrace{\frac{K A \Delta T}{L}}_{\text{Heat Transfer by Conduction (through air)}} + \underbrace{E_T \sigma A \left[\left(\frac{T_2}{100} \right)^4 - \left(\frac{T_1}{100} \right)^4 \right]}_{\text{Heat Transfer by Radiation}}$$

Given: Heat source and heat sink diameter = 4.5 inches

Area of heat transfer = $\frac{\pi D^2}{4} = 15.9 \text{ inches}^2$
= 0.1105 Ft²

$K_{\text{air}} @ 1200^\circ \text{ F} = 0.03691 \frac{\text{BTU}}{\text{hr-ft}^\circ \text{ F}}$

where K = the thermal conductivity factor

Assume Grey Bodies where:

$$Q = E_T \sigma A \left[\left(\frac{T_2}{100} \right)^4 - \left(\frac{T_1}{100} \right)^4 \right]$$

Where: $\sigma = \frac{173 \text{ BTU}}{^\circ \text{R} - \text{ft}^2 \text{ hr}}$ (When T has denominator of 100)

E_T is total emissivity factor

$$E_{\text{total}} = \frac{1}{\frac{1}{E_1} + \frac{1}{E_2} - 1}$$

A = Area of radiation interface.

T_2 = Heat source temperature ($^\circ \text{R}$) (Shield)

T_1 = Heat sink temperature ($^\circ \text{R}$) (Hot Frame)

(Where $^\circ \text{R} = 459^\circ + ^\circ \text{F}$)

The emissivity of some metals such as 304 stainless steel and Haynes-25 has and are being investigated by General Electric (Ref. - SNAP 17-B Power Supply Phase I Quarterly Report, March, 1964, pages 3-127).

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Also, the G. E. report (p. 3-126) references NASA Memo 1-20-59L, January, 1959, Field, Virginia, which gives a description of apparatus and methods of measuring total hemispherical emissivity of some metals. The metals include 347 stainless steel, tungsten, Haynes' alloys B, C, X and 25. The results indicate that strongly adherent oxide coated surfaces can be formed which possess a highly stable emittance up to 2000° F. Haynes' alloys oxidized to emittances ranging from 0.90 to 0.96 for alloy C; from 0.85 to 0.86 for alloy X, and from 0.85 to 0.89 for alloy 25. For this analysis, emissivities of both the generator hot frame and radiation disc will be assumed equal. Two values (0.8 and 0.9) were analyzed to yield results at each extreme of the emissivity range.

The following table shows the heat transfer of 165 watts by radiation and conduction through the air gap. Various air gaps (from 1/16 to 5/16 inch) and the method and amount of heat transfer by that method and the temperature of the radiating surface are shown for two total emissivity factors.

TABLE C-1
HEAT TRANSFER BY RADIATION
AND CONDUCTION (THROUGH AIR) FROM A RADIATION SHIELD
TO A HOT FRAME FOR 165 BTU OF HEAT TRANSFER

AIR GAP	Q THRU AIR GAP BY CONDUCTION (ASSUME $\Delta T = 200^{\circ}F$)	Q THRU AIR GAP BY RADIATION	SHIELD TEMP. WHEN $E_T = 0.667$	WHEN $E_T = 0.80$
1/16	45.6	119.4	1281° F	1251° F
1/8	22.8	142.2	1306° F	1279° F
3/16	15.3	149.8	1315° F	1285° F
1/4	11.4	153.6	1321° F	1291° F
5/16	9.1	155.9	1324° F	1292° F

As indicated in the table, the change of total emissivity of 0.667 to 0.8 lowers the temperature of the radiating surface approximately 30° F. Also, the closer the radiator and sink are, the lower the radiating temperature is (approximately 40° F lower when going from 5/16 inch gap down to 1/16 inch gap).

A 3/16 inch gap at sea level closing down to 1/8 inch due to the axial compression of the pressure vessel would give a radiation shield temperature of approximately 1306° F to 1315° F for an overall emissivity of 0.667 and

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approximately 1285° F to 1279° F for an overall emissivity of 0.8. This would give a 1100° F generator hot frame temperature while transmitting the heat required for generator operation.

Heat transfer due to convection currents in the 3/16 inch gap have been neglected because of the small volume of air under consideration.

Values Used for Determination of K Factor for Air

For Air @ 1070° F,	K =	Difference	where K = $\frac{\text{BTU}}{\text{hr-ft-}^\circ\text{F}}$
1160°	0.03483	0.00145	
1250°	0.03628	0.00142	
1340°	0.03770	0.00131	
	0.03901		

For conduction across air:

$$Q = \frac{KA\Delta T}{L}$$

where $Q = \frac{\text{BTU}}{\text{hr}}$

$$K = \text{Thermal conductivity of air} = \frac{(0.03691)\text{BTU}}{\text{hr-ft-}^\circ\text{F}}$$

$$A = \text{Area of conduction interface} = 0.110 \text{ ft.}^2$$

$$L = \text{Distance between surfaces in ft.}$$

$$Q = \frac{(0.03691)\text{BTU}}{\text{hr-ft-}^\circ\text{F}} (0.11 \text{ Ft}^2) (200^\circ\text{F})$$

$$\frac{\left(\frac{1}{16}\right) \left(\frac{1}{12}\right) \text{ Ft}}{\text{Ft}}$$

$$Q = 155.8 \text{ BTU;}$$

$$\text{Also } 3.42 \text{ BTU} = 1 \text{ watt,}$$

$$\text{Thus } 155.8 \text{ BTU} = 45.6 \text{ watts}$$

Therefore, the heat conducted across the 1/16 inch air gap is 45.6 watts for 200° F temperature difference.

$$Q = 45.6 \text{ watts when air gap} = 1/16 \text{ inch}$$

$$Q = 9.1 \text{ watts when air gap} = 5/16 \text{ inch}$$

$$Q = 22.8 \text{ watts when air gap} = 1/8 \text{ inch}$$

$$Q = 2.85 \text{ watts when air gap} = 1 \text{ inch}$$

$$Q = 15.2 \text{ watts when air gap} = 3/16 \text{ inch}$$

$$Q = 1.42 \text{ watts when air gap} = 2 \text{ inches}$$

$$Q = 11.4 \text{ watts when air gap} = 1/4 \text{ inch}$$

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Sample Calculation for Heat Source Temperature (T_2):

For the Given Conditions;

Total heat transfer = 165 watts

Given, heat transferred = 45.6 watts by conduction when air gap = 1/16 inch

Therefore, necessary heat transfer by radiation equals $165 - 45.6 = 119.4$ watts

$$119.4 \text{ watts} = 119.4 \text{ watts} \times \frac{3.42 \text{ BTU}}{\text{watt}} = 407 \text{ BTU}$$

$$Q = E_T \sigma A \left[\left(\frac{T_2}{100} \right)^4 - \left(\frac{T_1}{100} \right)^4 \right]$$

Case I $E_T = 0.667$

$$Q = 407 = (0.667) \frac{(0.174) \text{ BTU}}{^\circ \text{F-ft}^2\text{-hr}} (0.110 \text{ Ft.}^2) \left[\left(\frac{T_2}{100} \right)^4 - (15.6)^4 \right]$$

$$\text{Therefore: } (T_2)^{\circ \text{R}} = 1740 = (1740 - 459)^{\circ \text{F}}$$

$$T_2 = 1281^{\circ \text{F}}$$

Case II $E_T = 0.8$

$$Q = 407 = (0.80) \frac{(0.174) \text{ BTU}}{^\circ \text{F-ft}^2\text{-hr}} (0.110 \text{ Ft.}^2) \left[\left(\frac{T_2}{100} \right)^4 - (15.6)^4 \right]$$

$$\text{Therefore: } (T_2)^{\circ \text{R}} = 1710 = (1710 - 459)^{\circ \text{F}}$$

$$T_2 = 1251^{\circ \text{F}}$$

APPENDIX D

COLD ELECTRODE THERMAL ANALYSIS

INTRODUCTION

To allow for deformation without breakage of thermoelectric legs, a ball and socket joint can be incorporated into the leg assembly. For freedom of movement, the socket radius can be made slightly larger than the ball radius. This analysis determines the thermal characteristics of such a joint at various radii of the ball and socket.

Each configuration studied is specified by r_1 , the ball radius; r_2 , the socket radius, and θ , the angle through which heat is being transferred.

It was decided to investigate thermal characteristics for three different combinations of radii and angle.

Case I.	r_1	=	0.125 inches
	r_2	=	0.128 inches
	θ	=	90°
Case II.	r_1	=	0.325 inches
	r_2	=	0.329 inches
	θ	=	22.3°
Case III.	r_1	=	0.525 inches
	r_2	=	0.530 inches
	θ	=	13.7°

For each case the distance between the ball and socket at the top of the ball was allowed to vary from 0 to 0.003 inches to determine the effects of separation and to provide for a check on the method. (For Case I, if GAP = 0.003 inches this is the concentric case.) The heat to be transferred in each case was 2.5 BTU/hr, and the conductivity of the gap material was 0.014 BTU/hr-ft- $^\circ$ F.

Method:

Since the configuration does not fit any of the idealized cases found in the literature, it was decided to treat it as a thermal resistive network consisting of many thermal resistors and reduce this to an equivalent resistor.

ΔT was then calculated as $\Delta T = R_{eq} \cdot Q$.

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To obtain these resistors, each configuration was broken into a number of intervals and the results were compared against the same configuration with a greater number of intervals until convergence was attained.

Each resistor was calculated by assuming that the interval was small enough to use the flat plate heat transfer formula:

$$Q = \frac{KA\Delta T}{L}$$

Then, if:

A_i = surface area of the i th segment using R_2 equivalent radius

L_i = distance between surfaces at the midpoint of the i th segment

then:
$$R_1 = \frac{L_i}{A_i K}$$

A_i can be determined by using the formula for surface area of a sector and subtract the previous sector.

$$A_i = (2)(\pi)(R_2)(H_i - [H_i - Z])$$

where: Z is dependent on the specific previous sector.

L_i can be determined by calculating C_i and D_i and subtracting.

From the geometry:

$$D_i = \frac{R_3 \sin(\theta_i - \sin^{-1} [(R_3 - R_2) \sin(\pi - \theta_i)/R_3])}{\sin(\pi - \theta_i)}$$

$$C_i = \frac{R_1 \sin(\pi - \theta_i - \sin^{-1} [(R_2 - R_1) \sin(\theta_i)/R_1])}{\sin(\theta_i)}$$

$$L_i = D_i - C_i$$

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This method was incorporated into a computer program which was run for the three cases under study.

The following assumptions were made in this analysis:

1. The surfaces of the ball and socket are perfectly smooth.
2. When the Gap at the top = 0 the contact is a point contact with Area of Contact = 0.
3. Both surfaces are isothermal.

To check the accuracy of the method, the gap at the top of the ball (gap-top) was set equal to 0.003 inches in Case I and compared with the idealized concentric case. (See calculation.) The method was very accurate at this point.

CALCULATIONS

For 2 concentric spheres.

$$Q = \frac{4\pi K r_1 r_2 \Delta T}{r_2 - r_1}$$

For a hemisphere.

$$Q = \frac{2\pi K r_1 r_2 \Delta T}{r_2 - r_1}$$

Let $r_1 = 0.125$ inches

$r_2 = 0.128$ inches

$K = 0.014$ inches

$Q = 2.5$ BTU/hr

$$\Delta T = \frac{Q(r_2 - r_1)}{2\pi K r_1 r_2}$$

$$\Delta T = \frac{2.5 \times 0.003 \times 12}{2(0.014)(0.125)(0.128)} = 64^\circ \text{ F}$$

From computer program with GAP-TOP = 0.003

$$\Delta T = 63.9^\circ \text{ F}$$

RESULTS AND CONCLUSION

The results of the study are shown in the Table of Temperature Differences (Table D-1) and on the graph (Figure D-1). It can be seen from these that temperature difference is very sensitive when GAP-TOP is small

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and is very small at GAP-TOP = 0, in Case I. Case I and Case II are nearly linear because they closely approximate the flat plate condition. Normally, the isothermal surface assumption can be made even if temperature varies 10 degrees from one point to another. Here, however, since ΔT is so small when GAP-TOP is zero, this assumption may not be valid. The zero area contact assumption is, of course, conservative. The effects of smoothness of surfaces may be appreciable since ΔT is critical at low GAP-TOP values.

In reality the area of contact will be greater than zero, and the surfaces will be neither smooth nor isothermal. It is felt, however, that these assumptions are relatively valid. Actual effects of ball and socket joints will be experimentally determined in development hardware to either prove or disprove this analysis.

TABLE D-1

TEMPERATURE DIFFERENCES OF A BALL AND SOCKET
COLD JOINT AS A FUNCTION OF GAP-TOP

CASE	GAP AT TOP OF BALL (INCHES)							
Gap (Inches)	0	0.00001	0.00003	0.00005	0.0001	0.0002	0.0003	0.00035
I $\Delta T^{\circ}F$	4.5	11.17	13.74	15.35				
II $\Delta T^{\circ}F$	1.2				9.11		18.06	
III $\Delta T^{\circ}F$	0.8			4.5		11.3		17.8
Gap (Inches)	0.0005	0.0007	0.001	0.002	0.003			
I $\Delta T^{\circ}F$			38.8	52.6	63.9			
II $\Delta T^{\circ}F$	26.5	34.8	47.2	88.4	129.5			
III $\Delta T^{\circ}F$	24.2		45.5	88.0	130.5			

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Figure D-1 - Temperature Difference vs. Gap-Top

APPENDIX E

SNAP 21-B PRESSURE VESSEL QUALITY SPECIFICATIONS FOR 721 TITANIUM

CHEMICAL REQUIREMENTS

1. The chemical composition of all mill products (plates, bars, forgings, extrusions, etc.) shall conform to the following requirements:

<u>ELEMENT</u>	<u>CHEMICAL COMPOSITION *</u>
Aluminum	6.6 - 7.3
Columbium	2.0 - 2.75
Tantalum	0.8 - 1.2
Carbon	0.030
Nitrogen	0.012
Iron	0.180
Oxygen	0.080
Hydrogen	0.008
Other	0.50
Titanium	Remainder

* - Percent, maximum unless a range is shown or unless otherwise stated.

2. The vendor shall conduct or have conducted a certified chemical analysis for each mill product and shall supply 3M Company with a copy of all test results.

MECHANICAL REQUIREMENTS

1. All component parts shall meet the following minimum mechanical property requirements subsequent to final forming, heat treating, machining, and stress relieving operations:

<u>PROPERTY</u>	<u>MINIMUM VALUE</u>
Ultimate Tensile Strength	115,000 psi *
Compressive Yield Strength (0.2 percent offset)	110,000 psi *
Tensile Yield Strength (0.2 percent offset)	100,000 psi *
Elongation in two inches	10 percent
Reduction in area	20 percent
Charpy - V notch at - 80° F	25 ft. lb. **
Charpy - V notch at +32° F	35 ft. lb. **

* - Based on average of two specimens in each direction per location.

** - Based on an average of three specimens in each direction per location.

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2. All mechanical property requirements shall be certified by test results. The vendor shall conduct or have conducted all necessary mechanical testing and shall submit to 3M Company a copy of all test results including:
 - a. Portion of product from which test specimens were taken (location and direction).
 - b. Type and dimensions of test specimens.
 - c. Description of testing equipment.
 - d. Method of testing (rate of loading for tensile and compressive tests, impact velocity for Charpy tests).
 - e. Load-strain curves (tensile and compressive tests).
 - f. Mode of fracture (Charpy tests).
3. All tested specimens shall become the property of 3M Company and shall be delivered with the completed product.
4. The vendor shall certify that the properties of the test specimens are representative of the entire product.
5. All excess mill product material shall become the property of 3M Company and shall be delivered with the completed product. The vendor shall indicate the location of the excess material with respect to both the finished product and the mechanical test specimens.
6. All component parts' exterior surfaces shall be free of oxygen, nitrogen, and iron contamination.

SPECIMEN REQUIREMENTS

1. All mechanical test specimens shall be machined from each mill product component after final forming, heat treating, machining and stress relieving operations or after any final thermal or forming operation which may materially affect the mechanical properties. Specimens shall be taken longitudinal and transverse to the direction in which the material is most drawn or forged.

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2. All tensile specimens and testing procedures shall conform to the American Society for Testing of Materials' Tentative Methods of Tension Testing of Metallic Materials (ASTM-E 8-61T). Round specimens 1 or 2 shall be used. A strain rate of 0.003 ± 0.001 inch per minute shall be used throughout the test.
3. All compressive specimens and testing procedures shall conform to the American Society for Testing of Materials' Standard Methods of Compression Testing of Metallic Materials (ASTM-E 9-61). Medium length specimens shall be used. A loading rate of 250 psi per minute shall be used beyond the proportional limit.
4. All Charpy-V-notch impact specimens and testing procedures shall conform to the American Society for Testing of Materials' Standard Methods for Notched Bar Impact Testing of Metallic Materials (ASTM-E 23-64).

THERMAL REQUIREMENTS

1. The beta transus temperature shall not be exceeded during forging, extruding, or forming operations.
2. Formed components shall be stress relieved by soaking for one hour at 1400°F , plus 50°F , minus 0°F . Cooling shall be in air.

QUALITY ASSURANCE REQUIREMENTS.

1. All mill products shall be 100 percent ultrasonically inspected prior to rough and final machining operations (except those machining operations necessary to conform to the inspection specifications). Inspection procedures and techniques shall conform to Military Specification MIL-I-8950 (ASG), 28 August, 1964, Process for Ultrasonic Inspection of Wrought Metals. The minimum acceptable quality level shall be class AA.
2. The vendor shall conduct or have conducted all necessary ultrasonic inspections and shall submit a copy of all test results to 3M Company.

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3. All cracks, inclusions, etc., which are revealed by initial ultrasonic inspections may be removed providing dimensional specifications can be maintained. Additional ultrasonic inspections shall be conducted to substantiate defect removal.
4. All component parts shall be penetrant inspected prior to and after final machining operations. Inspection procedures and techniques shall conform to Military Specification MIL-I-6866B (ASG), 26 February, 1964, Penetrant Method of Inspection. Type I penetrants shall be used.
5. The vendor shall conduct or have conducted all necessary penetrant inspections and shall submit a copy of all test results to 3M Company.
6. All components shall be 100 percent penetrant inspected. No defects shall be acceptable except those which may be removed during final machining operations providing dimensional specifications can be maintained. Additional penetrant inspections shall be conducted to substantiate defect removal.

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APPENDIX F

IDENTIFICATION OF DC-DC CONVERTER TERMINOLOGY

1. IDENTIFICATION OF THE TERM "SERIES CONFIGURATION"

If a converter is designed to convert from a certain voltage, for instance 5 to 25 volts, the efficiency will be some number, assume 84%. Since the voltage ratio is not too large, the overall conversion efficiency can be increased by reducing the converter output voltage to 20 volts and putting the output in series with the input. This connection is called the series configuration.

Figure F-1 - Comparison of "Series Configuration" and "Basic Converter"

The output voltage is maintained at 25 volts, but only 20/25 of the power has to be converted. The remaining 5/25 of the power is obtained at near 100% efficiency. The overall efficiency obtained depends on the switching time of the converter. If the switching time is neglected, the series configuration will result in an 86.75 % efficiency, a net gain of 2.75 %. If the switching time is 2 % of the total cycle time, the series configuration will have an efficiency of 86.25 %, a net gain of 2.25 % through the series connection. Figure F-2 shows the expected improvement in conversion in series with the converter input. The values in the above example were obtained from the graph.

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Figure F-2 - Percent Gain in Efficiency Using a Converter in Series with Its Input

2. HIGH FREQUENCY GAIN CONSIDERATIONS

There are certain conditions that must be met before a DC-DC converter will switch from one state to the other. The principal condition that must be met is that the loop power gain be greater than unity at some point where the loop phase shift is exactly 0°. This condition must hold when the transformer is in a saturated state and full load is applied.

The loop power gain is calculated as the ratio of input power change to output power change for a particular portion of the circuit. These powers are usually measured by disconnecting the feed back winding from the transistor base (Point 1, Figure 6.2). A small change in power is applied to the base, and the change in power which is fed back to Point 1 through the transformer is measured. The ratio of the power change which is

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fed back to the power change which is put in; is the loop power gain. Note that these powers have nothing to do with the power put into or taken from the entire converter.

To see why this condition must be fulfilled, consider the following:

Twice during each cycle of the converter, a condition is set up that is nearly identical to the initial starting condition for the converter. This condition is with the transformer saturated in one state or the other. Both transistors receive starting bias and are drawing approximately equal current. Since the transformer is saturated, there is essentially no voltage across any winding. In order for the converter to switch so that the core flux is built up to saturation in the other direction, it is necessary for a high-frequency oscillation to begin. This oscillation interacts with the non-linear permeability of the core material to cause an average demagnetization of the transformer core. The average voltage produced by this demagnetization causes one transistor to conduct more current on the average and switching occurs.

In a severe case of difficult starting, these oscillations can be observed, but usually they persist for such a short time that switching follows the first portion of a cycle.

If these oscillations would normally occur at a frequency near or above the cut-off frequency of the transmitter, the converter will not always start, and the switching time will be quite long. This is because the loop gain is near unity, when the phase shift is 0° .

In order to minimize problems with high frequency cut-off of the transistors, the operating frequency is kept well below the cut-off frequency of the transistors. This ensures that the resonant frequency of the saturated transformer is within the pass-band of the transistors.

If the above relationships cannot be met, it is possible to lower the resonant frequency of the transformer by placing a capacitor across one of the windings. If excessive capacity is added, the converter efficiency will decrease.

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3. OPTIMUM FEED BACK VOLTAGE

Figure F-3 illustrates the effect of total feed back circuit voltage on the maximum feed back power loss. This graph is normalized so that it can be used with any converter. If a given transistor has a base-to-emitter voltage that might vary between 0.250 and 1.000 volts during its operating life and the input voltage is five (5) volts, the minimum peak loss will be 3.5 % at a base-to-emitter voltage of 0.250 volts. This value is obtained

Figure F-3 - Maximum Feed Back Circuit Power Loss

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Figure F-4 - Feed Back Voltage for Minimum Peak Feed Back Loss

by using the curve for $V_{be\ min} / V_{be\ max} = 0.25$ and observing that its minimum occurs at 17.5%. The 17.5% figure is divided by $E_{in} / V_{be\ max} = 5/1 = 5$ to obtain 3.5%. The effect of selecting a feed back voltage different from the optimum can be easily seen on this graph. This value of feed back voltage represents the sum of the starting voltage and the transformer feed back voltage.

Figure F-4 represents the result of differentiating the equation of Figure F-3. This graph is a plot of the normalized feed back voltage for minimum peak feed back loss. If a certain transistor has a base-to-emitter voltage that might vary between 0.250 and 1.00 volts, the feed back voltage for minimum peak feed back power is 1.7 volts. This is obtained by entering the graph at $V_{be\ min} / V_{be\ max} = 0.250/1.00 = 0.250$ and finding 1.7 on the vertical axis. This value is divided by the maximum feed back voltage of 1.00 volts to obtain the desired value of 1.7 volts.

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